COLD VESSEL BOLTED JOINT ANALYSIS

(Load: Test Pressure, 3.5 bar)

I. PRELIMINARY

The figures on the right show the free-body diagram of a bolted joint. Fig.1 shows the pre-load condition, and Fig. 2, the loaded condition. When an external load is applied to a closed and pre-loaded bolted joint assembly, the change in length of the bolt, ΔL_b, must equal to the total change in length, $\Sigma(\Delta L_{1,2..})$, of the compressed components.

Thus, $\Delta L_b = \Sigma(\Delta L_{1,2..})$ **< -- Eq. 1**

The change in length of an individual member is simply the force acting on that particular member divided by its spring constant, or stiffness.

So, if we let K_b be the stiffness of the bolt, and K_1, K_2,...etc., be the stiffness of the individual clamped components, we can re-write Eq. 1 as follows:

$$(W - W_i)/K_b = (W_i - (W - W_e)) / K_1 + (W_i - (W - W_e)) / K_2$$

Then, solving for W results in the following relationship:

$$\mathbf{W = W_i + W_e / (1 + K_j/K_b)} \quad , \quad \textbf{< --- Eq. 2}$$

where, $1/K_j = 1/K_1 + 1/K_2$ ....

Now, let's define a parameter, $r = 1 / (1 + K_j/K_b)$, and re-write Eq. 2 as:

$$\mathbf{W = W_i + rW_e} \quad \textbf{< -- Eq. 3}$$

We recognize right away that this is a slope-intercept function, and r is the slope of the load line in Fig. 3.

Separation of the clamped parts occur when W = We, represented by the 45 deg. line from the origin in Fig. 3. This does not necessarily mean immediate failure, but simply the point when the mating parts cease to share the applied load and the bolt carries the entire load the rest of the way until either the bolt or the joint fails.This situation should be avoided, however, since the resulting ***non-linear behavior*** makes failure prediction very difficult.

We will proceed to apply Eq. 3 to the bolted interfaces of the Atlas Barrel Cryostat cold vessel in conjunction with various Finite Element study that was done on this subject. We will begin by calculating the respective stiffness of the individual members of the bolted assembly.

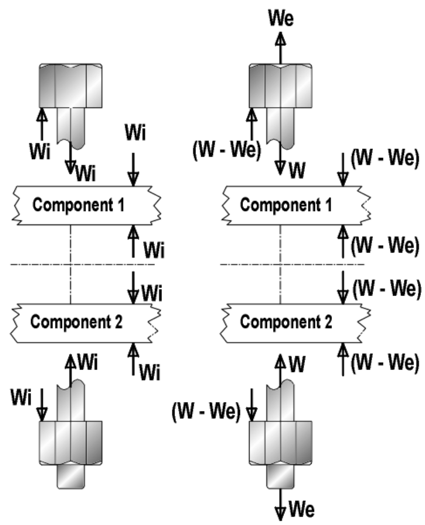


Fig. 1: Pre-Load Fig. 2: With Applied Load

Notation in the above figures:

Wi : Initial Pre-load (kN)
We : Effective applied load
W : Resultant load on Bolt

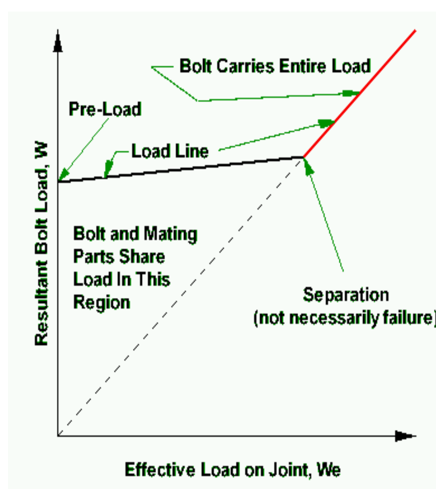


Fig. 3: W vs. We

II. STIFFNESS CALCULATIONS

1. Cold Bulkhead Outer & Inner Bolts: Nitronic, M16x2 (Coarse);
Central Bolts: Nitronic, M20x2.5 (Coarse)

Notation:

D = dia of unthreaded portion (Nominal dia)
D_m = minor dia at threaded section
D_{pd} = pitch diameter of thread
L_T = length of threaded portion
L_1 = length of unthreaded portion
S_i = Initial pre-stress
E_b = Elastic modulus of bolt material
S_y = yield point of bolt
A_1 = area of unthreaded portion
A_m = area of threaded portion
K_b = bolt spring constant

Note: *The first row in the 3x1 arrays below refers to the outer bolts, the 2nd row, to the inner bolts, and the third, to the central bolts.*

Given Data & Units Conversion Factors:

$$D := \begin{bmatrix} 16 \\ 16 \\ 20 \end{bmatrix} mm \qquad D_m := \begin{bmatrix} 13.546 \\ 13.546 \\ 16.933 \end{bmatrix} mm \qquad D_{pd} := \begin{bmatrix} 14.701 \\ 14.701 \\ 18.376 \end{bmatrix} mm$$

$$MPa := 10^6 \cdot Pa \qquad kN := 10^3\ N \qquad \mu m := 10^{-6}\ m$$

$$L_T := \begin{bmatrix} 42 \\ 40 \\ 50 \end{bmatrix} \cdot mm \qquad L_1 := \begin{bmatrix} 20 \\ 40 \\ 100 \end{bmatrix} \cdot mm \qquad S_y := 621 \cdot MPa \qquad S_i := 320 \cdot MPa \qquad E_b := 206000 \cdot MPa$$

