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Loads on Flanges - The ASME Way

ASME VIII-1 Appendix 2 provides a method of sizing flanges. The calculations use three loads - HT, HG & HD and two operating conditions - seating and operating. What are these loads, how are they calculated, and where are they applied to the flange?

A sample flange shown below will be calculated using ASME Appendix 2 methods and by finite element analysis (FEA) to illustrate the application of the loads and show the resulting stresses.

Sample flange (App 2 Fig 2-4(5)

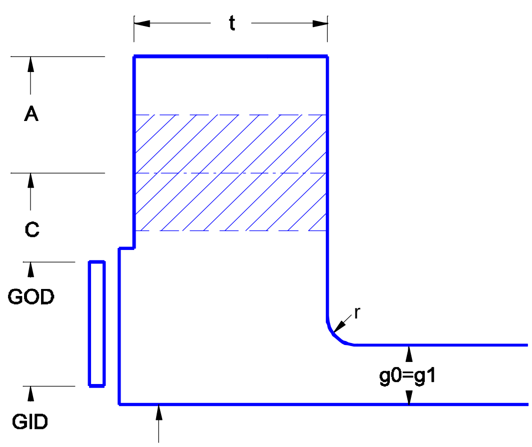


Fig 2-4 (5)

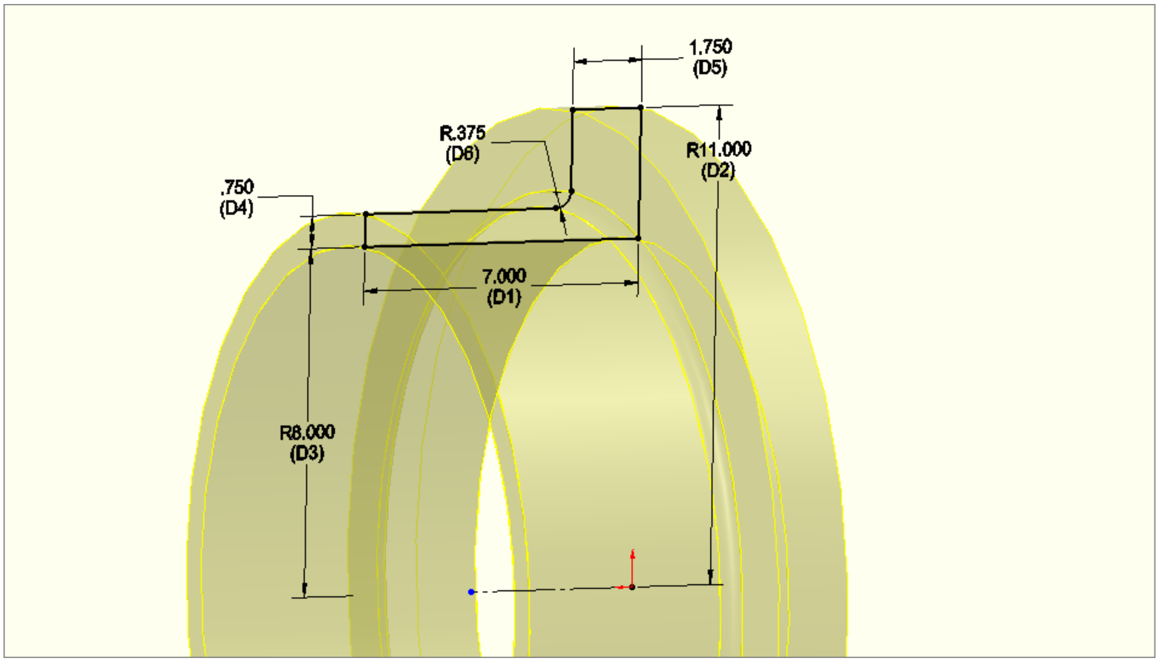
Sample Flange Dimensions

Inside Diameter = B = 16.00”
Outside Diameter = A = 22”
Thickness = t = 1.75
Hub radius = r = 0.375
Pipe thickness = g0 = 0.75

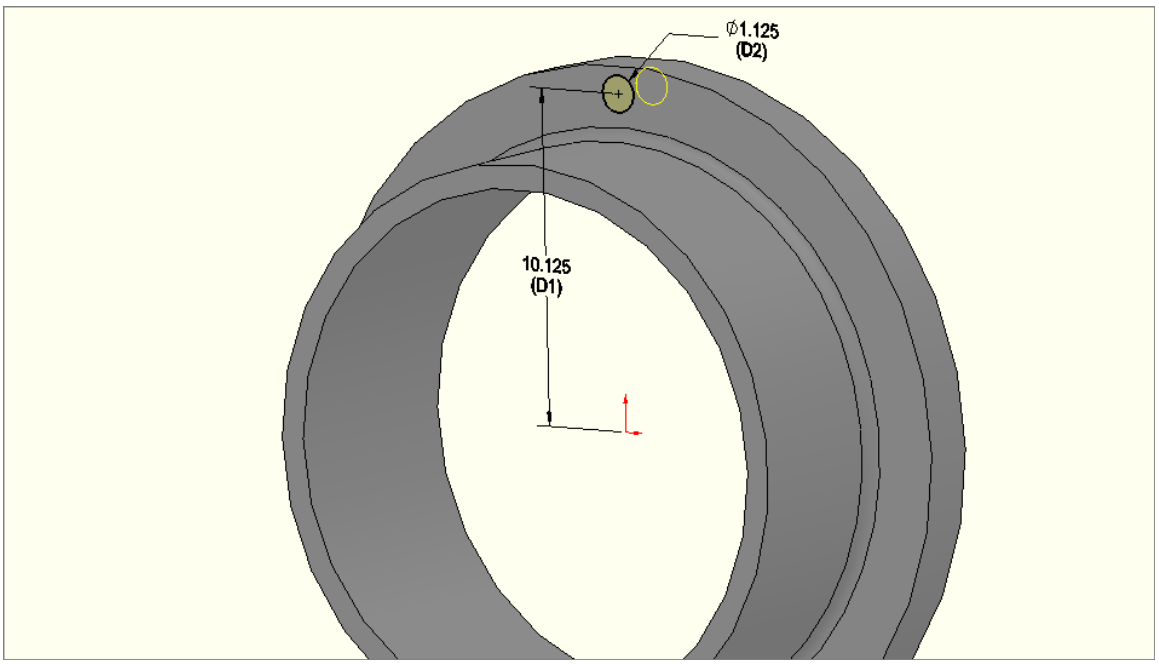
Gasket OD = 17.75”
Gasket ID = 16.25”
Gasket m = 3
Gasket y = 10,000

