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Particle Physics Division
Mechanical Department Engineering Note

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Author(s): E.D. LA VALLIE

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I Cover Sheet

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PPD/MSD
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CKM (E-921)

1m VACUUM WINDOW

AND

VACUUM WINDOW TEST FIXTURE

DIRECTION STATEMENT

AND

ENGINEERING CALCULATIONS

II. Contents

III Concept Development Statement (reference only)

CKM 1M UPSTREAM/DOWNSTREAM VACUUM WINDOW FLANGES AND VACUUM/HYDROSTATIC TEST FIXTURE PROPOSAL

It seems appropriate to start with an opening statement, to document the zigzag path that has been followed to establish the now hopefully acceptable inside diameter of both upstream and downstream vacuum flanges as well as prevent any backtracking.

My start date on CKM was 11/15/02 with the task scope on this assigned Vacuum Veto System being, to design a pair of vacuum window flanges for downstream end only, scaling down K-TEV 1.8 meter flanges (71" ID), using 58.88" for an inside dimension, which was the downstream CKM vacuum vessel flange inside diameter. My immediate question was to clarify scale down. Andrew Szymulanski requested that 1 ¾" thickness and 3 1/8" width of flange should if possible remain the same due to successful geometry testing form KTEV flanges. I put a preliminary drawing together for a meeting between Andrew, Don Goloski and myself on 11/19/02 to verify that we were all on the same parameter page after which we could hammer out additional parameters as necessary.

In that meeting it was stated that the flange clear window opening should be rounded to 59" but that consideration must be given to Straw Tube Detectors mounted within this last vessel. In an e-mail from Don he stated that the detectors were to be mounted via. 1/2" threaded rods with a bolt circle diameter was 62.205" which penetrated the end plates. Using stated 59" ID and putting seal outside bolt circle yielded a 5 ¼" flange width, far wider then 3 1/8", try again. Andrew, Don and I met again at which time it was agreed upon that the vacuum flanges, need not match the 59" flange ID of the vessel but the window in the deflected state must not contact this diameter which could cause rupture. It was also stated at this meeting that I should look into existing KTEV test vessel and define modifications necessary to test these new smaller flanges and to present this new geometry configuration at an upcoming experimental meeting on 12/04/02.

Just prior the meeting I showed Andrew what I had, at which time he expressed concern that with the larger vacuum flange ID of 61 3/8" driven by the rod circle might cause the deflected window (previously 9" deflection for KTEV) to come in contact with the smaller 59" ID of vessel flange. I recommended at that point that the tapped holes for the threaded rod do not penetrate vessel flange, which would allow us to return to the 59" ID. (Andrew and Don concurred). Now, with the flange inside diameter being 59"(1.5m), and the flange width being 3 1/8", the new outside diameter would be 65 1/4". This would, as previously stated require that the internal tapped holes for the threaded rods "must not" penetrate vessel end flange as they would be outside the vacuum seal.

Accepting the 3 1/8" flange width as a parameter, I have incorporated a couple of recommendations as stated in TM 1380, increasing the edge radius to 3/8" (15 times window material thickness) up from 1/8" to reduce stress concentration at edge and also increased space from groove edge to bolt hole to 2/3 x bolt diameter. At request of Andrew, I "have" maintained the o-ring seal/compression ring groove diameter proximities, groove sizes and ring materials (.250" diameter Orange Chord and 3/16" diameter 1100 Aluminum). I have further maintained the .005" Aluminized Mylar window material as well as .023" Kevlar window support material thickness pending finite element analysis of these materials. Since KTEV was built there is a newly instituted safety factor of 2 on stresses, which must be adhered to in addition to deflections causing detrimental contact

It should also be stated at this time that due to bolt clearances using same 3 degree angular spacing and 120 bolts on a smaller diameter flange, Ferry Head Cap Screw diameters have been decreased from 7/8-9 UNRC to 3/4-10 UNRC. This stated information was transmitted to Ang Lee on 12/13/02. Upon receipt of acceptable results and subject to CKM Experimental approval, fabrication drawings, costing for flanges can proceed.

During the course of implementing the aforementioned changes I discovered that there was a conflict between previously mentioned threaded rod and mounting tapped hole territories at which time I asked for information of angular timing of rod holes, which confirmed the conflict. At the experimental meeting of 12/04/02 it was stated that deflected window must not be allowed not only to interfere with flange ID but also with internal mounted detectors, although their exact size/space requirements were yet unknown. Given the tapped hole timing conflict and the probability of needing some sort of spacer to diminish intrusion of deflected window with detector, I used these needs to solve the conflicts by instituting a spacer flange whose thickness would be pending until Ang Lee calculates window deflection. If agreed upon this flange could also serve double duty, in the beam line as a spacer, and as an adapter flange to test vessel, where it could be oriented in either direction for vacuum and hydrostatic testing. With this adapter flange and new bolts, the remainder of the 1.8M Hydrostatic/Vacuum Test Vessel appears to be useable as is.