In conformance with Hooke's Law, the stiffness of a member is generally expressed as, $K = P/\Delta = AE / L$, where P is the force, Δ, the deformation, A, area, L, length, and E is the elastic modulus. This is directly derived from the basic definition of E = Stress/Strain = (P/A) / (Δ/L). From this basic expression, the overall stiffness of the bolt can be calculated by considering the threaded and unthreaded sections as well as the contributions of the bolt head and that portion that is screwed into the embedded Keensert; Ref. 1 describes this in a little more detail.
Let's start by calculating the following areas:

$$A_1 := \frac{\pi}{4} \cdot \overrightarrow{\left(D^2\right)} \qquad A_m := \frac{\pi}{4} \cdot \overrightarrow{\left(D_m^{\ 2}\right)}$$

$$A_1 = \begin{bmatrix} 201.062 \\ 201.062 \\ 314.159 \end{bmatrix} \circ mm^2 \qquad A_m = \begin{bmatrix} 144.116 \\ 144.116 \\ 225.194 \end{bmatrix} \circ mm^2$$

Now, considering all the terms mentioned above, the expression for the bolt stiffness, K_b, is:

$$K_b := \overrightarrow{\left[\frac{1}{\frac{1}{E_b}\cdot\left(0.4\cdot\frac{D}{A_1}+\frac{L_1}{A_1}+\frac{L_T}{A_m}+0.4\cdot\frac{D_m}{A_m}\right)}\right]}$$

< -- Eq. 4 (See Ref. 1)

Hence,

$$K_b = \begin{bmatrix} 447.5 \\ 377.3 \\ 345.7 \end{bmatrix} \circ \frac{kN}{mm}$$

< -- Bolt Stiffness

2. Flange and Washer: Al 5083, & Invar

Notation:

- D_w = dia of Invar washer, mm
- D_h = bolt hole dia, mm
- L_w = thickness of washer, mm
- L_f = thickness of flange, mm
- L_j = total thickness of joint, mm
- E_w = Elastic modulus of washer (Invar), MPa
- E_f = Elastic modulus of flange (Al 5083), MPa
- A_w = area of washer, mm.^{2}
- A_f = equivalent area of flange portion, mm.^{2}
- K_w = bolt spring constant, N/mm
- K_f = flange spring constant, N/mm
- K_j = joint spring constant, N/mm
- Nb = number of bolts

Note: *The first row in the 3x1 arrays below refers to the outer bolts, the 2nd row, to the inner bolts, and the third, to the central bolts.*

Given Data:

$$D_w := \begin{bmatrix} 34 \\ 36 \\ 44 \end{bmatrix}\cdot mm \qquad D_h := \begin{bmatrix} 17.5 \\ 20 \\ 24 \end{bmatrix}\cdot mm \qquad L_w := \begin{bmatrix} 10 \\ 15 \\ 16 \end{bmatrix}\cdot mm \qquad L_f := \begin{bmatrix} 50 \\ 25 \\ 75 \end{bmatrix} mm \qquad L_j := \begin{bmatrix} 85 \\ 90 \\ 150 \end{bmatrix} mm$$

$$N_b := \begin{bmatrix} 208 \\ 84 \\ 172 \end{bmatrix} \qquad E_w := 151724\cdot MPa \qquad E_f := 71018\cdot MPa$$

(a) Calculate the cross-sectional area of the invar washer, A_w:

$$A_w := \frac{\pi}{4}\cdot\overrightarrow{\left(D_w^{\,2} - D_h^{\,2}\right)}$$

, or

$$A_w = \begin{bmatrix} 667.392 \\ 703.717 \\ 1.068\cdot 10^3 \end{bmatrix} \circ mm^2$$

(b) Calculate the effective cross-sectional area of the flange, A_f:

At first glance, one would think that the area under the washer, A_w, determines the stress area of the flange due to the applied load.This approach ignores the thickness of the flange and will yield the same result no matter how thick it would be. Observations in actual practice, however, indicate that the load distribution on the flange is a function of the joint thickness, thus giving rise to the ***effective area*** concept.

According to Ref. 1, the effective flange cross-section, A_{ff}, that shares the load is given by the equation below. Notice the additional term (L_j / 10) added to the washer diameter; This term is a quick conservative way of making a rough estimate of the effective flange cross-sectional area.

Thus,

The equiv. area of the flange per bolt is:

$$A_{ff} := \frac{\pi}{4}\cdot\overrightarrow{\left[\left(D_w + \frac{L_j}{10}\right)^2 - D_h{}^2\right]} \quad \text{, or} \qquad A_{ff} = \begin{bmatrix} 1.178\cdot10^3 \\ 1.276\cdot10^3 \\ 2.282\cdot10^3 \end{bmatrix}\cdot mm^2$$

Comparing this area to the one under the washer (A_w) we can see the respective increases as indicated below.

$$\overrightarrow{\left(\frac{A_{ff}}{A_w}\right)} = \begin{bmatrix} 1.8 \\ 1.8 \\ 2.1 \end{bmatrix}$$