16 Bolts x 1” dia on a 20.25” BCD (C)

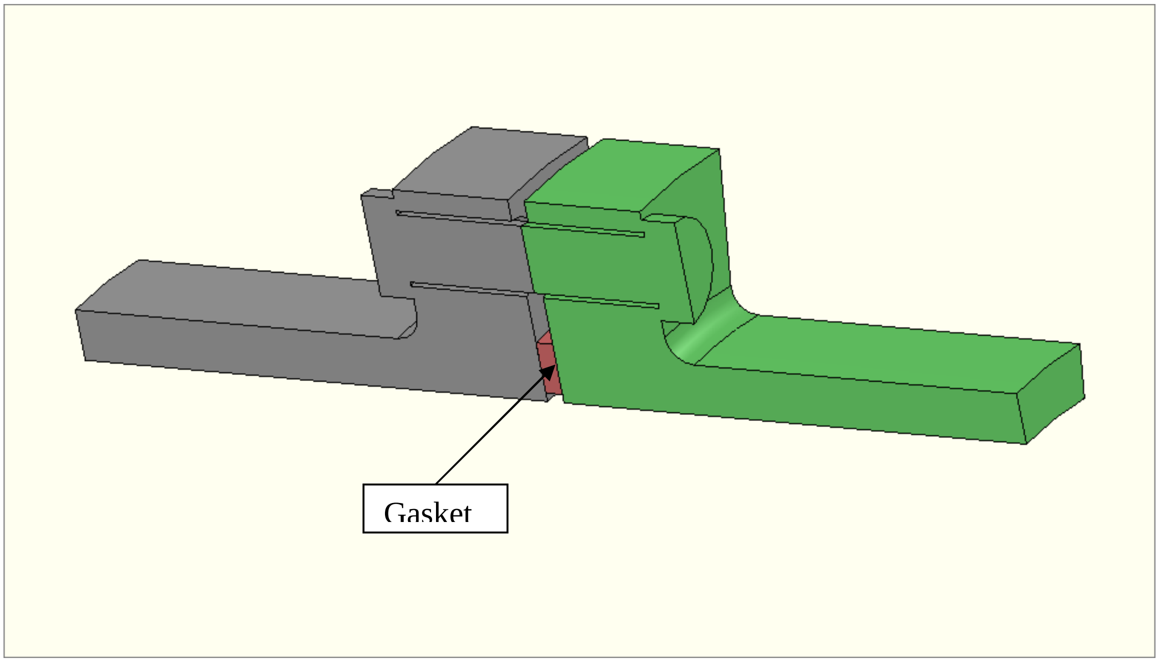
Sample flange - the FEA model



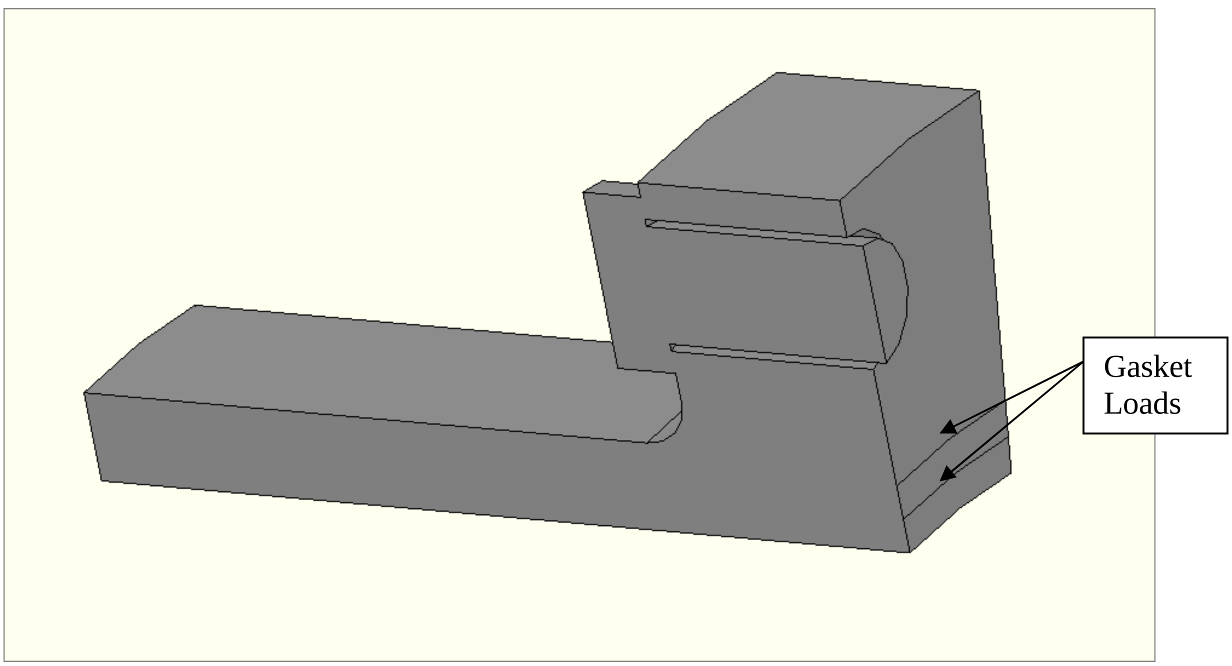
Flange Model



16 holes on 20.25” BCD, 1” bolt size - only 1/2 of one bolt will be used for the FEA model due to symmetry.



Half of the 1” bolt is added. A mirrored body creates a flanged pair.

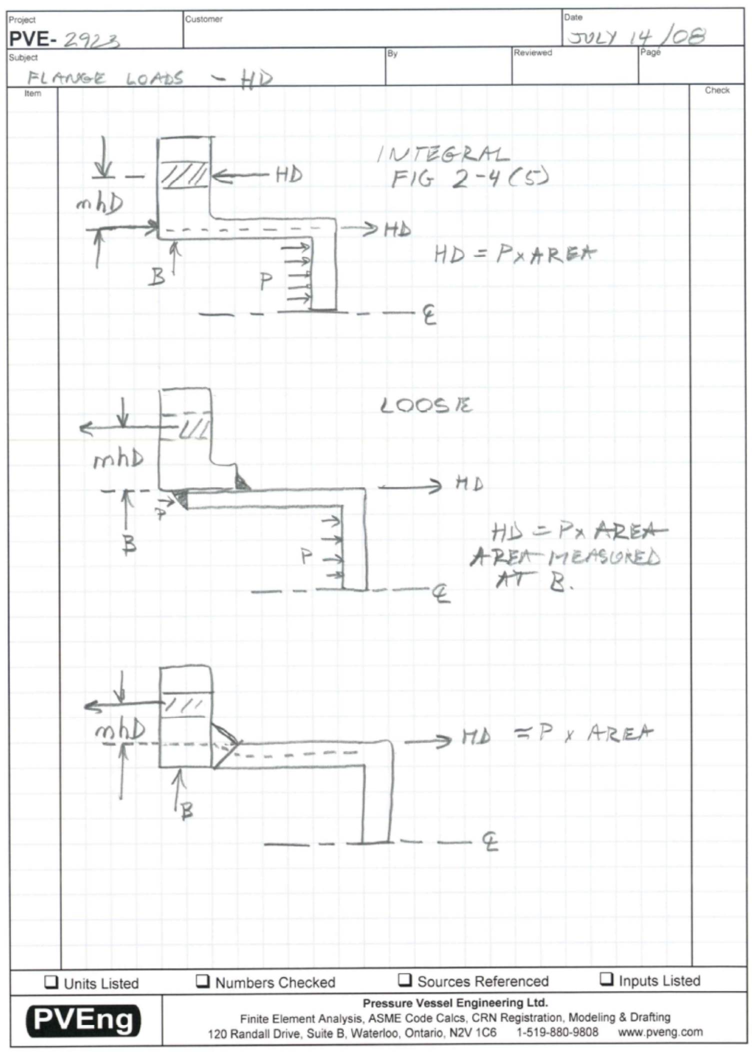


For FEA the bolt length is cut on the center of the gasket. The gasket is removed and is replaced by the loads it generates. Split lines can be seen where the gasket loads HT and HG are applied.