On 12/16/02 I received calculations from Ang Lee on the 1.5 Meter Vacuum Window, which yielded maximum stress of 151,123 psi and a deflection of 4.314" with a safety factor of 303,000/151,000 = 2, acceptable. With this information I re-configured the assembly, now illustrating a Straw Tube Detector Assembly of unknown cross section/length, which would be mounted directly to the vacuum vessel end flange and act as an adapter severing all ties to vessel. As this appeared to be a go on 12/12/02, I put in a request for drafting help to put together a package of drawings for costing purposes (no help is immediately available), and also began engineering note on flanges. Before this new geometry scheme was able to be presented and at the meeting of 12/19/02 the experiment decided that a 1.5 meter window was unnecessary and that it could be as small as 31.5"inside diameter. Before the day was done Don Goloski sent me an e-mail stating that we should go with 1 meter (39.375" inside diameter). At this point I decided to create a new model file (archiving old) as this was a major deviation from original design parameters and could yet be retrieved along with calculations to date.

This begins the further downsize to 1 meter ID Vacuum Windows, yet retains flange width and thickness and window materials. Angular bolt spacing however is now 5 degrees up from 3 although bolt quantity is decreased from 120 to 72, which again allows for 7/8-9 UNRC bolts. I have again put all of this information together using Straw Tube Detector as the adapter/spacer as is necessary to accept window deflection, which I presented at experimental meeting of 1/9/03 for geometry approval. It has in principle been accepted without spacer, which may or may not be needed pending stress and deflection evaluation I also stated in this meeting that now the now much smaller window size it would be less expensive to design/fabricate a new test vessel using ASME flanged and dished head. After the meeting Andrew and I requested that Ang supply a new set of calculations for the 1meter inside diameter window which is hopefully final scheme and as stated previously incorporating new design for test vessel utilizing ASME flanged and dished head and much thinner spacer flange if necessary at all. The following are the calculations for new test vessel with mating flange appropriate for testing vacuum flange assembly, bolting etc.

IV. Engineering Note Cover Sheet

EXHIBIT A-1

Vacuum Vessel Engineering Note
(per Fermilab ES&H Manual Chapter 5033)

Prepared by	Ed LaVallie	Date	2/02/04	Div/Sec	PPD/MSG
Reviewed by		Date		Div/Sec	
Div/Sec Head		Date		Div/Sec	

1. <u>Identification and Verification of Compliance</u>

Fill in the Fermilab Engineering Conformance Label information below:

This vessel conforms to Fermilab ES&H Manual Chapter 5033

Vessel Title	CKM 1 meter (39.37") Mylar/Kevlar Vac.Window
Vessel Number	
Vessel Drawing Number	
Internal MAWP	<45 psig
External MAWP	1 atmosphere
Working Temperature Range	Ambient °F °F
Design/Manufacturer	Fermilab
Date of Manufacture	n/a
Acceptance Date	n/a

Director's signature (or designee) if vessel is for manned area and requires an exception to the provisions of this chapter.

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Amendment No.	**Reviewed by:**	**Date:**

Laboratory location code	Installed in MP-9. Built and tested in MAB
Laboratory property number	Not applicable

Purpose of vessel CKM Experiment Vacuum Veto System

List all pertinent drawings
Drawing No: Location of original

2. Design Verification

Provide design calculations in the Note Appendix.

3. System Venting Verification

Can this vessel be pressurized either internal or external? [ ] Yes [X] No
If "Yes", to what pressure?

List all relief's and settings. Provide a schematic of the relief system components, and appropriate calculations or test results to prove that the vessel will not be subjected to pressures greater than 110% beyond the maximum allowable internal or external pressure.

Manufacturer	**Relief**	**Pressure Setting**	**Flow Rate**	**Size**

4. Operating Procedure Section

Is an operating procedure necessary for the safe operation of this vessel?
[ ] Yes [X] No *(If "Yes", it must be appended)*

Is a testing procedure necessary for the safe acceptance testing (acceptance testing) of this vessel? [X] Yes [ ] No
If "Yes", the written procedure must be approved by the division head prior to testing and supplied with this Engineering Note.

5. Welding Information

Has the vessel been fabricated in a Fermilab shop? [X] Yes [ ] No
If "Yes," append a copy of the welding shop statement of welder qualification.

6. Exceptional, Existing, Used and Non-Manned Area Vessels

Is this vessel or any part thereof in the above categories? [ ]Yes [X] No
If "Yes" follow the Engineering Note requirements for documentation and append to note.