Photoelastic stress analysis methods, however, have shown a better solution to the effective area problem (Ref. 2). It gave rise to a graphical method of determining the effective flange area. In Fig. 4 below, the dimension "a" determines the effective flange area; For the outer bolted joint assembly of the cold vessel it can be calculated as follows:

$$d := D \qquad \rho := \frac{D_w}{2} \qquad g := L_w$$

$$h := \begin{bmatrix} 20 \\ 25 \\ 32 \end{bmatrix} mm$$ < -- thickness of alum. flange under Invar washer

$$R := \begin{bmatrix} 12 \\ 12 \\ 15 \end{bmatrix} mm$$ < -- half-distance between flats of the bolt head

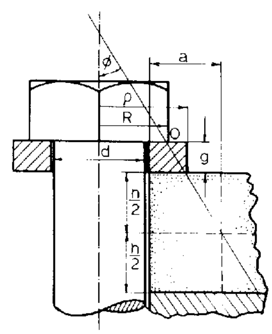


Fig. 4: Effective Area Method

From the geometry as shown in Fig. 4:

$$\phi := \overrightarrow{\mathrm{atan}\left[\frac{(\rho - R)}{g}\right]} \quad \text{, or} \qquad \phi = \begin{bmatrix} 26.57 \\ 21.8 \\ 23.63 \end{bmatrix}\cdot deg$$

$$a := \vec{\rho} + 0.5\cdot\overrightarrow{(h\cdot\tan(\phi) - d)} \quad \text{, or} \qquad a = \begin{bmatrix} 14 \\ 15 \\ 19 \end{bmatrix}\cdot mm$$

Therefore the diameter of the effective flange area is:

$$D_e := D_h + 2 \cdot a \qquad \text{, or} \qquad D_e = \begin{bmatrix} 45.5 \\ 50 \\ 62 \end{bmatrix} \cdot mm$$

As I mentioned previously, according to Ref. 1, a quick estimate of the effective flange diameter can be done easily by simply adding (L_j/10) to the washer diameter, D_w. Comparing this approach to D_e from the graphical approach above, shows a very close correlation:

$$\left(D_w + \frac{L_j}{10}\right) = \begin{bmatrix} 42.5 \\ 45 \\ 59 \end{bmatrix} \cdot mm$$

Thus, the effective flange cross-sectional area per bolt is:

$$A_f := \frac{\pi}{4} \cdot \overrightarrow{\left(D_e^{\,2} - D_h^{\,2}\right)} \qquad \text{, or} \qquad A_f = \begin{bmatrix} 1.385 \cdot 10^3 \\ 1.649 \cdot 10^3 \\ 2.567 \cdot 10^3 \end{bmatrix} \cdot mm^2 \qquad \text{, and} \qquad \overrightarrow{\left(\frac{A_f}{A_w}\right)} = \begin{bmatrix} 2.08 \\ 2.34 \\ 2.4 \end{bmatrix}$$

The effective area is more than double that of the area under the washer as indicated by the ratio (A_f /A_w) !

Hence, the stiffness of the washer and the flange are:

$$K_w := \overrightarrow{\left(\frac{A_w \cdot E_w}{L_w}\right)} \qquad K_w = \begin{bmatrix} 1.013 \cdot 10^4 \\ 7.118 \cdot 10^3 \\ 1.013 \cdot 10^4 \end{bmatrix} \cdot \frac{kN}{mm}$$

< -- Invar Washer Stiffness

$$K_f := \overrightarrow{\left(\frac{A_f \cdot E_f}{L_f}\right)} \qquad K_f = \begin{bmatrix} 1.968 \cdot 10^3 \\ 4.685 \cdot 10^3 \\ 2.43 \cdot 10^3 \end{bmatrix} \cdot \frac{kN}{mm}$$

< -- Aluminum Flange Stiffness

Let "K_o" (see below) be the stiffness of the outer bolted joint, "K_i", the inner joint, and "K_c" the central joint ; There's only one invar washer each in the outer and inner joints, and two in the central joint. The mating flange in the outer joint has a thinner section due to the omega seal groove, so I assumed only 20% more than the bulkhead thickness. On the other hand, the inner joint has a pretty thick mating flange - roughly about 3 times that of the bulkhead flange. So for the stiffness calculations below I used the factor 4 to account for both mating flanges. The central joint is straightforward, since it is basically symetrical.

$$K_o := \frac{1}{\left(\frac{1}{K_{w_{0,0}}} + \frac{1.2}{K_{f_{0,0}}}\right)} \qquad K_i := \frac{1}{\left(\frac{1}{K_{w_{1,0}}} + \frac{4}{K_{f_{1,0}}}\right)} \qquad K_c := \frac{1}{\left(\frac{2}{K_{w_{2,0}}} + \frac{2}{K_{f_{2,0}}}\right)}$$

Thus, the combined joint assembly stiffness is:

$$K_j := \begin{bmatrix} K_o \\ K_i \\ K_c \end{bmatrix} \quad , \text{ or } \quad K_j = \begin{bmatrix} 1.411 \cdot 10^3 \\ 1.006 \cdot 10^3 \\ 980.045 \end{bmatrix} \cdot \frac{kN}{mm}$$

< -- Joint Stiffness

Since the stiffness of each member has been determined, the joint-to-bolt stiffness ratio is:

$$\overrightarrow{\left(\frac{K_j}{K_b}\right)} = \begin{bmatrix} 3.154 \\ 2.666 \\ 2.835 \end{bmatrix}$$

< -- Stiffness ratio

Subsequently, the slope, r, of the load line in Eq. 3 would be :

$$r := \overrightarrow{\left(\frac{1}{1 + \frac{K_j}{K_b}}\right)} \quad , \text{ or } \quad r = \begin{bmatrix} 0.241 \\ 0.273 \\ 0.261 \end{bmatrix}$$

III. BOLT STRENGTH

The yield strength of the bolt can be calculated from the bolt stress area, and the specified yield point of the bolt material. The yield point as given in the KHI specification is, S_y = 621 MPa; This puts the bolt classification within metric class 8.8 which is equivalent to SAE Grade 5. Commercial bolts of this class should have a mark on the bolt head indicating its strength.