Load HD - Operating

HD is created by the pressure on the pipe attached to the flange. Force = Pressure x Area.
HD = P * B^2 / 4

The load is generated on center line of the pipe, but the ASME rules change the moment arm depending on the attachment method. When FEA is performed, the load should be applied to the attached pipe - the FEA program will determine how the load is distributed.



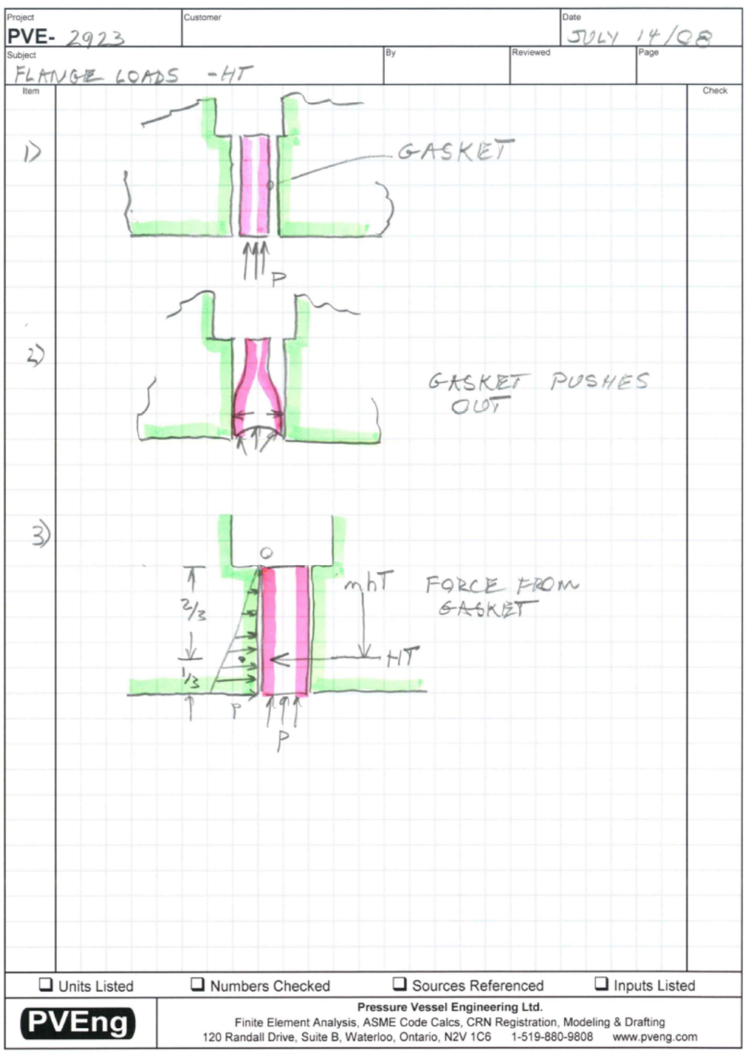
Load HT - Operating

HT is created by the internal pressure acting on the gasket:

1) Pressure is applied to the exposed edge of the gasket
2) The gasket tries to expand but is held in place by the flange faces
3) The flange faces push back

The force between the gasket and the flange is shown as a triangle. The force is zero at the OD of the gasket (there is no pressure at the gasket OD and thus no leakage). At the inside edge, the pressure is the pressure in the pipe. HT is the average pressure along the length. mhT is measured at the point 1/3 up the triangle, the centroid of the force.

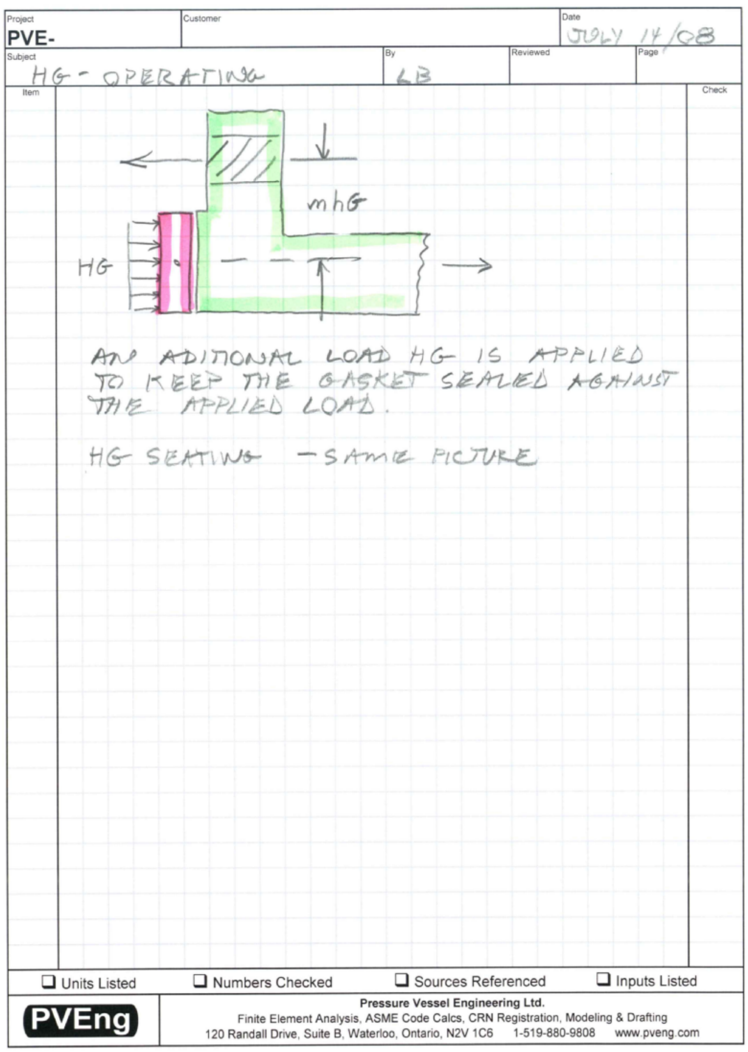
The ASME rules reduce the width of the gasket. This load is a design rule, not a predictor of actual flange stresses. For FEA analysis, the load HT is applied at the moment arm mhT away from the bolt centerline.



Load HG - Operating

HG operating is the force required to keep the flange sealed against the operating pressure. It is generated by tightening the bolts. Load = effective area x gasket factor m x Pressure. If the flange is self energizing (does not need additional force to seal such as an o-ring) then HG operating = 0

Load HG operates through the center of the gasket, but the gasket size is reduced by the ASME rules to create an effective area. Correlation to real gasket properties is difficult - this load and its moment arm is a design rule, not a predictor of actual flange stresses.



Load HG - Seating

HG seating is the force required to seat the gasket into the flange gasket face and be leak tight against a pressure of 0 psi. (HG operating provides the load required to keep the seal as the operating pressure is increased).

The force HG is loosely based on gasket physical properties, but the gasket area used is modified (reduce) from the actual gasket width because the code y factors are too high. Correlation to real gasket properties is impossible - this load and its moment arm are a design rule, not a predictor of actual flange stresses.

