NON-PROPRIETARY VERSION

ENCLOSURE 2

RESPONSE TO NRC REQUEST FOR ADDITIONAL INFORMATION USE OF ASME CODE CASES N-756 AND N-757 SECTION III DIVISION 1 DESIGN REPORTS

DOMINION ENERGY KEWAUNEE, INC. (DEK)
DOMINION NUCLEAR CONNECTICUT, INC. (DNC)
VIRGINIA ELECTRIC AND POWER COMPANY (DOMINION)

KEWAUNEE POWER STATION UNIT 1
MILLSTONE POWER STATION UNITS 2 AND 3
NORTH ANNA POWER STATION UNITS 1 AND 2
SURRY POWER STATION UNITS 1 AND 2

Swagelok Company
29450 F.A. Lennon Drive
Solon, Ohio
44139

Design Report

ASME BP&V Section III, Class 3
SS-45S8-18622-NSR Ball Valve
SS-45XS8-18623-NSR Ball Valve

Created/Approved by:

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05-19-07

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1.0 Design Report Certification Statement

CERTIFICATION

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of design and using the certified Design Specification and the drawings identified below as a basis for design, do hereby certify that to the best of my knowledge and belief the Design Report is complete and accurate and complies with the design requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division 1, 2004 Edition.

Design Specification and Revision: BSPEC-04940-00004, Rev. 001 (BP)

Design Report and Revision: SCS-00684, Rev. **DRAFT-2**

Certified by: Michael T. Gallagher , P.E.

Registration No. E-50154 State: Ohio

Date: 05/22/2007

2.0 Scope

The Swagelok 45S8 and 45XS8 valve has been analyzed to verify conformance to ASME Boiler and Pressure Vessel Code, Section III, Subsection ND, Class 3 components, 2004 edition. Code case N-757 will be invoked in this design report. These two valves are included in this design report because of their similarity.
This report includes the following: a description of the nuclear code components and their materials of construction; a stress analysis of the valve features determined to be critical to pressure containment; a diagram of the valve showing the location of these features.

3.0 References

3.1 ASME Boiler and Pressure Vessel Code, Section III, 2004 edition
3.2 ASME Boiler and Pressure Vessel Code, Section III, 2004 edition – Appendix XIII
3.3 ASME Boiler and Pressure Vessel Code, Section III, Code Case 757

4.0 ASME Section III Code Components

Following the guidelines of ND-1100(d) the following components are classified as ASME section III nuclear code components:

Description	Part number
Body (45XS8)	SS-1-45XS8-18623-NSR
Body (45S8)	SS-1-45S8-18622-NSR
Packing Bolt	SS-4A-45-18622-NSR
Stem (45XS8)	SS-3-45X-K-18623-NSR
Stem (45S8)	SS-3-45-K-18622-NSR

See Figure 1 for part schematics

5.0 Material Specifications

Body – 316 SS extruded bar UNS S31600 ASTM A276 / A479
Packing Bolt – 316 SS – UNS S31600 -ASME SA479 Level 2 Strain hardened
Stem – 316 SS – UNS S31600 -ASTM A276 condition B

Material Strength (psi /1000)				
Part Description	**Type of Stress**	**100°F**	**150°F**	**Notes**
Body	S_m (ksi)			(1)(4)
	S_y (ksi)			(1)(3)
Packing Bolt	S_m (ksi)			(1)(4)
	S_y (ksi)			(1)(3)
Stem	S_m (ksi)			(1)(4)
	S_y (ksi)			(1)(3)

Notes:
(1) Section ND2121(d) was implemented to allow the use of materials not listed in ASME Section II – Part D.
(2) Sm values obtained from ASME Section II, Part D, Table 2A
(3) Sy values obtained from ASME Section II, Part D, Table Y-1
(4) Sm values obtained from ASME Section II, Part D, appendix 2 criteria

6.0 Stress Analysis - Discussion

The stress analysis portion of this report is organized based on locations of the valve that are considered essential to pressure containment in the valve.

Several valve locations/features are analyzed in this report. The analyzed locations are shown on the drawing on the following page. The applicable loading scenarios and resulting stresses are calculated at each location. Calculated stress components are categorized into the appropriate design and loading level. Principal stresses are then calculated.

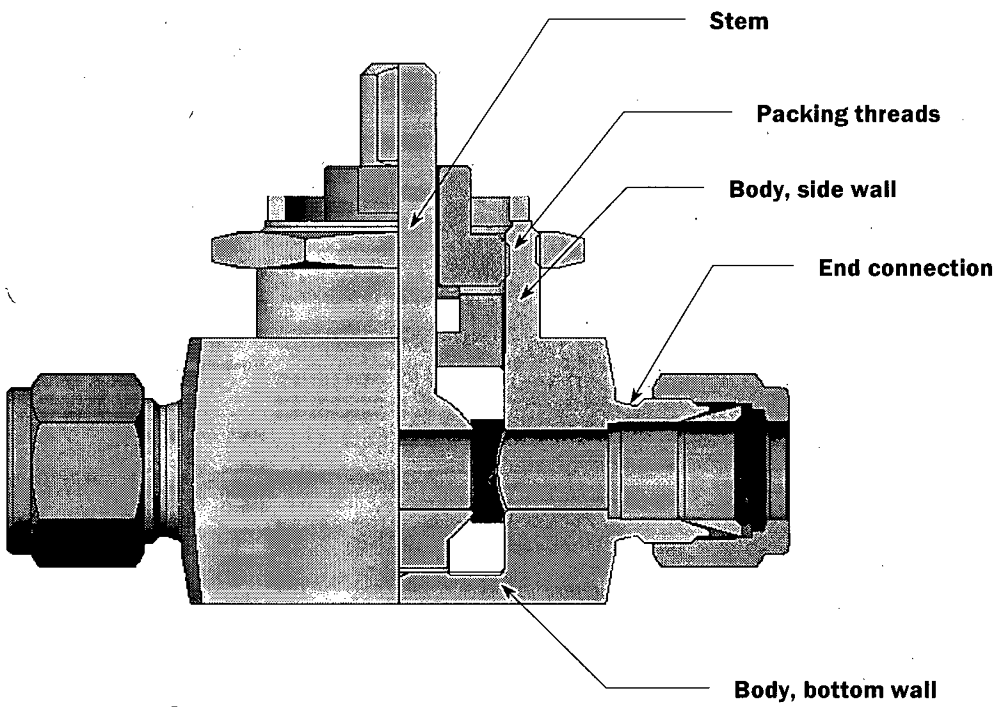
Code Case N-757 specifies ASME Section III, Appendix XIII as one of the options that can be used to analyze a valve product, and Swagelok has selected this option for the analysis in this design report. This appendix contains Five Basic Stress Intensity Limits that must be satisfied. Based on the type of loading the valve will be seeing, not all of the five limits may be applicable for a given feature.

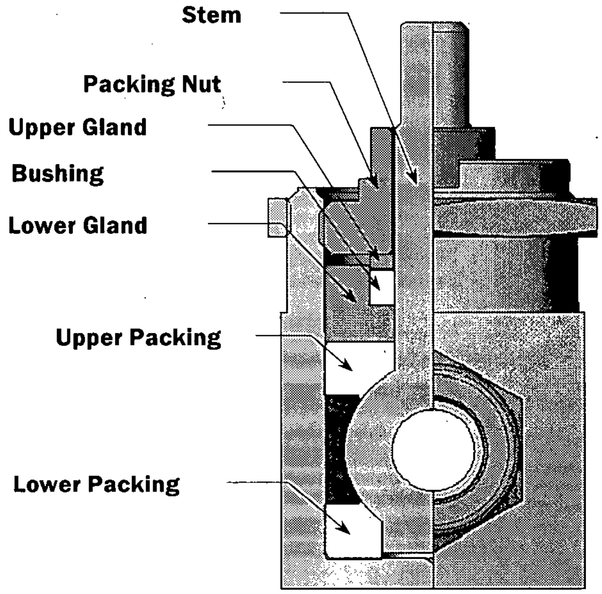
ASME Section III, Appendix XIII, paragraph XIII-1145 - Primary plus Secondary Stress Intensity requires that the combination of primary and secondary stress not exceed $3S_m$. Swagelok will use this allowable stress intensity for cases when the secondary loading results from a displacement load, i.e. a threaded joint assembled to a particular angular displacement.
There are cases where a secondary type loading results from a non-displacement type load (such as a threaded joint loaded to a given torque). Swagelok will treat these the same way that bolt stress intensities are addressed in ASME Section III Appendix XIII: Total loading cannot exceed 3 S_m. Bolting material S_m values typically equal 1/3 x yield strength, therefore $3S_m$ will typically equal the minimum yield strength at temperature.

7.0 Conclusion

The ball valve part numbers SS-45S8-18622-NSR and SS-45XS8-18623-NSR meet the requirements of ASME Section III, Subsection ND, as described in this design report.