Now, let us evaluate the diameter, D_s, associated with the stress area; This is simply the mean between the root diameter, D_m, and the pitch diameter, D_{pd}, thus:

$$D_s := \frac{\left(D_{pd} + D_m\right)}{2} \quad , \text{ or } \quad D_s = \begin{bmatrix} 14.124 \\ 14.124 \\ 17.654 \end{bmatrix} \cdot mm$$

So, the stress area will be:

$$A_s := \frac{\pi}{4} \cdot \overrightarrow{D_s^{\,2}} \quad , \text{ or } \quad A_s = \begin{bmatrix} 156.666 \\ 156.666 \\ 244.794 \end{bmatrix} \cdot mm^2$$

And the yield strength is:

$$F_y := A_s \cdot S_y \quad , \text{ or } \quad F_y = \begin{bmatrix} 97 \\ 97 \\ 152 \end{bmatrix} \cdot kN$$

< -- Bolt Yield Strength

IV. BOLT PRE-LOAD

The pre-load force, W_i, that corresponds to the pre-stress, S_i = 320 MPa, is:

$W_i := A_s \cdot S_i$, or $W_i = \begin{bmatrix} 50 \\ 50 \\ 78 \end{bmatrix} \cdot kN$ **< -- Bolt Pre-load Force**

To get a rough estimate of the assembly torque, we will assume the coeff., k, in the formula below, to be 0.2 which corresponds to a friction factor of about 0.15. Some experts call this as the "nut factor", since it is quite "nutty engineering" to rely on the torque wrench to determine the initial clamping force especially on critical joints.

$k := 0.2$

From the pre-load, W_i, above, the required torque will be:

$T := k \cdot \overrightarrow{\left(D \cdot W_i\right)}$ **<-- standard formula for calculating bolt torque**

$T = \begin{bmatrix} 160 \\ 160 \\ 313 \end{bmatrix} \cdot N \cdot m$ or, ($T = \begin{bmatrix} 118 \\ 118 \\ 231 \end{bmatrix} \cdot ft \cdot lbf$) **< -- Required torque**

Average stress on the Al5083 flange:

$S_a := \overrightarrow{\left(\frac{W_i}{A_f}\right)}$, or $S_a = \begin{bmatrix} 36 \\ 30 \\ 31 \end{bmatrix} \cdot MPa$ **< -- Bearing stress on Al5083 under invar washer**

Note that the yield stress of alum5083 is 117 MPa. The bending of the flange under the invar washer is not significant unless there is separation between the clamped parts. As long as there is enough preload to keep the joint closed, the dominant bulkhead bending stress will be somewhere else away from this joint since the combined section modulus is high at this clamped region.

V. SERVICE LOADS

A preliminary estimate of the expected service load on the assembly due to internal pressure alone was done by FEA with Ansys. The "piston" force alone due to the internal pressure of 3.5 bar would require less than 10 kN per outer bolt, and 17 kN per inner bolt. But due to the design of the outer joint interface which is essentially an ASME Appendix Y flange assembly, severe moments generated on the bulkhead drastically increases the service load on the outer bolts. Ansys results indicated that it would be about 30 kN per outer bolt, more than 3 times that of the normal "piston" force. On the other hand, the inner bolted assembly will not be affected by any additional moment load by virtue of the interface design which is more like an ASME Appendix 2 assembly. The central bolt has only 6.3 kN each since the inner cold cylinder shares the load. A drastic increase in this service load however would occur on some bolts when the 110-ton EM calorimeter is loaded, and furthermore, when there will be 60-tons of liquid argon.

Thus, the effective service loads on each bolt due to 3.5 bar of pressure alone are:

$F_p := \begin{bmatrix} 30 \\ 17 \\ 6.3 \end{bmatrix} \cdot kN$ **< -- Service Load per bolt**

VI. OMEGA SEAL (Original)

Let R_w = mean radius of the omega seal

$$R_\omega := \begin{bmatrix} 2197 \\ 1385.25 \\ 0 \end{bmatrix} mm$$

From the test data (attached):

$F_t := (1.2 \cdot 9.8) \cdot kN$ $\quad d_t := 300 \cdot \mu m$ $\quad L_t := 50 \cdot mm$

where, F_t and d_t are the force and deflection on a 50 mm. long test piece.
Hence, the omega seal stiffness, K_t, in kN per mm deflection per mm. length is:

$$K_t := \frac{F_t}{d_t \cdot L_t} \qquad \text{or,} \qquad K_t = 0.784 \circ \frac{kN}{mm \cdot mm}$$

Although the omega seal is designed nominally for a 0.2 mm compression, fabrication tolerances allow it to have a max. deflection, δ = 0.4 mm. Assuming this worst case, the additional forces required from each bolt due to the omega seal would be:

So, for $\delta := 0.4\ mm$,

$$F_\omega := \overrightarrow{\left[(2 \cdot \pi \cdot R_\omega) \cdot \frac{K_t \cdot \delta}{N_b} \right]} \qquad \text{or,} \qquad F_\omega = \begin{bmatrix} 21 \\ 32 \\ 0 \end{bmatrix} \circ kN \qquad K_t = 1.137 \cdot 10^5 \circ psi$$

It follows that the Effective Applied Load per bolt is:

$$W_e := F_\omega + F_p \qquad \text{or,} \qquad W_e = \begin{bmatrix} 51 \\ 49 \\ 6 \end{bmatrix} \circ kN$$

, per bolt

VII. OMEGA SEAL (NEW DESIGN)

From the test data (attached) of the new omega seal design:

$F_{t1} := 10 \cdot kN$ $\quad d_{t1} := 400 \cdot \mu m$ $\quad L_{t1} := 50 \cdot mm$

$$K_{t1} := \frac{F_{t1}}{d_{t1} \cdot L_{t1}} \qquad \text{or,} \qquad K_{t1} = 0.5 \circ \frac{kN}{mm \cdot mm} \qquad K_{t1} = 7.252 \cdot 10^4 \circ \frac{lbf}{in \cdot in}$$

$$F_{\omega 1} := \overrightarrow{\left[(2 \cdot \pi \cdot R_\omega) \cdot \frac{K_{t1} \cdot \delta}{N_b} \right]} \qquad \text{or,} \qquad F_{\omega 1} = \begin{bmatrix} 13.3 \\ 20.7 \\ 0 \end{bmatrix} \circ kN$$

, per bolt

Effective Applied load per bolt:

$$W_{e1} := F_{\omega 1} + F_p \qquad \text{or,} \qquad W_{e1} = \begin{bmatrix} 43 \\ 38 \\ 6 \end{bmatrix} \circ kN$$

, per bolt

$$\overrightarrow{\frac{W_e - W_{e1}}{W_e}} = \begin{bmatrix} 15 \\ 24 \\ 0 \end{bmatrix} \circ \%$$

VIII. BOLT RESULTANT LOAD ("TEXTBOOK-BASED" SOLUTION)

Finally, by applying Eq. 3, we can determine the resultant bolt loads:

$$W := W_i + \overrightarrow{\left(r \cdot W_e\right)} \qquad W = \begin{bmatrix} 62 \\ 64 \\ 80 \end{bmatrix} \cdot kN$$

< --Resultant Load with Original Omega Seal

$$W_1 := W_i + \overrightarrow{\left(r \cdot W_{e1}\right)} \qquad W_1 = \begin{bmatrix} 61 \\ 60 \\ 80 \end{bmatrix} \cdot kN$$

< --Resultant Load with New Omega Seal

In practical terms, the new omega seal design doesn't really change much the resultant bolt load despite a decrease in the effective applied load. In fact, without any omega seal at all, the resultant bolt loads are only slightly less, as can be seen below, where F_p is the effective applied force due to pressure alone since $F\omega$ is zero.

$$W_0 := W_i + \overrightarrow{\left(r \cdot F_p\right)} \qquad W_0 = \begin{bmatrix} 57 \\ 55 \\ 80 \end{bmatrix} \cdot kN$$

< --Resultant Load without Omega Seal

This "textbook- based" calculation is corroborated by the Finite Element results below.

IX. "TEXTBOOK-BASED" SOLUTION VS. FEA WITH ANSYS

An axi-symmetric FE model using Ansys (rev. 4) was developed in order to verify the textbook-based calculation above. The bolt elements were given an initial strain corresponding to the initial pre-stress, S_i, and then an internal pressure of 3.5 bar (the specified test pressure of the cryostat) was applied.

Here's a summary of the Ansys results (Please see also Appendix B for more details).

1. Preload Only (With original design of Omega seal):

(W$_{ia}$ is the Bolt Preload)

$$W_{ia} := \begin{bmatrix} \dfrac{0.10428 \cdot 10^8}{N_{b_{0,0}}} \\ \dfrac{0.42154 \cdot 10^7}{N_{b_{1,0}}} \\ \dfrac{0.13497 \cdot 10^8}{N_{b_{2,0}}} \end{bmatrix} \cdot N \qquad \text{, or} \qquad W_{ia} = \begin{bmatrix} 50 \\ 50 \\ 78 \end{bmatrix} \cdot kN$$

2. Preload and 3.5 bar Pressure (With original design of Omega seal):

(W_{rap} is the Bolt Resultant Force)

$$W_{rap} := \begin{bmatrix} \dfrac{0.13034 \cdot 10^{8}}{N_{b_{0,0}}} \\ \dfrac{0.53618 \cdot 10^{7}}{N_{b_{1,0}}} \\ \dfrac{0.12978 \cdot 10^{8}}{N_{b_{2,0}}} \end{bmatrix} \cdot N \qquad \text{, or} \qquad W_{rap} = \begin{bmatrix} 63 \\ 64 \\ 75 \end{bmatrix} \circ kN$$

3. Preload and 3.5 bar Pressure (No Omega seal):

(W_{rns} is the Bolt Resultant Force, no omega seal)

$$W_{rns} := \begin{bmatrix} \dfrac{0.11529 \cdot 10^{8}}{N_{b_{0,0}}} \\ \dfrac{0.46036 \cdot 10^{7}}{N_{b_{1,0}}} \\ \dfrac{0.12979 \cdot 10^{8}}{N_{b_{2,0}}} \end{bmatrix} \cdot N \qquad \text{, or} \qquad W_{rns} = \begin{bmatrix} 55 \\ 55 \\ 75 \end{bmatrix} \circ kN$$

X. GRAPHICAL REPRESENTATION

Let us construct a graphical representation using the parameters described above.

