Analysis of OH Bolted Ear Connection

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Introduction

The D0 endcap calorimeter outer hadronic (OH) modules play a major structural role in the calorimeter assembly. The discrete modules, once connected together, form a ring within which other massive calorimetry will reside. It has been proposed that the connection of the OH at the downstream end be accomplished by extending the downstream endplates in the radial direction to form "ears", and then through-bolting between adjacent ears as shown in Fig. 1. A single 2 1/4 in. dia. bolt is used, and previous calculations have determined that the design load on this joint should be 130000 lbs tension. The high load and serious consequences of failure make this a critical component in the calorimeter assembly. The purpose of this analysis is to investigate the stresses in the connection and other mechanical characteristics which determine joint performance.

Joint Design

The joint was designed based on the requirement that the bolting material be a material similar to that out of which the downstream calorimeter endplate is made, thus eliminating thermal contraction considerations. The bearing stress under preload was limited to $0.9S_y$, where S_y is the yield strength of the plate material. In addition to these requirements the plate thickness is also influenced by the stresses developed by external loads. The geometry of the joint is shown in Fig. 2.

A approximate hand calculation of joint performance is included in Appendix A.

Analysis

A two dimensional finite element model was made to examine the performance of the joint. This model, shown in Fig. 3, will estimate joint behaviour in the plane of the plate with sufficient accuracy to judge the viability of the design. The plate was assumed to be 4.33 in thick. Only half of the plate was modeled, and symmetry conditions were imposed at the bolted interface. The load was applied as shown in Fig. 3, with distributed nodal forces at the plate centerline which, in conjunction with the pivot constraint at the inner radius, provide a reaction at the joint of 130000 lbs. The compressive area between the ears was modeled with interface elements which could withstand compression only. The bolt head was coupled with the washer surface for movement in the direction normal to the surfaces; The same coupling was used to connect the washer surface with the ear surface. This coupling allowed no shear to develop between the compressed members, which is a conservative way to model this type of contact.

All components modeled must be represented by rectangular blocks 4.33 in thick. This requires that the appropriate stiffness be achieved by adjusting material moduli and, in the case of the bolt, in-plane dimensions. The bolt hole was represented by a portion of material with modulus reduced according to the proportion of total thickness occupied by the hole in the actual plate. The bolt has two stiffnesses which are important to this work, namely the axial stiffness and the in-plane bending stiffness. In order to correctly approximate these, both the in-plane height and the modulus of the bolt were adjusted such that both stiffnesses were achieved. The values used are shown in Fig. 3.

The joint preload was established by applying a temperature difference to the bolt material, which, through its nodal coupling with the washer, transmitted the contraction forces into compression along the interface. The preloading forces were calculated from the desired bolt stress based on a bolt stress area of 3.56 in^2

Results

Three preloading forces were applied. These forces corresponded to bolt stresses of 30, 60, and 90 ksi. The bolt and ear stresses for the 60 ksi preload case were looked at in detail, since these correspond most closely to appropriate preloads for the class of bolting materials being considered. The ear stresses at the minimum ear section do not vary significantly with the bolt preload, since all of the external load must pass through this section to reach the bolted connection.

The ear stresses were linearized across the narrow ear section and compared with the stress limits specified by the AISC Specification for the Design, Fabrication and Erection of Structural Steel for Buildings. These limits are: Maximum membrane fiber stress less than or equal to $0.6S_y$; Maximum combined membrane and bending fiber stress less than or equal to $0.66S_y$; Maximum nominal shear stress less than or equal to $0.4S_y$. S_y is the minimum specified yield strength for the ear material, which for SA-240 SS304 is 30 ksi. The stresses in the ear are given in Table I, and can be seen to be below the maximum allowable values

The bolt stresses were also linearized across the bolt for the 60 ksi preload case. The only stress of significance is the normal stress in the bolt. These stresses are given in Table II. No comment can be made on suitability as no steel has been specified for this application.

The effects of bolt preload were investigated by extracting for each of the three runs the amount of external load carried by the bolt and the amount carried by relief of compression in the ears. This information is summarized in Table III.

Discussion

The stresses in the SS304 ear as calculated by the finite element model are less than the allowable stresses for this material. There is a stress concentration at the fillet of the ear, but this is very highly localized and is of no significance to this static load case. A slightly smaller shear area is used in the finite element analysis than was used in the calculation in the appendix, because the slant of the plate is not modeled. The resulting shear stress is therefore larger than that calculated in the appendix.

The bending stress calculated by the finite element model is much smaller than that calculated in the appendix. The calculation in the appendix assumed that the 130000 lb external force was acting through the centerline of the bolt. The finite element model shows that due to deformation at the joint, the line of force is actually much closer to the minimum ear section, as so produces a much smaller moment than predicted by the simpler calculation.

The calculated stiffness distribution of this joint predicts that 20% of the external load will be carried by the bolt, and 80% by the members based on the ratio of their individual stiffness to the total joint stiffness. The finite element model shows that this is achieved for preloads in the range of 90 ksi, where the bolt carries 18% of the external load. In the range of 60 ksi preload stress, the bolt carries 35% of the external load. Lower preloads increase the bolt load further. A conservative design must take into account this variation of load carrying in determining the proper bolt material and preload stress.

**Table I. Stresses in Ear
at Minimum Section**

Stress Catagory	Stress	Allowable Stress
Membrane	0.0 psi	18000 psi
Combined Membrane + Bending	6780	20100
Shear	7016	12000

**Table II. Stresses in Bolt
at 60 ksi Preload Stress**

Stress Catagory	Stress	Allowable Stress
Membrane	72640 psi	Unspecified
Combined Membrane+ Bending(max)	95000	Unspecified

**Table III. External Load Sharing
of Bolt and Member**

Preload Stress	External Force on Bolt	External Force on Members
30 ksi	78500 lbs	51500 lbs
60	45000	85000
90	23800	106200

Fig 1. Downstream OH Ear Connection

Fig 2. Geometry of OH Downstream Bolted Connection

Moduli

E_{2r} : EX = 28.3(10⁶)
EY = 28.3(10⁶)

Bolt Hole
EX = 14.5(10⁶)
EY = 14.5(10⁶)

Bolt
EX = 14.15(10⁶)
EY = 1000

ANSYS 4.3
DEC 30 1987
14:28:59
PLOT NO. 1
POST1 ELEMENTS

ORIG
ZV = 1
DIST = 17.4
XF = 8.5
YF = 13.4

Fig 3. Finite Element Model

Approximate OH Ear/Bolt Analysis

Find stiffness of joint

$$\text{Bolt stiffness} = K_b = \frac{AE}{L} = \frac{\pi (2.25)^2 (29)(10^6)}{4 (2)(4.3+0.5)} = 1.2 (10^7) \text{ lbs/in}$$

Member stiffness

Model as a beam of length $2(4.3+0.5)$ and cross section as shown above

$$K_m = \frac{AE}{L} = \left[\left(1 - \frac{6(4)}{4} \right) - \frac{\pi (2.32)^2}{4} \right] \frac{29(10^6)}{2(4.3+0.5)}$$

$$K_m = 4.28 (10^7) \text{ lbs/in}$$

$$\boxed{\text{Total stiffness of joint} = 1.2(10^7) + 4.28(10^7) = 5.48(10^7) \text{ lbs/in}}$$

Deflection of joint under design load of 130000 lbs

$$\delta = \frac{30000 \text{ lbs (in)}}{5.48 (10^7) \text{ lbs}} = 0.00237''$$

This must equal or be a member compression if joint is to remain in contact. So minimum preload is

$$F_p = 4.28 (10^7) (0.00237) = 101436 \text{ lbs}$$

Resulting bolt stress is

$$\sigma_b = \frac{F_p}{A_{\text{stress}}} = \frac{101436}{3.56} = 28493 \text{ psi}$$

At separation,

$$\sigma_b = \frac{130000}{3.56} = 36517 \text{ psi}$$

Minimum bolt strength requirements (using ASME approach) is

$$\sigma_y = \frac{40044}{0.66} = 60673 \text{ psi} \quad (\sigma_u > 120132)$$

or

$$\sigma_u = 3(40044) = 120132 \text{ psi} \quad (\sigma_y > 60673)$$

Ear Stresses

For section A-A,

$$Area = 4.3(4.6) = 19.78 \text{ in}^2$$

$$I = \frac{4.6(4.3)^3}{12} = 30.4 \text{ in}^4$$

$$\text{Nominal shear} = \frac{130000}{19.78} = 6572 \text{ psi}$$

$$\text{Nominal bending} = \frac{130000(2)(2.15)}{30.4} = 18388 \text{ psi}$$

The shear and bending need not be combined since they don't occur in the same place. (Shear is max at B, 0 at C, bending is max at C, 0 at B)

This ear is safe for design loads. Bending stress calculation is conservative, since overlapping with adjacent ear is assumed.