Force HG has an additional load added to it - the “gasket destroying” or “gasket crushing” force. The computed seating load on the gasket is increased to the average of the required bolt strength and the available bolt strength. This code disaster greatly increases the required thickness of flanges far beyond the loads that the gasket can handle.

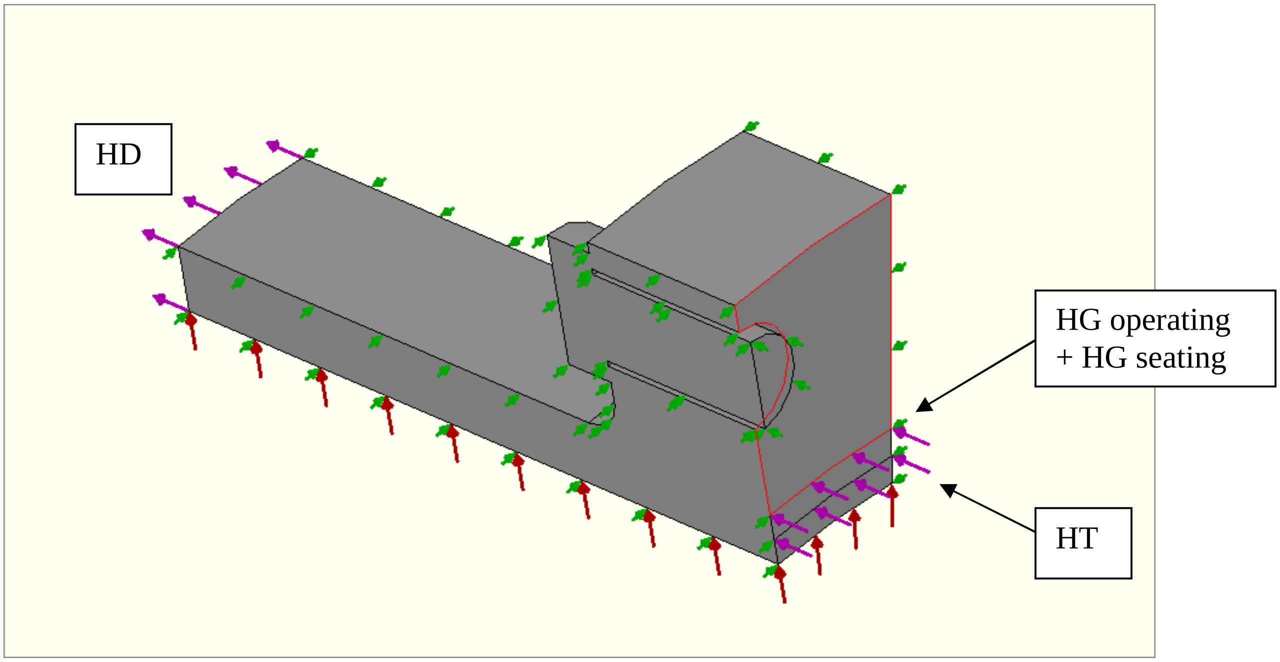
As a designer, when the seating loads are too large and are caused by extra bolt area, several options are available:

1) make the bolts smaller in diameter or fewer in number. Reducing the effective area of the bolts reduces this theoretical gasket crushing force.

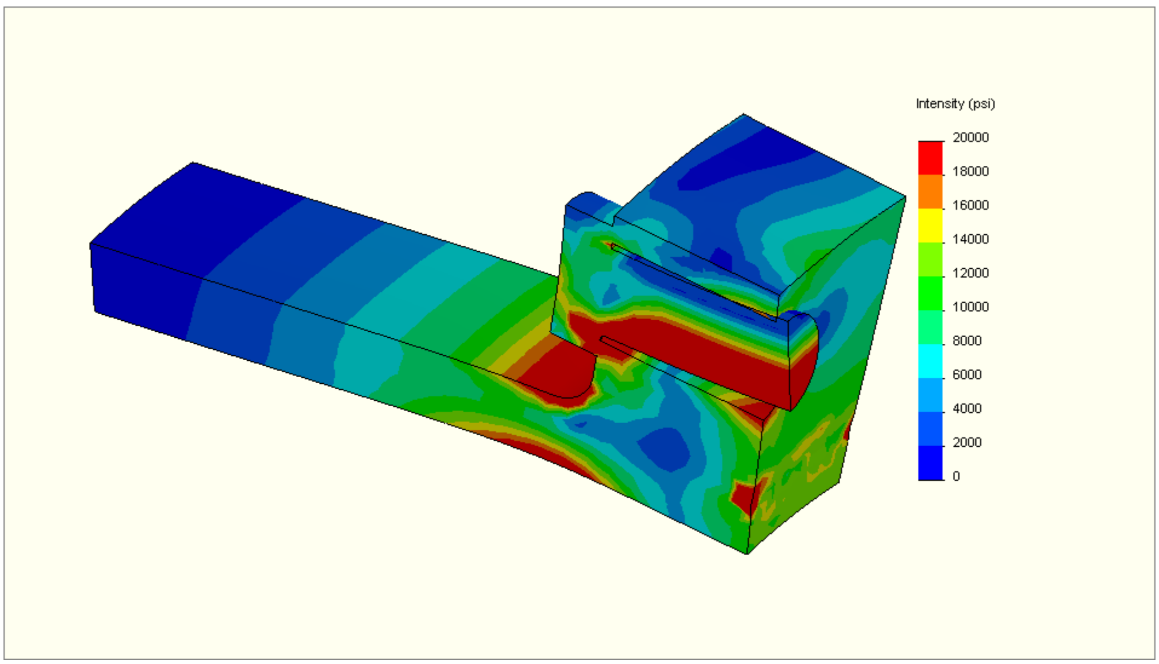
2) use weaker bolts - same idea as above.

3) if material waste and cost are no object, make the flange thicker. This route often is used when a custom appendix 2 flange must mate up to standard flanges such as B16.5 series which seldom calculate to appendix 2 rules.