8.0 Figure 1: SS-45S8 diagram





9.0 Body – main bore

Pressure Load: tangential (hoop) and longitudinal stress, ND-3324.3

$$S_h = \frac{PR}{t} + .6P \qquad S_L = \frac{PR}{2t} - .2P$$

Where:
- P = pressure =[ ]
- R = inside radius =[ ]
- t = wall thickness =[ ]
- S_h = hoop (tangential) stress
- S_L = longitudinal stress

S_h =[ ]
S_L =[ ]

[ ]

Radial stress is also present:

S_r = - pressure (acting at the ID) =[ ]
S_r = 0 (at the OD)

Assembly load, tangential pressure from packing

The assembly/adjustment torque on the packing bolt creates an axial force that acts on the packing components. In turn, this axial force results in a radial force in the packing. A conservative ratio of radial packing pressure / axial packing pressure for the axial stress at[ ]is 0.6, ref. "Valve Packings that Don't Leak", Lyons Valve Designer's Handbook, 1982.

Calculate assembly force from the empirical relationship: F = T / k x d
where:

Fa = Axial assembly Force lb
T = Assembly Torque, max. =[ ]
k = thread friction factor =[ ]
d = nominal thread size =[ ]
Ro = radius: packing bore, min.,[ ]
R = radius: stem, max. =[ ]

A_P = pressure area, packing; = π ($Ro^2 - R^2$) =[ ]

Fa =[ ]

Axial packing pressure = Fa / A_P (packing pressure area) = [ ]

Radial packing pressure = .6 x Axial packing pressure = [ ]

Stress components from resulting packing pressure:

Use thin wall equations, since t < .5 R

Tangential (hoop) and longitudinal stress from equation above:

Where: P = radial packing pressure = [ ]
R = inside radius = [ ]
t = wall thickness = [ ]

S_h from radial packing pressure = [ ]
S_L from radial packing pressure = [ ]

Radial Stress:

S_R = - Radial Packing Pressure, P (from above) = [ ]

Design Loading

Primary membrane stresses, resulting from pressure, ref XIII -1142.
Since there is no shear stress in this region the 3 principal stresses equal S_H; S_L, and S_R

The 3 principal stresses, stress intensity, and allowable stress intensity are tabulated below.

Temperature (°F)	Principal 1	Principal 2	Principal 3	S_I	Sm allowable
150	[				]

Service Level A Loading

Primary + secondary stresses, ref. XIII-1145 resulting from pressure and assembly loads.
Since there is no shear stress in this region the 3 principal stresses equal $\sum S_h$, $\sum S_L$, and $\sum S_R$

Where:
$\sum S_h$ = hoop stress from system pressure and assembly force
$\sum S_L$ = longitudinal stress from system pressure and assembly force
$\sum S_R$ = radial stress from system pressure and assembly force

Note: Adding the pressure and assembly stresses is a conservative step: Typically the packing pressure is sufficiently high to allow system pressure to be acting in all of the areas that are included in the calculation.

The 3 principal stresses, stress intensity, and allowable stress intensity are tabulated below.

Temperature (°F)	Principal 1	Principal 2	Principal 3	S_I	Sm allowable
150	[				]

10.0 Body – bottom

Pressure Load: Bending and shear stress

Bottom of the bore will be modeled as a circular plate with fixed edge constraints. From Roark, Formulas for Stress and Strain, table 24, case 10b, 5th ed.

$$S_{BP} = \frac{3}{4}\frac{Pr^2}{t^2}$$

Where: P = max. system pressure =[ ]
r = radius of main bore, max =[ ]
t = wall thickness, =[ ]
S_{BP} = Bending stress, from pressure

S_B =[¾ x 500 x .5325^2 / .141^2 =]

Shear:

$\tau_P = P \times A_P / A_S$

where: A_P = Pressure area, bottom of bore =[ ]
A_S = Shear area, bottom of bore =[ ]

τ_P =[ ]

Assembly load - bottom of bore

The assembly/adjustment torque applied to the packing nut results in a distributed force (pressure) at the bottom of the bore. The radial packing pressure and the resulting friction will cause a reduction in actual packing pressure at the bottom of the bore, as calculated below:

Packing force, bottom = Packing force – radial packing pressure, mean x packing surface area x coefficient of friction, bore to packing
Packing force, bottom=[ ]

Packing Pressure, bottom of bore =[ ]

Bending:
As is the case above, bottom of bore will be modeled as a circular plate with fixed edges.

S_{BA} = bending stress from assembly =[ ]

Shear:

τ_A = Packing pressure x A_P / A_S

where: A_P = Pressure area, packing =[ ]

A_S = Shear area, bottom of bore = [ ]

τ_A = [ ]

Design Loading

Primary membrane stresses, resulting from pressure, ref XIII -1142.
Principal stresses are calculated using the standard equation for a general 2D state of stress

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\tau^2{}_{xy} + (\frac{\sigma_x - \sigma_y}{2})^2}$$

Where:
$\sigma_x = S_{BP}$ = bending stress from pressure
$\tau_{xy} = \tau_P$ = shear stress from pressure

The 3 principal stresses, stress intensity, and allowable stress intensity are tabulated below.

Temperature (°F)	Principal 1	Principal 2	Principal 3	S_I	Sm allowable
150	[				]

Service Level A Loading

Primary + secondary stresses, ref. XIII-1145 resulting from pressure and assembly loads.
Principal stresses are calculated using the standard equation for a general 2D state of stress

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\tau^2{}_{xy} + (\frac{\sigma_x - \sigma_y}{2})^2}$$

Where:
$\sigma_x = S_{BA}$ = bending stress from assembly
$\tau_{xy} = \tau_A$ = shear stress from assembly
Note: Packing pressure is sufficient to seal system pressure.

The 3 principal stresses, stress intensity, and allowable stress intensity are tabulated below.

Temperature (°F)	Principal 1	Principal 2	Principal 3	S_I	Sm allowable
150	[				]

The 45X body has a minimum bottom wall thickness of [ ], thicker than the 45 body analyzed above, therefore no analysis will be required for the 45X bottom bore.

11.0 Body and Packing Bolt – packing threads

Pressure Load

System pressure will result in forces that cause shear stress in the packing bolt and body threads, and axial stress in the body adjacent to the threads.

Shear:

$\tau_P = P \times A_P / A_S$

where: A_P = Pressure area, packing bore =[ ]
A_S = Shear area, threads=[ ], **body (internal) thread**
=[ ], **packing bolt (external) thread**

τ_P (Body thread)=[ ]
τ_P (packing bolt)=[ ]

Axial:

$S_P = P \times A_P / A_T$

Where: A_P and P as noted above
A_T = Tensile area, upper body threads =[ ]

S_P =[ ]

Assembly load

Shear:

$\tau_A = F_A / A_S$

where: F_A = Assembly axial force =[ ]
A_S = Shear area, threads, from above

τ_A (Body thread) =[ ]
τ_A (Packing bolt) =[ ]

Axial:

$S_A = F_A / A_T$

Where: F_A and A_T as noted above

S_A =[ ]

Design Loading

Primary membrane stresses, resulting from pressure, ref XIII -1142.
Body thread principal stresses are calculated using the standard equation for a general 2D state of stress, at the thread root. Stresses at body thread and packing bolt thread are pure shear.

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\tau^2{}_{xy} + (\frac{\sigma_x - \sigma_y}{2})^2}$$

Where:

σ_Y = S_P = axial tensile stress from pressure

τ_{xy} = τ_P = shear stress from pressure

The 3 principal stresses, stress intensity, and allowable stress intensity are tabulated below.

Feature	Temperature (°F)	Principal 1	Principal 2	Principal 3	S_I	Sm allowable
Body Thread	150					
Packing bolt	150					

Service Level A Loading

Primary + secondary stresses, ref. XIII-1145 resulting from pressure and assembly loads.
Principal stresses are calculated using the standard equation for a general 2D state of stress
Body thread principal stresses are calculated using the standard equation for a general 2D state of stress, at the thread root. Stresses at body thread and packing bolt thread are pure shear.

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\tau^2{}_{xy} + (\frac{\sigma_x - \sigma_y}{2})^2}$$

Where:

σ_Y = S_P = axial tensile stress from assembly

τ_{xy} = τ_P = shear stress from assembly

Note: Packing pressure is sufficient to seal system pressure.

The 3 principal stresses, stress intensity, and allowable stress intensity are tabulated below.

Feature	Temperature (°F)	Principal 1	Principal 2	Principal 3	S_I	Sy allowable
Body Thread	150					
Packing bolt	150					

12.0 Stem

Design Loading

Stresses created by pressure loads are negligible.

Operation Loading

Torsional shear will result from the torque required to operate (rotate the ball) the valve. The stem region analyzed here is the minimum section within the pressure containing region

The ball valve will experience it's maximum torque value at the start of stem/ball rotation, the initial operating torque. This is the initial torque required to overcome static friction. This torque will be present for approximately 1 degree of rotation. This torque value will be included in the primary + secondary analysis (service level A).
The stem will experience a lower value of operating torque for the remainder of the stem/ball rotation, the run torque. This torque value is included with the primary membrane +primary bending loading analysis.