1. OUTER BOLTS:

The following functions of W_e are plotted below:

$W_i := 50$ < -- Pre-Load

$W_o := 27$ < --Original design Pre-Load

$W_e := 0, 10 .. 150$ < -- Range variable, horizontal axis

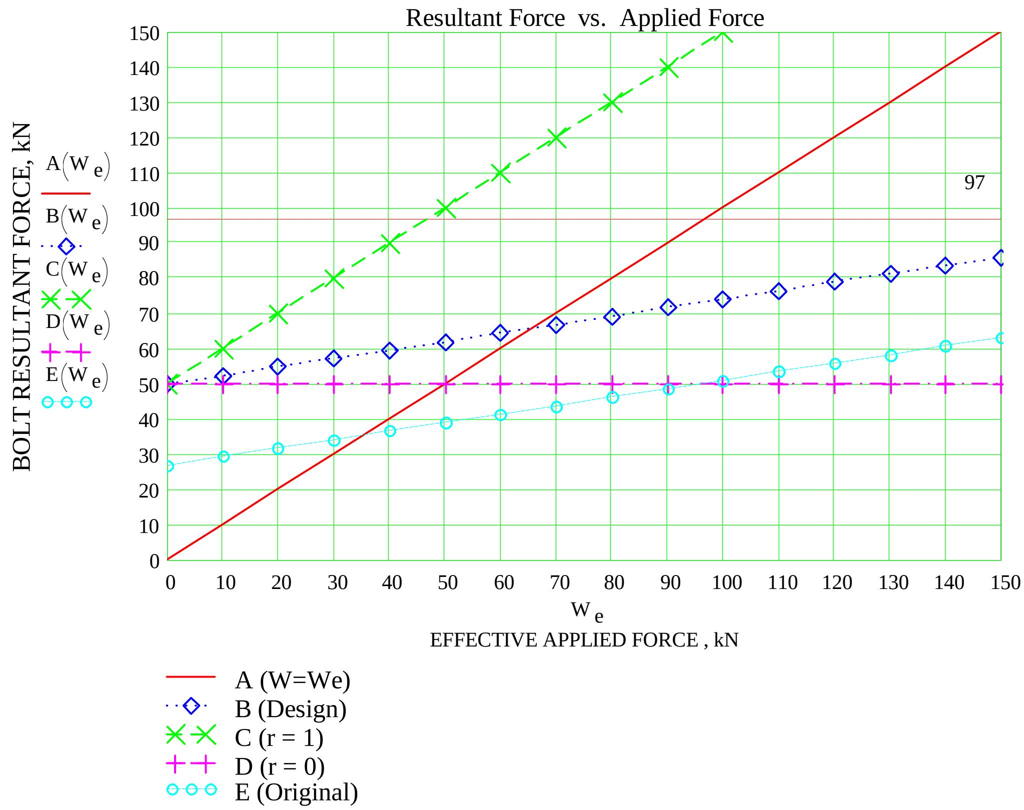
$A(W_e) := W_e$ < -- Line A, "Separation" line, 45° slope

$B(W_e) := W_i + r_{0,0} \cdot W_e$ < -- Line B, Load line, current design

$C(W_e) := W_i + W_e$ < -- Line C, Load line slope, r = 1

$D(W_e) := W_i$ < -- Line D, Load line slope, r = 0

$E(W_e) := W_o + r_{0,0} \cdot W_e$ < -- Line E, Load line with original design preload



1. INNER BOLTS:

$W_i := 50$ < -- Pre-Load

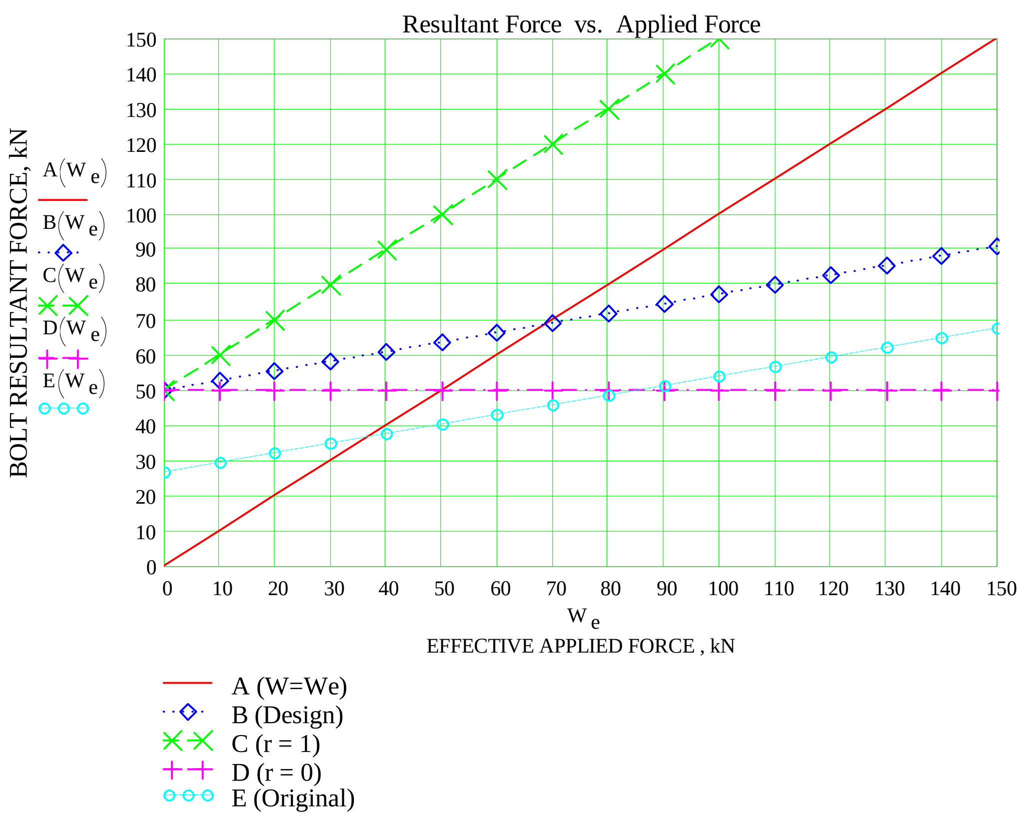
$W_o := 27$ < --Original design Pre-Load