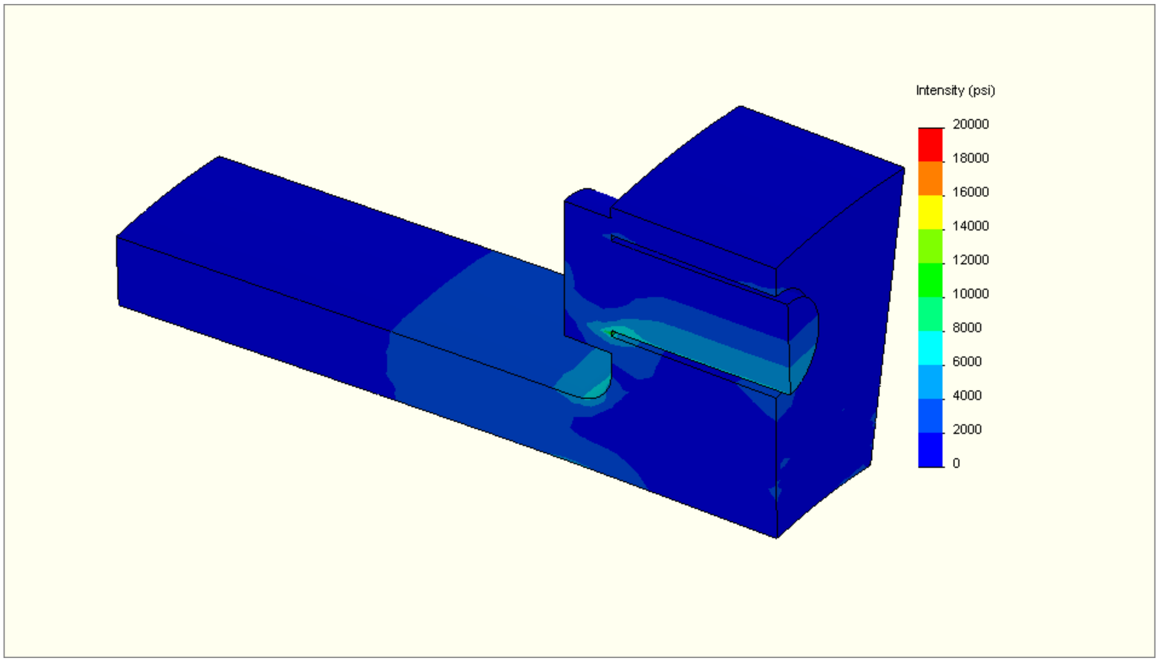
ASME Loads Applied to the FEA Model



The flange model with the HD, HG and HT loads applied.



Combined operating and seating stresses case stresses. Higher stresses can be seen at the pipe to flange discontinuity. Bending stresses can also be seen in the bolt. Although the stresses look high compared with the 20,000 psi membrane allowable stress for the flange and pipe, the stresses are minor if compared with a local discontinuity limit of 3x20,000 psi. This flange design although loaded to the maximum ASME allows can be considered to be lightly loaded and wasteful of materials.



Operating loads only - used for cycle life calculations (seating HG is removed). The gasket gets seated once, this is the load that the flange sees with each application and removal of pressure. The flange loads are extremely light for this flange that was designed around the gasket seating case.

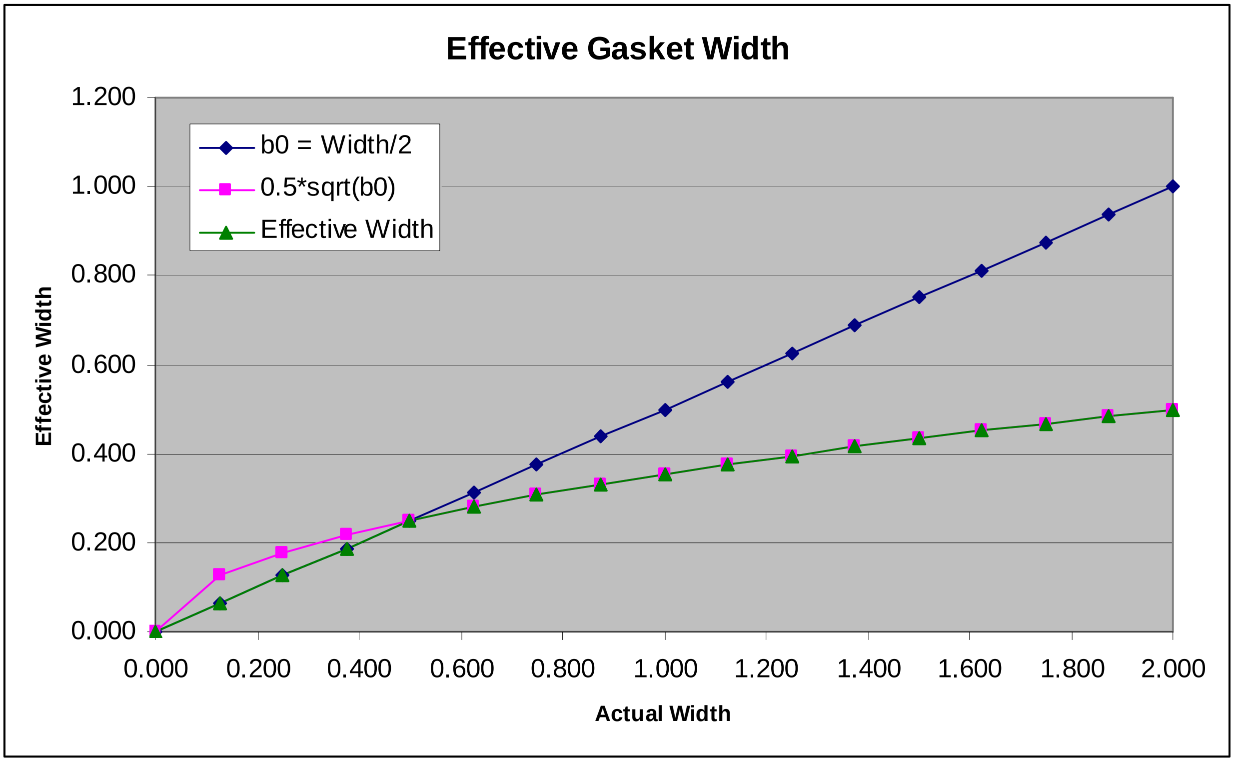
The Effective Seating Width of the Gasket

The effective seating width of the gasket removes the correlation between the physical properties of the gasket material, and the calculated gasket loads. The seating width is typically 1/2 * the square root of the actual gasket width (see table 2-5.2 for actual formulas which vary depending on the gasket seating arrangement and the gasket width). Traditionally, this was done to allow for rotation of the flanges under load which reduced the actual width of the gasket in contact with the flange faces (it was presumed that the inside edge of the gasket was not in contact). In reality, the ASME rules, including the flange rotation limits in 2-14, do not allow enough flange rotation for the gasket to be partially in contact. This effective width calculation removes any possible correlation between ASME flange calculation methods and flange manufacturers provided m and y values. It was probably introduced because the table 2-5.1 gasket factors are too high.

The seating and operating loads are design rules and should not be expected to predict actual flange stresses. They can be used in FEA analysis to simulate loads in a manner similar to App 2 methods as required by U-2(g).

Width	b0 = Width/2	0.5*sqrt(b0)	Effective Width
0.000	**0.000**	0.000	***0.000***
0.125	**0.063**	0.125	***0.063***
0.250	**0.125**	0.177	***0.125***
0.375	**0.188**	0.217	***0.188***
0.500	**0.250**	0.250	***0.250***
0.625	0.313	**0.280**	***0.280***
0.750	0.375	**0.306**	***0.306***
0.875	0.438	**0.331**	***0.331***
1.000	0.500	**0.354**	***0.354***
1.127	0.563	**0.375**	***0.375***
1.250	0.625	**0.395**	***0.395***
1.375	0.688	**0.415**	***0.415***
1.500	0.750	**0.433**	***0.433***
1.625	0.813	**0.451**	***0.451***
1.750	0.875	**0.468**	***0.468***
1.875	0.938	**0.484**	***0.484***
2.000	1.000	**0.500**	***0.500***

Effective width for a common gasket arrangement - Table 2-5.2 sketches (1a) and (1b)



Attachments

Attached are calculation sheets for:

- ASME code calculation for this flange. This flange is limited by the seating case - in this case seating of a high strength spiral wound gasket - m=3, y = 10,000.
- FEA loads for the operating and seating case
- FEA loads for the operating only case