$\tau_{Ox} = T_{Ox} \times r / J$
where:
T_{OI} = Initial (maximum) operating torque, [ ]
T_{OR} = Run torque, [ ]
J = polar moment of inertia = [ ]
r = radius of stem [ ]
τ_{OI} = [ ]
τ_{OR} = [ ]

Primary loading (membrane + bending)
Shear and axial stresses resulting from pressure and operation (run torque) will be combined to solve for principal stresses and stress intensity.

Temperature °F	Principal 1	Principal 2	Principal 3	S_I	1.5 Sm allowable
150	[				]

Service Level A Loading

Primary + secondary stresses, ref. XIII-1145 resulting from operation loads.

The only component stress present is the torsional shear from operation.

The 3 principal stresses, stress intensity, and allowable stress intensity are tabulated below.

Temperature (°F)	Principal 1	Principal 2	Principal 3	S_I	Sy allowable
150	[				]

Fatigue loading

Fatigue loading will be considered for the stresses resulting from valve operation.

Sa will be calculated, Sa is the alternating stress component. When the ball valve is fully cycled (open to close followed by close to open) the state of stress will be totally reversed. The stress that alternates during operation is the torsional stress.

The initial (maximum) stress resulting from the[ ] initial torque will be used in this analysis.

S_A =[ ]

Note: .6 factor is used to convert shear stress to principal stress since alternating stress is pure shear.

Peak Stress

S_A will be multiplied by K_S, the stress concentration factor, to find the peak stress. Peak stress will be compared to the tabulated values of N, number of cycles, and S_{AP}, peak alternating stress from the fatigue tables in ASME Section III, Appendix I to determine cycle life. Figure I-9.2.1 applies to austenitic stainless steels, and values are tabulated in table I-9.1

$S_{AP} = K_S \times S_A$

where: K_S = **2.3**, ref. Stress, Strain and Strength, Robert Juvinall, figure 13.8.[ ]

S_{AP} =[ ]

The tables can be interpolated to solve for number of cycles:

[ ]

End Connection

Pressure Load, tangential (hoop) from ND-3324

$$S_h = \frac{PR}{t} + .6P$$

Where: P = pressure,[ ]
R = inside radius ,[ ]
t = wall thickness, see below
S_h = hoop (tangential) stress

t = (ext. root diam – max. int throat diam) / 2 - pos. tolerance
=[ ]

S_h =[ ]

Longitudinal stress

$S_L = P \times A_P / A_T$

Where: R = radius @ end of 20° flare:[ ]
A_P = pressure area, PIxR^2 =[ ]
A_T = thread cross-section area, ext. =[ ]
S_L = longitudinal stress

S_L =[ ]

Radial stress, S_R is also present:

S_R @ ID =[ ]

S_R @ OD =[ ]

[ ]

Pressure Load, thread shear stress

External (end connection) thread shear area, min = [ ]

$$\tau_p = \frac{PA_P}{A_S}$$

Where:

P = pressure
R = radius @ end of 20° flare: [ ]
A_P = pressure area, PIR^2 = [ ]
A_s = thread shear area, ext., noted above

τ_p (thread shear) = [ ]

Assembly Loading

Longitudinal (axial) stress, threaded area

The axial force, Fa, generated by the assembly of the Swagelok fitting is calculated from the empirical equation:

Fa = T / (kd)

where:
T = assembly torque = [ ]
k = thread friction factor [ ]
d = fitting thread size = [ ]

Fa = [ ]

Longitudinal stress

S_L= Fa / A_L

where:
Fa = from above
A_L = [ ]

S_L = [ ]

Thread Shear Stress, threaded area:

$$\tau_A = \frac{F_A}{A_S}$$

Where: A_s = shear area, fitting thread, ext = [ ]
Fa = Axial make-up Force lb (from above)

τ_A (thread) = [ ]

Design Loading

Primary membrane stresses, resulting from pressure, ref XIII -1142.

Principal stresses are calculated using the standard equation for a general 2D state of stress

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\tau^2{}_{xy} + (\frac{\sigma_x - \sigma_y}{2})^2}$$

Where:

$\sigma_x = S_r$ = radial stress from pressure

$\sigma_y = S_l$ = axial (longitudinal) stress from pressure

τ_{xy} = Ss = shear stress from pressure

Note that the tangential pressure stress S_h is orthogonal to the other stresses and is therefore a principal stress.

The 3 principal stresses, stress intensity, and allowable stress intensity are tabulated below.

Feature	Temperature (°F)	Principal 1	Principal 2	Principal 3	Sm	Sm allowable
Thread	150	[				]

Service Level A Loading

Threaded area

Primary + secondary stresses, ref. XIII-1145 resulting from pressure and assembly loads.

Principal stresses will be calculated using this standard equation for a general 2D state of stress, equation above

Where:

$\sigma_x = S_r$ = radial stress from system pressure

$\sigma_y = S_l + S_A$ = axial (longitudinal) stress from system pressure and assembly force

τ_{xy} = Ss = shear stress from assembly and pressure

The 3 principal stresses, stress intensity, and allowable stress intensity are tabulated below.

Feature	Temperature (°F)	Principal 1	Principal 2	Principal 3	Sm	3Sm allowable
Thread	150	[				]

13.0 ND-3521 Analysis

To satisfy ND-3521, both cross sectional area and section modulus of the component in question must be greater than 1.1 times that of the attached tube member. Tubing tolerances are taken from ASTM A1016/A 1016M – 04A, Standard Specification for General Requirements for Ferritic Alloy Steel, Austenitic Alloy Steel, and Stainless Steel Tubes, page 766, Table 2.

$Z_B = \frac{bh^2}{6} - \frac{\pi d_i^4}{32h}$ section modulus of cross section, body crotch

where: b = body width = [ ]
h = body height = [ ]
d_i = body port ID = [ ]

$A_B = bh - \frac{\pi}{4} d_i^2$ area of cross section, body crotch

$Z_T = \frac{\pi}{32D_O}(D_O^4 - D_i^4)$ section modulus of connecting tubing (1/2" OD, .083 wall)

where: D_O = tubing OD = **.505 in**
D_i = tubing ID = **.3058 in** (min ID, based on max. wall tolerance)

$A_T = \frac{\pi}{4}(D_O^2 - D_i^2)$ area of cross section, connecting tubing

Therefore,

[ ] = .452 in^3 $Z_T = \frac{\pi}{32x.505}(.505^4 - .3058^4)$ = .0109 in^4

[ ] = 1.879 in^2 $A_T = \frac{\pi}{4}(.505^2 - .3058^2)$ = .1268 in^2

Z_B = .452 in^3 > .012 in^3 = 1.1 x Z_T

A_B = 1.879 in^2 > .1395 in^2 = 1.1 x A_T

14.0 Appendix – Code Case N-757

BC06-872

Approval Date January 22, 2007

Case N-757
Alternative Rules for Acceptability for Section III,
Division 1, Class 2 and 3 Valves, NPS 1 (DN 25) and Smaller
With Welded and Nonwelded End Connections other than Flanges

Inquiry: Under what rules may instrument, control and sampling line valves, NPS 1 (DN 25) and smaller, with welded and nonwelded end connections other than flanges, meet the design requirements of Section III, Division 1, Class 2 and 3 rules of NC/ND-3512, when the valve minimum wall thickness does not meet the t_m requirements of ASME B16.34?

Reply: It is the opinion of the Committee that instrument, control and sampling line valves, NPS 1 (DN 25) and smaller, having valve minimum wall thickness not in accordance with the t_m requirements of ASME B16.34, with welded and nonwelded end connections other than flanges, may meet the design requirements of Section III, Division 1, Class 2 and 3 rules of NC/ND-3500, provided the following additional requirements are met:

(a) Valves not meeting the t_m wall thickness requirements of ASME B16.34, shall meet the pressure design rules of NC/ND-3324; an experimental stress analysis (Section III, Division 1, Appendix II); or Design Based on Stress Analysis (Section III, Division 1, Appendix XIII). The design shall be qualified in accordance with the requirements of MSS-SP-105-2005, Section 5.
(b) The end connections shall meet the requirements of NC/ND-3661, -3671.3 or -3671.4, for welded, threaded, and flared, flareless and compression type fitting tube ends.
(c) Valve bonnets threaded directly into valve bodies shall have a lock weld or locking device to assure that the assembly does not disengage either through stem operation or vibration.
(d) This Code Case number shall be identified on the NPV-1 Data Report Form.

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4UW Design Report

Revisions

Revision	Date	Scope of the Revision
-		

Index:

1.0 SCOPE

The Swagelok 4 UW valve has been analyzed to verify design conformance to the ASME Boiler and Pressure Vessel Code Section III, Division I, Sub-section NB, class 1 components, year 2004.

Swagelok bellows valve products typically do not conform to ASME B16.34, mainly the result of conscious design decisions made on the basis of the industries Swagelok serves. Most of the design rules of NB-3500, Valve Design, require conformance to B16.34 as a starting point. Accordingly, Swagelok is using alternative design rule NB-3512.2(d), a rule that does not require conformance to B16.34. Rather, it relies on the in-depth stress analysis required per NB-3200, Design by Analysis.