$W_e := 0, 10 .. 150$ < -- Range variable, horizontal axis

$B(W_e) := W_i + r_{1,0} \cdot W_e$ < -- Line B, Load line, current design

$E(W_e) := W_o + r_{1,0} \cdot W_e$ < -- Line E, Load line with original design preload

(All other lines are the same as the outer bolts plot)



3. CENTRAL BOLTS:

$W_i := 78$ < -- Pre-Load

$W_o := 27$ < --Original design Pre-Load

$W_e := 0, 10 .. 160$ < -- Range variable, horizontal axis

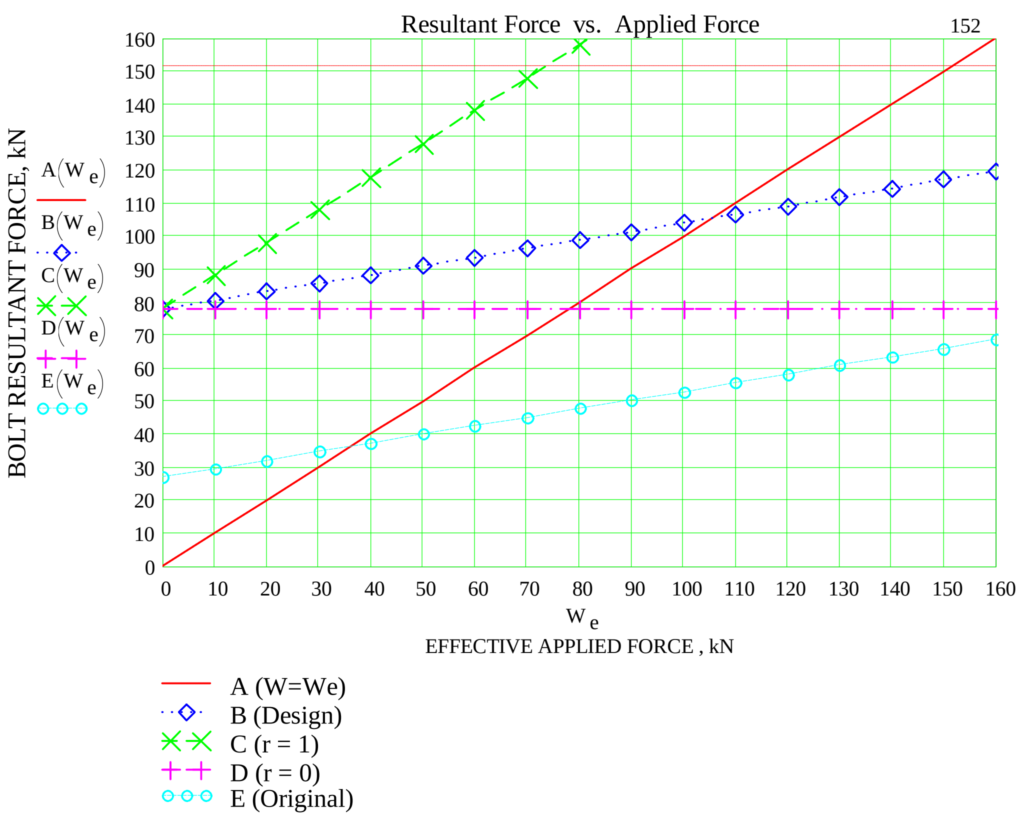
$A(W_e) := W_e$ < -- Line A, "Separation" line, 45º slope

$B(W_e) := W_i + r_{2,0} \cdot W_e$ < -- Line B, Load line, current design

$C(W_e) := W_i + W_e$ < -- Line C, Load line slope, r = 1

$D(W_e) := W_i$ < -- Line D, Load line slope, r = 0

$E(W_e) := W_o + r_{2,0} \cdot W_e$ < -- Line E, Load line with original design preload



XI. FURTHER DISCUSSION

Particular care should be given in applying the assembly torque as calculated previously. Considering the number of bolts involved, a broad range of variation in the actual clamping force can be expected. For instance, the coeffecient of friction for a steel bolt and nut under dry condition can be between 0.15 and 0.25. The assembly torque that was calculated earlier in this report was based on a coeff. of friction of 0.15. If a thread lubricant is used such as silicon grease, or molybdenum disulfide, the friction coef. may become less than half of the 0.15 that was initially assumed; In such a case, the yield strength of the bolt may well be exceeded if the same torque is applied.