Flanges ver 4.26

ASME VIII Div I Appendix 2

Code Flange Calculations Description

Dimensions:

Fig2-4(5)	**fd?** - Select a flange design
22.000	**A** [in] - flange OD
16.000	**Bn** [in] - ID, uncorroded
1.750	**t** [in] - flange thickness
0.375	**rf** [in] - hub corner radius
0.750	**g0f** [in] - hub thickness
0.750	**g1** [in] - hub base thickness

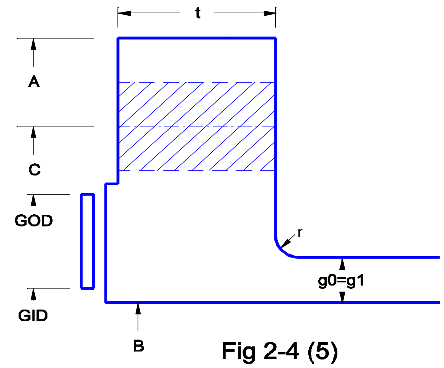


Fig 2-4 (5)

Gasket:

17.750	**GOD** [in] - gasket OD
16.250	**GID** [in] - gasket ID
3.00	**m** - gasket factor
10,000	**gy** - gasket factor y

Bolting:

20.250	**varC** [in] - bolt circle dia
1.000	**BoltOD** [in] - bolt size
16.0	**Nbolt** - number of bolts

Operating Conditions:

0.000	**Corr** [in] - corrosion allowance
100.0	**P** [psi] - internal operating pressure
0.0	**Pe** [psi] - external operating pressure

Material Properties:

NonCast	**CastMaterial?** - Cast Or NonCast
20,000	**Sf** [psi] - allowable flange stress at DESIGN temp.
20,000	**Sfa** [psi] - Allowable Flange Stress at ASSEMBLY temp.
27,900,000	**Efo** [psi] -Operating Flange Modulus
27,900,000	**Efs** [psi] - Seating Flange Modulus
20,000	**Sb** [psi] - allowable bolt stress at DESIGN temp
20,000	**Sba** [psi] - allowable bolt stress at ASSEMBLY temp

Geometry Constraints:

rMin =	max(1/4*g1,0.188)	MAX(1/4*0.75,0.188) =	**0.188**
NutG [in] =	PVELookup("TEMATableD5","Lookup","NutWidth",BoltOD)		1.796
Rh [in] =	PVELookup("TEMATableD5","Lookup","Rh",BoltOD)		1.375
E [in] =	PVELookup("TEMATableD5","Lookup","E",BoltOD)		1.063
WrenchClearance =	varC/2-B/2-g0-Rh TEMA Table D-5	20.25/2-16/2-0.75-1.375 =	0.000
CkWrenchClr =	WrenchClearance > 0	0 > 0 =	Acceptable
NutClearance =	varC/2-B/2-g0-rf-NutG/2 TEMA Table D-5	20.25/2-16/2-0.75-0.375-1.796/2 =	0.102
CkNutClr =	NutClearance > 0	0.102 > 0 =	Acceptable
EdgeClearance =	(A-E)-varC TEMA Table D-5	(22-1.063)-20.25 =	0.687
ckEdge =	EdgeClearance > 0	0.687 > 0 =	Acceptable

Calculated Dimensions:

g0 =	g0f-Corr	0.75-0 =	**0.750**
gOne =	g1-Corr	0.75-0 =	**0.750**
B =	Bn+2*Corr	16+2*0 =	**16.000**
varR =	(varC-B)/2 - gOne Gasket width in contact	(20.25-16)/2 - 0.75 =	**1.375**
varN =	(GOD-GID)/2 Gasket width in contact	(17.75-16.25)/2 =	**0.750**
b0 =	varN / 2 Gasket seating width	0.75 / 2 =	**0.375**

varb = IF(b0>0.25,Sqrt(b0)/2,b0) Effective seating width
IF(0.375>0.25,SQRT(0.375)/2,0.375) = **0.306**
varG = IF(b0>0.25,GOD-2*varb,(GOD-GID)/2 + GID)
IF(0.375>0.25,17.75-2*0.306,(17.75-16.25)/2 + 16.25) = **17.138**
hub = rf Length of Hub 0.375 = **0.375**

Bolt Loads: (VIII App 2-5)

Bolt size and class: 1-8 UNC 2A
H = 0.785*varG^2*P end load 0.785*17.138^2*100 = **23,055**
He = 0.785*varG^2*Pe end load external pressure 0.785*17.138^2*0 = **0**
HP = 2*varb*3.14*varG*m*P contact load 2*0.306*3.14*17.138*3*100 = **9,886**
HD = pi()/4 * B^2 * P end load PI()/4 * 16^2 * 100 = **20,106**
HDe = pi()/4 * B^2 * Pe end load external pressure PI()/4 * 16^2 * 0 = **0**
HT = H - HD face load 23055 - 20106 = **2,949**
HTe = He - HDe face load external 0 - 0 = **0**
Wm1 = H + HP bolt load 23055 + 9886 = **32,941**
Wm2 = pi()*varb*varG*gy seating load PI()*0.306*17.138*10000 = **164,849**
Am = Max(Wm1/Sb, Wm2/Sba) Bolt area required
MAX(32941/20000, 164849/20000) = **8.242**
RootArea [sq. in] = PVELookup("BoltSizing","Lookup","Root Area",BoltOD) **0.566**
Ab = RootArea*Nbolt 0.566*16 = **9.056**
CheckExcess = Ab>=Am 9.056>=8.242 = **Acceptable**

Flange Loads: (App 2-5)

W [lb] = (Am + Ab)*Sba/2 seating conditions (8.242 + 9.056)*20000/2 = **172,984**
HG [lb] = Wm1 - H operating conditions 32941 - 23055 = **9,886**
TBoltLoad [lb] = (W+Wm1)/Nbolt (172984+32941)/16 = **12,870**

Flange Moment Arms: (Table App 2-6 - Integral flanges)

mhD [in] = varR+0.5*gOne 1.375+0.5*0.75 = **1.750**
mhT [in] = (varR+gOne+mhG)/2 (1.375+0.75+1.556)/2 = **1.841**
mhG [in] = (varC-varG)/2 (20.25-17.138)/2 = **1.556**

Flange Moments: (App 2-6)

MD [in-lb] = HD * mhD end pressure 20106 * 1.75 = **35,186**
MT [in-lb] = HT * mhT face pressure 2949 * 1.841 = **5,428**
MG [in-lb] = HG * mhG gasket load 9886 * 1.556 = **15,384**
Mo1e [in-lb] = HDe*(mhD-mhG)+HTe*(mhT-mhG) total operating external
0*(1.75-1.556)+0*(1.841-1.556) = **0**
Mo1 [in-lb] = Max(MD+MT+MG,Mo1e) total operating
MAX(35186+5428+15384,0) = **55,998**
Mo2 [in-lb] = W*(varC-varG)/2 total seating 172984*(20.25-17.138)/2 = **269,196**