Note: Alternative design rule NB-3512.2(d) applies to valves with welded ends. Code case N-756 must be used for valves with NPT and Swagelok compression tube end connections

2.0 REFERENCES

2.1 ASME Boiler and Pressure Vessel Code Section III "Rules for Construction of Nuclear Power Plant Facility Components"

2.2 ASME B16.34 "Valves – Flanged, Threaded, and Welding Ends"

Analysis Method:

The analysis includes the valve body, bonnet nut, bonnet, and gland nut. A finite element (FEA) model of the bonnet and body was developed and used to analyze the majority of the stresses in these two components. Established engineering equations and method were used to analyze the balance of the key features; this part of the analysis will hereafter be referred to as the "analysis by calculation".

The types of loading analyzed in this report include: Pressure, weight, thermal effects, seismic, external piping. When any one of these loading effects are not included in a particular analysis it is either not applicable and/or negligible for that region.

Design Conditions:

Pressure / Temperature conditions:

Pressure (psig)	Temperature °F
3600	-20 to 100
3095	200
2795	300
2570	400
2390	500
2255	600
2220	650
2170	700
2135	750
2110	800

Bonnet Nut Assembly Torque: [ ]

Handle Closure Torque: [ ]

Seismic Acceleration: 6g

Erosion, corrosion or other effects that would result in a loss of valve component material are not accounted for in this analysis.

Design Piping Loads:

Fitting loads must meet the following equation:

3/8" tube butt weld end fitting:

$$F_{shear} / 135 + F_{axial} / 275 + M_{torsion} / 20 + M_{bending} / 20 \leq 1$$

Where:
F_{shear} – force in pounds applied parallel to the face of the end connection
F_{axial} – force in pounds applied parallel to the flow axis of the end connection
$M_{torsion}$ – Torsional moment (in-lb) applied to the end connection
$M_{bending}$ – Bending moment (in-lb) applied to the end connection

Note: The absolute value of F_{shear}, F_{axial}, $M_{torsion}$, and $M_{bending}$ must be used in these equations.

Structure of this report:

As noted above, two methods were used to develop this report: Finite Element Analysis (FEA) and analysis by calculation.

The FEA results are reported by first describing the particular valve region analyzed, followed by a table that contains the actual values for stress intensity compared to the appropriate allowable stress intensity value. Values are tabulated from 100° F to 800° F. The FEA software has performed the calculations, therefore they are not shown in the report. The FEA software is ANSYS .

Analysis by calculations are also reported by first describing the particular valve region. Loading and stress calculations, at 100° F conditions, are then shown. Table(s) containing the calculated stress intensities and allowable stress intensities are then listed for 100° F to 800° F conditions.

Allowable Stress Intensity Criteria:

The 2004 Edition of the ASME Section III code will be includes a change to requirement NB-2121(d) that will allow 1" and under valves to be constructed of material made to specifications other than those listed in Section II, Part D, Subpart 1, Tables 2A and 2B, provided the valves meet the requirements of NB-3200 or NB-3500. This is the basis by which the allowable stress intensity values have been chosen for the analysis of the 4UW valve.

The ASME criteria of using the minimum of either 1/3 of the tensile strength at room temperature and 1.1 x 1/3 of the tensile strength at elevated temperatures or 2/3 of the yield strength at temperature forms the basis for the allowable stress intensities used in this report. Raw material product shape and size (hex, round, etc.) as well as proximity to weld joints dictate the allowable stress intensity value.

Here is a tabulation of the components/features and the room temperature tensile and yield strengths used to obtain the allowable stress intensity:

Bonnet, gland nut, bonnet nut
Body
Body, at pipe welds
Bonnet, close to weld

Minimum tensile and yield strength at elevated temperature are listed in Section II for 316 stainless steel at[ ]. The values not directly listed in Section II are obtained by interpolation. These minimum tensile and yield values are then used to obtain allowable stress intensity, as described above. The[ ] strength value, and is applied when the feature being analyzed includes a weld. The[ ]is a conservative value, chosen based on the feature being near the weld. Strength values for this feature are not expected to be affected by the nearby welding, but for the sake of conservatism are being reduced.

Code Components:

Part Number	Drawing	Description	Material	Rev.	Deviation / NCM
SS-4B-P0-xxxxx-NSR	4B-BW6-049-P0-NSR	Body	316 SS SA479	x	
	4U-UPPERBODY-NSR	Upper Body detail		x	
174PH-4BS-P3-xxxxx-NSR	4BS-P3	Stem Insert	174 PH SS ASTM A564	x	
SS-4U-P4-xxxxx-NSR	4U-P4-NSR	Weld Ring	316 SS SA479	x	
SS-4U-P6-xxxxx-NSR	4U-P6-NSR	Bonnet	316 SS SA479	x	
SS-4B-UA7A-xxxxx-NSR	4B-UA7A-NSR	Bonnet Nut	316 SS SA479	x	
SS-4U-P12-xxxxx-NSR	4U-P12-NSR	Gland Nut	316 SS SA479	x	

Stress Limit Criteria:

All regions analyzed include design loading and service level A limit analysis, with the exception of regions that experience special stress states as described in NB-3227.

The relatively small mass of this valve results in seismic loading and stress magnitudes that are quite low. Regions that contain non-negligible seismic stress are analyzed to service level C limits. treating seismic loading as a primary load.

A fatigue analysis per NB-3222.4 is included in this report. Thermal stresses are assumed to be general thermal stresses in the fatigue analysis, which allows the use of a constant value for Poisson's ratio, ref. NB-3227.6(b).

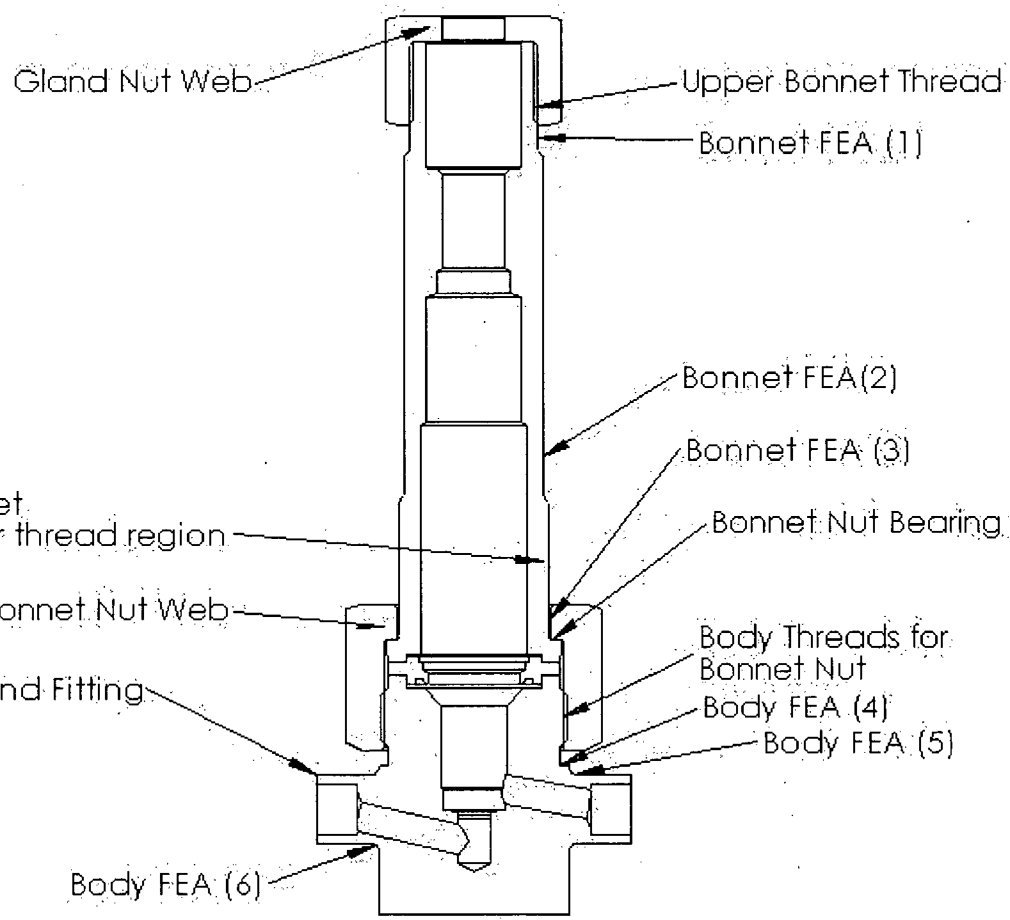
Thermal stress analysis is based on the valve being in the open position, with system temperature on the internal flow passages and ambient external conditions.

Conclusion:

The results of the FEA analysis and analysis by calculation show that all regions analyzed by these methods meet the requirements of ASME Section III.