V. Stress Tabulation

CKM Vacuum Window Test Fixture and Vacuum Flanges

Item	Calculation	Numerical Value	Remarks	Pg.
	Vacuum/Hydrostatic Test Vessel			
1	*VI.1.1 Flanged & Dished Head Torospherical 38"ODx 1/4"thk.*	*S=2927 psi stress*	*Vacuum Pressure @15psig x 1.67=25psig*	8
	VI.1.2 Flanged & Dished Head Torospherical 38"OD x1/4' thk.	*S=7525 psi stress*	*Hydrostatic Test @ 45 psig int. pressure*	9
2	*VI.2 Shell 38ODx.25"thk. x3.5"lg.*	*S=4860 psi stress*	*@45 psig int. pressure*	10
3	*VI.3.6 Operating Bolt Stress*	σ_b*=23321 psi stress*		15
4	*VI.3.7Design Pre-stress in bolts*	*Si=23271 psi stress*		15
5	*VI.3.8 Radial Flange Stress at BC*	S_R*=3253 psi stress*		16
6	*VI.3.9 Radial Flange Stress at ID*	*SR=3253 psi stress*		16
7	*VI.3.10 Tang. Flange Stress at ID*	S_T*=282 psi stress*		16
8	*VI.3.11 Longitudinal Hub Stress*	S_H*= 937 psi stress*		16
	Vacuum Window Assembly			
9	*VII.1* σ *Stress using tab. factors*	σ *=1865 psi stress*		20
10	*VII.2* σ *Stress for special case*	σ *=1744 psi stress*		20

VI. Test Vessel Analysis

General Design Data for Test Vessel, see assembly drawing #3921.260-MD-415054

Test Vessel Flange 38" ID x 45 5/8" OD x 1 3/4" thick with o-ring groove
Flanged and Dished Head torospherical, with outside diameter $D_0 = B$ = 38" OD
Thickness t = 1/4" chosen and verified below
Shell (head extension) = 1/4" x 38" OD x 3 1/2" long
*Outside crown radius L = D_0 * 0 .9 = 38 * 0.9 = 34.2 say L = 34 1/4"*
*Knuckle radius min. r = 6% D_0 = 0.06 * 38" = 2.28 say 2 1/4"*
Vacuum pressure (external) P = 15 psig
Hydrostatic test pressure (internal) P =45 psi
Room temperature use
Flange & Head Material ASTM SA-515 Grade 70, Sy=38000, St=70000, Sa = 17500 psi @ –20F to 650F per UCS-23 (ASME Boiler & Pressure Vessel Code Section II Part D)
Joint efficiency E = 0.7 Table UW-12
Radiography (none)
Corrosion allowance CA (none)
Seamless torispherical head, to 3 1/2" long shell to butt welded flange with o-ring groove and, 36 - 15/16" bolt holes to facilitate attachment of vacuum window assembly

VI.1. ASME Flanged and Dished Head Analysis (torispherical) #3921.260-MD-415036

The following calculations are for torispherical heads and follow Pressure Vessel Handbook (seventh edition) by Eugene F. Megyesy, referencing ASME Boiler and Pressure Code Section VIII: Division 1

VI.1.1ASME, Flanged and Dished Head (torispherical) with external pressure applied shall be computed by the procedures given for ellipsoidal heads per manual page 34. Maximum Allowable Pressure shall be the lesser of computed values in the following calculations.

*(1) Thickness computed using internal pressure formula @- 1.67 * external pressure or 1.67 * 15 psi = 25.05, say 25 psi w/ a joint efficiency of E = 1, L / r=34.25 / 2.25 = 15.2*

*When L/ r is < 16 2/3 use P = (2 * S * E * t) / (M * L -(t * (M -0 .2))) Formulas page 24 w/ Factor M =1.72 Factor M @ 15.2 page 24*

*P =(2 * 17500 * 0.55 * 0.25) / (1.72 * 34 -(0.25 * (1.72 - 0.2)))*
= 104 psig, if calculating to maximum allowable stress
=2927 psi stress, @ recommended 25 psig pressure

Input	Name	Output	Unit	Comment
25	P		psig	Design Pressure = 1.67 x 15 psig external pressure, using internal pressure formula page 34 & 24 from Handbook
	S	2927	psi	Stress
1	E		factor	Joint Efficiency
0.25	t		in.	Head Thickness
1.72	M		factor	for L / r = 34 / 2.25 = 15.1 (less then 16 2/3) thus factor M
34.25	L(R_0)		in.	Outside radius of dish

Input	Name	Output	Unit	Comment
	P	104	psig	Maximum pressure @ max. allowable stress
17500	S		psi	Max. Allowable Stress ASME table UCS 23
1	E		Factor	Joint Efficiency Factor
0.25	t		in.	Head Thickness
1.72	M		Factor	For L / r = 34 / 2.25 =15.1(less than 16 2/3), thus factor M
34.25	L (R_0)		in.	Outside radius of dish