Graphs: App 2-7.1-6 Values of F, f, T, U, V, Y and Z

h0 = sqrt(B*g0) SQRT(16*0.75) = **3.464**
hh0 = hub/h0 0.375/3.464 = **0.108**
g1g0 = gOne/g0 0.75/0.75 = **1.000**
F = PVELookup("F","FlangeFactor",hh0,g1g0) **0.909**
V = PVELookup("V","FlangeFactor",hh0,g1g0) **0.550**
smallF = 1 1 = **1.000**
K = A/B 22/16 = **1.375**
T = PVELookup("T","FlangeFactorK",K) **1.765**
U = PVELookup("U","FlangeFactorK",K) **6.877**
Y = PVELookup("Y","FlangeFactorK",K) **6.258**
Z = PVELookup("Z","FlangeFactorK",K) **3.246**
d = (U/V)*h0*g0^2 (6.877/0.55)*3.464*0.75^2 = **24.360**
e = F / h0 0.909 / 3.464 = **0.262**
L = (t*e + 1)/T + t^3/d (1.75*0.262 + 1)/1.765 + 1.75^3/24.36 = **1.047**

Flange Seating Stress: (App 2-7,8)

SHs =	smallF*ABS(Mo2) / (L*gOne^2 * B)		
		1*ABS(269196) / (1.047*0.75^2 * 16) =	**28,577**
CheckSHs =	SHs <= 1.5*(Sfa)	28577 <= 1.5*(20000) =	**Acceptable**
SRs =	(1.33*t*e+1)*ABS(Mo2) / (L*t^2*B)		
		(1.33*1.75*0.262+1)*ABS(269196) / (1.047*1.75^2*16) =	**8,454**
CheckSRs =	SRs <= Sfa	8454 <= 20000 =	**Acceptable**
STs =	(Y*ABS(Mo2) / (t^2*B)) - Z*SRs		
		(6.258*ABS(269196) / (1.75^2*16)) - 3.246*8454 =	**6,943**
SAs =	(SHs + Max(SRs, STs))/2	(28577 + MAX(8454, 6943))/2 =	**18,515**
CheckSTs =	ABS(STs) <= Sfa	ABS(6943) <= 20000 =	**Acceptable**
CheckSAs =	SAs <= Sfa	18515 <= 20000 =	**Acceptable**

Flange Operating Stress: (App 2-7,8)

SHo =	smallF*Mo1/(L*gOne^2*B)	1*55998/(1.047*0.75^2*16) =	**5,945**
CheckSHo =	SHo <= 1.5*(Sf)	5945 <= 1.5*(20000) =	**Acceptable**
SRo =	(1.33*t*e+1)*Mo1/(L*t^2*B)		
		(1.33*1.75*0.262+1)*55998/(1.047*1.75^2*16) =	**1,759**
CheckSRo =	SRo <= Sf	1759 <= 20000 =	**Acceptable**
STo =	Y*Mo1/(t^2*B)-Z*SRo	6.258*55998/(1.75^2*16)-3.246*1759 =	**1,444**
CheckSTo =	STo <= Sf	1444 <= 20000 =	**Acceptable**
SAo =	(SHo+Max(SRo,STo))/2	(5945+MAX(1759,1444))/2 =	**3,852**
CheckSAo =	SAo <= Sf	3852 <= 20000 =	**Acceptable**

Flange Flexibility: (App 2-14)

Jseating =	(52.14*Mo2*V) / (L*Efs*g0^2*h0*0.3)		
		(52.14*269196*0.55) / (1.047*27900000*0.75^2*3.464*0.3) =	**0.452**
CheckJSt =	ABS(Jseating) <= 1	ABS(0.452) <= 1 =	**Acceptable**
Joperating =	(52.14*Mo1*V) / (L*Efo*g0^2*h0*0.3)		
		(52.14*55998*0.55) / (1.047*27900000*0.75^2*3.464*0.3) =	**0.094**
CheckJOp =	ABS(Joperating) <= 1	ABS(0.094) <= 1 =	**Acceptable**

Flange Loads for FEA ver 1.12

ASME VIII div 1 App 2

Combined Loads (Operating + Seating Conditions) <- Description

Dimensions and Conditions:

10.020 <- B - ID, uncorroded
0.990 <- g1 - hub thickness
0.000 <- Corr - corrosion allowance
13.0 <- P, internal operating pressure

13.375 <- GOD - gasket OD
10.750 <- GID - gasket ID
0.50 <- m - gasket factor
0 <- gy - gasket factor y

14.250 <- varC - bolt circle dia
0.875 <- BoltOD, bolt size
12.0 <- Nbolt, number of bolts

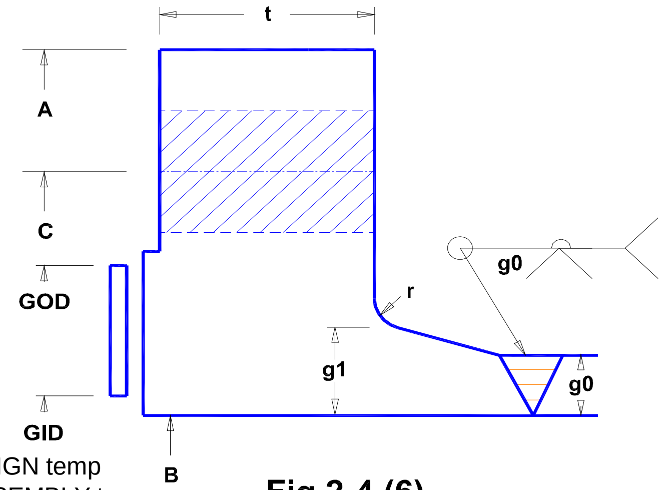


Fig 2-4 (6)

Material Properties:

<- Bolting Material
25,000 <- Sb - allowable bolt stress at DESIGN temp
25,000 <- Sba - allowable bolt stress at ASSEMBLY temp

Calculated Dimensions:

g0 =	g0-corr	= 0-0		g0 =	***0.000***
gOne =	g1 - corr	= 0.99-0		gOne =	***0.990***
B =	B+2*corr	= 10.02+2*0	*Corroded ID*	B =	***10.020***
varR =	(varC-B)/2 - gOne	= (14.25-10.02)/2 - 0.99		varR =	***1.125***
varN =	(GOD-GID)/2	= (13.375-10.75)/2	*Gasket Width in Contact*	varN =	***1.313***
b0 =	varN / 2	= 1.313 / 2	*gasket seating width*	b0 =	***0.656***
varb =	min(Sqrt(b0)/2,b0)	= min(Sqrt(0.656)/2,0.656)	*eff seating width*	varb =	***0.405***
varG =	max(GOD-2*varb,(GOD-GID)/2 + GID)		*gasket load reaction diameter*	varG =	***12.565***
=	max(13.375-2*0.405,(13.375-10.75)/2 + 10.75)				

Flange Loads (VIII App 2-5):

H =	0.785*varG^2*P	= 0.785*12.565^2*13	*end load*	H =	***1,611***
HP =	2*varb*3.14*varG*m*P	= 2*0.405*3.14*12.565*0.5*13	*contact load*	HP =	***208***
HD =	pi/4 * B^2 * P	= pi/4 * 10.02^2 * 13	*end load*	HD =	***1,025***
HT =	H - HD	= 1611 - 1025	*face load*	HT =	***586***
Wm1 =	H + HP	= 1611 + 208	*bolt load*	Wm1 =	***1,819***
Wm2 =	pi*varb*varG*gy	= pi*0.405*12.565*0	*seating load*	Wm2 =	***0***
Am =	max(Wm1/Sb, Wm2/Sba)	= max(1819/25000, 0/25000)	*req bolt area*	Am =	***0.073***
Ab =	Root*Nbolt	= 0.431*12 ***7/8-9 UNC 2A***		Ab =	***5.172***