4UW stress analysis diagram

This diagram shows the FEA and analysis by calculation regions that are analyzed in this report



4UW Body thread for Bonnet Nut analysis

Length of Engagement:

Shear Area:

1 - 20UNEF 2A/2B

Max minor, int .957
Max pitch, int .9734
Min major, ext .9905
Min pitch, ext .9616

Internal (bonnet nut) = SAbn =
External (body) = SAb =

Additional dimensions:

Body bore ID, max =
Tensile (axial) area of upper body =

Loading:

Pressure Loads

Pressure Area, based on ID of bonnet to weld ring seal weld =

Axial force resulting from pressure, Fay =

Thread shear stress due to pressure, τ_{xyp}
τ_{xyc} =

The pressure stress will be evaluated at the body thread shear diameter, equal to the minor diameter of the bonnet nut.

Use the general equation for tangential (hoop) stress

σ_{hoop} = *pressure x* ri^2 / ($ro^2 - ri^2$) x (1+ ro^2 / r^2)
ri = inner radius

ro = outer radius
r = radius at point of interest

Tangential (hoop) stress due to pressure, σ_{zp}
σ_{zp} (shear diam.) =[ ]

Axial stress due to pressure, σ_{yp}
σ_{yp} = axial pressure force / tensile area of upper body =[ ]

Radial stress due to pressure, σ_{xp}
σ_{xp} (shear diam.) =[ ]

Assembly

Stresses resulting from the assembly of the bonnet nut to the body

Assembly torque [,]
Use T =KFd to find F, empirical relationship between torque, friction and force in threaded assemblies.

K =[ ] thread friction factor,[ ]
d = 1.00, nominal thread size

Assembly Force, Fa = T/kd = [ ]

Thread Shear Stress, assembly (100F):
τ_{xya} = Fa / SAb =[ ]

Along with the shear stress generated by the assembly load there will also be a compressive stress
Compressive stress, adjacent to threads:
σ_{ya} = Fa / Aub =[ ]

where:
Aub = cross section area of the cylindrical region adjacent to the body threads
=[ ]

Handle closure Loads

Handle Torque =[ ]

Handle Closure Force, Fc = T /kd =[ ]

Where : k =[,] thread friction factor with[ ]
d = .375, nominal diameter of stem threads

Shear stress, due to Fc, τ_{xyc} = Fc / Sab = [ ]

Seismic Loading

Fxs = 6 x
Fys = 6 x
Fzs = 6 x
Total seismic moment, from Fxs and Fys, Mts = sqrt ($9.6^2 + 9.6^2$) x moment arm
Where moment arm = y distance from threads to valve c.g.
Mts =

Section Modulus of upper body =

Axial seismic stress =

Bending seismic stress =

Shear seismic stress, threads =

Shear seismic stress, across section =

Thermal Loading

Thermal stresses in this region were obtained from the Finite Element analysis and included in the Service level A calculations.

Relative Stiffness

The relative stiffness of this joint was conservatively set at 0.5, based on FEA results of a similar joint. This means a service load applied to the joint, such as pressure or closure, will be distributed such that 50% of the load will act on the threads, and increase the shear and axial stresses accordingly. The remaining 50% of the load will act to decompress the compressed portion on the joint.

Resolution of Stresses:

The stresses calculated above will be combined, and principal stresses will be calculated. The principal stresses are then used to determine stress intensities. Calculated stress intensities are then compared to the allowable stress intensity or allowable stress intensity multiple, per NB-3220.

Design Loadings: Primary Stresses

Principal stresses determined from pressure stresses

Service level A limits: Primary and secondary stresses:

Principal stresses determined from the combination of pressure, assembly, closure and thermal stresses.

Service level C limits: Primary stresses:

Principal stresses determined from the combination of seismic and pressure stresses

The first table below contains the appropriate combinations of stress intensities compared to the specific allowable stress intensities, for design conditions and service level A conditions. The alpha factor for the NB-3221.3 allowable stress intensity is conservatively chosen as 1.0. The second table contains the appropriate combinations of stress intensities compared to the specific allowable stress intensities, for service level C. Allowable stress intensities are conservatively based on annealed material, since this region is close to the weld ring to body weld.

Temp. °F	Design Loading				Level A Service Limits	
	NB-3221.1 General Primary Stress Intensity	NB-3221.1 allowable stress intensity , Sm	NB-3221.3 Primary membrane + primary bending	NB-3221.3 allowable stress intensity, 1.0 Sm	NB-3222.2 primary + secondary stress intensity	NB-3222.2 allowable stress intensity 3 Sm
100						
200						
300						
400						
500						
600						
650						
700						
800						

Temp. °F	**Level C Service Limits**	
	NB-3224.1 Primary Stress Intensity	NB-3224.1 allowable stress intensity , (greater of 1.2Sm or yield)
100		
200		
300		
400		
500		
600		
650		
700		
800		

4UW End Fitting analysis

3/8" x .049 tube ends are included in this analysis.

Minimum OD = [ ]
Maximum ID = [ ]
Area = [ ]
Moment of Inertia = [ ]
Polar Moment of Inertia = [ ]

[ ]

Pressure

Use the general equation for tangential (hoop) stress:

σ_{hoop} *(ID)* $= pressure \times ri^2 / (ro^2 - ri^2) \times (1 + ro^2 / ri^2)$

σ_{hoop} *(OD)* $= 2 \times pressure \times ri^2 / (ro^2 - ri^2)$

σ_{hoop} (ID) = [ ]

σ_{hoop} (OD) = [ ]

Axial stress due to pressure:

$\sigma_{axial} = pressure \times ri^2 / (ro^2 - ri^2) = pressure \times ID^2 / (OD^2 - ID^2)$

σ_{axial} = [ ]

Radial stress due to pressure:

σ_{radial} *(ID)* = *-pressure*

σ_{radial} *(OD)* = *0*

<u>Seismic</u>

6g acceleration values are employed to determine the seismic forces, which are simultaneously applied in three orthogonal directions, at the center of mass of the valve. The valve is assumed to be fully constrained at one of the end fittings.

Fx = Fy = Fz = [ ]

Distances from fitting centerline to center of mass:
Lx = [ ] (along fitting axis)
Ly = [ ] (along stem axis)
Lz = [ ] (perpendicular to x and y)

The moments that result from these loads and positions:

Bending:
Mx = 0
My = Fz x Lx = [ ]
Mz = Fx x Ly + Fy x Lx = [ ]

Torsion:
Tx= Fz x Ly = [ ]
Ty = Tz = 0

Weight

We will use the same assumption that was used in the seismic analysis, the valve is fully constrained at one end fitting.

Fyw = [ ]
Mzw = [ ]

Shear stress from Fyw = [ ]
Bending stress from Mzw (OD) = [
Bending stress from Mzw (ID) =]

External Tubing Loads, Design conditions:

Allowable force and moment values are based on not exceeding the Design loading allowable stress intensity.

Force axial = [
Force shear =
Moment bending =
Moment torsion =]

External Tubing Loads, service level A

The 3/8 x .049 tube loads are based on 30 ksi tensile yield strength and 24 ksi shear yield strength, per the requirements of NB-3512.2(d)(1)

Here are the loads at 100F, loads at elevated temperatures are based on the yield strength at temperature

Tube	Bending (in-lb)	Axial (lb)	Torsion (in-lb)
3/8 x .049	[		]

Design Load Analysis

This analysis will include the primary effects of pressure and weight and design pipe loadings:

The worst case for stress intensity occurs at the ID of the fitting.

Since this analysis includes the maximum primary stresses , NB-3221.1 and 3221.3 can be accounted for simultaneously. Alpha factor for NB-3221.3 is conservatively chosen as 1.0

Temp. °F	Design Loading		
	3/8" x .049		NB-3221.1, NB-3221.3 allowable stress intensity , Sm
	NB-3221.1 , NB-3221.3 General Primary Stress Intensity		
100			
200			
300			
400			
500			
600			
650			
700			
800			

Service Level A analysis

This analysis will include pressure, weight, thermal and the service level A tubng loads listed above. NB-3512.2(d)(1) requires that these piping loads be applied in four separate ways:

1. Direct tension (axial) in the x direction
2. Bending about the z axis (In the plane of the valve)
3. Bending about the y axis (perpendicular to the plane of the valve)
4. Torsion about the x-axis

Direct Tension:

σ_x , axial tube = axial piping load / area

σ_x , axial pipe =

To complete the analysis, σ_x , axial pipe, is added to the axial pressure stress, axial thermal stress, and bending stress from the weight to obtain the x normal stress. The other two normal stresses include the hoop, radial pressure, and thermal stresses. The x, y and z normal stresses are combined with the xy, xz, and yz shear stresses to find the principal stresses. Combinations including both tensile and compressive axial pipe forces are included in our spreadsheet analysis.

Bending about z axis:

σ_x, z bending pipe = pipe level A bending moment x distance from neutral axis / moment of inertia + axial pressure stress + bending stress from weight. The remainder of the analysis is similar to Direct Tension, above Analysis will include the effects of compressive and tensile bending, to capture the worst case stress intensity.

From the spreadsheet review of the above combinations, the worst case is the tensile bending case, at the fitting OD. The stress intensities are compiled in the table below

Bending about y axis:

Due to the symmetry of the fitting, this analysis would be identical to bending about the z axis.