(2) P = B / (R_0 / t) w/ Factor A= 0.125 / (R_0/ t) = 0.125 / (34.25/0.25) = 0.0009 then B for E = 29x 10^6 =11800, from charts page 40 (UGO-28.0) & 41(UCS-28.2) (Pressure Vessel Handbook) P = 11,800 / (34.25 / 0.25) = 86 psig, if calculating to allowable

	Name	Output	Unit	Comment
	P	86.13	psig	Max. Allowable Pressure (external)
11800	B		Factor	UGO-28 factor A & UCS-28.2 factor B pages 40 & 41 Pressure Vessel Handbook
34.25	L (R_0)		in.	Inside radius dish
0.25	t		in.	Head thickness

Allowable external pressure is well over vacuum pressure of 15 psig

*VI.1.2 ASME Flanged and Dished Head w/ Internal Pressure 45 psig hydrostatic pressure test L / r =34.25 / 2.25 = 15.2, which is less then 16 2/3 thus the following formula P = (2 * S * E * t) / (M * L – (t*(M -0.2))) w / M =1.72 page 24 Pressure Vessel Handbook. As the maximum allowable working pressure has been established using the external pressure procedure, using the same formula, the following are solving for thickness using joint efficiency of 0.7 at 45 psig internal test pressure, t minimum = 0.108". Using 0.250" thickness, stress = 7525 psi.*

Input	Name	Output	Unit	Comment
45	P		psig	Hydrostatic Test Pressure (internal)
	S	7525	psi	Stress @ 45 psig
0.7	E		factor	Joint efficiency
0.25	t		in.	Head Thickness
1.72	M		factor	L / r = 34.25 / 2.25 = 15.1(less then 16 2/3)
34.25	L		in.	Inside radius of dish

Input	Name	Output	Unit	Comment
45	P		psig	Allowable Pressure (internal)
17500	S		psi	Max. Allowable Stress ASME table UCS-23
0.7	E		Factor	Joint Efficiency Factor
	t	0.108	in.	Head Thickness
1.72	M		Factor	L / r = 34 / 2.25 = 15.1(less than 16 2/3), thus factor M
34.25	L		in.	Inside radius of dish

VI.2. Shell Analysis (extension to head)

*Required shell wall thickness using ASTM A-36 carbon steel w/ allowable of 12700 psi t = P * R / S * E + 0.4 * P = 45 * 19 / ((12700 * 0.7) + (0.4 * 45)) = 0.096" minimum thickness working to allowable, 4860 psi stress for 0.250" thickness and test pressure.*

Input	Name	Output	Unit	Comment
	t	0.096	in.	Minimum shell thickness
45	P		psig	Test pressure
19	R		in.	Outside radius of pipe = 38 / 2
12700	S		psi	Allowable stress for A-36 carbon steel
0.7	E		factor	Joint efficiency

Input	Name	Output	Unit	Comment
0.25	t		in.	shell thickness
45	P		psig	working pressure
19	R		in.	OD pipe
	S	4860	psi	Allowable stress
0.7	E		factor	Joint efficiency

VI.3. Test Vessel Flange Analysis #3921.260-MD-415037 Item 1

Classification of the flange/head/shell connection:

As per ASME Code Section VIII, Division 1 Appendix Y (See figure #1 page 9)

Flange: Optional Type, calculated as an integral type as per Appendix 2 Figure 2-4.8
Class 1 Assembly as per Appendix Y-5.1.a
Category 1 Flange as per Appendix Y-5.2a
Calculation: as an integral type

See Appendix Y, Table Y-6.1 for Summary of Applicable Formulas
In this case Formulas, (5a), (7)-(13), (14a), (15a), (16a) apply.

VI.3.1Numerical Data Tabulation

$A = 45.625"$

AR =0.243

$A_b = 15.084\ in^2$

$B = 37.5"$

$B_1 = 37.75"$

$C = 44.125"$

$D = 0.937"$

$E_f = 29.5 \times 10^6\ psi$

$E_b = 29.3 \times 10^6\ psi$

$E0_B = 5470.335$

$F = 0.9$

$F' = 0.69$

$G = 40.75"$

$H = 58659\#$

$H_C = 293119.53\#$

$H_D = 49676\#$

$H_G = 0\#$

$H_T = 8983\#$

$ID = 38"$

$J_P = 0.096$

$J_S = 0.170$

$M_P = 172932\ in.lb.$

$M_S = 2081\ in.lb.$

$P_V = 15\ psig$

$P_H = 45\ psig$

$R = 2.709"$

$S_i = 2371.24\ psi$

$S_R = 3252.696\ psi$

$S_T = 282\ psi$

$S_H = 937\ psi$

$V = 0.42$

$W_{m1} = 351778\#$

$a = 1.189$

$d = 0.731"$

$d_b = 0.875"$

$f = 4.6$

$g_0 = 0.250"$

$g_1 = 0.60"$

$h_c\ min. = 0.438"$

$h_c\ max. = 0.75"$

$h_c\ mean = 0.594"$

$h_D = 3.011"$

$h_G = 1.6875"$

$h_0 = 3.062"$

$h_T = 2.25"$

$l = 5.907$

$n = 36$

$r_B = 0.003$

$r_E = 1.007$

$t = 1.75"$

$t_S = 0.219"$

$\beta = 1.084$

$\sigma_b = 23321\ psi$

$S_R = 30.489\ psi$

$Z = 10.217$

VI.3.2 Flange Moment due to hub interaction form Y-6 (7)

$M_s = (J_p * F' * M_p) / (t^3 + (J_s * F'))$
$= (0.096 * 0.690 * 172032) / (1.75^3 + (0.170 * 0.690))$
$= 2081$

$J_p = (1 / B_1)*((h_D / \beta)+(h_C / a))+(\pi * r_B)$
$= (1 / 37.75) * ((3.011 / 1.084) + (0.594 / 1.189)) + (\pi*0.003)$
$= 0.096$

$B_1 = B + g_0$ *with f (hub stress correction) greater then 1*
$= 37.5 + 0.250$
$= 37.750"$

$h_D = R + (0.5 * g_1)$ *from table2-6 Appendix 2 (moment arms for flange loads)*
$=2.709 + (0.5 * 0.604)$
$= 3.011"$

$R =((C - B) / 2) - g_1$ *(for integral and hub flanges)*
$= ((44.125 - 37.5) / 2) - 0.604$
$= 2.709"$

$g_1 = g_0 + {}_{AC\,(weld)} = 0.604"$

$\beta =$ *shape factor* $= (C + B_1) / (2 * B_1)$
$= (44.125 + 37.75) / (2 * 37.75)$
$= 1.084$

h_C *mean value* $= (h_C \text{ min.} + h_C \text{ max}) / 2$ *(see figure #2)*
$= (0.438 + 0.750) / 2$
$= 0.594"$

h_C *min.* $= d_b / 2$
$= 0.875 / 2$
$= 0.438"$

h_C *max.* $= (A - C) / 2$
$= (45.625 - 44.125) / 2$
$= 0.75"$

a (shape factor) $= (A + C) / (2 * B_1)$
$= (45.625 + 44.125) / (2 * 37.75)$
$= 1.189$

$r_B = 1/n * ((4 / \sqrt{(1- AR^2)})^{*} (\tan^{-1}(\sqrt{(1+ AR) / (1 - AR)})) - \pi - (2 * AR))$
$= 1/36*((4 / (\sqrt{(1- 0.243^2)})*(\tan^{-1}(\sqrt{((1+ 0.243)/(1- 0.243)})))-\pi-(2* 0.243)$
$= 1/36 * (4.124* 0.908$ *radians* $(52.032^{0)} -\pi - 0.486) = 0.003$

(Convert degrees to radians $52.032\,° / (360 / (2*\pi)) = 0.908)$

$n = 36$ *(number bolts)*

$AR = (n * D) / (\pi * C)$
$= (36 * 0.9375) / (\pi * 44.125)$
$= 0.243$ *(bolt hole aspect ratio)*

F' for Category I Class 1
$F' = g_0^2 * (h_0 + F * t) / V$
$= .250^2 * (3.062 + (0.9 * 1.75)) / 0.42$
$= 0.690$

$h_0 = \sqrt{(B * g_0)}$
$= \sqrt{(37.5 * 0.25)}$
$= 3.062"$

F from code Fig. 2-7.2 for, $g_1/g_0 = 2.416$ *and* $h/h_0 = 0.116$
$= 0.9$

V = Factor for integral type flanges from code Fig. 2-7.3, for
$h/h_0 = 0.116$, *and* $g_1 / g_0 = 2.416$
$= 0.42$

$M_P = H_D h_D + H_T h_T + H_G h_G$
$= (49676 * 3.011) + (8983 * 2.500) + (0 * 1.688)$
$= 172032$ *in.-lb*

$H_D = 0.785 * B^2 * P$
$= 0.785 * 37.5^2 * 45$
$= 49676\#$

$H_T = H - H_D$
$= 58659 - 49676$
$= 8983\#$

$h_T = (R + g_1 + h_G) / 2$
$= (2.709 + 0.604 + 1.6875) / 2$
$= 2.500"$

$H_G =$ *(gasket load due to seating pressure)*
in this case the gasket is very soft,
we can neglect the reaction force of gasket
$= 0\#$

$h_G = (C-G) / 2$
$= (44.125 - 40.750) / 2$
$= 1.6875"$

$J_s = (1/B_1) * ((2h_D / \beta) + (h_C / a)) + (\pi * r_B)$
$= (1 / 37.75) * ((2 * 3.011 / 1.084) + (0.594 / 1.189)) + (\pi * 0.003)$
$= 0.170"$

VI.3.3 Slope of Flange at Inside Diameter Times E_ Y-6.1(8)

$E\theta_B = (5.46 / (\pi * t^3)) * ((J_S * M_S) + (J_P * M_P))$
$= (5.46 / (\pi * 1.75^3)) * ((0.170 * 2081) + (0.096* 172032))$
$= 5470.335 \qquad \theta = .010°$

VI.3.4 Contact Force Between Flanges at h_C_Y-6.1(9)

$H_C = (M_P + M_S) / h_C$
$= (172032 + 2081) / 0.594$
$= 293119.529\#$

VI.3.5 Bolt Load at Operating Conditions_Y-6.1 (10)

$W_{m1} = H + H_G + H_C$
$= 58659 + 0 + 293119.529$
$= 351778.529\#$

$H = 0.785 * G^2 * P$
$= 0.785 * 40.750^2 * 45$
$= 58659\#$

VI.3.6 Operating Bolt Stress_Y-6.1 (11)

$\sigma_b = W_{m1} / A_b$
$= 351778.529 / 15.084$
$= 23321$ psi

$A_b = (\pi * d^2) / 4 * n$
$= (\pi * (0.731)^2 / 4) * n$
$= 0.419\ in^2 * 36$ bolts
$= 15.084\ in^2$

d = minor of 7/8-9 thread =.738 minus tol. 0.007 = 0.731

VI.3.7 Design Prestress in Bolts_Y-6.1 (12)

$S_i = \sigma_b - ((1.159 * h_C^2 * (M_P + M_S)) / (a * t^3 * l * r_E * B_1))$
$= 23321 - ((1.159 * 0.594^2 * (172032 + 2081)) / (1.189 * 1.75^3 * 5.907 * 1.007 * 37.75)) = 23321 - (71201.235 / 1430.9)$
$= 23271.24$ psi

$l = (2 * t) + t_s + (0.5 * d_b)$ *Value l calc. in this case, (2 * t) is really (3 * t)*
$= (3*1.75) + 0.219 + (.5 * .875)$
$= 5.907$

$t_s = 0.219"$ *(washer thickness)*

$d_b = 0.875"$ *(bolt diameter)*

Flange materials for test vessel and vacuum windows are same
A515 Grade 70 with both, having carbon content> 30%
Bolts are Ferry Cap Countr-bor Screws SA-574 4140, 170,000 psi tensile,
135,000 psi Yield w/ carbon content> 30% w/ allowable of 33,800 psi from
–20F to 550 F from ASME Boiler and Pressure Vessel Code Section II Part D

$r_E = Ef\,(\,flanges) / Eb\ (bolts)$
$= 29.5*10^6 / 29.3*10^6$
$= 1.007$

VI.3.8 Radial Flange Stress at Bolt Circle_Y-6.1 (13)

$S_R = (6 * (M_P + M_S)) / (t^2 * ((\pi * C) - (n * D))$
$= (6 * (172032 + 2081)) / (1.75^2 * ((\pi * 44.125) - (36 * 0.9375)))$
$= 1044678 / 321.173$
$= 3252.696$ psi

VI.3.9 Radial Flange Stress at Inside Diameter_Y-6.1 (14a)

$S_R = -[((2 * F * t) / (h_0 + (F * t)) + 6] * [M_S / (\pi * B_1 * t^2)]$
$= -[((2 * 0.9 * 1.75)/(3.062 + (0.9 * 1.75)) + 6] * [2081 / (\pi * 37.75 * 1.75^2)]$
$= 5.321 * 5.73$
$= 30.489$ psi

VI.3.10 Tangential Flange Stress at Inside Diameter_Y-6.1 (15a)

$S_T = [(t * E\theta_B) / B_1] + [(2 * F * t * Z) / (h_0 + (F * t)) - 1.8] * [M_S / (\pi * B_1 * t^2)]$
$[(1.75*5470.335)/37.75] + [(2*0.9*1.75*10.217)/(3.062+(0.9*1.75)) - 1.8]$
$*[2081/(\pi*37.75*1.75^2)]$
$=252.592+5.141*5.73$
$= 282.05$ psi

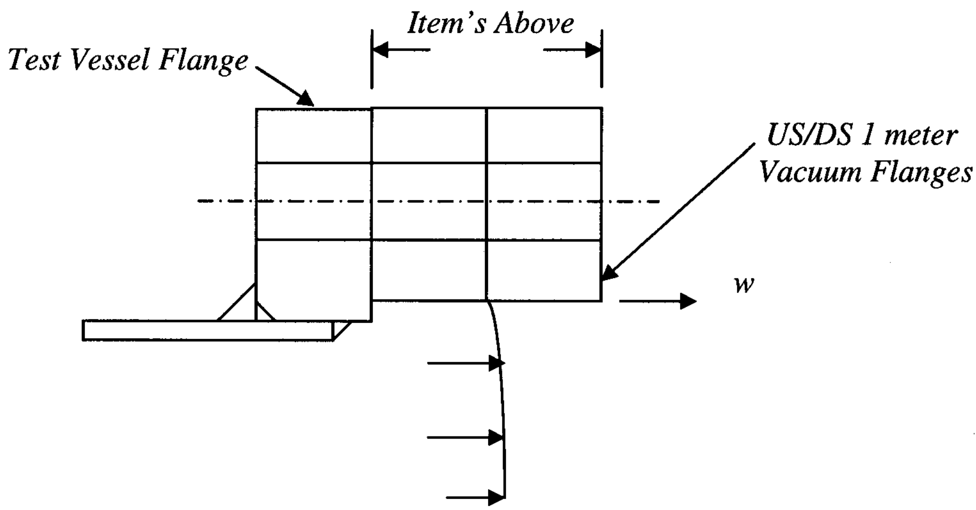
Z---- Form Fig. 1-7.1 for $K = A / B = 45.625 / 37.5 = 1.217$
$Z = (K^2 + 1) / (K^2 - 1)$
$= (1.217 + 1) / (1.217 - 1)$
$= 10.217$

VI.3.11 Longitudinal Hub Stress_Y-6.1 (16a)

$S_H = (h_0 * E\theta_B * f) / (0.91 * (g_1 / g_0)^2 * B_1 * V)$
$(3.062 * 5470.335 * 4.6) / (0.91 * 2.416^2 * 37.75 * 0.41)$
$= 937.219$ psi

f---- From fig. 2-7.6 w/ $h / h_0 = 0.116$ *and* $g_1 / g_0 = 2.41$
$f = 4.6$

VII. Flange Analysis drawing # 3921.260-MD-415039 items #1 and #3



B.1 Classification
The analyses in subsection A (General Requirements) of this calculation showed very marginal angular deflection of the Test Vessel Flange. Having that as a base we will treat the window assembly as a "fixed case and formulas for flat circular plates of constant thickness will apply. Roark's Formulas for stress and strain seventh edition

The flange/flange metal contact will occur with, angular dislocation, (contact at outer flange diameter) and with washer installation between flanges at bolt centerline. Both cases will be investigated during tests with the second being the base with flange fixed.

Design Data 1 meter Flange

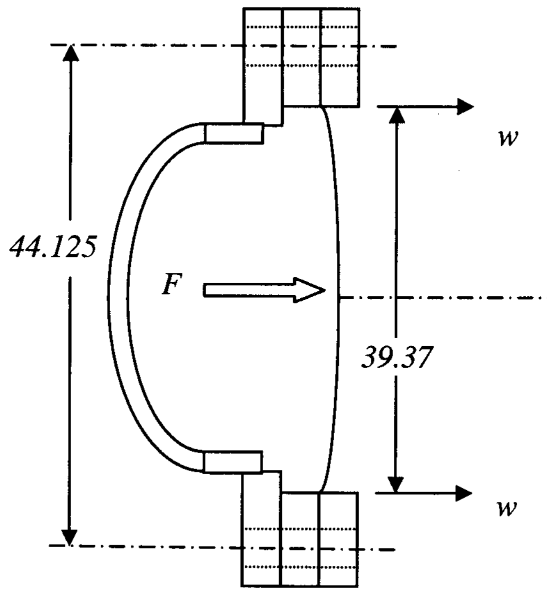
P_H = 45 psi (maximum hydrostatic pressure)

a = C / 2
= 44.125" / 2
= 22.063" (bolt circle radius)

b = Window Flange Inside Radius / 2
= 39.37" Dia. / 2
= 19.685" (inside radius)

r_0 = b = 19.685"

w =find force at inside radius of flange



$$
\begin{aligned}
F &= A * P_H \\
&= ((\pi * diameter^2) / 4) * \ 45\ psi \\
&= ((\pi * 39.37^2) / 4) * 45 \\
&= 54781\ psi.\ pressure\ on\ entire\ window
\end{aligned}
$$

$$
\begin{aligned}
w &= F / (\ \pi * diameter) \\
&= 54781 / (\pi * 39.37) \\
&= 442.9\ lbs.\ lineal\ in.\ say\ 443
\end{aligned}
$$

Rule
if or(b<=0,r0a,a<=b,ra) then caution1='Dim_Err else caution1='_
if t>(a-b)/4 then caution2='t_Thick else caution2='_
D=E*t^3/(12*(1-nu^2))
call get_tab(matnum,matl,E,nu)
call case(a,D,nu,b,r0,r;ya,yb,Mra,Mrb,Qa,Qb,tha,thb,case)
call load(nu,r0,a,b,r,D,Mra,Mrb,tha,thb,Qa,Qb,ya,yb;y,th,Mr,Mt,Q)
sigma_r=6*Mr/(t^2)
sigma_t=6*Mt/(t^2)
call clear()
plot=given('plot,plot,'y)
if and (solved(),plot<>'n) then call genplot()

Input	Name	Output	Unit	Comment
				Case 1 - Annular plate with a uniform
				annular line load
443	w		lbf/in	Uniform annular line load
19.685	R_0		in	Radius to annular line load
				Table 11.2: Roark's Formulas
				Formulas for flat circular plates of
				constant thickness
	case	'Case_1e		Reference number
	caution1	'_		Caution Message
	caution2	't_Thick		
	matnum			Material Number (See Material Table)
	matl			Material name
	plot	'y		Generate plots? 'n=no (Default=yes)
2.95E7	E		psi	Young's Modulus
0.3	nu			/cadwhs/server01/ms_lavallie/CKM.mf1
22.062	a		in	Outer Radius
19.685	b		in	Inner Radius
1.75	t		in	Plate Thickness
	D	14478165	lbf-in	Plate Constant
				AT RADIUS:
22.062	r		in	Radius
	y	0	in	Deflection
	th	0	rad	Radial Slope Angle
	Mr	-952.051	lbf-in/in	Radial Bending Moment
	Mt	-285.615	lbf-in/in	Tangential Bending Moment
	Q	-395.252	lbf/in	Shear Force
	sigma_r	-1865.242	psi	Radial Bending Stress
	sigma_t	-559.573	psi	Tangential Bending Stress
				AT OUTER EDGE:
	ya	0	in	Deflection
	tha	0	rad	Radial Slope Angle
	Mra	-952.051	lbf-in/in	Radial Bending Moment
	Qa	-395.252	lbf/in	Shear Force

				AT INNER EDGE:
	yb	-.0001284	in	Deflection
	thb	.0000825	rad	Radial Slope Angle
	Mrb	0	lbf-in/in	Radial Bending Moment
	Qb	0	lbf/in	Shear Force

Using formula for special case when: $r_0 = b$ *(load at inner edge)*

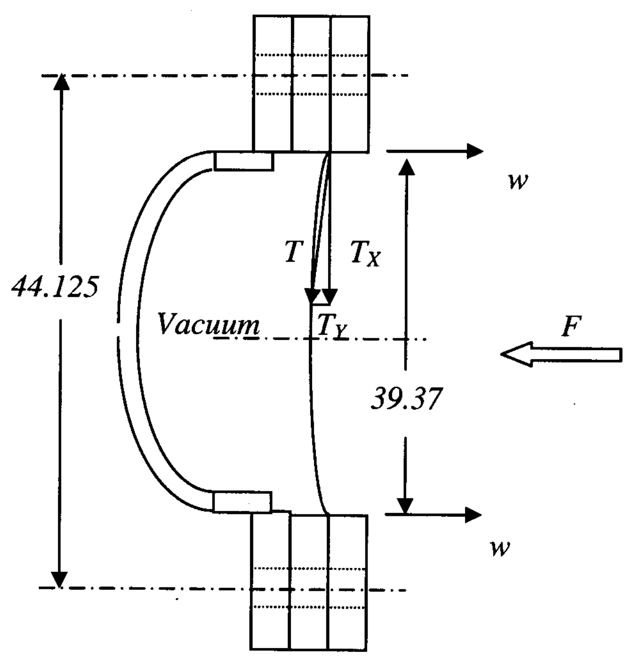
$b / a = 19.685 / 22.063$
$\quad = 0.8922,$ *for* $b / a = 0.9 \ldots Kmra = 0.0911$ *coefficient*

$Mra = Kmra * w * a$
$Mra = 0.0911 * 443 * 22.063$
$Mra = 890$ *in. lb.*

$Max.\sigma = (6 * Mra) / t^2$
$\quad = (6 * 890) / 1.75^2$
$\quad = 1744$ *psi*

Both methods of analysis give safe range of stresses.

VIII. Checking for Hoop Stresses in the window assembly #3921.260-MD-415039



VIII.1 Find the Radial Force T_X *(Vacuum Case)*

This computation needs to be done using ANSYS Finite Analysis Program by Ang Lee due to historical knowledge of all large vacuum windows built in recent times.