Total Bolt Loads - lbs - (app 2-5):

W =	(Am + Ab)*Sba/2	= (0.073 + 5.172)*25000/2	*seating conditions*	W =	***65,559***
HG =	Wm1 - H	= 1819 - 1611	*operating conditions*	HG =	***208***

Flange Moment Arms - inch - (Table App 2-6 - Integral flanges):

mhD =	varR+0.5*gOne	= 1.125+0.5*0.99	*end pressure*	mhD =	***1.620***
mhT =	(varR+gOne+mhG)/2	= (1.125+0.99+0.843)/2	*face pressure*	mhT =	***1.479***
mhG =	(varC-varG)/2	= (14.25-12.565)/2	*gasket load*	mhG =	***0.843***

Summary of Loads and Locations - Combined Operating and Seating - (lbs, inch) for *ONE HALF* bolt

Operating + Seating Conditions			
Load (lbs)		Acting Rad	**Loads for a model using ONE HALF bolt only**
HT/(Nbolt*2)	*24*	*5.646*	Gasket face pressure (Operating)
(HG+W)/(Nbolt*2)	*2,740*	*6.282*	Gasket load (Seating + Operating)
HD/(Nbolt*2)	*43*	*5.505*	End pressure (Operating)
(Wm1+W)/(Nbolt*2)	*-2,807*	*7.125*	Bolt reaction balancing load (Seating +Operating)

Do not apply Wm1 + W - use boundary conditions on the bolt to apply this balancing load
This model has a sweep of 360º/(Nbolt*2) = 15º for one half bolt
Omit end pressure HD if a closed pipe end is modelled and pressurized

Flange Loads for FEA ver 1.12

ASME VIII div 1 App 2

Operating Load Only <- Description

Dimensions and Conditions:

16.000 <- B - ID, uncorroded
0.750 <- g1 - hub thickness
0.000 <- Corr - corrosion allowance
100.0 <- P, internal operating pressure

17.750 <- GOD - gasket OD
16.250 <- GID - gasket ID
3.00 <- m - gasket factor
10,000 <- gy - gasket factor y

20.250 <- varC - bolt circle dia
1.000 <- BoltOD, bolt size
16.0 <- Nbolt, number of bolts

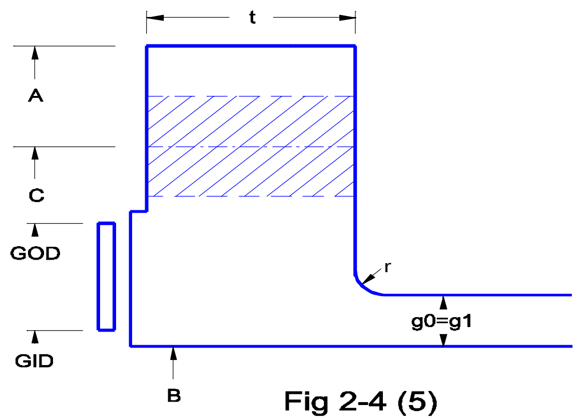


Fig 2-4 (5)

Material Properties:

<- Bolting Material
20,000 <- Sb - allowable bolt stress at DESIGN temp
20,000 <- Sba - allowable bolt stress at ASSEMBLY temp

Calculated Dimensions:

g0 =	g0-corr	= 0-0		g0 = ***0.000***
gOne =	g1 - corr	= 0.75-0		gOne = ***0.750***
B =	B+2*corr	= 16+2*0	*Corroded ID*	B = ***16.000***
varR =	(varC-B)/2 - gOne	= (20.25-16)/2 - 0.75		varR = ***1.375***
varN =	(GOD-GID)/2	= (17.75-16.25)/2	*Gasket Width in Contact*	varN = ***0.750***
b0 =	varN / 2	= 0.75 / 2	*gasket seating width*	b0 = ***0.375***
varb =	min(Sqrt(b0)/2,b0)	= min(Sqrt(0.375)/2,0.375)	*eff seating width*	varb = ***0.306***
varG =	max(GOD-2*varb,(GOD-GID)/2 + GID)		*gasket load reaction diameter*	varG = ***17.138***
=	max(17.75-2*0.306,(17.75-16.25)/2 + 16.25)			
hub =	r	= 0	*length of hub*	hub = ***0.000***

Flange Loads (VIII App 2-5):

H =	0.785*varG^2*P	= 0.785*17.138^2*100	*end load*	H = ***23,055***
HP =	2*varb*3.14*varG*m*P	= 2*0.306*3.14*17.138*3*100	*contact load*	HP = ***9,886***
HD =	pi/4 * B^2 * P	= pi/4 * 16^2 * 100	*end load*	HD = ***20,106***
HT =	H - HD	= 23055 - 20106	*face load*	HT = ***2,949***
Wm1 =	H + HP	= 23055 + 9886	*bolt load*	Wm1 = ***32,941***
Wm2 =	pi*varb*varG*gy	= pi*0.306*17.138*10000	*seating load*	Wm2 = ***164,849***
Am =	max(Wm1/Sb, Wm2/Sba)	= max(32941/20000, 164849/20000)	*req bolt area*	Am = ***8.242***
Ab =	Root*Nbolt	= 0.566*16 ***1-8 UNC 2A***		Ab = ***9.056***

Total Bolt Loads - lbs - (app 2-5):

W =	(Am + Ab)*Sba/2	= (8.242 + 9.056)*20000/2	~~*seating conditions*~~	W =
HG =	Wm1 - H	= 32941 - 23055	*operating conditions*	HG = ***9,886***

Flange Moment Arms - inch - (Table App 2-6 - Integral flanges):

mhD =	varR+0.5*gOne	= 1.375+0.5*0.75	*end pressure*	mhD = ***1.750***
mhT =	(varR+gOne+mhG)/2	= (1.375+0.75+1.556)/2	*face pressure*	mhT = ***1.841***
mhG =	(varC-varG)/2	= (20.25-17.138)/2	*gasket load*	mhG = ***1.556***

Summary of Loads and Locations - Combined Operating and Seating - (lbs, inch) for *ONE HALF* bolt

Operating + Seating Conditions				
Load (lbs)		**Acting Rad**	**Loads for a model using ONE HALF bolt only**	
HT/(Nbolt*2)	***92***	***8.284***	Gasket face pressure (Operating)	
(HG+W)/(Nbolt*2)	***309***	***8.569***	Gasket load (Operating)	*W removed, HG remains*
HD/(Nbolt*2)	***628***	***8.375***	End pressure (Operating)	
(Wm1+W)/(Nbolt*2)	***-1,029***	***10.125***	Bolt reaction balancing load (Operating)	*W removed, Wm1 remains*

Do not apply Wm1 + W - use boundary conditions on the bolt to apply this balancing load
This model has a sweep of 360º/(Nbolt*2) = 11.25º for one half bolt
Omit end pressure HD if a closed pipe end is modelled and pressurized