Torsion about the x axis:

Pressure stresses σ_{hoop}, σ_{axial}, and σ_{radial} are combined with torsional shear and thermal to find principal stresses.

Thermal stresses obtained from the FEA are included in all of the Level A service limit stresses tabulated below.

The summary table below contains the calculated stress intensities compared to the allowable limits. Stress intensities are based on annealed material, due to the welded joint

Temp. °F	Level A Service Limits						
	3/8 x .049 end						
	NB-3222.2 primary + secondary stress intensity, axial pipe loading	NB-3222.2 primary + secondary stress intensity, bending pipe loading	NB-3222.2 primary + secondary stress intensity, torsion pipe loading				NB-3222.2 allowable stress intensity 3 Sm
100							
200							
300							
400							
500							
600							
650							
700							
800							

Service level C analysis

This analysis will include seismic effects, along with weight, pressure and design piping loads

The state of stress resulting from these loads has all six components of stress, x, y ,and z normal stresses etc. We will conservatively add the shear stresses together directly, creating a slightly more conservative model, but much simpler. Also, the seismic bending stresses will be added together directly: This is a conservative step compared to locating the theoretical maximum combined moment.
(For reference: the location of the maximum bending stress, resulting from the combined effects of Mz and My is: $\tan \theta = Mz/My$, θ measured from z axis.)

The summary table below contains the calculated stress intensities compared to the allowable limits. Stress intensities are based on annealed material, due to the welded joint.

Temp. °F	Level C Service Limits		
	3/8 x .049		NB-3224.1 allowable stress intensity , (greater of 1.2Sm or yield)
	NB-3224.1 Primary Stress Intensity		
100			
200			
300			
400			
500			
600			
650			
700			
800			

4U Bonnet Nut bearing stress

The contact area between the inside face of the bonnet and the bonnet lip will experience bearing stress:

Bearing area = {(bonnet lip OD – bonnet lip chamfer $)^2$ - (bonnet nut ID – bonnet nut ID chamfer $)^2$ } x PI / 4

Bearing Area = [ ]

Bearing Loads, from bonnet nut thread analysis

Assembly load
Pressure load:
Closure load:

These loads are combined, based on NB-3227.1, including the relative stiffness factor of 0.5 described above in the "12UW Body thread for Bonnet" analysis.

Bearing stress =[ ]

The table below compares the bearing stress to the yield strength, the allowable bearing stress per NB-3227.1

Temp. °F	NB-3227.1 Bearing Stress	NB-3227.1 Allowable bearing stress
100		
200		
300		
400		
500		
600		
650		
700		
800		

The top of the bonnet lip is not expected to be in the heat affected zone following the bonnet to weld ring weld. However, the allowable bearing stress has been reduced, conservatively, to account for the possibility of some strength loss in this region.

4U Bonnet Nut web stress

Stresses are analyzed in the inner corner of the web. Thermal stresses, obtained from the FEA analysis, are included

Shear area = web thickness x Bonnet nut ID x PI

Shear area =

Tensile area =

Temp. °F	NB-3221.1 General Primary stress intensity	NB-3221.1 allowable General Primary stress intensity	NB-3222.2 primary + secondary stress intensity	NB-3222.2 allowable stress intensity 3 Sm
100				
200				
300				
400				
500				
600				
650				
700				
800				

4U Gland Nut Web Analysis

Relevant Dimensions:

Bonnet Packing Bore ID = [ ]
Stem OD = [ ]
Bonnet Nut Web Thickness = [ ]
Gland Nut Radius = [ ]
Load Radius = [ ]
Gland Nut Internal Radius = [ ]
Poisson's Ration (316 SS) = .27
Gland Nut Thread = 11/16-28 UN-2B
Nominal Thread Size = 11/16
Thread Coefficient of Friction = [ ]

Design Loads:

1. Primary:
    a. 3600 psi internal pressure

Pressure Area = π {(Bonnet Packing Bore ID)^2 – (Stem OD)^2}/4
= [ ]
= [ ]

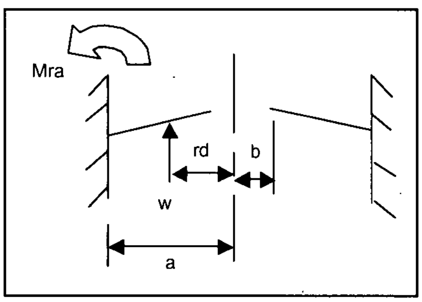
Pressure Load = [ ]
= [ ]

2. Secondary:
[ ]

Axial load due to gland nut torque = F_a

F_a = Gland Nut Torque / (Thread Coefficient of Friction * Nominal Thread Size)
F_a = [ ]
F_a = [ ]

Bending Stress Formula:
Source: Roark & Young



M_{ra} = unit radial bending moment per inch of circumference

$M_{ra} = w*a\ (L_9 - C_7L_6/C_4)$
Where L_9, C_7, L_6, and C_4 are defined by:

$L_9 = (r_d/a)[.5(1+\nu) \ln (a/r_d) + .25(1-\nu)\{1-(r_d/a)^2\}]$
$C_7 = .5(1-\nu^2)(a/b - b/a)$
$L_6 = (r_d/4a)[(r_d/a)^2 - 1 + 2 \ln (a-r_d)]$
$C_4 = .5[(1+\nu)(b/a) + (1-\nu)(a/b)]$

Substituting variables,

$L_9 = .256$
$C_7 = .722$
$L_6 = .052$
$C_4 = 1.057$

4U Gland Nut Analysis- 100 in/lbs Assembly Torque:

Temp. °F	Design Loading				Level A Service Limits	
	NB-3221.1 General Primary Stress Intensity	NB-3221.1 allowable stress intensity , Sm	NB-3221.3 Primary membrane + primary bending	NB-3221.3 allowable stress intensity, 1.5 Sm	NB-3222.2 primary + secondary stress intensity	NB-3222.2 allowable stress intensity 3 Sm
100						
200						
300						
400						
500						
600						
650						
700						
800						

4UW Gland Nut / Bonnet Thread analysis

Length of Engagement:

Min. Engage. = ( )

Chamfer size: ( )

Shear Area:

11 / 16 - 28UN 2

Max minor, int	.657 in	Internal (gland nut)	= ( )
Max pitch, int	.6692	External (bonnet)	= ( )
Min major, ext	.6799		
Min pitch, ext	.6594		

Additional Dimensions:

Pressure seal area of packing = ( )

Loading

Pressure Loads

Valve pressure will result in primary hoop, axial, and shear stress.

Use the general equation for tangential (hoop) stress

σ_{hoop} = *pressure x* $ri^2 / (ro^2 - ri^2)$ x $(1 + ro^2 / r^2)$

σ_{hoop} ( )

Axial pressure stress:

Axial pressure stress = pressure x seal area / upper bonnet tensile area

= ( )

Shear stress: = pressure x seal area / thread shear area

=

Assembly

Forces resulting from assembly of gland nut to bonnet

Nominal assembly torque ; use T =KFd to find F
Assume a maximum assembly torque of

K =
d = .6875, nominal thread size

F = T/kd =
=

This force will generate a shear stress in the threads and axial stress in the bonnet, and the axial force in the packing will create radial sealing pressure

Shear Stress
Shear Stress

Axial stress in bonnet:
Bonnet tensile area =

Axial stress in packing :
Seal area cross-section =

Use the stress ratio, K, defined in "Valve Packings that Don't Leak" ref. Valve Design Handbook
At 1333 psi axial stress, K = .8 , upper bound
At 5322 psi axial stress, K = .5 , upper bound
At 9313 psi axial stress, K = .5 , upper bound

Radial packing pressure = K x Axial packing pressure

Radial packing pressure at
Radial packing pressure at

Apply this radial packing pressure:

Use the general equation for tangential (hoop) stress

σ_{hoop} = *pressure x* $ri^2 / (ro^2 - ri^2) \times (1 + ro^2 / r^2)$

ri = inner radius
ro = outer radius
r = radius at point of interest

tang stress due to packing pressure,

σ_{hoop}
σ_{hoop}

note: radial stress is negligible in this region

Resolution of Stresses

Design conditions: Primary stresses

Principal stresses are determined from pressure stresses

Service level A conditions: Primary and secondary stresses

Principal stresses are determined from the combination of pressure, assembly and thermal stresses.

The table contains the appropriate combinations of stress intensities compared to the specific allowable stress intensities. The alpha factor for NB-3221.3 allowables is conservatively set to 1.0

Temp. °F	**Design Loading**				**Level A Service Limits** (using 350 in-lb gland nut torque)	
	NB-3221.1 General Primary Stress Intensity	NB-3221.1 allowable stress intensity , Sm	NB-3221.3 Primary membrane + primary bending	NB-3221.3 allowable stress intensity, 1.0 Sm	NB-3222.2 primary + secondary stress intensity	NB-3222.2 allowable stress intensity 3 Sm
100						
200						
300						
400						
500						
600						
650						
700						
800						

4U Bonnet analysis – lower thread region

The 13/16 –20 threaded region of the bonnet is analyzed in this part of the report. The 13/16 threads are used for the panel mounting nut, and the load effects of that connection are included here.

Relevant geometry:

Bonnet ID [ ]
Bonnet OD
Tensile (axial) area of bonnet [ ]
Shear area of bonnet thread at panel nut [ ]

Loading:

Pressure Loads

Valve pressure will produce tangential (hoop), axial and radial stresses in the bonnet wall

Use the general equation for tangential (hoop) stress

σ_{hoop} = *pressure x* ri^2 */* $(ro^2 - ri^2)$ *x* $(1 + ro^2 / r^2)$

ri = inner radius
ro = outer radius
r = radius at point of interest

σ_{hoop} (ID) = [ ]
σ_{hoop} (OD) = [ ]

σ_{axial} = *pressure x bonnet* ID^2 *x .785 / tensile (axial) area of bonnet*

σ_{axial} = [ ]

σ_{radial} (ID) = [ ]
σ_{radial} (OD) = 0, negligible

Handle closure Loads

Handle Torque =[ ]

Handle Closure Force, Fc = T /kd =[ ]

Where : k =[ ]
d = .375, nominal diameter of stem threads

Axial bonnet stress, closure =[ ]

Panel Nut installation load

Maximum panel nut torque =[ ]

Panel nut force, Fpn = T / kd =[ ]

Where : k =[ ], thread friction factor, lubed
d = .8125, nominal diameter of bonnet threads

Panel nut axial stress (in bonnet) =[ ]
Panel nut thread shear stress = [ ]

Resolution of Stresses:

The stresses calculated above will be combined, and principal stresses will be calculated. The principal stresses are then used to determine stress intensities. Calculated stress intensities are then compared to the allowable stress intensity or allowable stress intensity multiple, per NB-3220.

The table below contains the appropriate combinations of stress intensities compared to the specific allowable stress intensities. Alpha (α) factor for NB-3221.3 is conservatively chosen as 1.0 The table contains the worst case stress intensity states: located at the bonnet ID for design loading and at the bonnet OD for level A service limits:

Temp. °F	Design Loading				Level A Service Limits	
	NB-3221.1 General Primary Stress Intensity	NB-3221.1 allowable stress intensity , Sm	NB-3221.3 Primary membrane + primary bending	NB-3221.3 allowable stress intensity, 1.0 Sm	NB-3222.2 primary + secondary stress intensity	NB-3222.2 allowable stress intensity 3 Sm
100						
200						
300						
400						
500						
600						

650						
700						
800						

4U Body & Bonnet Finite Element Analysis

Design Loads:

1. Primary:
   a. 3600 psi internal pressure

2. Secondary:
   a. Thermal Gradient:
      i. Ambient Temperature = 40°F
      ii. All internal wetted surfaces are at the design temperature
      iii. The bellows is assumed to be ruptured (bonnet contains pressure)
      iv. External heat transfer coefficient =[ ]
   b. Union Nut Assembly Preload =[ ]
   c. Handle Closure Torque =[ ]

3. Service Level A Piping Loads:
   a. [ ] in-lbs bending moment (My)
   b. in-lbs bending moment (Mz)
   c. in-lbs torsion
   d. lbs tension

The analysis is divided into 6 sections (3 bonnet, 3 body).

Reference page 4 for diagram

Results and comparison to allowable stresses are as follows:

Bonnet Section 1

Temp. °F	Design Loading			
	NB-3221.1 General Primary Stress Intensity	NB-3221.1 allowable stress intensity , Sm	NB-3221.3 Primary membrane + primary bending	NB-3221.3 allowable stress intensity, 1.0 Sm
100				
200				
300				
400				
500				
600				
650				
700				
800				

Temp. °F	Level A Service Limits				
	NB-3222.2 primary + secondary stress intensity				NB-3222.2 allowable stress intensity 3 Sm
	My	Mz	Torsion	Tension	
100					
200					
300					
400					
500					
600					
650					
700					
800					

Bonnet Section 2

Temp. °F	Design Loading			
	NB-3221.1 General Primary Stress Intensity	NB-3221.1 allowable stress intensity , Sm	NB-3221.3 Primary membrane + primary bending	NB-3221.3 allowable stress intensity, 1.0 Sm
100				
200				
300				
400				
500				
600				
650				
700				
800				

Temp. °F	Level A Service Limits				
	NB-3222.2 primary + secondary stress intensity				NB-3222.2 allowable stress intensity 3 Sm
	My	Mz	Torsion	Tension	
100					
200					
300					
400					
500					
600					
650					
700					
800					

Bonnet Section 3

Temp. °F	Design Loading			
	NB-3221.1 General Primary Stress Intensity	NB-3221.1 allowable stress intensity , Sm	NB-3221.3 Primary membrane + primary bending	NB-3221.3 allowable stress intensity, 1.0 Sm
100				
200				
300				
400				
500				
600				
650				
700				
800				

Temp. °F	Level A Service Limits				
	NB-3222.2 primary + secondary stress intensity				NB-3222.2 allowable stress intensity 3 Sm
	My	Mz	Torsion	Tension	
100					
200					
300					
400					
500					
600					
650					
700					
800					

Body Section 4

Temp. °F	Design Loading			
	NB-3221.1 General Primary Stress Intensity	NB-3221.1 allowable stress intensity , Sm	NB-3221.3 Primary membrane + primary bending	NB-3221.3 allowable stress intensity, 1.0 Sm
100				
200				
300				
400				
500				
600				
650				
700				
800				

Temp. °F	Level A Service Limits				
	NB-3222.2 primary + secondary stress intensity				NB-3222.2 allowable stress intensity 3 Sm
	My	Mz	Torsion	Tension	
100					
200					
300					
400					
500					
600					
650					
700					
800					

Allowable stress intensity values are based on annealed material, due to welding in this region.

Body Section 5

Temp. °F	Design Loading			
	NB-3221.1 General Primary Stress Intensity	NB-3221.1 allowable stress intensity , Sm	NB-3221.3 Primary membrane + primary bending	NB-3221.3 allowable stress intensity, 1.0 Sm
100				
200				
300				
400				
500				
600				
650				
700				
800				

Temp. °F	Level A Service Limits				
	NB-3222.2 primary + secondary stress intensity				NB-3222.2 allowable stress intensity 3 Sm
	My	Mz	Torsion	Tension	
100					
200					
300					
400					
500					
600					
650					
700					
800					

Body Section 6

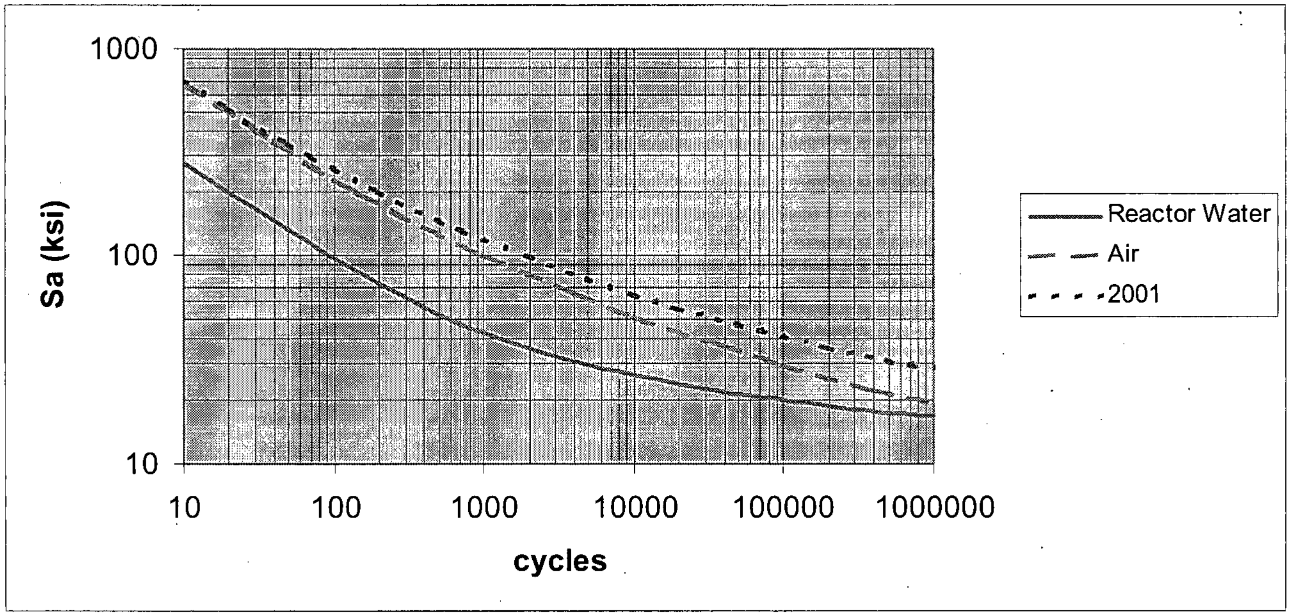
Temp. °F	Design Loading			
	NB-3221.1 General Primary Stress Intensity	NB-3221.1 allowable stress intensity , Sm	NB-3221.3 Primary membrane + primary bending	NB-3221.3 allowable stress intensity, 1.0 Sm
100				
200				
300				
400				
500				
600				
650				
700				
800				

Temp. °F	Level A Service Limits				
	NB-3222.2 primary + secondary stress intensity				NB-3222.2 allowable stress intensity 3 Sm
	My	Mz	Torsion	Tension	
100					
200					
300					
400					
500					
600					
650					
700					
800					

Fatigue (Cyclic) Analysis

A fatigue analysis has been performed on the 4UW valve. Cyclic loading resulting from valve operation and packing nut adjustment will be shown here.

Analysis Method: Alternating stress (Salt) is calculated per NB-3216. The Salt values are adjusted for the modulus of elasticity variation with temperature. The Salt value for each transient is used to obtain N, the allowable number of cycles, from the austenitic stainless steel fatigue curves in ASME Section III, appendix I, tables I-9.1 and I-9.2.2. It is important to note that Swagelok has obtained advance copies of new fatigue strength data and fatigue curves for stainless steels, working with a member of the ASME sub-group on fatigue strength. There are two new curves, one for air and the other for reactor water. They are both more restrictive than the existing 2001 curve, therefore Swagelok has taken the proactive step of including them in this analysis. The curves can be compared in the plot below. Usage factors for each condition are calculated by dividing the actual number of cycles, n, by the allowable number of cycles, N.. The cumulative usage factor is the total of all usage factors. This value must be less than or equal to 1.0.



Tabulated Sa vs. Cycle Values

	cycles	10	20	50	100	200	500	1000	2000	5000
2001	**Sa (ksi)**	708	512	345	261	201	148	119	97	76
2004	**cycles**	10000	20000	50000	100000	200000	5E+05	1.0E+06		
	Sa (ksi)	64	55.5	46.3	40.8	35.9	31	28.3		

Air	**Cycles, N**	10	40	100	400	1000	4000	10000	40000	100000	400000	1E+06
	Sa (ksi)	700	358	232	137	99	65	50.5	36.6	29.9	22.7	19.5
Reactor water	**Cycles, N**	10	40	100	400	1000	4000	10000	40000	100000	400000	1E+06
	Sa (ksi)	280	146	94.9	56.6	42.6	31.1	26.6	22.2	20	17.9	17

The fatigue calculations below include analysis of the Gland nut web undergoing packing adjustments, and the bonnet and bonnet nut experiencing the effects of on-off operation of the valve. Typically, additional features will be analyzed, based on the design specification requirements.

Packing adjustment – Gland Nut web

Calculate Salt

Stress concentration factor Kt from Peterson tables: Kt = 2.2

Si max = [ ]
Si min = 0, conservatively assume that packing load totally relaxes between adjustments

Salt = [ ]

Calculate Sa & N

Packing nut should not normally be exposed to any system media, therefore it is appropriate to use the Sa "air" values.
[ ]

Calculate U, usage factor

[ ]

On / Off Operation – Lower Bonnet (FEA bonnet section 2)

Calculate Salt

Stress concentration factor Kf, from Lipson,Noll, & Clock: Kf = 2.8

Si max = [ ]
Si min = 0 conservatively assume that valve is fully open with no pressure load

Salt = [ ]

Calculate Sa & N

Assuming that the bellows is intact during normal operation the bonnet will not be exposed to system media, however, in the event of a bellows rupture the bonnet will see system media. Therefore Sa values for air and reactor water exposure will be used to calculate N.

Reactor water
[ ]

Air
[ ]

Calculate U, usage factor

[ ]
- and -
[ ]

Therefore U = U1 + U2 =[ ]

On / Off Operation – Bonnet Nut Web

Calculate Salt

Stress concentration factor Kt from Peterson tables: Kt = 2.9

Si max =[ ]
Si min =[ ]
No closure force and thermal stress, conservatively assume that thermal and mechanical cycles coincide

Salt =[ ]

Calculate Sa & N

Bonnet nut should not normally be exposed to any system media, therefore it is appropriate to use the Sa "air" values.
[ ]

Calculate U, usage factor

[ ]

Correlation of Strain Gage Experimental Stress Analysis to FEA

Background: The UW Series bellows valves are presently being analyzed for conformance to requirements for safety related applications in the nuclear power industry. Per the ASME Boiler and Pressure Vessel Code, Section III, Division I – NB-3512.2d3), adequacy of stress analysis of the valve body and bonnet shall be verified by experimental stress analysis conducted in accordance with the requirements of II-1100 through II-1400.

Objective:

1) Conduct experimental stress analysis on a 4UW bellows valve per requirements above.
2) Conduct finite element stress analysis with ANSYS Mechanical using boundary conditions similar to those from experimental stress analysis.
3) Compare results of stress analysis from both methods.

Results: The correlation between the two methods is detailed in the correlation of experimental results to FEA spreadsheet, shown on the next page. It should be noted that all strain measurements less than 50 micro strain (corresponding to stress of 1500 psi) are not considered for correlation with FEA results. This was done because the accuracy of the strain gages is uncertain below this level.

Procedure: Principal strain results (epsilon1 and epsilon3) were taken from the FEA. These results were then corrected for load difference from the experimental analysis and multiplied by the appropriate Young's modulus to get principal stresses in the correlation spreadsheet. For the equations used in the correlation spreadsheet for calculation of principal strains from strain gauge data refer to Mechanics of Materials, 4th Ed., Higdon, Ohlsen, Stiles, Weese and Riley, 1985, pp. 73-84.

Discussion: In general, the correlation of stress intensities from the FEA with those derived from the experimental stress analysis was very good. The differences are consistent with the expected accuracies of strain gage experimental stress analysis and finite element analysis.

NOTE: All principal strains less than 50micro strain (principal stresses less than 1500 psi) are removed. The accuracy of the strain gauges at this low range is questionable.

E (psi) 2.83E+07

Max. principal stress **Min. principal stress**

Load Case	Location	FEA Results S1	Experimental Results avg. S1	FEA Results S3	Experimental Results avg. S3	FEA Results Sint	Experimental Results avg. Sint	Percent Difference (Experimental vs. FEA)
axial								
	Tube Diameter from Weld							
bending x-plane								
	Tube Diameter from Weld							
bending z-plane								
	Tube Diameter from Weld							
torque								
	Bottom Right							
	Front Right							
	Bottom Left							
	Front Left							
	Tube Diameter from Weld							
internal pressure								
	Bottom Bonnet							
	Middle Bonnet							
	Top Bonnet							
	Tube Diameter from Weld							

ATTACHMENT 2

APPLICATION FOR WITHHOLDING FROM PUBLIC DISCLOSURE AND AFFIDAVIT OF DAVID A. PEACE

DOMINION ENERGY KEWAUNEE, INC. (DEK)
DOMINION NUCLEAR CONNECTICUT, INC. (DNC)
VIRGINIA ELECTRIC AND POWER COMPANY (DOMINION)

KEWAUNEE POWER STATION UNIT 1
MILLSTONE POWER STATION UNITS 2 AND 3
NORTH ANNA POWER STATION UNITS 1 AND 2
SURRY POWER STATION UNITS 1 AND 2

10 CFR § 2.390

<u>APPLICATION FOR WITHHOLDING</u>
<u>AND</u>
<u>AFFIDAVIT OF *DAVID H. PEACE*</u>

I, David H. Peace, Vice President, Engineering, state that:

1. I am authorized to execute this affidavit on behalf of Swagelok Company

2. Swagelok is submitting two documents for NRC review and approval: 4UW Design Report Template and SS-45S8-18622-NSR / SS-45XS8-18623-NSR Design Report. These two documents contain proprietary commercial information that should be held in confidence by the NRC pursuant to the policy reflected in 10 CFR §§ 2.390(a)(4) because:

a. This information is being held in confidence by Swagelok.

b. This information is of a type that is held in confidence by Swagelok, and there is a rational basis for doing so because the information contains sensitive commercial information concerning manufacturing of Swagelok valve components.

c. This information is being transmitted to the NRC in confidence.

d. This information is not available in public sources and could not be gathered readily from other publicly available information.

e. Public disclosure of this information would create substantial harm to the competitive position of Swagelok by disclosing confidential Swagelok manufacturing information to other parties whose commercial interests may be adverse to those of Swagelok. Furthermore, Swagelok has expended significant engineering resources in the development of the information. Therefore, the use of this confidential information by competitors would permit them to use the information developed by Swagelok without the expenditure of similar resources, thus giving them a competitive advantage.

3. Accordingly, Swagelok requests that the designated document be withheld from public disclosure pursuant to the policy reflected in 10 CFR §§ 2.390(a)(4).

Swagelok Company

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David H. Peace
Vice President, Engineering

STATE OF Ohio
COUNTY OF Cuyahoga

Subscribed and sworn to me, a Notary Public, in and for the County and State above named, this 2 day of October, 2007.

____

My Commission Expires: April 12, 2010

LORI A. SARVER - NOTARY PUBLIC
STATE OF OHIO - SUMMIT COUNTY
MY COMMISSION EXPIRES APRIL 12, 2010