There is, of course, an advantage in applying a higher preload as can be seen clearly in the plots above. In the construction industry, direct tension indicating washers are normally guaranteed to achieve a clamping force of 15% above the specified nominal value. But, in our case, we have to be careful that we won't end up with a lot of broken bolts, especially that the mating flanges have blind tapped holes; It will be a hassle to remove broken bolts from these blind tapped holes.

With respect to the central bolts, it would seem that the specified preload is much too high than what is required. But as mentioned earlier, some of the central bolts, especially those near the calorimeter rail region, will see additional loads due mostly to the massive weight of the EM calorimeter, as well as the weight of liquid argon; The central bolt preload is designed to handle those additionlal forces.

Finally, just a quick comment about embedment. This is the term given to local yielding under the bolt head, or under the washer due to high pre-loading. This is the manifestation of a high P/A, where A is the area under the bolt head (or, washer), not the *effective flange area* as described earlier in this report. This is a local deformation and is quite common and normal in most bolted joints; when the required preload is very high, hardened washers are always preferable.

In our case, we will probably see some embedment between the bolt head and the Invar washer, or between the Invar and the aluminum. But I don't think it will be significant enough to affect the results of the our calculations.

XII. REFERENCES

(1) Machine Design Calculations Reference Guide, p. 115, edited byTyler Hicks
McGraw Hill, 1985: "Selecting Bolt Diameter For Bolted Pressurized Joint"

(2) Practical Stress Analysis In Engineering Design, by Alexander Blake, 1982

(3) An Introduction to the Design and Behavior of Bolted Joints by John H. Bickford

(4) Axisymmetric Static Analysis: Cold Vessel Bolted Flanges, Rev. 1 at
http://www.collider.bnl.gov/cv-axi-rev1.html

XIII. ATTACHED FIGURES

Fig. A: Omega Seal (Original) Test Data:

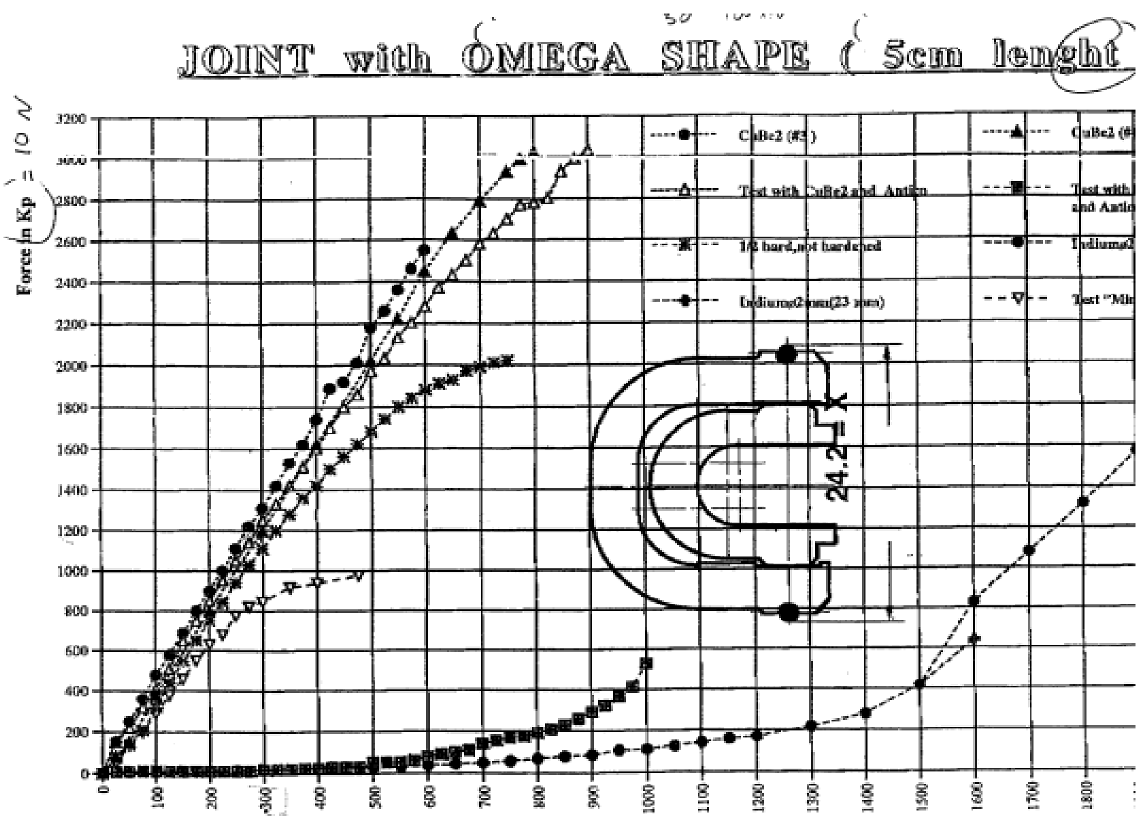


Fig. B: Omega Seal (New) Test Data: