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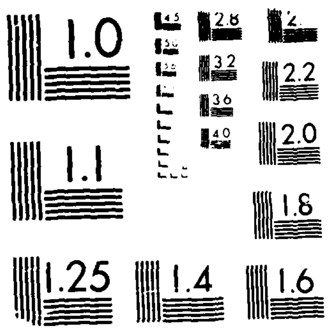

MICROCOPY RESOLUTION TEST CHART

TN 2036-1310 NR/01

FEB STRUCTUAL ANALYSIS

Rev. H

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by

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FEB STRUCTURAL ANALYSIS

1. Scope

The purpose of this analysis is to document the expected simple loads on the cold plate through the tie down bolts around the perimeter of the FEB. Finite element analysis has been performed to determine the specific vibration modes and natural frequencies. The simple static analysis is also repeated when one bolt is missing. Flange analysis of the baseplate is performed as well as analysis of the attachment and cross bracing of the internal modules.

2. Relevant Documents

2.1 MSFC-JA-418

2.2 SPAH-Appendix B

2.3 FEB Configuration Drawing 2036-131001.00A

2.4 MSFC-JA-595, ATLAS 1, Integrated Payload Requirements Document, Feb 1987

2.5 Machine Design Calculations Reference Guide, Tyler G. Hicks, Editor, McGraw-Hill

2.6 Fastener Preload, Machine Design, Nov. 13, 1986

2.7 An Introduction to the *Design and* Behavior of Bolted Joints, by John H. Bickford, Marcel Dekker, 1981.

2.8 MIL-HDBK-5D

2.9 STRUCTURAL SAFETY VERIFICATION PLAN, PL 2036-1000 DS/01, Issue D

2.10 Structural Analysis for MAS-Sensor Package, TN 2036-1000 DS/09

3. Assumptions and input conditions

3.1 Rigidity of bottom plate

The underlying coldplate is assumed "soft" in that the net moments are assumed centered around the center of gravity of the FEB and the coldplate does not force any specific moment axis. Since all perimeter bolts on the 700 mm x 420 mm grid are used, 32 tie down bolts are used. For the analysis of the distribution of quasi-static loads on the tie down bolts, the FEB bottom plate will be assumed to be perfectly rigid around its perimeter. The thick-walled outer cover attaches to the FEB with the same bolt pattern, which contributes to the rigidity of the outer perimeter of the base plate.

3.2 Center of Gravity

As various subsections of the FEB have been received, minor modifications of the internal x-y layout of the subsections on the base plate have been necessary to maintain the center of gravity in the x-y plane at the center of the coldplate. This analysis uses the latest estimates of these module positions. The center of gravity along the z axis is 135 mm above the x-y plane. The mass of the FEB is assummed to be 60.8 kg.

3.3 Thermal load

The underlying coldplate support structure is expected to be machined from 6061-T6 aluminum as is the FEB bottom plate, so lateral x-y thermally induced shear stresses are not a problem. Thermally induced axial stresses in the tie down bolts during landing are considered however. Such analysis typically considers a range from room temperature (70°F) during assembly to 150°F during landing. For this analysis we will start from the 20° C temperature to a landing temperature cargo bay temperature of 66°C, for a change of temperature of 46°C.

It is expected that 10-32 English thread, corrosion resistant steel (A286) tie down bolts will be used around the perimeter of the baseplate. Standard handbooks indicate a stress area of 0.020 in^2 for this size screw.

3.4 Quasi-static/vibrational load factors

The FEB mounts on the MSFC orthogrid so the expected quasi-static load factors are:

	X	Y	Z
LIFT-OFF	-3.0/+6.0	-4.0/+4.0	-2.7/+2.7
LANDING	-10.0/+10.0	-2.4/+2.4	-5.9/+2.8

From JA-595 for experiments mounted on the orthogrid coldplate support structure, the random vibration input to the FEB depends on the axis. On the axis normal to the orthogrid (Z-axis)

20 Hz -	93 Hz	0.006 g**2/Hz
93 Hz -	200 Hz	+9 dB/oct
200 Hz -	356 Hz	0.06 g**2/Hz
356 Hz -	2000 Hz	-9 dB/oct
-	2000 Hz	0.00034 g**2/Hz

For axis in the plane of the orthogrid, (X and Y axis)

20 Hz -	82 Hz	0.006 g**2/Hz
82 Hz -	150 Hz	+6 dB/oct
150 Hz -	350 Hz	0.02 g**2/Hz
350 Hz -	2000 Hz	-7 db/oct
above	2000 Hz	0.00034 g**2/Hz

4. Overall Analysis Results

4.1 Vibrational analysis

As shown in the FEB configuration drawing, the outer enclosure consists of five ribbed plates which are assembled into a rectangular box. Preliminary analysis of the vibration modes of these plates yielded results which suggested the need for ribs. As can be seen in the drawing, the top cover has the largest x-y physical dimensions and the widest rib spacing, which produces the greatest vibrational compliance. FFT analysis of the forced impulse response of the completed top cover (clamped to the sides) has found a lowest resonant frequency of 1074 Hz. The prime resonances were observed to be at 1562 Hz and 1782 Hz. The side walls and end walls are physicaly smaller and have narrower rib spacing so their resonant frequencies will be higher.

A finite element analysis has also been performed for the vibrational modes of the bottom mounting plate, using additional out-of-plane elements to represent internal substructures. With the bottom plate simply supported (ignoring any additional stiffness produced by the coldplate support structure or the side covers), the lowest natural resonant frequency in each direction has been found. Animation of the resulting deflections has identified the following vibration modes. For the first frequency of 216 Hz the filter modules (the main mass elements) are moving in the Y direction as shown in the first figure of the appendix. At the next eigen-frequency, 269 Hz, the filter modules are moving in the z direction, as shown in the second figure. For the third eigen-frequency, 341 Hz, the filters are moving in the X direction while the synthesizers are moving in the Y direction.

4.2 Random Vibration Load

At the X and Y axis eigen-frequencies, the Power Spectral Density is 0.02 g**2/Hz. At the Z axis eigen-frequecny, the Power Spectral Denisty is .06 g**2/Hz, using the "normal to the orthogrid" values from JA-595

The random vibration load factor (RV) is then calculated by

$$RV = +/- \ 3 * \left(\frac{PI}{2} * f_n * Q * psd\right)^{1/2}$$

where the amplifcation factor, Q, depends on the damping and is assumed to be 10

	X	Y	Z
RV =	+/- 31.1 g	+/- 24.7 g	+/- 47.8 g

During lift off the quasi-static load and random vibration load terms add to produce the following combined load factors. The random load term is added one axis at a time as indicated in JA-418, section 5.1.2.3

LOAD CASE	XG	YG	ZG	
1	-34.1/+37.1	-4.0/+4.0	-2.7/+2.7	
2	-3.0/+6.0	-28.7/+28.7	-2.7/+2.7	
3	-3.0/+6.0	-4.0/+4.0	-50.5/+50.5	
4	-10.0/+10.0	-2.4/+2.4	-5.9/+2.8	(Landing)

The maximum worst case combination of accelerations for each load case will be used.

For the following analysis, load case #1 will be used to illustrate the calculations. The margin of safety will then be summarized for all load cases.

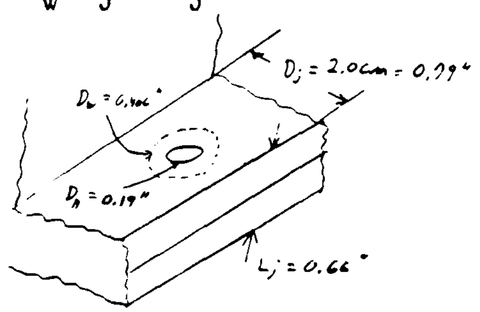
4.3 Thermal load

The thermal expansion coefficient for Al is 2.36E-05 and for stainless steel the thermal expansion coefficient is 1.6E-05. The worst case temperature excursion (room temperature assembly to cargo bay landing) is 46°C. The net expected axial strain is

$$e_{net} = (2.36E{-}05 - 1.6E{-}05) * 46 = 3.5E{-}04.$$

This strain divides between the aluminum flange and the steel bolt according to the relative cross section and E-modulus for each. The bearing surface under the bolt head is due to the flat washer with an outside diameter of 0.406 inch. The effective flange area can be calculated with the design equations of reference 2.5 (Case 2--page 117)

$$A_{FL} = PI * (D_W^2 - D_H^2) / 4 + PI * (D_J/D_W - 1)*(D_W*L_J/5+L_J^2/100)/8$$



$$= 3.14 * (0.406^2 - 0.19^2) / 4 + 3.14 * (0.79/0.406 - 1)*(0.406 * 0.66/5 + 0.66^2/100) / 8$$

$$= 0.115 \text{ in}^2 = 74.4 \text{ mm}^2$$

While for the bolt, a handbook shows a stress cross section for a 10-32 screw of

$$A_B = 0.020 \text{ in}^2 = 12.9 \text{ mm}^2$$

Also,

$$E_{FL} = 1.0E+07 \text{ psi} = 68.9 \text{ GPa}$$

and

$$E_B = 2.91E+07 \text{ psi} = 201 \text{ GPa}.$$

The net strain then divides between the flange and bolt according to

$$e_{BOLT} = e_{net} \frac{A_{FL} * E_{FL}}{A_B * E_B + A_{FL} * E_{FL}}$$

$$e_{BOLT} = 2.32E-04.$$

This bolt strain corresponds to a bolt axial load of

$$2.32E-04 * 201E+09 * 1.29E-05 = 602 \text{ N}$$

4.4 Coldplate mounting bolt preload

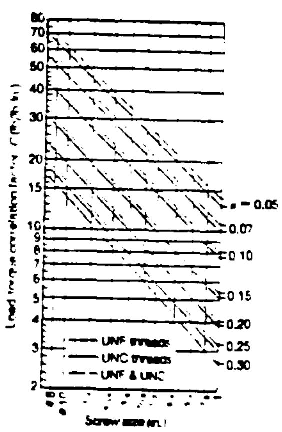
Each bolt will typically be torqued between 2.7 N-m and 3.1 N-m (section 4.3.1.3 of the SPAH Appendix B). The minimum cross section diameter of a 10-32 screw is

0.1517 inches = 3.85 mm.

From standard handbooks, the expected maximum preload on each bolt is then

$$F_{PRELOAD} = 5 * T / D_{min}$$
$$= 5 * 3.1 / 3.85 E-03 = 4026 \text{ N}$$

A more precise calculation considers the minimum expected coefficient of friction with the maximum torque. For a 10-32 (0.19) screw, and a coefficient of friction of 0.0784, an expression for the preload can be found from the figure of reference 2.6



Predicting initial bolt load

Graph shows the correlation between load and torque. An estimate of preload on nuts, bolts, and socket screws with UNF or UNC threads is provided by the graph. The graph accounts for the effect of thread helix angle, but assumes that the coefficients of friction between mating threads are the same as that between abutment and fastener head.

To use the graph, select a nominal bolt size on the horizontal axis and read vertically to the curve that represents the correct friction coefficient and thread type. Then read horizontally to find correlation factor C. To estimate preload multiply C by the torque applied to the fastener.

$$F_{maxPRE} = 1748 * T = 1748 * 3.1 = 5419 \text{ N}$$

The same figure can be used to find the relation between torque and preload for a maximum expected coefficient of friction of 0.1323. After conversion to metric units, the minimum preload is found to be

$$F_{minPRE} = 1134 * T = 1134 * 2.7 = 3061 \text{ N}$$

4.5 Lift off produced loading, all bolts present

As descibed in section 4.2, four different load cases will be considered. Load case #1 will be used to illustrate the calculations and all load case margins of safety will be summarized

All 32 bolts share the load due to Z-axis acceleration, the net axial loading per bolt is

$$F_{zz} = \text{mass} * \text{acg} * ZG / N_{bolts}$$

$$F_{zz} = 60.8 \text{ kg} * 9.81 * 2.7 / 32 = 50.3 \text{ N}.$$

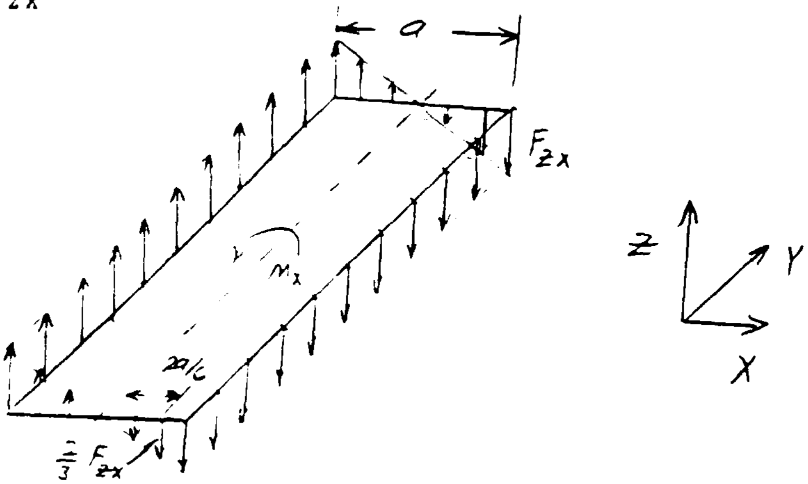
The global X-axis is along the short axis of the FEB so the conversion of X-axis acceleration loads through the CG above the mounting plate is through a short lever arm. The moment produced by the X-axis acceleration is

$$M_x = \text{mass} * \text{acg} * XG * \text{length from c.g.}$$

$$M_x = 60.8 \text{ kg} * 9.81 * 37.1 * 0.135 \text{ m} = 2985 \text{ Nm}.$$

This is counterbalanced by the bolts in tension on one side and the compression against the cold plate on the other side.

$$M_x = 2 * F_{zx} * [11 \text{ bolts} * a/2 + 2 \text{ sides} *(1/3 * a/6 + 2/3 * 2a/6)]$$



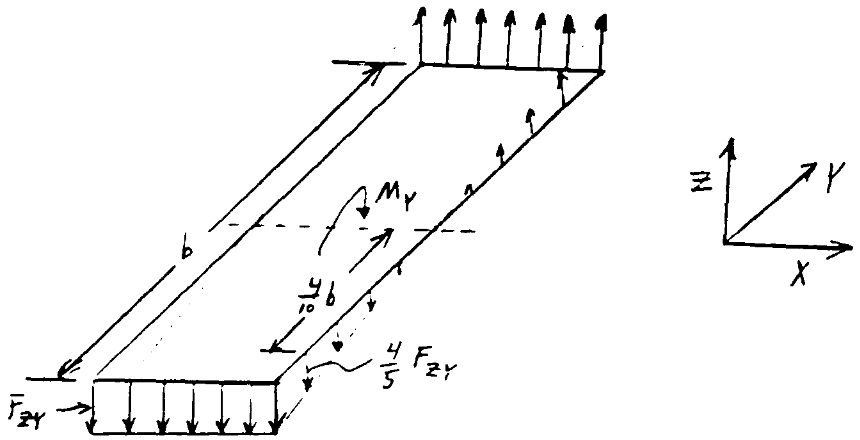
where a = length of the short side of the FEB bolt mounting centers = 0.4 m

$$F_{zx} = 2985 / 4.84 = 616 \text{ N}$$

The global Y-axis is along the long axis of the FEB. The Y-axis acceleration induced moment is

$$M_y = 60.8\ kg * 9.81 * 4.0 * 0.135\ m = 322\ Nm.$$

This is counterbalanced by the bolts in tension along one side and compression against the cold plate on the other side



$$M_y = 2 * F_{zy} * [7\ bolts * b/2 + 2\ sides * b\ (4/5 * 4/10$$
$$+ 3/5 * 3/10 + 2/5 * 2/10 + 1/5 * 1/10)]$$
$$= 2 * F_{zy} * b * 4.7.$$

Where b = 0.7 meter, so

$$F_{zy} = 48.9\ N.$$

Adding the three components (F_{zz}, F_{zy}, F_{zx}) produces a maximum loading on the one corner bolt of

$$F_{zmax} = 715\ N.$$

From a calculation similar to that of section 4.4, the minimum torque preload can be expected to be 3061 N. Thus the bolt will remain in tension

4.6 Lift off produced loading, one bolt missing

If one of the corner bolts fails, the total load redistributes slightly. An analysis similar to that of section 4.5 shows that the next nearest bolt on the short axis has a loading of 554 N. The nearest bolt on the long axis has a loading of 767 N. The corner bolt opposite the missing one has a loading of 717 N. The remaining corner bolt on the short axis has a loading of 722 N And the final corner bolt on the long side has a loading of 772 N.

4.7 Landing produced loading, all bolts present

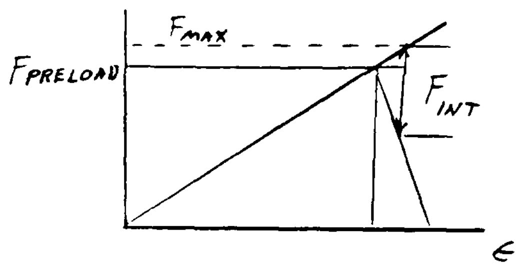
During landing, random vibration loading is not present, but the quasistatic loads are higher. Also the thermally induced load (602 N, see section 4.3) must be added to the initial bolt preload. Using the landing load factors and an analysis similar to that of the above sections the worst case bolt loading due to landing vibration is 306 N.

4.8 Landing produced loading, one bolt missing

If one corner bolt fails, the landing loads also redistribute slightly. Further similar analysis shows that the corner bolt along the same long side suffers the highest loading due to landing vibrations of 324 N.

4.9 Mounting bolt margin of safety

The maximum loading on a mounting bolt is 715 N. Part of this interface load is absorbed by the flange and part is absorbed by the bolt. If all of this interface force is assumed acting at the outer surface of the flange, then the external interface force divides between the flange and bolt according to a flange or "screw" diagram. The actual interface force will probably be effective part-way into the flange, but assuming the external force acts at the outer surface is the most conservative calculation.



In section 4.3 it was shown that the relative spring constants of the flange and bolt produces a relative division factor of 0.66. Thus the portion of the external load which is taken by the bolt is

$$F_{bolt} = 0.34 * 715 = 240 \text{ N}$$

This should be added to the maximum bolt pretension which occurs for the maximum torque and minimum friction

$$F_{boltmax} = 240 + 5419 = 5659 \text{ N}$$

For the A286 stainless steel bolts

$$\sigma_{su} = 91 \text{ E+03 psi} = 627 \text{ MPa (ultimate shear stress)}$$

$$\sigma_{tu} = 140 \text{ E+03 psi} = 965 \text{ MPa (ultimate tensile stress)}$$

Following the guidelines of Reference 2.9, the bolt pretension is first calculated with the minimum friction and maximum torque, as above. Then the margin of safety is calculated with the a safety factor of two on only the interface forces only.

$$MS_{tu} = \frac{\text{allowable load}}{2 * F_{bolt} + F_{pre}} - 1$$

$$MS_{tu} = (965 * 12.9)/(2 * 240 + 5419) - 1 = 1.11$$

When one bolt is missing, from the above earlier calculations the maximum interface load is 623 N. Then,

$$MS_{tu} = (965 * 12.9)/(2 * 0.34 * 772 + 5419) - 1 = 1.10.$$

Again following reference 2.9, the margin of safety should be also calculated using an average friction coefficient and with the application of a 1.4 safety factor to the bolt preload.

$$MS_{tu} = (965 * 12.9)/(2 * 0.34 * 715 + 4454) - 1 = 0.85$$

While for one bolt missing,

$$MS_{tu} = (965 * 12.9)/(2 * 0.34 * 772 + 1.4 * 4454) - 1 = 0.84$$

Gapping of the joint under minimum mounting torque and maximum friction must also be considered. From section 4.4 of this document, the minimum bolt preload is 3061 N. There will also be a relaxation of the initial bolt preload which is estimated to be 20%, which leaves a residual bolt preload of 2449 N. If the flange's fraction of the external load were to exceed the minimum preload, gapping would occur.

$$MS = 2449 / (2 * 0.66 * 715) - 1 = 1.6$$

The shear loading comes from the X and Y axis directed accelerations.

$$F_{sx} = \text{mass} * \text{accel} /32 \text{ bolts}$$

$$= 60.8 \text{ kg} * 9.81 * 37.1 /32 = 691 \text{ N}$$

$$F_{sy} = \text{mass} * \text{accel} /32 \text{ bolts}$$

$$= 60.8 \text{ kg} * 9.81 * 4 /32 = 74.5 \text{ N}$$

The maximum shear forces is thus

$$F_s = ((691)^2 + (74.5)^2)^{1/2} = 695 \text{ N}$$

The above calculated minimum clamping force will resist this shear force through the friction between the surface of the FEB and the coldplate. However the clamping force is not sufficient, so the shear forces must be taken by the bolts.

$$MS_s = (627 * 12.9) / (2 * 695) - 1 = 4.8$$

The bolt has both a tensile loading and a shear loading at the same ıme, following the guidelines of Reference 2.10, the corresponding factor of limit load capabilities" are first computed and then ombined.

$$FLC_t = MS_{tu} + 1 = 1.11 + 1 = 2.11$$

$$FLC_s = MS_s + 1 = 4.8 + 1 = 5.8$$

'hen,

$$FLC_{comb} = \frac{1}{[(1/FLC_t)^2 + (1/FLC_s)^2]^{1/2}}$$

$$FLC_{comb} = \frac{1}{[(1/2.11)^2 + (1/5.8)^2]^{1/2}} = 0.99$$

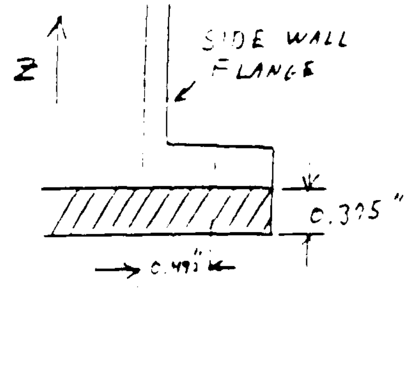
.oad case #4 (landing) considers the landing quasistatic loads and the hermal load calculated in section 4.3

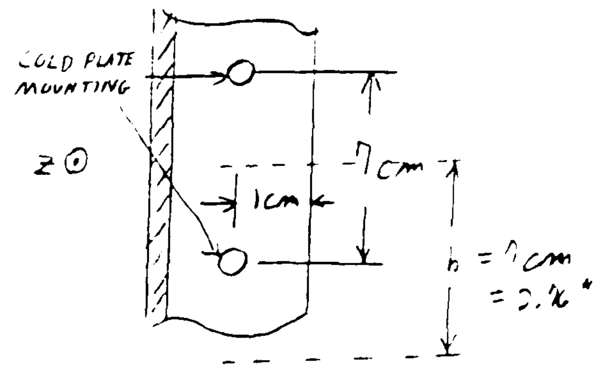
largin of Safety Summary for FEB-Coldplate mounting bolts

.OAD ;ASE	2.0 SF * INTERFACE ONLY ALL BOLTS	ONE MISS	2.0 SF * INTERFACE 1.4 SF * PRELOAD ALL BOLTS	ONE MISS	SHEAR MS	COMB MS	GAPPING MS
1	1.11	1.09	0.85	0.84	4.8	0.99	1.6
2	1.16	1.16	0.89	0.89	6.4	1.07	2.7
3	1.02	1.02	0.79	0.78	29.1	1.02	0.7
4	1.06	1.06	0.82	0.82	20.1	1.06	5.1

1.10 Baseplate mounting flange bending

For simplicity, the mounting flange is considered to consist of the 3/8 inch thick baseplate only. Additional strength supplied by the bottom flange of the outer sidewalls is not considered. For the bending analysis the flange is modeled as a simple beam in pure bending. This bending torque is produced only by the quasistatic and random vibration loading, not the bolt preload. Thus the maximum bending torque is produced by the maximum bolt loading of 715 N (161 lb)





$$M_o = 0.492 \text{ inch} * 161 \text{ lb} = 79.1 \text{ in-lb}$$

en the stress is

$$\sigma_{max} = 6 * M_o / b\ t^2$$

$$= 6 * 79.1 / (2.76 * (.375)^2)$$

$$= 1223 \text{ psi.}$$

r the 6061-T6 aluminum which is used for the sidewalls and baseplate
eference 2.8)

$$\sigma_{ty} = 35\ E{+}03 \text{ psi}$$

$$\sigma_{su} = 27\ E{+}03 \text{ psi}$$

$$\sigma_{tu} = 42\ E{+}03 \text{ psi.}$$

erefore the bending stress margin of safety is

$$MS_{bfbend} = \sigma_{tu} / (2 * \sigma_{max}) - 1$$

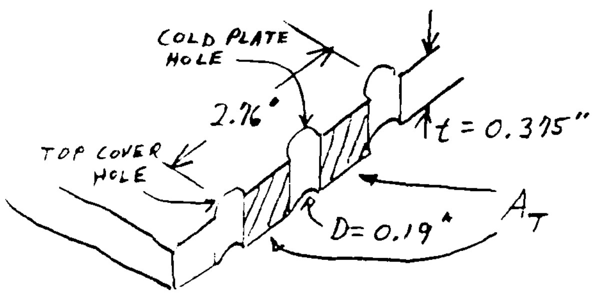
$$MS_{bfbend} = 42\ E{+}03 / (2 * 1223) - 1 = 16.2$$

ecking for yield, a safety factor of 1.25 is used.

$$MS_{ybfbend} = 35\ E{+}03 / (1.25 * 1223) - 1 = 21.9$$

11 Baseplate mounting flange tension failure

The flange geometry for tension failure is



ice again we consider only the thickness of the baseplate, neglecting
dewall flange thickness, for stress calculations. The appropriate
pression for tension tear out is

$$P_u = \sigma_{tu} * A_t$$

iere $A_t = (2 * R - 2 * D) * t$

$$= (2.76 - 2 * 0.19) * .375 = 0.89 \text{ in}^2$$

$$P_u = 42\ E{+}03 * 0.89 = 37.4\ E{+}03 \text{ lb}$$

The maximum shear load per bolt from section 4.9 is 695 N (155 lb),

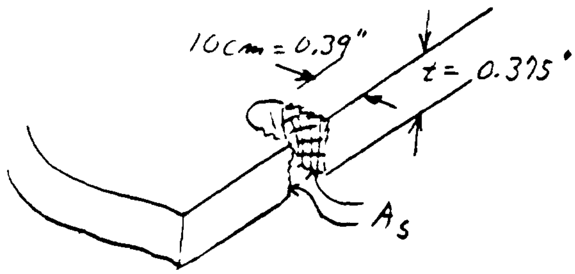
therefore $MS_{bft} = 37.4\ E{+}03\ /\ (2 * 155) - 1 = 120.$

Checking for yield by using a safety factor of 1.25,

$$MS_{ybft} = (35\ E{+}03 * 0.89)\ /\ (1.25 * 155) - 1 = 160$$

4.12 Baseplate mounting flange shear tear out

The flange geometry for shear tear out at the mounting holes is



$$A_s = 2 * 0.39 * 0.375 = 0.29\ in^2$$

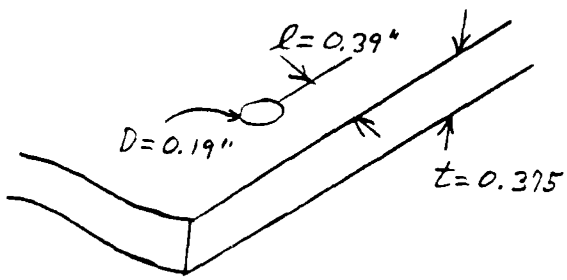
$$P_u = \sigma_{su} * A_s$$

$$= 27E{+}03 * 0.29 = 7.9E{+}03\ lb.$$

Therefore $MS = 7.9E{+}03\ /\ (2 * 155) - 1 = 24.$

4.13 Baseplate mounting flange bearing strength

The geometry for the bearing strength of the flange is



From JA-418 the expression for the bearing stress is

$$P_{bru} = k_{bru} * \sigma_{tu} * A_{br}$$

where $A_{br} = D * t = 0.19\ inch * 0.375\ inch = .071\ in^2$

To obtain the constant from the table of page 10 of the "example" analysis shown in the appendix of JA-418, the following ratios are needed

$e/D = 0.375 / 0.19$ inch $= 2.0$

$D/t = 0.19$ inch $/ 0.375$ inch $= 0.51$

Then $k_{bru} = 2.0$, approximately

and $P_{bru} = 2.0 * 42 E+03 * .071 = 5985$ lb

Therefore MS = 5985 / (2 * 155) -1 = 18

Summary of baseplate edge safety margins

LOAD CASE	MNT FLANGE BEND TENS	YIELD	MNT FLANGE TEAR TENS	YIELD	MNT FLANGE SHEAR OUT	MNT FLANGE BEARING
1	16.2	21.9	120	160	24	18
2	23.5	31.6	155	206	32	24
3	10.3	14.0	744	993	156	118
4	39.2	53.6	446	595	93	70

4.14 Outer cover mounting bolts

For all subsequent analysis in sections 4.14 to 4.18, the baseplate is assumed to be fixed rigidly to the orthogrid. The shear bending loads calculated in these sections result from the accelerations acting only on the outer cover assembly (four sidewalls and topwall). The mass of the outer cover is comprised of

4 sides and top, 29.9 lb,

doghouses, pwr bridge, 1.5 lb,

TOTAL 31.4 lb, 14.3 kg

The center of mass of the complete cover is 223 mm above the plane of the baseplate. In addition to the 32 attachment bolts which pass through to the cold plate support structure, an additional 32 bolts interleaved between the cold plate bolts attach the cover to the baseplate. These additional bolts are 10-32 English thread, A286 stainless steel. When the FEB is being carried by the FEB lifting device which attaches to the top cover, these bolts carry the weight of the FEB baseplate and internal modules. When the FEB is being carried in this way the accelerations are much lower than during lift-off or landing. Since these bolts are of the same size and type as the coldplate attachment bolts, the analysis of the coldplate attachment bolts (sections 4.2-4.9) is sufficient for these additional bolts during ground operations.
For lift-off and landing, these additional bolts are analyzed in this section.
As in section 4.5, the X-axis and Y-axis accelerations act on the

center of mass to produce tension loading which adds to the Z-axis loading. 64 bolts carry this loading from the top cover. The present analysis is for the 32 interleaved bolts.

Analysis similar to that of section 4.5 shows that for load case #1 the tension loading on these attachment bolts is

$$F_{zx} = 120.1 \text{ N}$$

$$F_{zy} = 9.5 \text{ N}$$

$$F_{zz} = 5.9 \text{ N}$$

for a total loading of 135.5 N. Due to the preload of the bolt, part of this interface load is taken by the bolt and the remainder is taken by the flange. For the margin of safety calculation, a saftey factor of two is applied to this interface force but not the maximum bolt preload.

$$MS_{tu} = (964 * 12.9)/(2 * 0.34 * 135.5 + 5419) - 1 = 1.26.$$

Following reference 2.9, the margin of safety should also be calculated with a safety factor of 1.4 applied to the bolt preload that results from an average bolt friction.

$$MS_{tu} = (964 * 12.9)/(2 * 0.34 * 135.5 + 1.4 * 4454) - 1 = 0.97$$

Gapping of the joint under minimum mounting torque and maximum bolt friction must also be considered. From section 4.4 of this document, the minimum bolt preload is 3061 N. There will also be a relaxation of the initial bolt preload which is estimated to be 20%, which leaves a residual bolt preload of 2449 N. If the flanges fraction of the external load were to exceed the minimum preload, gapping would occur.

$$MS = 2449 / (2 * 0.66 * 135.5) - 1 = 12.6.$$

Shear loadings are produced by the X and Y axis accelerations.

$$P_{sx} = (14.3 \text{ kg} * 9.81 * 37.1) / 64 = 81. \text{ N}$$

$$P_{sy} = (14.3 \text{ kg} * 9.81 * 4.0) / 64 = 8.7 \text{ N}$$

Considering the vector sum, $P_s = 81.5$ N

If no relative motion between the top cover and the baseplate is to occur, the resistance produced by friction must be greater than this amount. Assuming a friction coefficient of 0.15

$$F_{fric} = 0.15 * 2449 = 367 \text{ N}$$

Then $MS_{slip} = 367 /(2 * 81.5) - 1 = 1.3.$

If the friction between the cover and baseplate were not enough, the bolts would have to absorb the lateral loading. Using the strength of the bolts and the stress cross-section

$$MS_s = (627 * 12.9) / (2 * 81.5) - 1 = 48.6$$

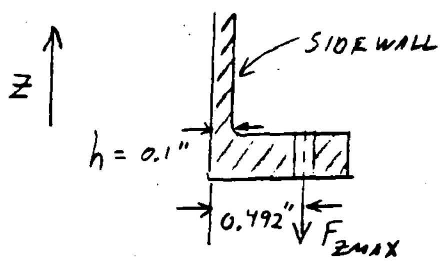
The combined "factor of limit load capabiltiy" is then calculated as in section 4.9. Since the shear safety margins are so large, the combined safety margin is very close to that due to tension only.

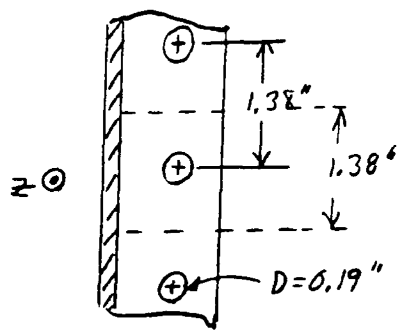
Margin of Safety Summary for FEB outer cover mounting bolts

LOAD CASE	2.0 SF * INT ONLY ALL BOLTS	2.0 SF * INT 1.4 SF * PRE ALL BOLTS	SLIP MS	SHEAR MS	COMB MS	GAPPING MS
1	1.26	0.97	1.3	48.6	1.25	12.6
2	1.27	0.98	1.9	62.1	1.27	18.7
3	1.26	0.97	10.7	256	1.26	12.2
4	1.13	0.86	7.2	179	1.12	35.1

4.15 Sidewall mounting flange bending

The maximum bending momement acting on the sidewall flange is produced by the external interface load at the joint.





$$M_o = F_{zmax} * 0.492 \text{ in} = 15.0 \text{ in-lb}$$

Then the maximum bending stress, assumed tensile, is

$$\sigma_{max} = 6 * M_o/(1.38 * 0.1^2) = 6516 \text{ psi}$$

The margin of safety (using $_{tu}$ = 42 E+03 psi) is

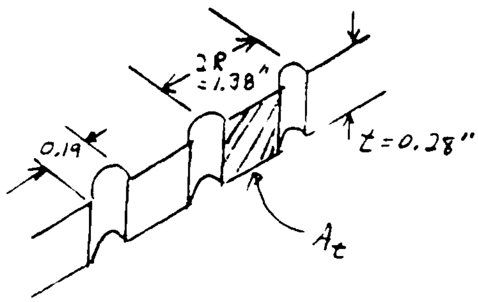
$$MS_{sfbend} = 42 \text{ E+03} /(2* 6516) -1 = 2.2$$

Of course, the stiffing added by the ribs in the sidewall will increase the margin of safety.

4.16 Sidewall flange tension failure

The margin of safety calculated in section 4.14 indicates that no slipping between the top cover and baseplate should occur. If slipping were to somehow occur, the following calculations demonstrate that the sidewall flange has sufficient strength.

The flange geometry of the sidewalls for tension failure is:



The appropriate expression for tension failure is

$$P_u = \sigma_{tu} * A_t$$

where $A_t = (2 * R - D) * t$

$$= (1.38 - 0.19) * 0.28 = 0.33 \text{ in}^2$$

$$P_u = 42\text{ E+03} * 0.33 = 14\text{ E+03 lb}$$

The maximum shear load per bolt from section 4.15 is 81.5 N (18.2 lb),

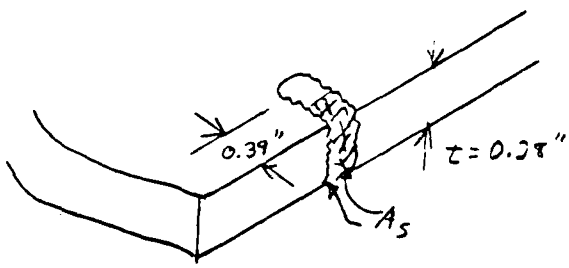
therefore $MS_t = 14\text{ E+03} / (2 * 18.2) -1 = 383$

Checking for yield with a safety factor of 1.25,

$$MS_y = (35\text{ E+03} * 0.33) / (1.25 * 18.2) - 1 = 511$$

4.17 Sidewall mounting flange shear tear out

The flange geometry for the shear tear out is



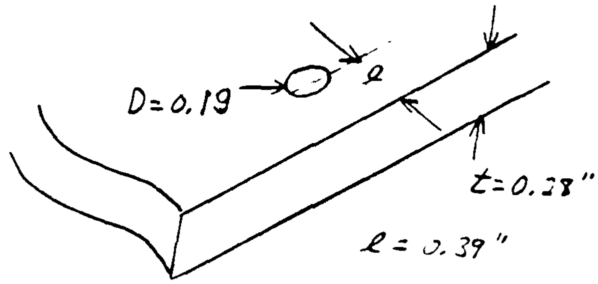
$$P_u = \sigma_{su} * A_s$$

$$= 27\text{ E+03} * (2 * 0.39 * 0.28) = 5897\text{ lb.}$$

Therefore $MS = 5897 / (2 * 18.2) -1 = 161.$

4.18 Sidewall mounting flange bearing strength

The geometry for the bearing strength of the flange is



From JA-418 the expression for the bearing stress is

$$P_{bru} = k_{bru} * \sigma_{tu} * A_{br}$$

where $A_{br} = D * t = 0.19 \text{ inch} * 0.28 \text{ inch} = .053 \text{ in}^2$

To obtain the constant from the table of page 10 of the "example" analysis shown in the appendix of JA-418, the following ratios are needed

$$e/D = 0.375 \text{ inch} / 0.19 \text{ inch} = 2$$

$$D/t = 0.19 \text{ inch} / 0.28 \text{ inch} = 0.68$$

Then $k_{bru} = 2.0$, approximately.

and $P_{bru} = 2 * 42\text{ E+03} * 0.053 = 4468 \text{ lb}$

Therefore $MS = 4468 / (2 * 18.2) - 1 = 122$

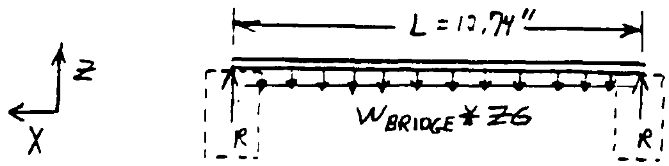
Summary of sidewall flange safety margins:

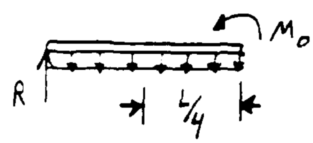
LOAD CASE	SIDE FLNGE TENS	BEND YIELD	SIDE FLNGE TENS	TEAR YIELD	SIDE FLNGE SHEAR OUT	SIDE FLANGE BEARING
1	2.2	3.3	383	511	161	122
2	3.7	5.2	495	660	208	157
3	2.1	3.2	2373	3164	999	757
4	7.6	10.4	1423	1898	599	454

5 Internal module analysis

5.1 Power distribution bridge

The power distribution bridge attaches at each end to the top of the signal and power doghouses with two #6-32 type A286 stainless steel screws.





The power distribution bridge is a thin U-channel that carries the power supply connectors. Z-axis loading on the FEB will produce tension loading in the attachment bolts. X-axis and Y-axis accelerations produce shear in the attachment bolts but little additional tension since the bridge is thin.

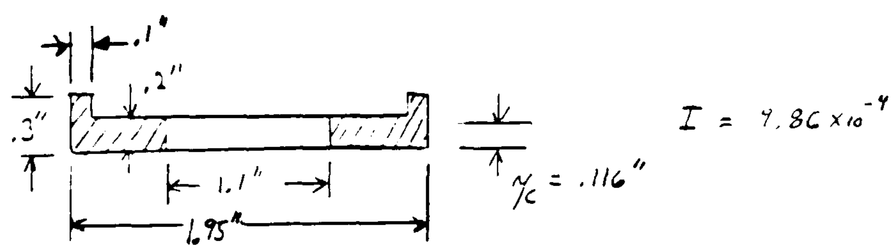
Due to connector mounting holes, the effective cross section area is reduced near the center of the bridge where the maximum bending stress occurs. Load case #3 (Z-axis accelerations) will produce the maximum bending moment.

$$R = 0.5 * W_{bridge} * ZG = 0.5 * 0.5 * 50.5 = 12.6$$

$$M_o = R * L/2 - 0.5 * W_{bridge} * ZG * L/4$$

$$= 12.6 \text{ lb} * 6.37 \text{ in} - 12.6 \text{ lb} * 3.19 \text{ in} = 40.2 \text{ in-lb}$$

For the effective cross section



y_c = 0.116 in and I = 9.86 E-04

Then $\sigma_{max} = M_o * y_c / I$

$$= 40.2 * 0.116 / 9.86 \text{ E-04} = 4723 \text{ psi}$$

Thus, $MS_{tbend} = 42\ E{+}03\ /\ (2 * 4723) - 1 = 3.4$

Checking for yield,

$$MS_{ybend} = 35\ E{+}03\ /\ (1.25 * 4723) - 1 = 4.9$$

The maximum shear occurs at the end of the beam due to reaction R. The shear area is the cross section of the beam at the doghouse.

$$P_s = 27\ E{+}03 * 0.25\ in^2 = 6750\ lb$$

Then $MS_s = 6750\ /\ (2 * 12.6) - 1 = 266.$

The power distribution bridge attaches to the each doghouse with two 6-32 screws at each end. Since the bridge is relatively thin, X and Y axis moments do not produce additional Z-axis loading.

$$F_{zz} = W_{bridge} * ZG\ /(4\ \text{bolts}) = 6.3\ lbs \qquad (LC\ \#3)$$

The bolt preload must first be calculated. The expected range of tightening torque is 10 to 12 in-lb. Using the figure of reference 2.6, for the minimum friction value of 0.0784 and the maximum torque

$$F_{maxPRE} = 60 * 12 = 720\ lb.$$

Also calculating the average preload and minimum preload

$$F_{avgPRE} = 50 * 12 = 600\ lb$$

and $F_{minPRE} = 40 * 10 = 400\ lb.$

The stress area for a 6-32 bolt is 0.00909 in^2, thus

$$P_{ut} = 140\ E{+}03 * 0.00909 = 1270\ lb$$

The fraction of the external interface load that is take by the flange is determined by the flange geometry which fits case 2 of reference 2.5 (page 117)

$$A_j = PI * (D_w^2 - D_h^2)\ /\ 4 + PI * (D_j / D_w - 1) * (D_w * L_j / 5 + L_j^2 / 100)\ /\ 8$$

$$= 3.14 * (0.375^2 - 0.22^2)\ /\ 4 + 3.14 * (0.29 / 0.375 - 1) * (0.375 * 0.2 / 5 + 0.2^2 / 100)\ /\ 8$$

$$= 0.0765\ in^2$$

The area and modulus of elasticity for the flange and bolt are used to determine the joint or "screw" diagram.

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$

$$= \frac{0.0765 * 1E+07}{0.00909 * 2.91E+07 + 0.0765 * 1E+07} = 0.74$$

Thus most of the external interface load is taken by the flange.

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (1270)/(2 * (1-0.74) * 6.3 + 720) - 1 = 0.77$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension.

$$MS_t = (1270)/(2 * (1-0.74) * 6.3 + 1.4 * 600) - 1 = 0.51$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the initial preload, a preload of 0.8 * 400 = 320 lb results. If the flange's fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 320 / (2 * 0.74 * 6.3) - 1 = 33.1$$

The shear loading comes from the X-axis and Y-axis accelerations.

$$F_{sx} = W_{bridge} * XG / 4 \text{ bolts}$$

$$= 0.5 * 37.1 / 4 = 4.6 \text{ lb} \qquad \text{(LC \#1)}$$

$$F_{sy} = W_{bridge} * YG / 4 \text{ bolts}$$

$$= 0.5 * 4.0 / 4 = 0.5 \text{ lb} \qquad \text{(LC \#1)}$$

and $F_{sxy} = ((F_{sx})^2 + (F_{sy})^2)^{1/2} = 4.7$ lb

To avoid slipping between the bridge and the top of the doghouse, the maximum possible shear force is calculated using a friction value of 0.15 between the members and the relaxed minimum bolt preload

$$F_{fric} = 0.15 * 320 \text{ lb} = 48 \text{ lb}$$

Thus $MS_{slip} = 48 / (2 * 4.7) - 1 = 4.2$

If the holding friction was not sufficient, the mounting bolts can absorb the shear load

$$MS_s = (91E+03 * 0.00909) / (2 * 4.7) - 1 = 88$$

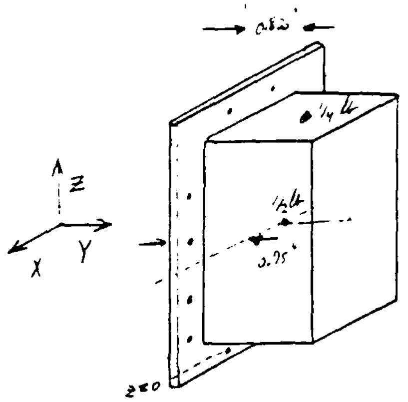
These shear loads are small enough that tension and shear tear out of the 0.2 inch thick flange of the power distribution bridge do not need to be calculated

Margin of safety summary of power distribution bridge

MS BRIDGE TENS	BEND YIELD	BRIDGE SHEAR	BOLT 2.0	TENSION 2.0 & 1.4	GAPPING	SLIPPING	BOLT SHEAR
3.4	4.9	266	0.77	0.51	33.1	4.2	88

5.2 Signal and power doghouse

The signal and power doghouses attach to the connector endwall of the FEB. Y-axis loading of the FEB produces direct axial tension in the mounting bolts. X-axis and Z-axis loading produce shear in the mounting bolts and tension through the moments produced on the CG. The loading of the power distribution bridge is modeled by adding a point mass of $W_{bridge}/2$ at the bridge attachment point.



COMPOSITE CG:

$\frac{1}{2}$ lb $*$ 2.1" + $\frac{1}{4}$ lb $*$ 4.2 = 2.1

$Z_{CG} * (\frac{1}{2}+\frac{1}{4}) = 2.1$

$Z_{CG} = 2.8"$

$\frac{1}{2}$ lb $*$ 0.75 + $\frac{1}{4}$ $*$ 0.325 = 0.58

$Y_{CG} * (\frac{1}{2}+\frac{1}{4}) = 0.58$

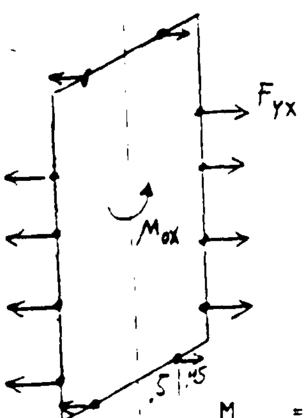
$Y_{CG} = 0.78"$

For tension calculations, the effective center of gravity, CG, must be first found. The moment calculations are straighforward.

$$M_{ox} = (W_{doghouse} + W_{bridge/2}) * XG * 0.78 \text{ inch}$$

$$= (0.5 + 0.5/2) * 37.1 * 0.78 = 21.7 \text{ in-lb}$$

This is counterd by the bolt reaction, F_{yx}.

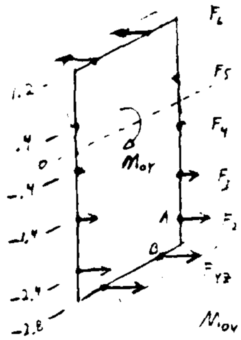


$$F_{Yx} * 2 * \left(4 + 2 * \frac{0.5}{0.95}\right) * 0.95 = M_{ox}$$

$$M_{ox} = 2 * (4 + 2 * 0.5/0.95) * 0.95 \text{ inch} * F_{yx}$$

$$F_{yx} = 21.7 / (9.6) = 2.26 \text{ lb}$$

The Z-axis accelerations also produce a moment around the CG which is countered by the distributed perimeter bolts.



$$F_2 = \frac{2.4}{2.8} F_{Yz}$$
$$F_3 = \frac{1.4}{2.8} F_{Yz}$$
$$F_4 = \frac{.4}{2.8} F_{Yz}$$
$$F_5 = \frac{.4}{2.8} F_{Yz}$$
$$F_6 = \frac{1.2}{2.8} F_{Yz}$$
$$M_{oy} = F_{Yz} * 2 * \left(1 + \frac{2.4}{2.8} + \frac{1.4}{2.8} + \frac{.4}{2.8} + \frac{.4}{2.8} + \frac{1.2}{2.8}\right) * 2.8$$

$$M_{oz} = (W_{doghouse} + W_{bridge}/2) * ZG * 0.78 \text{ inch}$$
$$= (0.5 + 0.5/2) * 2.7 * 0.78 = 1.58 \text{ in-lb}$$

$$F_{yz} = M_{oz} / (17.2) = 0.092 \text{ lb}$$

The Y-axis induced load acts through the new CG, and directly produces tensile loading

$$F_{yy} = ((W_{doghouse} + W_{bridge}/2) * YG) / 12 \text{ bolts}$$
$$= (0.5 + 0.5/2) * 4.0 / 12 = 0.25 \text{ lb}$$

Bolt positions A or B may suffer the maximum loading depending on the load case

$$F_A = F_{yy} + F_{yz} * 2.4/2.8 + F_{yx}$$
$$F_B = F_{yy} + F_{yz} + F_{yx} * 0.5/0.95$$

LOADCASE	F_A	F_B
1	2.59	1.52 lb
2	2.24	2.08
3	2.09	2.16
4	0.93	0.67

These loads are all very small compared to the bolt preload. The bolt preload will essentialy control the total bolt load. A joint diagram is not needed. (Also the soft elastomer gasket which is used for EMC control fully compresses and does not enter into the joint diagram calculations.)

The 6-32 bolts used to attach the doghouse to the sidewall are type 316 stainless steel for which

$$\sigma_s = 37.5 \text{ E+03}$$
$$\sigma_t = 75 \text{ E+03}$$

The ultimate bolt strength is then

$$P_{tu} = 75\ E+03 * 0.00909 = 682\ lb$$

$$P_{su} = 37.5\ E+03 * 0.00909 = 341\ lb$$

The expected tigthening torque is 7 to 8 in-lb. The maximum bolt preload is then

$$F_{maxPRE} = 60 * 8 = 480\ lb$$

and

$$F_{avgPRE} = 50 * 8 = 400\ lb$$

$$F_{minPRE} = 40 * 7 = 280\ lb$$

The shear loads in these bolts are produced by X-axis and Z-axis acceleration. The maximum shear load occurs for load case #3.

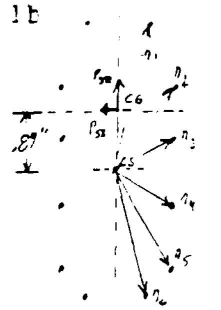
$$P_{sx} = (W_{doghouse} + W_{bridge}/2) * XG = 4.5\ lb$$

$$P_{sz} = (W_{doghouse} + W_{bridge}/2) * ZG = 37.9\ lb$$

The primary shear loading for each of the 12 bolts is simply

$$P_{sxz} = ((P_{sx})^2 + (P_{sz})^2)^{1/2}\ /12$$

$$= 3.2\ lb$$



$r_1 = \sqrt{(2.07)^2 + .5^2} = 2.13$

$r_2 = 1.59 \quad r_3 = 1.06$

$r_4 = 1.09 \quad r_5 = 1.80 \quad r_6 = 1.99$

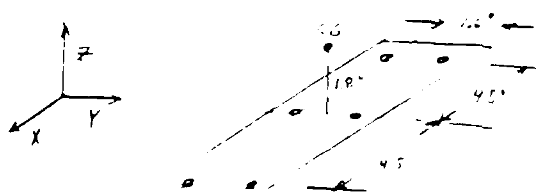
$$F_{s1} = \frac{P_{sx} * .87 * r_1}{2 * \Sigma r_i^2}$$

$$= P_{sx} * 0.056$$

Since P_{sz} passes through the center of screws, only P_{sx} produces an eccentric shear loading on the screws. The worse case additional loading is 0.25 lb. The loads are so small that continued detailed analysis is not necessary

5.3 Power supply

The power supply is fastened to the FEB baseplate through an adapter plate so that it can be removed from the top without accessing any screw heads on the underneath side of the FEB baseplate. The commerical power supply is attached to the adapter plate through the following hole pattern with 6-32, type A286 stainless screws.



The CG is assumed directly overtop the center of the screws. Z-axis acceleration produces tension only.

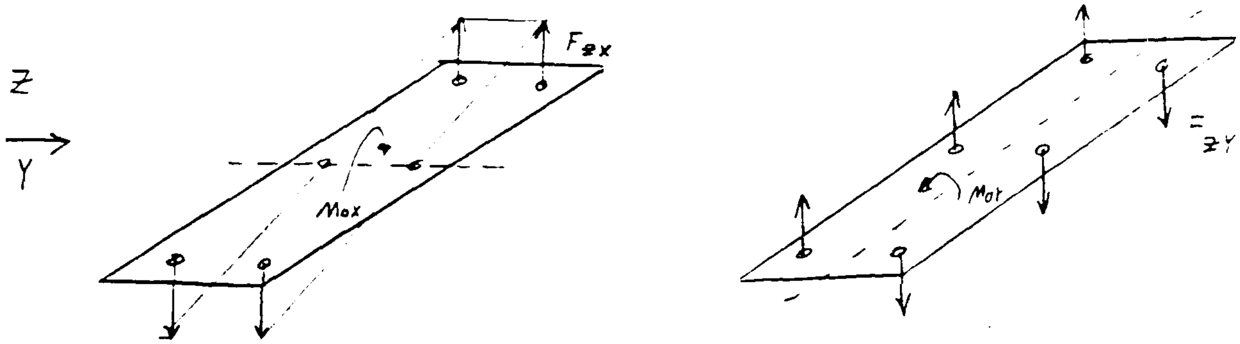
$$F_{zz} = W_{pwrsup} * ZG / 6 \text{ bolts}$$

$$= 3.9 * 2.7 / 6 = 1.76 \text{ lb}$$

X-axis acceleration produces a moment of

$$M_{ox} = W_{pwrsup} * XG * 1.8 \text{ inch}$$

$$= 3.9 * 37.1 * 1.8 = 260 \text{ in-lb}$$



This is countered by

$$F_{zx} = M_{ox}/(4 * 4.5 \text{ inch}) = 14.5 \text{ lb}$$

The Y-axis acceleration also produces a moment,

$$M_{oy} = W_{pwrsup} * YG * 1.8 \text{ inch}$$

$$= 3.9 * 4 * 1.8 = 28.1 \text{ in-lb}$$

which is countered by

$$F_{zy} = M_{oy}/(6 * 1.125/2) = 8.3 \text{ lb}$$

The maximum tensile load is then

$$F_{max} = F_{zx} + F_{zy} + F_{zz}$$

$$= 14.5 + 8.3 + 1.76 = 24.5 \text{ lb}$$

Part of this external interface load in taken by the flange. These bolts are countersunk into the underneath side of the 0.15 inch thick adapter plate. The effective flange area for such a situation is not described in the references. Therefore the effective flange area will be calculated using the diameter of the flat-head screw (0.307 inch)

$$A_j = PI * ((0.307)^2 - (0.14)^2) / 4 = 0.059 \text{ in}^2$$

The fraction of the external interface load taken by the adapter plate is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$

$$= \frac{0.059 * 1 E+07}{0.0090 * 2.91 E+07 + 0.059 * 1 E+07} = 0.69$$

hus most of the external interface load is taken by the flange.

he margin of safety can then be calculated using a safety factor of wo on the external interface load.

$$MS_t = (1270)/(2 * (1-0.69) * 24.5 + 720) - 1 = 0.73$$

sing an average value for the coefficent of friction, and a safety actor of 1.4 on the resulting pretension.

$$MS_t = (1270)/(2 * (1-0.69) * 24.5 + 1.4 * 600) - 1 = 0.49$$

apping of the joint under minimum mounting torque and maximum torque riction must be considered. After the 20 % relaxation from the nitial preload, a preload of 0.8 * 400 = 320 lb. If the adapter late fraction of the external load were to exceed this minimum reload, gapping would occur.

$$MS_{gap} = 320 / (2 * 0.69 * 24.5) - 1 = 8.4$$

he shear loading comes from the X-axis and Y-axis loading.

$$F_{sx} = W_{pwrsup} * XG / 6 \text{ bolts}$$

$$= 3.9 * 37.1 / 6 = 24.1 \text{ lb} \qquad (LC \#1)$$

$$F_{sy} = W_{pwrsup} * YG / 6 \text{ bolts}$$

$$= 3.9 * 4.0 / 6 = 2.6 \text{ lb} \qquad (LC \#1)$$

nd $$F_{sxy} = ((F_{sx})^2 + (F_{sy})^2)^{1/2} = 24.2 \text{ lb}$$

o avoid slipping between the power supply and the adapter plate, the aximum possible shear force is calculated using a friction value of .15 between the members and the relaxed minimum bolt preload.

$$F_{fric} = 0.15 * 320 \text{ lb} = 48 \text{ lb}$$

hus $$MS_{slip} = 48 / (2 * 24.2) - 1 = -0.01$$

ince the holding friction is not sufficient, the mounting bolts must bsorb the shear load

$$MS_s = (.91E+05 * 0.00909) / (2 * 24.2) - 1 = 16.1$$

ince these mounting bolts sustain both tension and shear, the effect f both stress are combined through calculation of the corresponding factor of "limit load capabilities" which are then combined

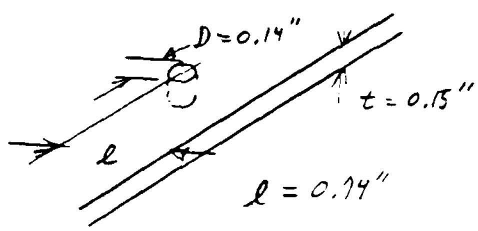
$FLC_t = MS_t + 1 = 0.73 + 1 = 1.73$

and $FLC_s = MS_s + 1 = 16.1 + 1 = 17.1$.

Then $$FLC_{comb} = \frac{1}{[(1/1.73)^2 + (1/17.1)^2]^{1/2}} = 1.72$$

The power supply attachment holes are located far enough from the edges of the adapter plate, that tension tear out or shear tear out are very unlikely.

Checking for the bearing strength:



$e/D = 0.74 / 0.14 = 5.3$

$D/t = 0.14 / 0.15 = 0.93$

yield a $k_{bru} > 3$.

$A_{br} = D * t = 0.14 * 0.15 = 2.1\text{ E-02 in}^2$

Then $F_{bru} = k_{bru} * \sigma_t * A_{bru}$

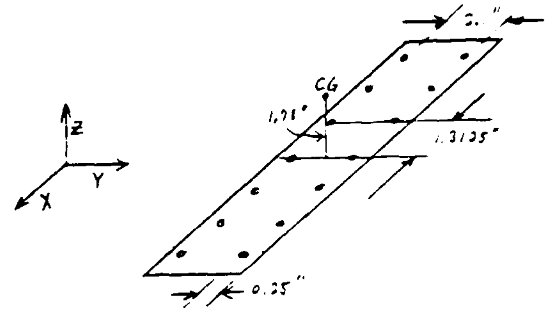
$= 3.0 * 42\text{ E+03} * 2.1\text{ E-02} = 2646$

$MS_{bru} = 2646 / (2 * 24.2) - 1 = 54$

Margin of safety summary, power supply to adapter plate interface

LOAD CASE	PWR<-->ADAPT 2.0 SF	2.0 & 1.4	MS GAP	MS SLIP	MS SHEAR	MS COMB	MS BEARING
1	0.73	0.49	8.4	-0.01	16.1	0.72	54
2	0.66	0.44	2.6	0.25	20.7	0.67	68
3	0.70	0.47	4.3	4.12	87.2	0.70	282
4	0.74	0.50	17.2	2.59	60.9	0.74	196

The power supply adapter plate attaches to the FEB baseplate with fourteen, 10-32 type A286 screws in the following pattern. The total weight is the sum of the power supply and the adapter plate (3.9 + 0.4).



Since the CG is directly over the center of the screws, Z-axis loading produces tension only.

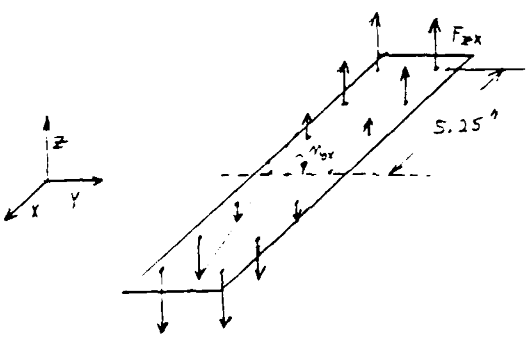
$$F_{zz} = (W_{pwrsup} + W_{adapt}) * ZG / 14 \text{ bolts}$$

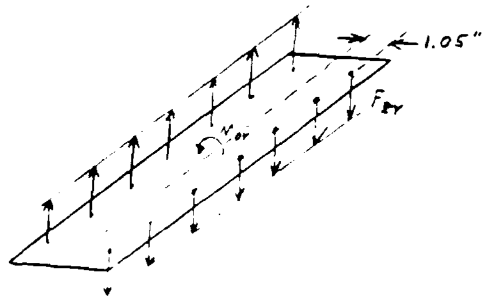
$$= (3.9 + .4) * 2.7 / 14 = 0.83 \text{ lb.}$$

X-axis acceleration produces a moment of

$$M_{ox} = (W_{pwrsup} + W_{adapt}) * XG * 1.78 \text{ inch}$$

$$= (3.9 + .4) * 37.1 * 1.78 = 284 \text{ in-lb}$$





This is countered by

$$F_{zx} = M_{ox} / (4 * (1 + 2/3 + 1/3) * 5.25)$$

$$= 284 / (8 * 5.25) = 6.76 \text{ lb}$$

The Y-axis acceleration also produces a moment,

$$M_{oy} = (W_{pwrsup} + W_{adapt}) * YG * 1.78 \text{ inch}$$

$$= (3.9 + .4) * 4.0 * 1.78 = 30.6 \text{ in-lb}$$

This is countered by

$$F_{zy} = M_{oy} / (14 * 1.05) = 2.1 \text{ lb}$$

The maximum tensile load is then

$$F_{max} = F_{zx} + F_{zy} + F_{zz}$$
$$= 6.76 + 2.1 + 0.8 = 9.7 \text{ lb}$$

Part of this external interface load in taken by the flange. The geometery of the flange is closest to case 2 of reference 2.5. The calculation of the effective flange area is

$$A_j = PI * (D_w^2 - D_h^2) / 4 + PI * (D_j / D_w - .1) * (D_w * L_j / 5 + L_j^2 / 100) / 8$$
$$= 3.14 * (0.375^2 - 0.25^2) / 4 + 3.14 * (0.5/0.375 - .1) * (0.375 * 0.15/5 + 0.15^2/100) / 8$$
$$= 0.067 \text{ in}^2$$

The fraction of the external interface load taken by the adapter plate is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$
$$= \frac{0.067 * 1 E{+}07}{0.02 * 2.91 E{+}07 + 0.067 * 1 E{+}07} = 0.54$$

A joint diagram will thus show that 0.54 of the external interface force is taken by the flange and 0.46 of the interface force is taken by the bolt which adds to the preload of the bolt to produce the maximum bolt stress.

The expected tightening torque is 30 to 34 in-lb. Thus for minimum friction (u =0.0784) and maximum torque (using the "nut factors" from reference 2.6)

$$F_{maxPRE} = 44.4 * 34 = 1510 \text{ lb}$$

For an average friction value of u = 0.1,

$$F_{avgPRE} = 36.5 * 34 = 1241 \text{ lb}$$

and for the maximum friction value and minimum torque,

$$F_{minPRE} = 28.8 * 30 = 864 \text{ lb.}$$

This minimum preload will be reduced an additional 20 % for the gapping analysis which will follow

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (2800)/(2 * (1-0.54) * 9.7 + 1510) - 1 = 0.84$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension.

$$MS_t = (2800)/(2 * (1-0.54) * 9.7 + 1.4 * 1241) - 1 = 0.60$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the initial preload, a preload of 0.8 * 864 = 691 lb. If the adapter plate fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 691 / (2 * 0.54 * 9.7) - 1 = 66.$$

The shear loading comes from the X-axis and Y-axis acceleration.

$$F_{sx} = (W_{pwrsup} + W_{adapt}) * XG / 14 \text{ bolts}$$

$$= (3.9 + 0.4) * 37.1 / 14 = 11.4 \text{ lb}$$

$$F_{sy} = (W_{pwrsup} + W_{adapt}) * YG / 14 \text{ bolts}$$

$$= (3.9 + .4) * 4.0 / 14 = 1.2 \text{ lb}$$

and $F_{sxy} = ((F_{sx})^2 + (F_{sy})^2)^{1/2} = 11.5 \text{ lb}$

To avoid slipping between the adapter and the FEB baseplate, the maximum possible shear force is calculated using a friction value of 0.15 between the members and the relaxed minimum bolt preload.

$$F_{fric} = 0.15 * 691 \text{ lb} = 104 \text{ lb}$$

Thus $MS_{slip} = 104 / (2 * 11.5) - 1 = 3.5$

If the holding friction was not sufficient, the mounting bolts can absorb the shear load

$$MS_s = (91E+03 * 0.02) / (2 * 11.5) - 1 = 78.$$

Since these mounting bolts sustain both tension and shear, the effect of both stress are combined through calculation of the corresponding "factor of limit load capabilities" which are then combined

$$FLC_t = MS_t + 1 = 0.84 + 1 = 1.84$$

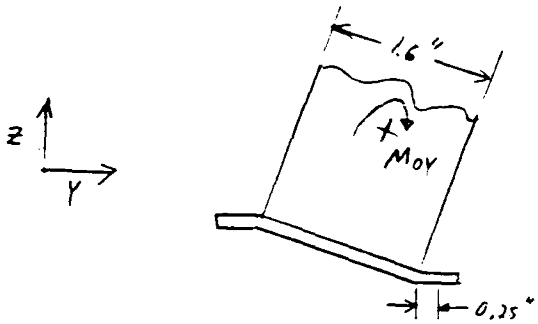
and $FLC_s = MS_s + 1 = 78 + 1 = 79.$

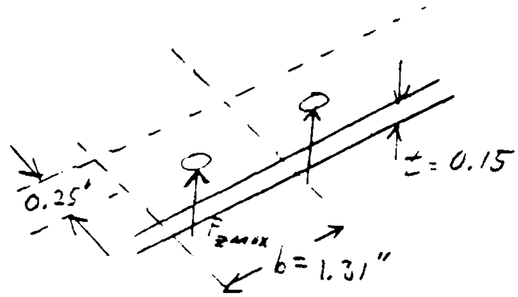
Then $$FLC_{comb} = \frac{1}{[(1/1.84)^2 + (1/79)^2]^{1/2}} = 1.84$$

Margin of safety summary, power supply to adapter plate bolts:

LOAD CASE	BOLT 2.0 SF	ADAPT<->BASE 2.0 & 1.4	GAP	SLIP	BOLT SHEAR	MS COMB
1	0.84	0.60	66	3.5	78	0.84
2	0.84	0.60	37	4.8	100	0.83
3	0.83	0.60	34	22.4	410	0.83
4	0.85	0.61	131	15.4	287	0.85

The adapter plate must withstand the bending produced on its flange by the Y-axis loading.





$$M_o = 0.25 \text{ inch} * F_{zmax} = 0.25 * 9.7 = 2.42 \text{ in-lb}$$

then the stress

$$\sigma_{bend} = 6 * M_o / b\ t^2$$
$$= 6 * 2.42 / (1.31 * (0.15)^2)$$
$$= 493 \text{ psi}$$

Shear stress is also present,

$$\sigma_{shear} = F_{zmax} / A_{shear}$$
$$= 9.7 / (1.31 * 0.15) = 49.2 \text{ psi}$$

The combined stress is then

$$\sigma_{max} = ((\sigma_{bend})^2 + 3 * (\sigma_{shear})^2)^{1/2}$$
$$= ((493)^2 + 3 * (49.2)^2)^{1/2} = 499 \text{ psi}$$

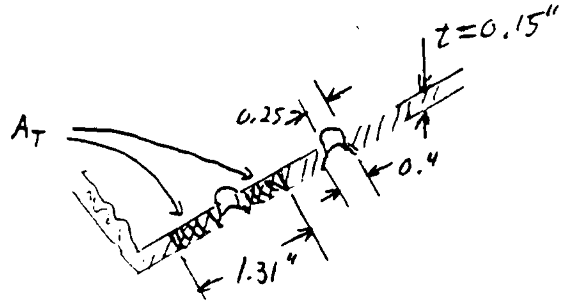
The margin of safety is

$$MS_{ubend} = 42 E+03 / (2 * 499) - 1 = 41.1$$

Checking for yield,

$$MS_{ybend} = 35\ E{+}03\ /\ (1.25 * 499) - 1 = 55.8$$

The flange geometery for tension failure is



The ultimate strength for tension tear out is

$$P_u = \sigma_{tu} * A_t$$

where $$A_t = (2 * R - (D_1 + D_2)/2) * t$$

$$= (1.31 - (0.25 + 0.4)/2) * 0.15 = 0.148\ in^2$$

$$P_u = 42\ E{+}03 * 0.148 = 6205\ lb$$

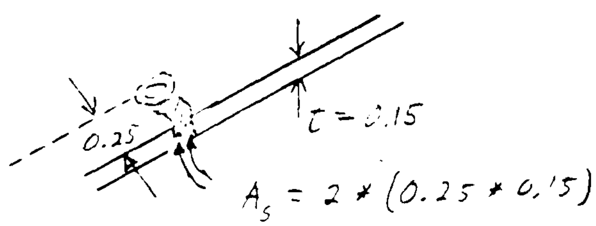
The maximum shear load was 11.5 lb, thus

$$MS_{utear} = 6205\ /\ (2 * 11.5) - 1 = 270.$$

Checking for yield by using as safety factor of 1.25,

$$MS_{ytear} = 35\ E{+}03 * 0.148\ /\ (1.25 * 11.5) - 1 = 360$$

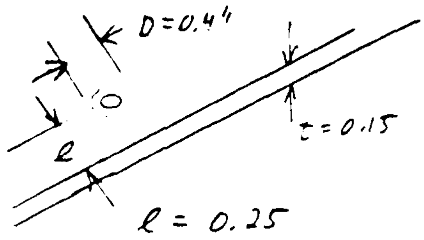
The geometry for shear tear out is



$$A_s = 2 * (0.25 * 0.15) = 0.075\ in^2$$

Then $$MS_{shear} = 27\ E{+}03 * 0.075\ /\ (2 * 11.5) - 1 = 87.4$$

Checking for the bearing strength.



$e/D = 0\ 25\ /\ 0\ 4 = 0\ 63$

$D/t = 0\ 4\ /\ 0\ 15 = 2\ 7$

yield a $k_{bru} = 0\ 3$

$A_{br} = D * t = 0\ 4 * 0\ 15 = 0\ 06\ in^2$

Then $P_{bru} = k_{bru} * \sigma_t * A_{bru}$

$= 0\ 3 * 42\ E+03 * 0\ 06 = 756$

$MS_{bru} = 756\ /\ (2 * 11\ 5) - 1 = 32$

Margin of safety summary for power supply adapter plate:

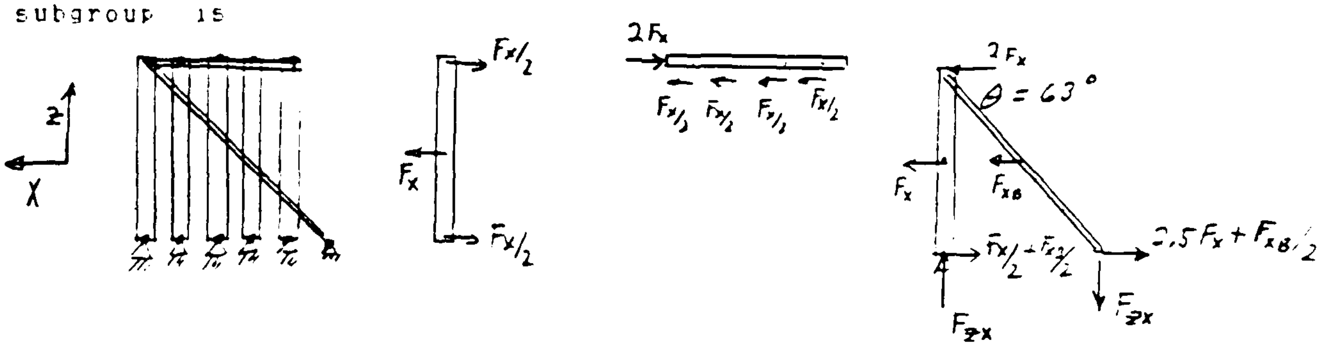
LOAD CASE	MS BEND TENS	MS BEND YIELD	MS TENS TEAR TENS	MS TENS TEAR YIELD	MS SHEAR	MS BEARING
1	41	55	270	360	87	32
2	23	31	343	458	111	41
3	21	28	1400	1867	456	170
4	82	110	981	1309	320	119

5 4 Filters and computer

The "wide" and "narrow" filter modules ("wide" and "narrow" RF bandwidth) and computer mount directly to the base plate with two 10-32, stainless steel screws (A286) Additional cross ties bridge across the tops of these units and diagonal braces prevent gross rotation of the assemblies about the X-axis Each module's weight is 4 95 lb except for the computer whose weight is 4 lb (This weight includes 0 05 lb for the prorated portion of the top U-channel) The analysis of this section will consider a group of five filter modulues which is the worst case compared to the other group of four filters and one computer

The U-channel bracing across the filter tops prevent individual filter movement along the X-axis. Each subgroup of five filter modules has a diagonal cross brace on each side for stiffness in the X-axis. These diagonal braces have a separate attachment to the FEB baseplate.

The free body diagrams for each filter module and for the five filter subgroup is



From the geometry shown above, the Z-axis load produced by the X-axis acceleration is

$$F_{zx} = \tan(\theta) * (2.5 * F_x + F_{xb})$$

where $F_x = 4.95 \text{ lb} * XG / 2 \text{ sides} = 4.95 * 37.1 / 2 = 91.8 \text{ lb}$

and $F_{xb} = 0.28 \text{ lb} * XG * 2 \text{ braces} / 2 \text{ sides} = 10.4 \text{ lb}$

thus $F_{zx} = \tan(63^0) * (2.5 * 91.8 + 10.4) = 471 \text{ lb}$

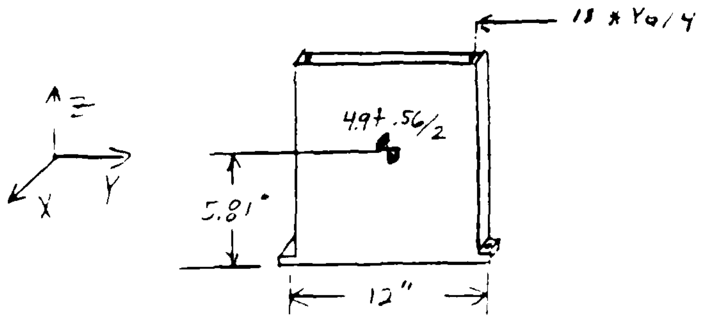
The Z-axis acceleration directly produces tension in the attachment bolts

$$F_{zz} = (W_{filt} + 2 * W_{brace}) * ZG / 2 \text{ sides}$$
$$= (4.95 + 2 * 0.28) * 2.7 / 2 = 7.4 \text{ lb}$$

The Y-axis acceleration produces a moment

$$M_{oy} = (4.95 + 2 * 0.28/2) * YG * 11.625/2 \text{ inch} + (11 \text{ lb} * YG / 4) * 13.25 \text{ inch} = 267 \text{ in-lb}$$

One-half of each brace is allocated this outside filter module; the other four modules would have only the 4.95 lb of their own weight. A cross brace to the IF module adds to the Y-axis load. Analysis of this cross brace in a later section will yield one fourth of the load of the IF module as a conservative case.



This moment is countered by F_{zy}.

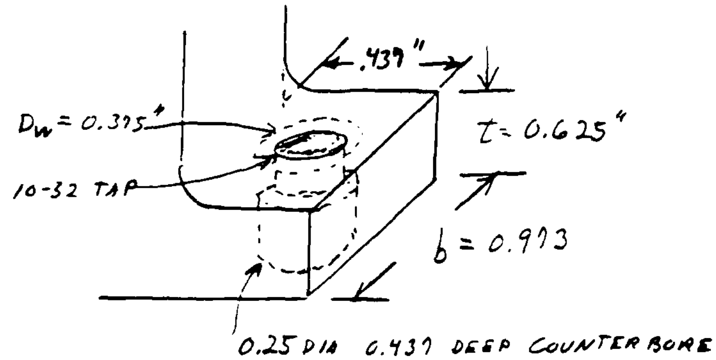
$$F_{zy} = M_{oy}/(2 * 6 \text{ inches}) = 267 / (2 * 6) = 22.3 \text{ lb}$$

The maximum tension load is

$$F_{zmax} = F_{zx} + F_{zy} + F_{zz}$$
$$= 471 \text{ lb} + 22.3 \text{ lb} + 7.4 \text{ lb} = 500 \text{ lb}$$

To determine the division of the external interface load between the mounting flange and the bolt, the effective joint area of the flange must be calculated.

For the mounting feet at the end of the filter module,



The joint geometry fits case 2 of reference 2.5

$$A_j = PI * (D_w^2 - D_h^2) / 4$$
$$+ PI * (D_j/D_w - .1) * (D_w * L_j/5 + L_j^2/100) / 8$$
$$= 3.14 * (0.375^2 - 0.25^2) / 4$$
$$+ 3.14*(0.437/0.375 - .1)*(0.375 * 0.625 /5 + 0.625^2/100) / 8$$
$$= 0.083 \text{ in}^2$$

The fraction of the external interface load taken by the mounting flange is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$
$$= \frac{0.083 * 1 E+07}{0.018 * 2.91 E+07 + 0.083 * 1 E+07} = 0.61$$

The tie down bolts are made "captive" by reducing the shank diameter to that of the thread root. Thus the effective stress area will be slightly smaller than the typical handbook value for a 10-32 screw

$$A_{cap} = PI * dia^2 / 4 = 3.14 * (0.152)^2 /4 = 0.018 \text{ in}^2$$

Thus $P_{us} = 91 E+03 * 0.018 = 1651$ lb

and $P_{ut} = 140 E+03 * 0.018 = 2540$ lb

The expected tightening torque is 30 to 34 in-lb. Thus for minimum friction (u =0.0784) and maximum torque (using the "nut factors" from reference 2.6)

$$F_{maxPRE} = 44.4 * 34 = 1510 \text{ lb}$$

For an average friction value of u = 0.1,

$$F_{avgPRE} = 36.5 * 34 = 1241 \text{ lb}$$

and for the maximum friction value and minimum torque,

$$F_{minPRE} = 28.8 * 30 = 864 \text{ lb.}$$

This minimum preload will be reduced an additional 20 % for the gapping analysis which will follow.

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (2540)/(2 * (1-0.61) * 500 + 1510) - 1 = 0.34$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension.

$$MS_t = (2540)/(2 * (1-0.61) * 500 + 1.4 * 1241) - 1 = 0.19$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the initial preload, a preload of 0.8 * 864 = 691 lb. If the mounting flange fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 691 / (2 * 0.61 * 500) - 1 = 0.13.$$

The shear loading comes from the X-axis and Y-axis loading.

From the previous free body diagram,

$$F_{sx} = F_x/2 + F_{xb}/2$$

$$= 91.8 / 2 + 10.4 / 2 = 51.1 \text{ lb}$$

$$F_{sy} = (W_{filt} + W_{brace}) * YG / 2 \text{ bolts}$$

$$= (4.95 + 0.28) * 4.0 / 2 = 10.5 \text{ lb}$$

and $$F_{sxy} = ((F_{sx})^2 + (F_{sy})^2)^{1/2} = 52.1 \text{ lb}$$

To avoid slipping between the filter and the FEB baseplate, the maximum possible shear force is calculated using a friction value of 0.15 between the members and the relaxed minimum bolt preload

$$F_{fric} = 0.15 * 691 \text{ lb} = 104 \text{ lb}$$

Thus $MS_{slip} = 104 / (2 * 52.1) - 1 = -0.01$

For load case #1 and #2 the margin of safety for slipping is negative for the worst case minimum preload, however the mounting bolts can absorb the shear load

$$MS_s = (91E+03 * 0.018) / (2 * 52.1) - 1 = 14.7$$

(For load case #2 the bolt shear margin of safety is 9.8.)

Since these mounting bolts sustain both tension and shear, the effect of both stress are combined through calculation of the corresponding "factor of limit load capablilities" which are then combined.

$$FLC_t = MS_t + 1 = 0.34 + 1 = 1.34$$

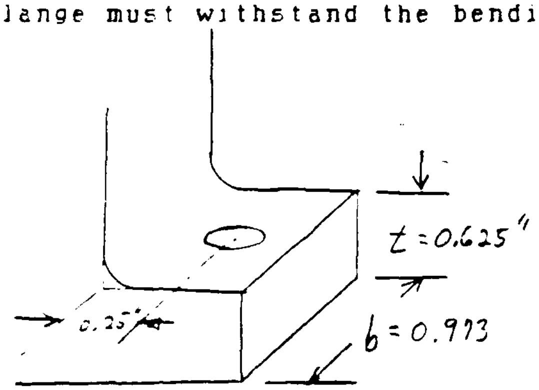
and $FLC_s = MS_s + 1 = 14.7 + 1 = 15.7.$

Then $$FLC_{comb} = \frac{1}{[(1/1.34)^2 + (1/15.7)^2]^{1/2}} = 1.33$$

Margin of safety summary, filter module mounting bolts:

LOAD CASE	FILTER 2.0 SF	MOUNTING 2.0 & 1.4	MS GAP	MS SLIP	BOLT SHEAR	MS COMB
1	0.34	0.19	0.13	-0.01	18.7	0.33
2	0.49	0.32	1.32	-0.31	9.8	0.48
3	0.50	0.32	1.38	2.89	60.4	0.50
4	0.56	0.37	2.61	2.42	53.1	0.56

The mounting flange must withstand the bending produced by F_{zmax}.



$$M_c = 0.25 \text{ inch} * F_{zmax} = 0.25 * 500 = 125 \text{ in-lb}$$

then the stress

$$\sigma_{bend} = 6 * M_c / b\, t^2$$
$$= 6 * 125 / (0.973 * (0.625)^2)$$

$= 1973$ psi

Shear stress is also present.

$$\sigma_{shear} = F_{zmax} / A_{shear}$$
$$= 500 / (0.973 * 0.625) = 822 \text{ psi}$$

The combined stress is then

$$\sigma_{max} = ((\sigma_{bend})^2 + 3 * (\sigma_{shear})^2)^{1/2}$$
$$= ((1973)^2 + 3 * (822)^2)^{1/2} = 2435 \text{ psi}$$

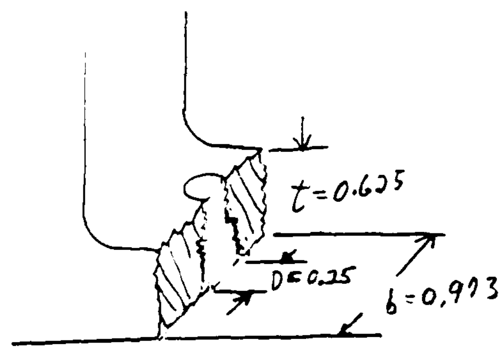
The margin of safety is then

$$MS_{ubend} = 42\text{ E+03} / (2 * 2435) - 1 = 7.6.$$

Checking for yield,

$$MS_{ybend} = 35\text{ E+03} / (1.25 * 2435) - 1 = 10.5.$$

The flange geometery for tension failure is



The ultimate strength for tension tear out is

$$F_u = \sigma_{tu} * A_t$$

where $A_t = (b - D) * t$

$$= (0.973 - 0.25) * 0.625 = 0.45 \text{ in}^2$$

$$P_u = 42\text{ E+03} * 0.45 = 1.9\text{ E+04 lb}$$

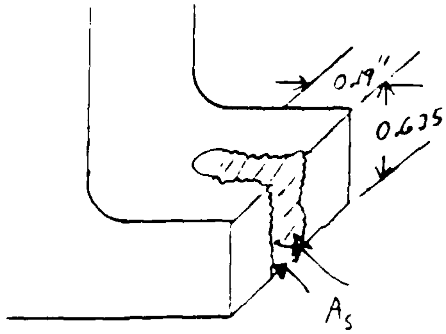
The maximum shear load was 52.1 lb. (While this resultant shear does not all act to produce the tension tear out, using the total shear load is the conservative approach.)

Thus $MS_{utear} = 1.9\text{ E+04} / (2 * 52.1) - 1 = 181.$

Checking for yield by using as safety factor of 1.25,

$$MS_{ytear} = 35\text{ E+03} * 0.49 / (1.25 * 52.1) - 1 = 242$$

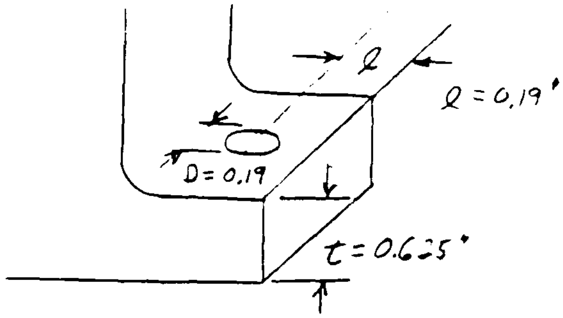
The geometry for shear tear out is



$A_s = 2 * (0.19 * 0.625) = 0.24 \text{ in}^2$

Then $MS_{shear} = 27 E+03 * 0.24 / (2 * 52.1) - 1 = 61$

Checking for the bearing strength



$e/D = 0.19 / 0.19 = 1.0$

$D/t = 0.19 / 0.625 = 0.30$

yield a $k_{bru} = 0.8$

$A_{br} = D * t = 0.19 * 0.188 = 0.036 \text{ in}^2$

Then $P_{bru} = k_{bru} * \sigma_t * A_{bru}$

$= 0.8 * 42 E+03 * 0.036 = 1200$

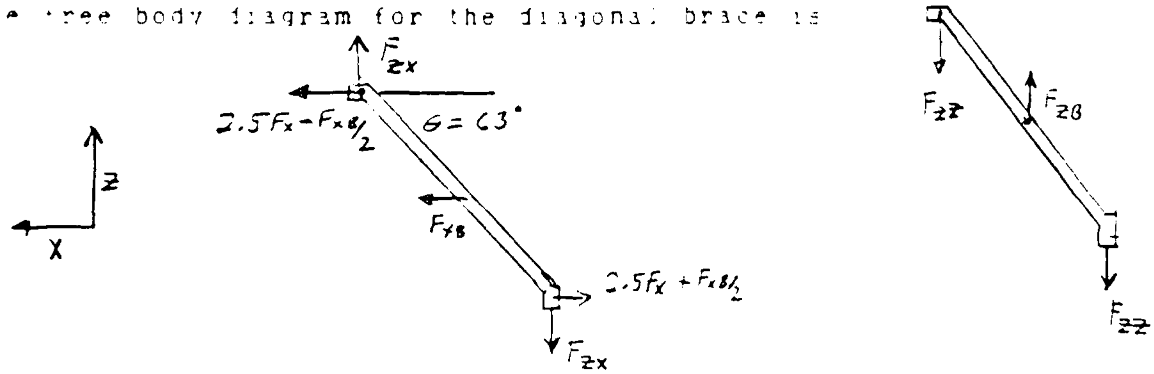
$MS_{bru} = 1200 / (2 * 52.1) - 1 = 10.5$

Margin of safety summary for filter mounting flange:

LOAD CASE	MS BEND TENS	YIELD	MS TENS TEAR TENS	YIELD	MS SHEAR	MS BEARING
1	7.6	10.5	181	242	61	10.4
2	16.7	22.6	124	166	41	7.0
3	17.2	23.2	711	948	240	44.4
4	26.6	35.7	626	835	211	39.0

5.5 Diagonal braces

The free body diagram for the diagonal brace is



Since the upper end A is simply pinned, reaction forces at this end sum vectorialy to produce an axial force in the brace. Due to the X-axis acceleration,

$$F_{axialx} = F_{zx} / \sin(63^0) = 471.0/.89 = 528 \text{ lb}$$

Z-axis acceleration also produces loading in the brace,

$$F_{zz} = F_{zb} / 2 = W_{brace} * ZG / 2$$

$$= 0.28 * 2.7 / 2 = .38 \text{ lb}$$

The corresponding axial force is

$$F_{axialz} = F_{zz} / \sin(63^0) = 0.42 \text{ lb}$$

Y-axis acceleration does not produce any axial loading in the brace.

$$F_{axial} = F_{axialx} + F_{axialz} = 529 \text{ lb.}$$

The 10-32, type A286, bolt at the upper end, A, has shear loading only.

$$P_{us} = 91 E{+}03 * A_s = 91 E{+}03 * 0.02 = 1820 \text{ psi}$$

Thus the margin of safety is

$$MS_s = P_{us} / (2 * F_{axial}) - 1$$

$$= 1820 / (2 * 529) - 1 = 0.72$$

The tenison loading on bolt A comes from Y-axis accelerations on the weight of the brace.

$$F_{yy} = (W_{brace} / 2) * YG = (0.28/2) * 4 = 0.56 \text{ lb}$$

For all load cases this tensile loading is so small that the total tension loading is essentially that of due to the bolt preload.

The expected tightening torque is 30 to 34 in-lb. Thus for minimum friction (u =0.0784) and maximum torque (using the "nut factors" from reference 2 6)

$$F_{maxPRE} = 44.4 * 34 = 1510 \text{ lb}$$

For an average friction value of u = 0.1,

$$F_{avgPRE} = 36.5 * 34 = 1241 \text{ lb}$$

and for the maximum friction value and minimum torque,

$$F_{minPRE} = 28.8 * 30 = 864 \text{ lb}.$$

This minimum preload will be reduced an additional 30 % for the slipping analysis which will follow

The external interface load is so small that a joint diagram is not needed. For the maximum preload

$$MS_t = 140\ E{+}03 * 0.02 / (2 * 0 + 1510) - 1 = 0.85$$

and for the average preload

$$MS_t = 140\ E{+}03 * 0.02 / (2 * 0 + 1.4 * 1241) - 1 = 0.61$$

The shear loading a bolt A is large enough that "slipping" at this joint will occur. But from the above analysis the bolt itself can absorb the load.

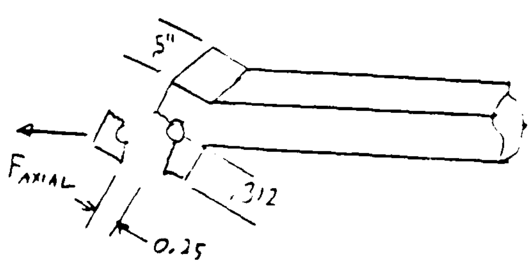
Since these mounting bolts sustain both tension and shear, the effect of both stress are combined through calculation of the corresponding "factor of limit load capablilities" which are then combined.

$$FLC_t = MS_t + 1 = 0.85 + 1 = 1.85$$

and $$FLC_s = MS_s + 1 = 1.72 + 1 = 1.72.$$

Then $$FLC_{comb} = \frac{1}{[(1/1.85)^2 + (1/1.72)^2]^{1/2}} = 1.26$$

The flange geometery for tension failure at bolt connection A



The ultimate strength for tension tear out is

$$P_u = \sigma_{tu} * A_t$$

where $A_t = (0.25 - D/2) * t + (0.312 - D/2) * t$

$$= (0.25 + 0.312 - 0.25) * 0.5 = 0.16 \text{ in}^2$$

$$P_u = 42\text{ E+03} * 0.16 = 6552 \text{ lb}$$

The maximum axial load was 529 lb.

Thus $MS_{utear} = 6552 / (2 * 529) - 1 = 5.2.$

Checking for yield by using as safety factor of 1.25,

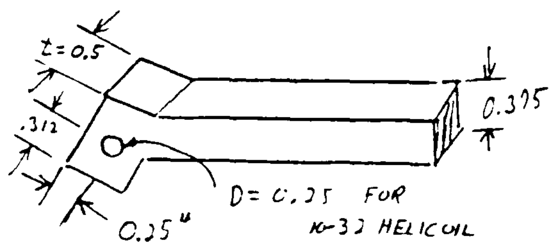
$$MS_{ytear} = 35\text{ E+03} * 0.16 / (1.25 * 529) - 1 = 7.3$$

The geometry for shear tear out is essentially the same as for tension failure.

$$A_s = A_t = 0.16 \text{ in}^2$$

Then $MS_{shear} = 27\text{ E+03} * 0.16 / (2 * 529) - 1 = 3.0.$

Checking for the bearing strength:



$A = 0.5 * 0.375$
$= 0.1875 \text{ in}^2$
$r = \dfrac{0.375}{\sqrt{12}} = 0.108 \text{ in}$

$$e/D = 0.25 / 0.25 = 1.0$$

$$D/t = 0.25 / 0.5 = 0.5$$

yield & $k_{bru} = 0.8$

$$A_{br} = D * t = 0.25 * 0.5 = 0.125 \text{ in}^2$$

Then $P_{bru} = k_{bru} * \sigma_t * A_{bru}$

$$= 0.8 * 42\text{ E+03} * 0.125 = 4200$$

$$MS_{bru} = 4200 / (2 * 529) - 1 = 3.0$$

Margin of safety summary, bolt A of diagonal brace

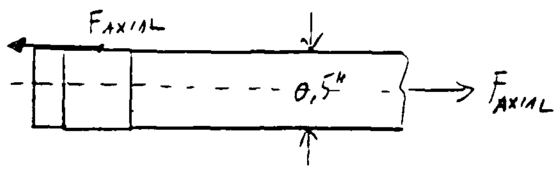
LOAD CASE	BOLT A TENS MAX	BOLT A TENS AVG PRE	BOLT A SHEAR	MS COMB	TEAR OUT TENS	TEAR OUT YIELD	SHEAR OUT	BEARING
1	0.85	0.61	0.7	0.26	5.2	7.3	3.0	3.0
2	0.85	0.61	9.6	0.83	37.1	49.9	23.5	23.4
3	0.85	0.61	8.7	0.82	34.1	45.8	21.5	21.5
4	0.85	0.61	5.3	0.78	21.8	29.5	13.7	13.6

The main body of the diagonal brace must withstand tension, bending, and compression buckling

$$A_t = (0.375 * 0.5) = 0.188 \text{ in}^2$$

Then $\sigma_t = F_{axial} / A_t = 529 / 0.188 = 2819$ psi

The reaction at bolt A is applied to one side of the brace, so a bending moment is produced in the brace.



$$M_o = F_{axial} * 0.25 \text{ inch} = 529 * 0.25 = 132 \text{ in-lb}$$

Then $\sigma_{bend} = 6 * M_o/(0.375 * (0.5)^2)$

$$= 6 * 132/(0.375 * (0.5)^2) = 8456 \text{ psi}$$

The total tensile stress is then

$$\sigma_{tens} = \sigma_t + \sigma_{bend} = 2818 + 8456 = 11274 \text{ psi}$$

Then $MS_{ut} = 42 \text{ E+03} /(2 * 11274) - 1 = 0.86$

Checking for yield

$$MS_{yt} = 35 \text{ E+03} /(1.25 * 11274) - 1 = 1.5$$

When the brace undergoes compression buckling may occur. For the diagonal brace the cross section area is 0.1875 in^2 and the radius of gyration is 0.108 inch. For the compression load the bolt A end is simply pinned. The end at bolt B which attaches to the baseplate will also be assumed to be simply pinned as a conservative case. Then from Euler's formula for column buckling, the critcal buckling load is

$$P_c = (PI)^2 * E * A * (r/L)^2$$

$$= (3.14)^2 * 1 E+07 * 0.1875 * (0.108/13.3)^2$$

$$= 1220 \text{ lb}$$

The margin of safety for buckling is then

$$MS_{buck} = 1220 / (2 * F_{axial}) - 1$$

$$= 1220 / (2 * 529) - 1 = 0.15$$

(If the end at the baseplate were considered fully clamped the margin of safety would be 1.35)

Margin of safety summary for diagonal brace body:

LOAD CASE	MS TENSION ULTIMATE	MS TENSION YIELD	MS BUCKLING
1	0.86	1.48	0.15
2	10.5	14.3	6.1
3	9.5	13.1	5.5
4	5.9	8.2	3.3

At bolt connection B of the brace to the baseplate, both tension and shear is carried by the bolt. From the previous free body diagram, the tension in bolt B is

$$F_{zmax} = F_{zx} + F_{zz}$$

$$= 470.5 + 0.38 = 470.9 \text{ lb}$$

Part of this interface load is taken by the mounting flange of the brace. The joint geometry fits case 2 of reference 2.5

$$A_j = PI * (D_w^2 - D_h^2) / 4 + PI * (D_j/D_w - 1) * (D_w * L_j/5 + L_j^2/100) / 8$$

$$= 3.14 * (0.375^2 - 0.19^2) / 4 + 3.14*(0.375/0.375 - 1)*(0.375 * 1.3 /5 + 1.3^2/100) / 8$$

$$= 0.103 \text{ in}^2$$

The fraction of the external interface load taken by the mounting flange is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$

$$= \frac{0.103 * 1\ E{+}07}{0.018 * 2.91\ E{+}07 + 0.103 * 1\ E{+}07} = 0.7$$

The tie down bolts are made "captive" by reducing the shank diameter to that of the thread root. Thus the effective stress area will be slightly smaller than the typical handbook value for a 10-32 screw

$$A_{cap} = PI * dia^2 / 4 = 3.14 * (0.152)^2 / 4 = 0.018\ in^2$$

Thus $P_{us} = 91\ E{+}03 * 0.018 = 1651$ lb

and $P_{ut} = 140\ E{+}03 * 0.018 = 2540$ lb

The expected tightening torque is 30 to 34 in-lb. Thus for minimum friction (u =0.0784) and maximum torque (using the "nut factors" from reference 2.6)

$$F_{maxPRE} = 44.4 * 34 = 1510 \text{ lb}$$

For an average friction value of u = 0.1,

$$F_{avgPRE} = 36.5 * 34 = 1241 \text{ lb}$$

and for the maximum friction value and minimum torque,

$$F_{minPRE} = 28.8 * 30 = 864 \text{ lb.}$$

This minimum preload will be reduced an additional 20 % for the gapping analysis which will follow.

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (2540)/(2 * (1-0.7) * 471 + 1510) - 1 = 0.42$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension

$$MS_t = (2540)/(2 * (1-0.7) * 471 + 1.4 * 1241) - 1 = 0.26$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the initial preload, a preload of 0.8 * 864 = 691 lb. If the mounting flange fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 691 / (2 * 0.7 * 471) - 1 = 0.05$$

The shear loading comes from the X-axis and Y-axis accelerations

From the previous free body diagram,

$$F_{sx} = 5 * F_x/2 + F_{xb}/2$$

$= 5 * 91.8 / 2 + 10.4 / 2 = 235$ lb

$F_{sy} = W_{brace} * YG / 2$ bolts

$= 0.28 * 4.0 / 2 = 0.56$ lb

and $F_{sxy} = ((F_{sx})^2 + (F_{sy})^2)^{1/2} = 235$ lb

This large shear force will produce slipping between the brace and the bottom plate. However bolt B can absorb the shear load.

$MS_s = (91E+03 * 0.018) / (2 * 235) - 1 = 2.5$

Since these mounting bolts sustain both tension and shear, the effect of both stress are combined through calculation of the corresponding "factor of limit load capablilities" which are then combined.

$FLC_t = MS_t + 1 = 0.42 + 1 = 1.42$

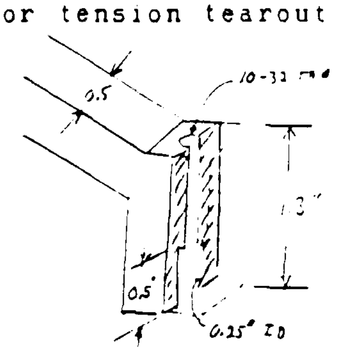
and $FLC_s = MS_s + 1 = 2.5 + 1 = 3.5$

Then $FLC_{comb} = \dfrac{1}{[(1/1.42)^2 + (1/3.5)^2]^{1/2}} = 1.31$

Margin of safety summary, diagonal brace mounting bolt B

LOAD CASE	BOLT B OF BRACE 2.0 SF	2.0 & 1.4	MS GAP	BOLT SHEAR	MS COMB
1	0.42	0.26	0.05	3.4	0.31
2	0.63	0.42	5.45	20.5	0.63
3	0.63	0.42	4.93	20.6	0.62
4	0.60	0.40	2.86	11.9	0.59

The flange geometry for tension tearout at bolt B of the brace is



The ultimate strength for tension tear out is

$P_u = \sigma_{tu} * A_t$

where $A_t = 2 * (0.8 * (0.5 - 0.19) + 0.5 * (0.5 - 0.25))$

$$= 0.75 \text{ in}^2$$

$$P_u = 42 \text{ E+03} * 0.75 = 3.1 \text{ E+04 lb}$$

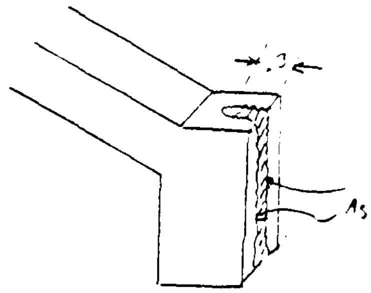
The maximum shear load was 235 lb (While this resultant shear does not all act to produce the tension tear out, using the total shear load is the conservative approach.)

Thus $MS_{utear} = 3.1 \text{ E+04} / (2 * 235) - 1 = 66$

Checking for yield by using as safety factor of 1.25,

$$MS_{ytear} = 35 \text{ E+03} * 0.75 / (1.25 * 235) - 1 = 88$$

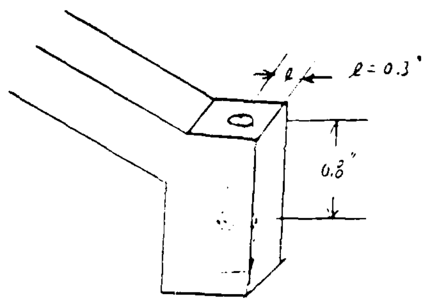
The geometry for shear tear out is



$$A_s = 2 * (0.3 * 1.3) = 0.78 \text{ in}^2$$

Then $MS_{shear} = 27 \text{ E+03} * 0.78 / (2 * 235) - 1 = 44.$

Checking for the bearing strength.



$$e/D = 0.3 / 0.19 = 1.6$$

$$D/t = 0.19 / 0.8 = 0.24$$

yield a $k_{bru} = 1.5$

$$A_{br} = D * t = 0.19 * 0.8 = 0.15 \text{ in}^2$$

Then $P_{bru} = k_{bru} * \sigma_t * A_{bru}$

$= 1.5 * 42\ E{+}03 * 0.15 = 9576$

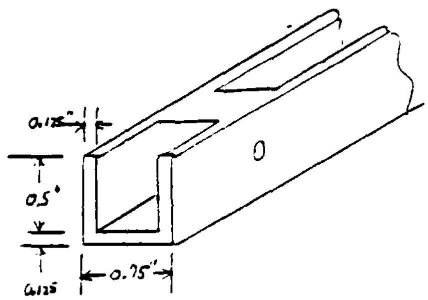
$MS_{bru} = 9576 / (2 * 235) - 1 = 19$

Margin of safety summary, brace mounting flange at bolt B

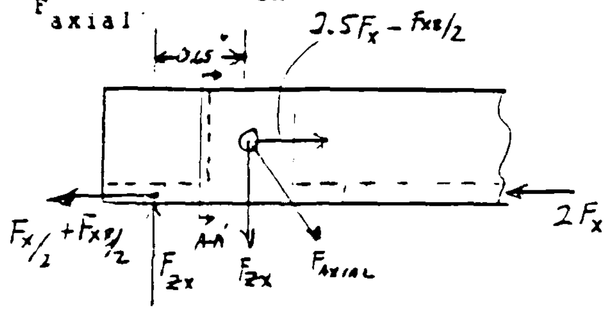
LOAD CASE	MS TENS TENS	TEAR YIELD	MS SHEAR	MS BEARING
1	66	88	44	19
2	409	546	275	124
3	412	549	276	125
4	247	329	165	75

5.7 Top filter straps

The top filter straps tie the top corners of the filter modules together. The geometry is a U-channel with a solid area where the diagonal braces attach.



The cross section at A-A′ has shear loading produced by F_{zx}, tension loading produced by $(F_x/2 + F_{xb}/2)$, bending stress produced by F_{zx} and bending stress by the lateral component of F_{axial}.



DUE TO
X-AXIS ACCELERATIONS

The shear produced stress is

$$\sigma_s = F_{zx} / A_{chan}$$

$$= 471 / (0.75 * 0.125 + 2 * (0.125 * 0.5))$$

$$= 471 / (0.219) = 2151 \text{ psi}$$

(This ignores the weight of the channel itself.)

Shear is also produce by the torque on the channel by F_{zx}:

$$T_z = F_{zx} * 0.375 = 471 * 0.375 = 177 \text{ in-lb}$$

From section 5.9, the worst case loading from the cross strap to the IF module is

$$F_{yy} = 11 \text{ lb} * YG / (4) = 11 \text{ lb}$$

which also produces a torque

$$T_y = F_{yy} * 1.3125 \text{ inch} = 14.4 \text{ in-lb}$$

for a total torque of 191 in-lb. One half of this torque is resisted by the channel on each side of the attachment. The torque produced stress is appoximated by considering the channel cross section area to be distributed at an effective radius of 0.375 inch.

$$\sigma_s = (191 / 2) / (A_{chan} * 0.375)$$

$$= (191 / 2) / (0.219 * 0.375) = 1163 \text{ psi}$$

The tension produced stress is

$$\sigma_t = (F_x/2 + F_{xb}/2) / A_{chan}$$

$$= (91.8 / 2 + 10.4 / 2) / (0.219) = 233 \text{ psi}$$

The bending stress produced by the moment from F_{zx} depends upon the moment of inertia of the U-channel around the z centroid.

$$\sigma_{zb} = M_{oz} * z_c / I_z$$

$$= (F_{zx} * 0.3125) * z_c / I_z$$

$$= 471 * 0.3125 * 0.384 / 7.96E-03 = 7093 \text{ psi}$$

The bending stress produced by the moment from F_{axial} depends upon the moment of inertia of the U-channel around the y centroid.

The moment produced by the lateral force from the diagonal brace is resisted by the channel on either side of the attachment point.

$$M_{oy} = (F_{axial} * \text{Cos}(63) * (0.65-0.3125)) / 2 \text{ sides}$$

$$= (529 * 0.45 * 0.3375) / 2 = 40.2 \text{ in-lb}$$

Thus $\sigma_{yb} = M_{oy} * y_c / I_y$

$$= 40.2 * 0.375 / 1.676E-02 = 906 \text{ psi}$$

Adding the three components of tensile stress and then combining with the shear stress

$$\sigma_{tot} = ((233 + 7093 + 903)^2 + 3 * (2151+1163)^2)^{1/2}$$

$$= 1.0E+04 \text{ psi}$$

The ultimate margin of safety is then

$$MS_{tu} = 42 E+03 /(2 * 1.0E+04) - 1 = 1.1$$

Checking for yield with a safety factor of 1.25

$$MS_{ty} = 35 E+03 /(1.25 * 1.0E+04) - 1 = 1.8$$

Margin of safety summary, U-channel bending at diagonal brace

LOAD CASE	SECTION A-A′ TENS	YIELD
1	1.1	1.8
2	7.7	10.7
3	11.3	15.3
4	6.7	9.3

The captive screws that attach the U-channel to the top of each filter are full diameter, type A286, 8-32 screws.

Thus $P_{us} = 91 E+03 * 0.014 = 1274 \text{ lb}$

and $P_{ut} = 140 E+03 * 0.014 = 1960 \text{ lb}$

The expected tightening torque is 24 to 26 in-lb. Thus for minimum friction ($u = 0.0784$) and maximum torque (using the "nut factors" from reference 2.6)

$$F_{maxPRE} = 47.6 * 26 = 1238 \text{ lb}$$

For an average friction value of $u = .1$,

$$F_{avgPRE} = 39.5 * 26 = 1027 \text{ lb}$$

and for the maximum friction value and minimum torque,

$$F_{minPRE} = 31.4 * 24 = 754 \text{ lb}$$

This minimum preload will be reduced an additional 20 % for the gapping analysis which will follow.

To determine the division of the external interface load between the top brace and the bolt, the effective joint area of the brace is first calculated.

The joint geometry fits case 2 of reference 2.5

$$A_j = PI * (D_w^2 - D_h^2) / 4 + PI * (D_j/D_w - 1) * (D_w * L_j/5 + L_j^2/100) / 8$$

$$= 3.14 * (0.375^2 - 0.164^2) / 4 + 3.14*(0.75/0.375 - 1)*(0.375 * 0.125 /5 + 0.125^2/100) / 8$$

$$= 0.096 \text{ in}^2$$

The fraction of the external interface load taken by the top brace is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$

$$= \frac{0.096 * 1 E+07}{0.014 * 2.91 E+07 + 0.096 * 1 E+07} = 0.7$$

The maximum filter attachment screw load was determined from the free body diagrams of section 5.6.

$$F_{zmax} = 500 \text{ lb}$$

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (1960)/(2 * (1-0.7) * 500 + 1238) - 1 = 0.28$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension.

$$MS_t = (1960)/(2 * (1-0.7) * 500 + 1.4 * 1027) - 1 = 0.13$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the intial preload, the preload is 0.8 * 754 = 603 lb. If the mounting flange fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 603 / (2 * 0.7 * 500) - 1 = -0.14$$

This negative margin of safety occurs only for load case #1. The free body diagram of section 5.4 assumed for simplicity that each filter was simply pinned at its attachment to the baseplate and that the resisting diagonal brace was perfectly rigid. The actual case is a statically indeterminate case, in that the filter flanges are clamped to the baseplate (no rotation) and the diagonal brace will have some strain displacement. Thus the actual loading at the top strap will be much less and a positive margin of safety should result.

From the free body diagrams of section 5.6 the maximum shear load produced by x-axis accelerations is

$$F_{sx} = F_x/2 + F_{xb}/2$$

$$= 91.8 / 2 + 10.4 /2 = 51.1 \text{ lb}$$

The shear produced by y-axis accelerations is

$$F_{sy} = W_{brace} * YG / 2 \text{ ends}$$

$$= 0.28 * 4 / 2 = 0.56 \text{ lb}$$

and $F_{sxy} = ((F_{sx})^2 + (F_{sy})^2)^{1/2} = 51.1$ lb

To avoid slipping between the top strap and the top of the filter the maximum possible shear force is calculated using a friction value of 0.15 between the members and the relaxed minimum bolt preload

$$F_{fric} = 0.15 * 603 \text{ lb} = 90.5 \text{ lb}$$

Thus $MS_{slip} = 90.5 / (2 * 51.1) - 1 = -0.11$

The margin of safety for load case #1 is negative; however, the mounting bolts can absorb the shear load.

$$MS_s = (91E+03 * 0.014) / (2 * 51.1) - 1 = 11.5.$$

Since these mounting bolts sustain both tension and shear the effect of both stress are combined through calculation of the corresponding "factor of limit load capabilities" which are then combined.

$$FLC_t = MS_t + 1 = 0.28 + 1 = 1.28$$

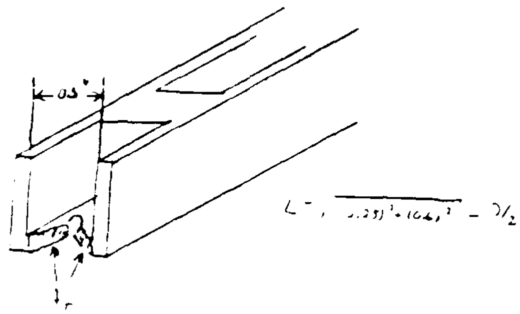
and $FLC_s = MS_s + 1 = 11.5 + 1 = 12.5.$

Then $FLC_{comb} = \dfrac{1}{[(1/1.28)^2 + (1/12.5)^2]^{1/2}} = 1.27$

Margin of safety summary, top strap to filter top bolts:

LOAD CASE	TOP STRAP<-->FILT 2.0 SF	2.0 & 1.4	MS GAP	MS SLIP	BOLT SHEAR	MS COMB
1	0.28	0.13	-0.14	-0.11	11.5	0.27
2	0.42	0.24	0.76	3.9	68	0.42
3	0.42	0.24	0.81	4.5	76	0.42
4	0.47	0.28	1.74	2.3	45	0.47

The geometry for tension tear out is



The ultimate strength for tension tear out is

$$P_u = \sigma_{tu} * A_t$$

where $A_t = 2*[((0.25)^2 + (0.6)^2)^{1/2} - D/2] * t$

$= 2 * [0.65 - 0.375/2] * 0.125 = 0.116 \text{ in}^2$

$P_u = 42 \text{ E+03} * 0.116 = 4856 \text{ lb}$

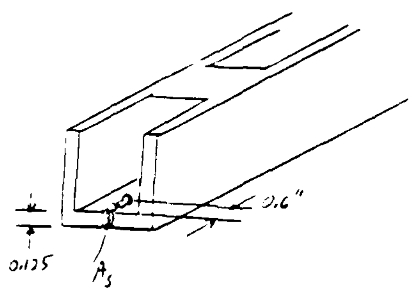
The maximum shear load was 51.1 lb. (Not all of this resultant shear load acts to produce the tension failure, but using the total is the conservative approach.)

Thus $MS_{utear} = 4856 / (2 * 51.1) - 1 = 47.$

Checking for yield by using as safety factor of 1.25,

$MS_{ytear} = 35 \text{ E+03} * 0.116 / (1.25 * 51.1) - 1 = 62$

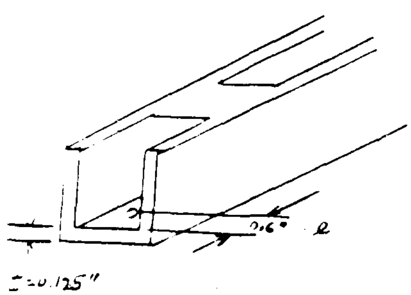
The geometry for shear tear out is



$A_s = 2 * (0.125 * 0.6) = 0.15 \text{ in}^2$

Then $MS_{shear} = 27 \text{ E+03} * 0.15 / (2 * 51.1) - 1 = 39$

Checking for the bearing strength



$$e/D = 0.6 / 0.375 = 1.6$$

$$D/t = 0.375 / 0.15 = 2.5$$

yield a $k_{bru} = 1.5$

$$A_{br} = D * t = 0.375 * 0.15 = 0.0563 \text{ in}^2$$

Then $P_{bru} = k_{bru} * \sigma_t * A_{bru}$

$$= 1.5 * 42 \text{ E+03} * 0.0563 = 3544$$

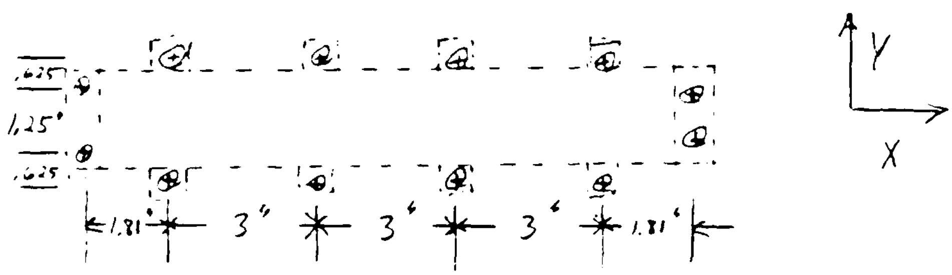
$$MS_{bru} = 3544 / (2 * 51.1) - 1 = 34$$

Margin of safety summary, top strap attachment to filters:

LOAD CASE	TENS ULT	TEAR YIELD	SHEAR OUT	BEARING
1	47	62	39	34
2	263	351	219	192
3	292	390	243	213
4	175	234	146	128

5.8 IF module

The IF module is a double-width module with twelve 10-32 attachment bolts. Its weight is 11 lb. The CG is 6.625 inches above the center of the attachment screws. Bracing from the top of the IF module is provided over to the top of the filter modules. This results in a statically indeterminate case which is difficult for hand analysis. The following analysis will show that the IF module attachment to the baseplate by the mounting feet alone is sufficient. The actual loading on the mounting feet will be less than the values which result from this conservative analysis approach.



The tensile stress on each bolt due to Z-axis accelerations is

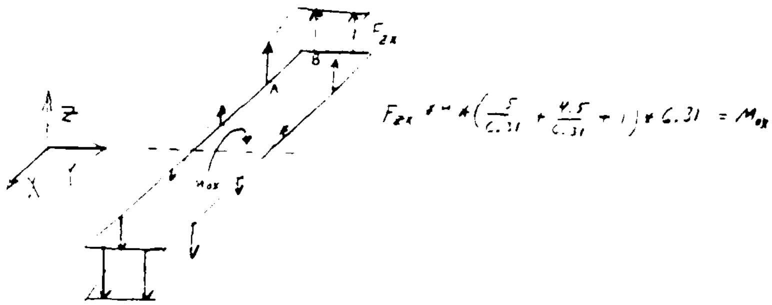
$$F_{zz} = W_{IF} * ZG / 12 \text{ bolts}$$

$$F_{zz} = 11 \text{ lb} * 2.7 / 12 \text{ bolts} = 2.5 \text{ lb}$$

The X-axis acceleration produces a moment, M_{ox}

$$M_{ox} = 6.625 \text{ inch} * 11 \text{ lb} * XG = 2701 \text{ in-lb}$$

This is countered by

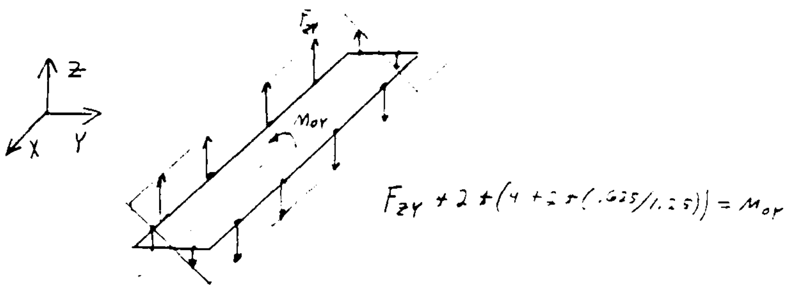


$$F_{zx} = M_{ox}/(4 * (1.95) * 6.31) = 54.9 \text{ lb}$$

For accelerations along the Y-axis, the moment produced is

$$M_{oy} = 6.626 \text{ inch} * 11 \text{ lb} * YG = 291.5 \text{ in-lb}$$

This is countered by



$$F_{zy} = M_{oy}/(2*(4+2*0.5)) = 29.2 \text{ lb}$$

The bolt at position A suffers the maximum loading for all load cases except load case #1.

$$F_{Amax} = F_{zx} * 4.5/6.31 + F_{zy} + F_{zz}$$

$$= 54.9 * 4.5/6.31 + 29.2 + 2.5 = 70.8 \text{ lb}$$

LOAD CASE	F_{Amax}	F_{Bmax}
1	70.8 lb	71.9
2	208.2	116.0
3	81.8	69.7
4	33.5	29.0

As expected by the geometry, load case #2 is the worst case

These tie down bolts are made "captive" by reducing the shank diameter to that of the thread root. Thus the effective stress area will be slightly smaller than the typical handbook value for a 10-32 screw

$$A_{cap} = PI * dia^2 / 4 = 3.14 * (0.152)^2 / 4 = 0.018 \ in^2$$

Thus $P_{us} = 91 E+03 * 0.018 = 1651 \ lb$

and $P_{ut} = 140 E+03 * 0.018 = 2540 \ lb$

The expected tightening torque is 30 to 34 in-lb. Thus for minimum friction (u =0.0784) and maximum torque (using the "nut factors" from reference 2.6)

$$F_{maxPRE} = 44.4 * 34 = 1510 \ lb$$

For an average friction value of u = .1.

$$F_{avgPRE} = 36.5 * 34 = 1241 \ lb$$

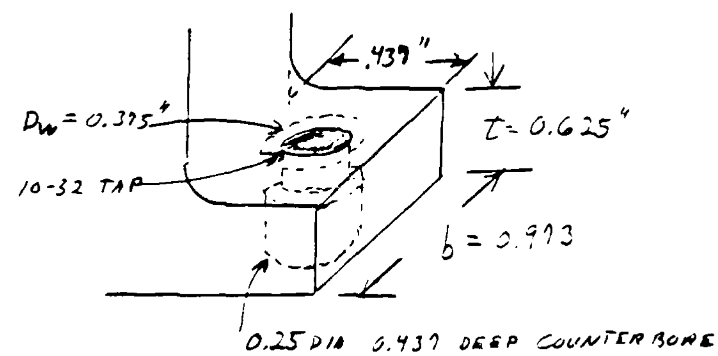
and for the maximum friction value and minimum torque,

$$F_{minPRE} = 28.8 * 30 = 864 \ lb$$

This minimum preload will be reduced an additional 20 % for the gapping analysis which will follow.

To determine the division of the external interface load between the mounting flange and the bolt, the effective joint area of the flange must be calculated.

For the mounting feet along the side of the IF module.



The joint geometry fits case 2 of reference 2.5

$$A_j = PI * (D_w^2 - D_h^2) / 4 + PI * (D_j / D_w - .1) * (D_w * L_j / 5 + L_j^2 / 100) / 8$$

$$= 3.14 * (0.375^2 - 0.25^2) / 4 + 3.14*(0.437/0.375 - .1)*(0.375 * 0.625 / 5 + 0.625^2/100) / 8$$

$$= 0.083 \ in^2$$

The mounting flange at the end of the module has two mounting bolts but is more than twice the width. The effective flange area is essentially the same.

The fraction of the external interface load taken by the mounting flange is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$

$$= \frac{0.083 * 1\ E+07}{0.018 * 2.91\ E+07 + 0.083 * 1\ E+07} = 0.61$$

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (2540)/(2 * (1-0.61) * 71.9 + 1510) - 1 = 0.62$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension.

$$MS_t = (2540)/(2 * (1-0.61) * 71.9 + 1.4 * 1241) - 1 = 0.42$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the intial preload, a preload of 0.8 * 864 = 691 lb. If the mounting flange fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 691 / (2 * 0.66 * 71.9) - 1 = 6.9.$$

The shear loading comes from the X-axis and Y-axis loading.

$$F_{sx} = W_{IF} * XG / 12 \text{ bolts}$$

$$= 11 * 37.1 / 12 = 34.0 \text{ lb}$$

$$F_{sy} = W_{IF} * YG / 12 \text{ bolts}$$

$$= 11 * 4.0 / 12 = 3.7 \text{ lb}$$

and $F_{sxy} = ((F_{sx})^2 + (F_{sy})^2)^{1/2} = 34.2 \text{ lb}$

The center of gravity is directly over the center of the screws, so that no eccentric loading occurs.

To avoid slipping between the IF module and the FEB baseplate, the maximum possible shear force is calculated using a friction value of 0.15 between the members and the relaxed minimum bolt preload.

$$F_{fric} = 0.15 * 691 \text{ lb} = 104 \text{ lb}$$

Thus $MS_{slip} = 104 / (2 * 34.2) - 1 = 0.5$

If the holding friction was not sufficient, the mounting bolts can absorb the shear load

$$MS_s = (91E+03 * 0.018) / (2 * 34.2) - 1 = 23$$

Since these mounting bolts sustain both tension and shear, the effect of both stress are combined through calculation of the corresponding "factor of limit load capabliliti[e]s" which are then combined.

$$FLC_t = MS_t + 1 = 0.62 + 1 = 1.62$$

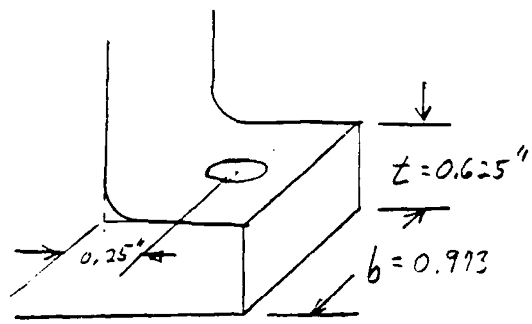
and $$FLC_s = MS_s + 1 = 23 + 1 = 24.$$

Then $$FLC_{comb} = \frac{1}{[(1/1.62)^2 + (1/24)^2]^{1/2}} = 1.62$$

Margin of safety summary, IF module mounting bolts

LOAD CASE	IF MODULE 2.0 SF	MOUNTING 2.0 & 1.4	MS GAP	MS SLIP	BOLT SHEAR	MS COMB
1	0.62	0.42	6.9	0.5	23	0.62
2	0.51	0.33	1.6	0.9	29	0.51
3	0.61	0.41	5.9	6.8	123	0.61
4	0.65	0.44	15.9	4.5	86	0.65

The mounting flange must withstand the bending produced by the Y-axis loading.



$$M_o = 0.25 \text{ inch} * F_{zmax} = 0.25 * 71.9 = 18 \text{ in-lb}$$

then the stress

$$\sigma_{bend} = 6 * M_o / b\, t^2$$
$$= 6 * 18 / (0.973 * (0.625)^2)$$
$$= 284 \text{ psi}$$

Shear stress is also present,

$$\sigma_{shear} = F_{zmax} / A_{shear}$$
$$= 71.9 / (0.973 * 0.625) = 118 \text{ psi}$$

The combined stress is then

$$\sigma_{max} = ((\sigma_{bend})^2 + 3 * (\sigma_{shear})^2)^{1/2}$$

$$= ((284)^2 + 3 * (118)^2)^{1/2} = 350 \text{ psi}$$

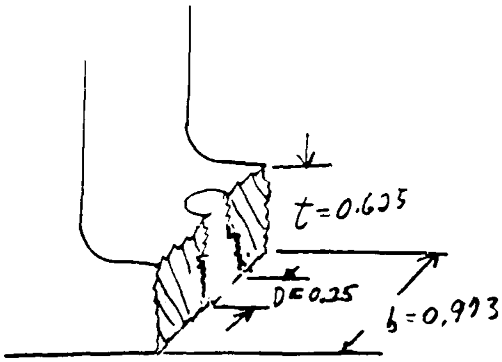
The margin of safety is then

$$MS_{ubend} = 42 \text{ E+03} / (2 * 350) - 1 = 59$$

Checking for yield,

$$MS_{ybend} = 35 \text{ E+03} / (1.25 * 350) - 1 = 79$$

The flange geometery for tension failure is



The ultimate strength for tension tear out is

$$P_u = \sigma_{tu} * A_t$$

where $A_t = (b - D) * t$

$$= (0.973 - 0.25) * 0.625 = 0.45 \text{ in}^2$$

$$P_u = 42 \text{ E+03} * 0.45 = 1.9 \text{ E+04 lb}$$

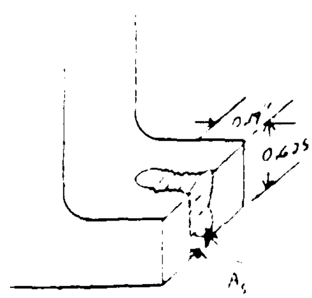
The maximum shear load was 27.1 lb. (Not all of this resultant shear load acts to produce the tension failure, but using the total is the conservative approach.)

Thus $MS_{utear} = 1.9 \text{ E+04} / (2 * 34.2) - 1 = 277$

Checking for yield by using as safety factor of 1.25,

$$MS_{ytear} = 35 \text{ E+03} * 0.45 / (1.25 * 34.2) - 1 = 369$$

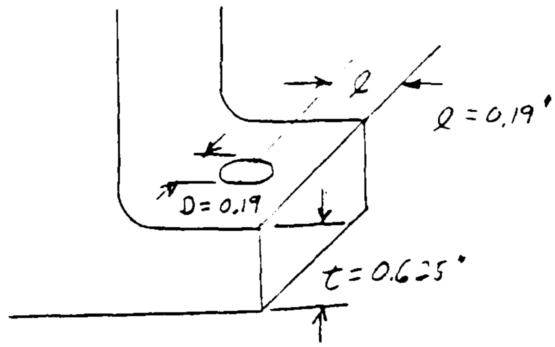
The geometry for shear tear out is



$$A_s = 2 * (0.19 * 0.625) = 0.24 \text{ in}^2$$

Then $MS_{shear} = 27 E+03 * 0.24 / (2 * 34.2) - 1 = 93.$

Checking for the bearing strength:



$$e/D = 0.19 / 0.19 = 1.0$$

$$D/t = 0.19 / 0.625 = 0.30$$

yield a $k_{bru} = 0.8$

$$A_{br} = D * t = 0.19 * 0.188 = 0.036 \text{ in}^2$$

Then $P_{bru} = k_{bru} * \sigma_t * A_{bru}$

$$= 0.8 * 42 E+03 * 0.036 = 1200$$

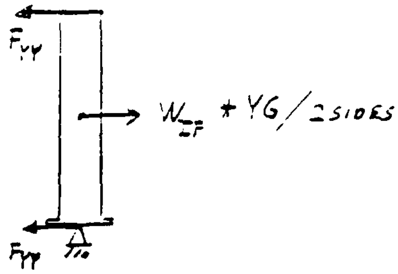
$$MS_{bru} = 1200 / (2 * 34.2) - 1 = 16.7.$$

Margin of safety summary for IF module mounting flange:

LOAD CASE	MS BEND TENS	YIELD	MS TENS TEAR TENS	YIELD	MS SHEAR	MS BEARING
1	59	79	277	369	93	16.7
2	19	25	352	469	118	21.5
3	52	69	1435	1913	484	90.5
4	128	171	1006	1341	339	63.2

5.9 Top braces to IF module

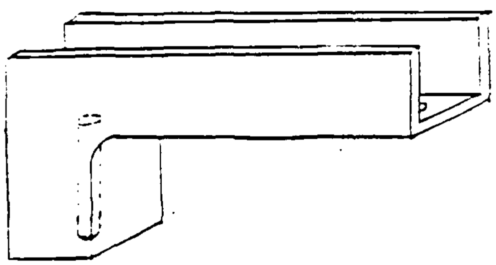
The two top braces which bridge from the top strap of the filter modules to the top of the IF module provide redundancy to the attachment of the IF module to the baseplate. The twelve feet of the IF module are sufficient to hold it in place. The actual loading of these top braces is a statically indeterminate case. As a worse case calculation of the loading in the brace, the IF module is assumed simply pinned at its base. Load case #2 is the worst case due to the physical layout of the IF module.

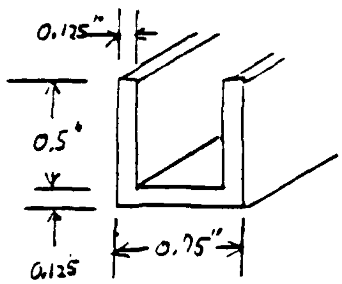


Thus F_{yy} = (W_{IF} * YG / 2 sides) / 2

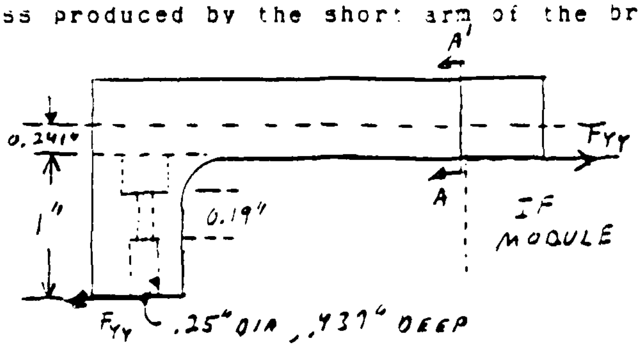
= (11 lb * 28.7 / 2) /2 = 78.9 lb (LC # 2)

The cross section of this brace is the same as the cross section of the filter top braces.





The cross section at A-A' has tension loading produced by F_{yy} and bending stress produced by the short arm of the brace.



The tension produced stress is

$$\sigma_t = F_{yy}' / A_{chan}$$

$$= 78.9 / (0.219) = 361 \text{ psi}$$

The bending stress produced by the moment from F_{yy} depends upon the moment of inertia of the U-channel around the z centroid

$$\sigma_{zb} = M_{oz} * z_c / I_z$$

$$= (F_{yy} * (1.0 + 0.241) * z_c / I_z$$

$$= 78.9 * 1.241 * 0.384 / 7.96E-03 = 4730 \text{ psi}$$

The total tensile stress is then

$$\sigma_{tot} = 361 + 4730 = 5091$$

The ultimate margin of safety is then

$$MS_{tu} = 42\ E+03 /(2 * 5091) - 1 = 3.1$$

Checking for yield with a safety factor of 1.25

$$MS_{ty} = 35\ E+03 /(1.25 * 5091) - 1 = 4.5$$

Margin of safety summary for top brace to IF module body

LOAD CASE	TENSION ULT	YIELD
1	28.6	38.5
2	3.1	4.5
3	28.6	38.5
4	48.4	64.8

The captive screws that attach the U-channel to the top of the IF module are full diameter, type A286, 8-32 screws.

Thus $P_{us} = 91\ E+03 * 0.014 = 1274$ lb

and $P_{ut} = 140\ E+03 * 0.014 = 1960$ lb

The expected tightening torque is 24 to 26 in-lb. Thus for minimum friction (u =0.0784) and maximum torque (using the "nut factors" from reference 2.6)

$$F_{maxPRE} = 47.6 * 26 = 1238 \text{ lb}$$

For an average friction value of u = .1,

$$F_{avgPRE} = 39.5 * 26 = 1027 \text{ lb}$$

and for the maximum friction value and minimum torque.

$$F_{minPRE} = 31.4 * 24 = 754 \text{ lb}$$

This minimum preload will be reduced an additional 20 % for the gapping analysis which will follow.

To determine the division of the external interface load between the top brace and the bolt, the effective joint area of the brace is first calculated.

The joint geometry fits case 2 of reference 2.5

$$\begin{aligned} A_j &= PI * (D_w^2 - D_h^2) / 4 \\ &\quad + PI * (D_j/D_w - .1) * (D_w * L_j/5 + L_j^2/100) / 8 \\ &= 3.14 * (0.375^2 - 0.164^2) / 4 \\ &\quad + 3.14*(0.75/0.375 - .1)*(0.375 * 0.125 /5 + 0.125^2/100) / 8 \\ &= 0.096 \text{ in}^2 \end{aligned}$$

The fraction of the external interface load taken by the top brace is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$

$$= \frac{0.096 * 1 \text{ E+07}}{0.014 * 2.91 \text{ E+07} + 0.096 * 1 \text{ E+07}} = 0.7$$

Very little Z-axis loading on these screws is expected. The only loading is due to the weight of the brace.

$$F_{zz} = W_{brace} * ZG / 2 \text{ ends}$$

$$= 0.14 \text{ lb} * 2.7 / 2 = 0.19 \text{ lb}$$

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (1960)/(2 * (1-0.7) * 0.2 + 1238) - 1 = 0.58$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension.

$$MS_t = (1960)/(2 * (1-0.7) * 0.2 + 1.4 * 1027) - 1 = 0.36$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the intial preload, the preload is 0.8 * 754 = 603 lb. If the mounting flange fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 603 / (2 * 0.7 * 0.2) - 1 = 2200$$

The maximum shear load produced by x-axis accelerations is

$$F_{sx} = W_{brace} * XG / 2 \text{ ends}$$
$$= 0.14 \text{ lb} * 6 / 2 = 0.42 \text{ lb}$$

The shear produced by y-axis accelerations is

$$F_{sy} = W_{brace} * YG / 2 \text{ ends}$$
$$= 0.14 * 28.7 / 2 = 79 \text{ lb}$$

and $F_{sxy} = ((F_{sx})^2 + (F_{sy})^2)^{1/2} = 79 \text{ lb}$

To avoid slipping between the top strap and the top of the IF module, the maximum possible shear force is calculated using a friction value of 0.15 between the members and the relaxed minimum bolt preload.

$$F_{fric} = 0.15 * 603 \text{ lb} = 90.5 \text{ lb}$$

Thus $MS_{slip} = 90.5 / (2 * 79) - 1 = -0.43$

The margin of safety for load case #2 is negative; however, the mounting bolts can absorb the shear load.

$$MS_s = (91E+03 * 0.014) / (2 * 79) - 1 = 7.1.$$

Since these mounting bolts sustain both tension and shear, the effect of both stress are combined through calculation of the corresponding "factor of limit load capablilities" which are then combined.

$$FLC_t = MS_t + 1 = 0.58 + 1 = 1.58$$

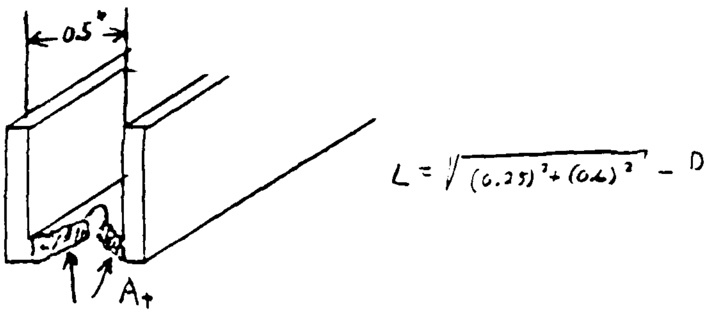
and $FLC_s = MS_s + 1 = 7.1 + 1 = 8.1.$

Then $FLC_{comb} = \dfrac{1}{[(1/1.58)^2 + (1/8.1)^2]^{1/2}} = 1.55$

Margin of safety summary, top strap to IF module top bolts:

LOAD CASE	TOP 2.0 SF	STRAP<-->IF 2.0 & 1.4	MS GAP	MS SLIP	BOLT SHEAR	MS COMB
1	0.58	0.36	2267	3.0	55	0.58
2	0.58	0.36	2267	-0.43	7	0.55
3	0.58	0.36	120	3.1	57	0.58
4	0.58	0.36	1038	5.8	95	0.58

The geometry for tension tear out is



The ultimate strength for tension tear out is

$$P_u = \sigma_{tu} * A_t$$

where $A_t = 2*[((0.25)^2 + (0.6)^2)^{1/2} - D/2] * t$

$$= 2 * [0.65 - 0.375/2] * 0.125 = 0.116 \text{ in}^2$$

$$P_u = 42 \text{ E+03} * 0.116 = 4856 \text{ lb}$$

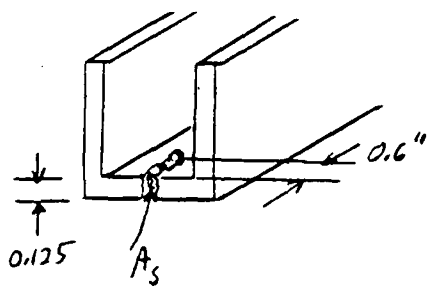
The maximum shear load was 79 lb.

Thus $MS_{utear} = 4856 / (2 * 79) - 1 = 30.$

Checking for yield by using as safety factor of 1.25,

$$MS_{ytear} = 35 \text{ E+03} * 0.116 / (1.25 * 79) - 1 = 40$$

The geometry for shear tear out is



$$A_s = 2 * (0.125 * 0.6) = 0.15 \text{ in}^2$$

Then $MS_{shear} = 27 \text{ E+03} * 0.15 / (2 * 79) - 1 = 25.$

Checking for the bearing strength:

$$e/D = 0.6 / 0.375 = 1.6$$

$$D/t = 0.375 / 0.15 = 2.5$$

yield a $k_{bru} = 1.5$

$$A_{br} = D * t = 0.375 * 0.15 = 0.0563 \text{ in}^2$$

Then $P_{bru} = k_{bru} * \sigma_t * A_{bru}$

$= 1.5 * 42\ E{+}03 * 0.0563 = 3544$

$MS_{bru} = 3544 / (2 * 79) - 1 = 21$

Margin of safety summary, top strap attachment from filters to IF module

LOAD CASE	TENS ULT	TEAR YIELD	SHEAR OUT	BEARING
1	214	285	178	156
2	30	40	25	21
3	220	293	183	160
4	365	487	304	266

A 10-32, type A286, stainless screw attaches the other end of this brace to the top of the filter cross straps. These tie down bolts are made "captive" by reducing the shank diameter to that of the thread root. Thus the effective stress area will be slightly smaller than the typical handbook value for a 10-32 screw.

$$A_{cap} = PI * dia^2 / 4 = 3.14 * (0.152)^2 /4 = 0.018\ in^2$$

Thus $P_{us} = 91\ E{+}03 * 0.018 = 1651$ lb

and $P_{ut} = 140\ E{+}03 * 0.018 = 2540$ lb

The expected tightening torque is 30 to 34 in-lb. Thus for minimum friction (u =0.0784) and maximum torque (using the "nut factors" from reference 2.6)

$$F_{maxPRE} = 44.4 * 34 = 1510 \text{ lb}$$

For an average friction value of u = .1,

$$F_{avgPRE} = 36.5 * 34 = 1241 \text{ lb}$$

and for the maximum friction value and minimum torque,

$$F_{minPRE} = 28.8 * 30 = 864 \text{ lb}$$

This minimum preload will be reduced an additional 20 % for the gapping analysis which will follow

To determine the division of the external interface load between the mounting flange and the bolt, the effective joint area of the flange must be calculated.

For the mounting feet along the side of the IF module, The joint geometry fits case 2 of reference 2.5

$$A_j = PI * (D_w^2 - D_h^2) / 4 + PI * (D_j / D_w - .1) * (D_w * L_j / 5 + L_j^2 / 100) / 8$$

$$= 3.14 * (0.375^2 - 0.25^2) / 4 + 3.14*(0.625/0.375 - .1)*(0.375 * 0.625 / 5 + 0.625^2 / 100) / 8$$

$$= 0.093 \text{ in}^2$$

The fraction of the external interface load taken by the mounting flange is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$

$$= \frac{0.093 * 1 E+07}{0.018 * 2.91 E+07 + 0.093 * 1 E+07} = 0.64$$

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (2540)/(2 * (1-0.64) * 0.2 + 1510) - 1 = 0.68$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension.

$$MS_t = (2540)/(2 * (1-0.64) * 0.2 + 1.4 *1241) - 1 = 0.46$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the intial preload, a preload of 0.8 * 864 = 691 lb. If the mounting flange fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 691 / (2 * 0.66 * 0.2) - 1 = 2800.$$

The maximum shear load produced by x-axis accelerations is

$$F_{sx} = W_{brace} * XG / 2 \text{ ends}$$

$$= 0.14 \text{ lb} * 6 / 2 = 0.42 \text{ lb}$$

The shear produced by y-axis accelerations is

$$F_{sy} = W_{brace} * YG / 2 \text{ ends}$$

$$= 0.14 * 28.7 / 2 = 79 \text{ lb}$$

and $$F_{sxy} = ((F_{sx})^2 + (F_{sy})^2)^{1/2} = 79 \text{ lb}$$

To avoid slipping between the top strap and the top of the filter, the maximum possible shear force is calculated using a friction value of 0.15 between the members and the relaxed minimum bolt preload

F_{fric} = 0.15 * 691 lb = 103.7 lb

Thus MS_{slip} = 103.7 / (2 * 79) - 1 = -0.34

The margin of safety for load case #2 is negative, however the mounting bolts can absorb the shear load

MS_s = (91E+03 * 0.018) / (2 * 79) - 1 = 9.4

Since these mounting bolts sustain both tension and shear the effect of both stress are combined through calculation of the corresponding "factor of limit load capablilities" which are then combined

$FLC_t = MS_t + 1 = 0.68 + 1 = 1.68$

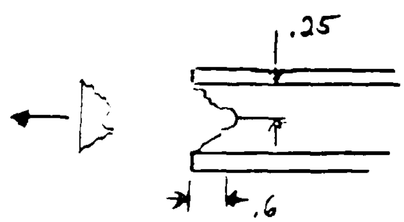
and $FLC_s = MS_s + 1 = 9.4 + 1 = 10.4$

Then $FLC_{comb} = \dfrac{1}{[(1/1.68)^2 + (1/10.4)^2]^{1/2}} = 1.658$

Margin of safety summary, top strap to IF module top bolts

LOAD CASE	TOP 2.0 SF	BRACE<-->FILT 2.0 & 1.4	MS GAP	MS SLIP	BOLT SHEAR	MS COMB
1	0.68	0.46	2862	3.6	71	0.58
2	0.68	0.46	2862	-0.34	9.4	0.55
3	0.68	0.46	152	3.7	73	0.58
4	0.68	0.46	1309	6.8	122	0.58

The geometry for tension tear out is



$L = \sqrt{(0.25)^2 + (.6)^2} - D$

The ultimate strength for tension tear out is

$P_u = {}_{tu} * A_t$

where $A_t = 2*[((0.25)^2 + (0.6)^2)^{1/2} - D] * t$

$= 2 * [0.65 - 0.25] * 0.625 = 0.5 \text{ in}^2$

$P_u = 42 \text{ E+03} * 0.5 = 21 \text{ E+03 lb}$

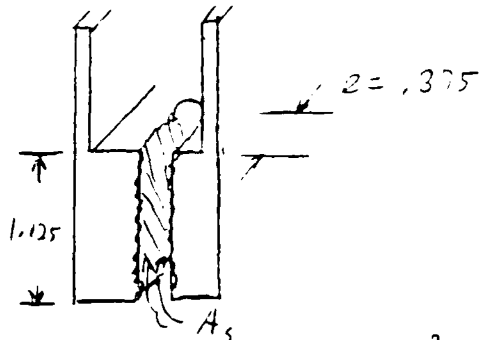
The maximum shear load was 79 lb

Thus $MS_{utear} = 21\ E{+}03\ /\ (2 * 79) - 1 = 132$

Checking for yield by using as safety factor of 1.25,

$$MS_{ytear} = 35\ E{+}03 * 0.5\ /\ (1.25 * 79) - 1 = 176$$

The geometry for shear tear out is



$$A_s = 2 * (0.375 * 1.125) = 0.84\ in^2$$

Then $MS_{shear} = 27\ E{+}03 * 0.84\ /\ (2 * 79) - 1 = 143$

Checking for the bearing strength:

$$e/D = 0.375\ /\ 0.19 = 2.0$$

$$D/t = 0.19\ /\ 0.19 = 1$$

yield a $k_{bru} = 1.8$

$$A_{br} = D * t = 0.19 * 0.19 = 0.036\ in^2$$

Then $P_{bru} = k_{bru} * \sigma_t * A_{bru}$

$$= 1.8 * 42\ E{+}03 * 0.036 = 2729$$

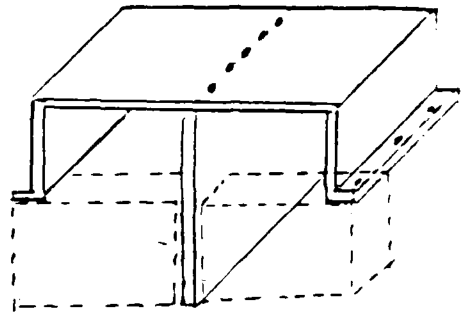
$$MS_{bru} = 2729\ /\ (2 * 79) - 1 = 16.2$$

Margin of safety summary, top strap attachment from filters to IF module:

LOAD CASE	TENS ULT	TEAR YIELD	SHEAR OUT	BEARING
1	928	1238	1006	119
2	132	176	143	16
3	953	12[illegible]	1034	123
4	1581	2108	1715	204

5.10 Synthesizer upper deck

The sysnthesizer subassembly consists of an upper deck, the synthesizers, and a mounting adapter plate. The upper deck contains the RF input attenuators and amplifiers. It attaches to a center rib with five 6-32 screws and to each synthesizer with one 6-32 screw.



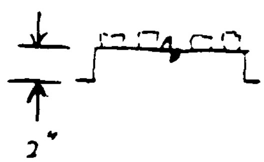
First considering the upper deck whose weight with components is 4.75 lb, the Z-axis accelerations produce a load of

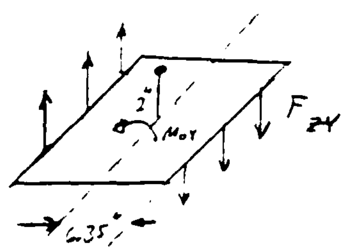
$$F_{zz} = 4.75 \text{ lb} * ZG / 11 \text{ screws}$$

$$= 4.75 * 2.7 / 11 = 1.2 \text{ lb}$$

The Y-axis acceleration produces a moment.

$$M_{oy} = 4.75 \text{ lb} * YG * 2 \text{ inch} = 38 \text{ in-lb}$$

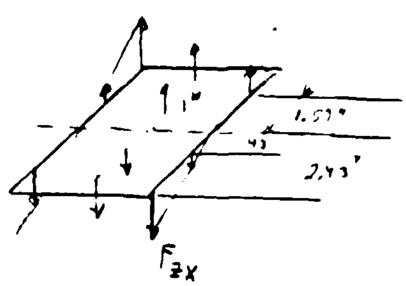




which is countered by F_{zy},

$$F_{zy} = M_{oy} / (6 * 6.35 \text{ inch}) = 1.0 \text{ lb}$$

X-axis acceleration also produces a moment,



$$M_{ox} = 4.75 * XG * 2 \text{ inch} = 4.75 * 37.1 * 2 = 352 \text{ in-lb}$$

Which is countered by F_{zx}

$$M_{ox} = 2 * F_{zx} * 2.43 *$$

$$(1 + 0.43/2.43 + 1.57/2.43 + 1.0/2.43 + 2.0/2.43)$$

$$F_{zx} = 352 / (2 * 2.43 * 3.06) = 23.7 \text{ lb}$$

The total tensile load is then

$$F_{zmax} = F_{zx} + F_{zy} + F_{zz}$$

$$= 23.7 + 1.0 + 1.2 = 25.9 \text{ lb}$$

Part of this external interface load in taken by the flange. The geometery of the flange is closest to case 2 of reference 2.5. The calculation of the effective flange area is

$$A_j = PI * (D_w^2 - D_h^2) / 4$$
$$+ PI * (D_j/D_w - .1) * (D_w * L_j/5 + L_j^2/100) / 8$$

$$= 3.14 * (0.312^2 - 0.14^2) / 4$$
$$+ 3.14*(0.4/0.312 - .1)*(0.312 * 0.2/5 + 0.2^2/100) / 8$$

$$= 0.067 \text{ in}^2$$

The fraction of the external interface load taken by the adapter plate is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$

$$= \frac{0.067 * 1 E+07}{0.00909 * 2.91 E+07 + 0.067 * 1 E+07} = 0.72$$

The 6-32 bolts used to attach the upper deck are type 316 stainless steel for which

$$\sigma_s = 37.5 E+03$$

$$\sigma_t = 75 E+03$$

The ultimate bolt strength is then

$$P_{tu} = 75 E+03 * 0.00909 = 682 \text{ lb}$$

$$P_{su} = 37.5 E+03 * 0.00909 = 341 \text{ lb}$$

The expected tigthening torque is 7 to 8 in-lb. The maximum bolt preload is then

$$F_{maxPRE} = 60 * 8 = 480 \text{ lb}$$

and $$F_{avgPRE} = 50 * 8 = 400 \text{ lb}$$

$$F_{minPRE} = 40 * 7 = 280 \text{ lb}$$

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (682)/(2 * (1-0.72) * 25.9 + 480) - 1 = 0.38$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension

$$MS_t = (682)/(2 * (1-0.72) * 25.9 + 1.4 * 400) - 1 = 0.19$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the initial preload, a preload of 0.8 * 280 = 224 lb. If the adapter plate fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 224 / (2 * 0.72 * 25.9) - 1 = 5.0$$

The shear loads in these bolts are produced by X-axis and Y-axis acceleration.

$$P_{sx} = (W_{deck}) * XG / 11 \text{ bolts} = 16 \text{ lb}$$

$$P_{sy} = (W_{deck}) * YG = 1.7 \text{ lb}$$

$$P_{sxy} = ((P_{sx})^2 + (P_{sy})^2)^{1/2}$$

$$= 16.1 \text{ lb}$$

To avoid slipping, the maximum possible shear force is calculated using a friction value of 0.15 between the members and the relaxed minimum bolt preload.

$$F_{fric} = 0.15 * 224 \text{ lb} = 33.6 \text{ lb}$$

Thus $$MS_{slip} = 33.6 / (2 * 16.1) - 1 = 0.04$$

If the holding friction was not sufficient, the mounting bolts must absorb the shear load.

$$MS_s = (37.5E+03 * 0.00909) / (2 * 16.1) - 1 = 9.6$$

$$FLC_t = MS_t + 1 = 0.38 + 1 = 1.38$$

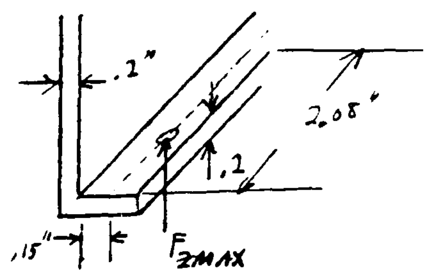
and $$FLC_s = MS_s + 1 = 9.6 + 1 = 10.6$$

Then $$FLC_{comb} = \frac{1}{[(1/1.38)^2 + (1/10.6)^2]^{1/2}} = 1.37$$

Margin of safety summary, upper deck mounting bolts

LOAD CASE	UPPPER DECK 2.0 SF	2.0 & 1.4	MS GAP	MS SLIP	MS SHEAR	MS COMB
1	0.38	0.19	5.0	0.04	9.6	0.37
2	0.40	0.20	11.8	0.33	12.5	0.39
3	0.38	0.19	4.9	4.40	53.7	0.38
4	0.41	0.21	15.4	2.78	37.4	0.40

The upper deck must withstand the bending produced on its flange by the Z-axis loading



$$M_o = 0.15 \text{ inch} * F_{zmax} = 0.15 * 25.9 = 3.89 \text{ in-lb}$$

then the stress

$$\sigma_{bend} = 6 * M_o / b\ t^2$$
$$= 6 * 3.89 / (2.08 * (0.2)^2)$$
$$= 280 \text{ psi}$$

Shear stress is also present,

$$\sigma_{shear} = F_{zmax} / A_{shear}$$
$$= 25.9 / (2.05 * 0.2) = 63 \text{ psi}$$

The combined stress is then

$$\sigma_{max} = ((\sigma_{bend})^2 + 3 * (\sigma_{shear})^2)^{1/2}$$
$$= ((280)^2 + 3 * (63)^2)^{1/2} = 300 \text{ psi}$$

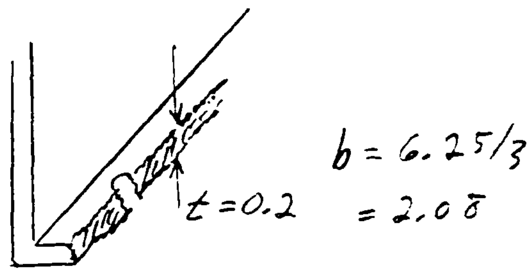
The margin of safety is

$$MS_{ubend} = 42 \text{ E+03} / (2 * 300) - 1 = 70$$

Checking for yield,

$$MS_{ybend} = 35 \text{ E+03} / (1.25 * 300) - 1 = 93$$

The flange geometery for tension failure is



The ultimate strength for tension tear out is

$$P_u = \sigma_{tu} * A_t$$

where A_t = 2.08 inch * 0.2 inch = .416 in^2

$$P_u = 42\ E{+}03 * 0.416 = 1.7E{+}03\ lb$$

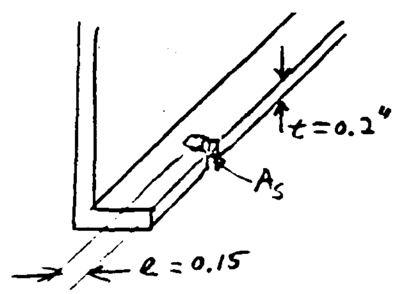
The maximum shear load was 16.1 lb, thus

$$MS_{utear} = 1.7E{+}03 / (2 * 16.1) - 1 = 541.$$

Checking for yield by using as safety factor of 1.25,

$$MS_{ytear} = 35\ E{+}03 * 0.416 / (1.25 * 16.1) - 1 = 722$$

The geometry for shear tear out is



$$A_s = 2 * (0.15 * 0.2) = 0.06\ in^2$$

Then MS_{shear} = 27 E+03 * 0.06 / (2 * 16.1) - 1 = 49

Checking for the bearing strength:

$$e/D = 0.15 / 0.14 = 1.07$$

$$D/t = 0.14 / 0.2 = 0.85$$

yield a k_{bru} = 0.9

$$A_{br} = D * t = 0.14 * 0.2 = 2.8\ E{-}02\ in^2$$

Then $P_{bru} = k_{bru} * \sigma_t * A_{bru}$

$= 0.9 * 42\ E{+}03 * 2.8\ E{-}02 = 1058$

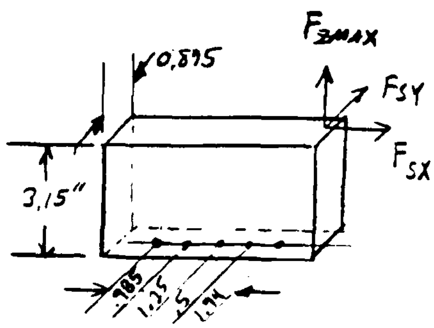
$MS_{bru} = 1058 / (2 * 16.1) - 1 = 32$

Margin of safety summary, upper deck assembly flange

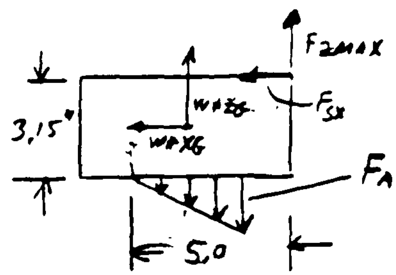
LOAD CASE	BENDING TENS	STRESS YIELD	TENSION TENS	TEAR YIELD	SHEAR OUT	BEARING MS
1	70	94	542	722	49	32
2	149	198	688	918	63	41
3	68	91	2804	3740	259	169
4	190	253	1966	2622	181	118

5.11 Synthesizer mounting

The synthesizer consists of two small bricks, the upper deck attaches to the larger whose weight is 1 lb. Each synthesizer is attached to the synthesizer adapter plate with five 4-40 screws.



The X-axis and Z-axis accelerations on the mass of the synthesizer combine with the moments produced by F_{zmax} and F_{sx} transferred from the upper deck to produce the maximum tension in bolt A



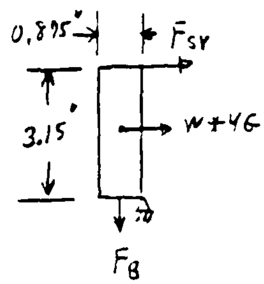
$F_A * (4.475+2.735*2.735/4.475+2.235*2.235/4.475+0.985*0.985/4.475)$

$= F_{zmax} * 5.0 + F_{sx} * 3.15 + 1\ lb * ZG * 1.8 + 1\ lb * XG * 1.58$

$F_A * 7.48 = 25.9 * 5.0 + 16 * 3.15 + 2.[illegible] * 1.[illegible] + 37.1 * 1.3[illegible]$

$F_A = 243.4 / (7.48) = 32.5 \text{ lb}$

Additional loading is produced by Y-axis accelerations



$$5 * F_B * 0.375 / 2 = F_{SY} * 3.15 + 1 \text{ lb} * YG * 1.58$$

$$F_B = (1.7 * 3.15 + 2.7 * 1.58) / (0.4375 * 5) = 5.4 \text{ lb}$$

The maximum tension load is

$$F_{AB} = 32.5 + 5.4 = 37.9 \text{ lb}$$

Part of this external interface load in taken by the flange. The synthesizer is attached to the adapter plate with 4-40 flat head screws. The geometery of the flange is not considered in reference 2.5. The calculation of the effective flange area performed using the diameter of the flat head.

$$A_j = PI * (D_w^2 - D_h^2) / 4$$

$$= 3.14 * (0.225^2 - 0.112^2) / 4$$

$$= 0.03 \text{ in}^2$$

The fraction of the external interface load taken by the adapter plate is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$

$$= \frac{0.03 * 1 \text{ E+07}}{0.00604 * 2.91 \text{ E+07} + 0.03 * 1 \text{ E+07}} = 0.63$$

The 4-40 bolts used to attach the upper deck are type 316 stainless steel for which

$$\sigma_s = 37.5 \text{ E+03}$$

$$\sigma_t = 75 \text{ E+03}$$

The ultimate bolt strength is then

$$P_{tu} = 75\ E+03 * 0.00604 = 453\ lb$$

$$P_{su} = 37.5\ E+03 * 0.00604 = 226\ lb$$

The expected tightening torque is 3 to 4 in-lb. Thus for minimum friction (u =0.0784) and maximum torque (for these small screws the "nut factors" are very approximate.)

$$F_{maxPRE} = 71 * 4 = 284\ lb$$

For an average friction value of u = 0.1,

$$F_{avgPRE} = 59 * 4 = 236\ lb$$

and for the maximum friction value and minimum torque,

$$F_{minPRE} = 47 * 3 = 141\ lb$$

This minimum preload will be reduced an additional 20 % for the gapping analysis which will follow.

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (453)/(2 * (1-0.63) * 37.9 + 284) - 1 = 0.45$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension.

$$MS_t = (453)/(2 * (1-0.63) * 37.9 + 1.4 * 236) - 1 = 0.26$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the initial preload, a preload of 0.8 * 141 = 113 lb. If the adapter plate fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 113 / (2 * 0.63 * 37.9) - 1 = 1.4.$$

The shear loading comes from the X-axis and Y-axis acceleration and the moment from the upper deck.

$$F_{synx} = (W_{syn} * XG + F_{sx}) / 5 \text{ bolts}$$

$$= (1.0 * 37.1 + 16) / 5 = 10.6\ lb$$

$$F_{syny} = W_{syns} * YG / 5 + F_{sy} * 5 /(7.48)$$

$$= 1 * 4.0 / 5 + 1.73 * 5 / 7.48 = 2.0\ lb$$

and $$F_{sxy} = ((F_{synx})^2 + (F_{syny})^2)^{1/2} = 10.8\ lb$$

To avoid slipping between the adapter and the FEB baseplate the maximum possible shear force is calculated using a friction value of 0.15 between the members and the relaxed minimum bolt preload

$$F_{fric} = 0.15 * 113 \text{ lb} = 17 \text{ lb}$$

Thus $$MS_{slip} = 17 / (2 * 10.8) - 1 = -0.22$$

Since the holding friction is not sufficient, the mounting bolts must absorb the shear load

$$MS_s = (37.5E+03 * 0.00604) / (2 * 10.8) - 1 = 9.5$$

Since these mounting bolts sustain both tension and shear, the effect of both stress are combined through calculation of the corresponding "factor of limit load capabilities" which are then combined

$$FLC_t = MS_t + 1 = 0.45 + 1 = 1.45$$

and $$FLC_s = MS_s + 1 = 9.5 + 1 = 10.5$$

Then $$FLC_{comb} = \frac{1}{[(1/1.45)^2 + (1/10.5)^2]^{1/2}} = 1.44$$

Since these attachment bolts are not near the edge of the adapter plate, flange tension tear out and shear tear out need not be checked

Checking for the bearing strength:

$$e/D = 0.767 / 0.112 = 6.8$$

$$D/t = 0.112 / 0.15 = 0.75$$

yield a k_{bru} > 3

$$A_{br} = D * t = 0.112 * 0.15 = 1.7 E-02 \text{ in}^2$$

Then $$P_{bru} = k_{bru} * \sigma_t * A_{bru}$$

$$= 3 * 42 E+03 * 1.7 E-02 = 2117$$

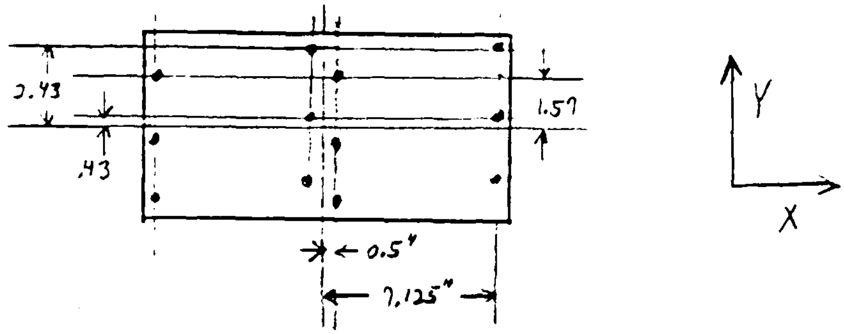
$$MS_{bru} = 2117 / (2 * 10.8) - 1 = 98$$

Margin of safety summary, synthesizer mounting bolts:

LOAD CASE	SYNTHESIZER BOLTS 2.0 SF	2.0 & 1.4	MS GAP	MS SLIP	MS SHEAR	MS COMB	MS BEARING
1	0.45	0.26	1.4	-0.22	9.5	0.44	98
2	0.41	0.23	0.8	-0.40	7.0	0.39	75
3	0.45	0.26	1.4	2.25	42.5	0.45	410
4	0.54	0.33	5.0	1.73	35.6	0.53	345

5.12 Synthesizer subassembly adapter plate mounting

The synthesizer subassembly is fastened to the FEB baseplate through an adapter plate so that it can be removed from the top without accessing any screw heads on the underneath side of the FEB baseplate. The total weight of the subassembly is 18 lb. Twelve 10-32, type A286, stainless steel screws attach the subassembly to the baseplate.



The CG is directly overtop the center of the screws. Z-axis acceleration produces tension only.

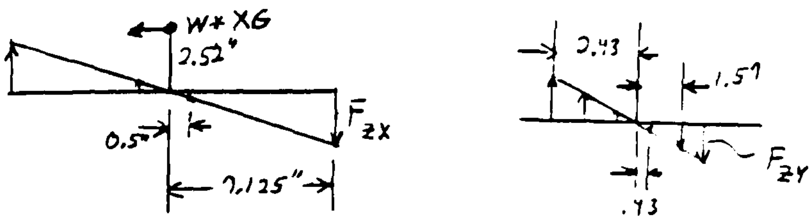
$$F_{zz} = W_{synsub} * ZG / 12 \text{ bolts}$$

$$= 18 * 2.7 / 12 = 4.05 \text{ lb}$$

X-axis acceleration produces a moment of

$$M_{ox} = W_{synsub} * XG * 2.52 \text{ inch}$$

$$= 18 * 37.1 * 2.52 = 1681 \text{ in-lb}$$



This is countered by

$$F_{zx} = M_{ox}/(6 * (7.125 + (0.5/7.125) * 0.5)) = 39.1$$

The Y-axis acceleration also produces a moment,

$$M_{oy} = W_{synsub} * YG * 2.52 \text{ inch}$$

$$= 18 * 4 * 2.52 = 181 \text{ in-lb}$$

which is countered by

$$F_{zy} = M_{oy}/(4 * (2.43 + 1.57/2.43 * 1.57 + 0.43/2.43 * 0.43)$$
$$= 12.9 \text{ lb}$$

The maximum tensile load is then

$$F_{max} = F_{zx} + F_{zy} + F_{zz}$$
$$= 39.1 + 12.9 + 4.0 = 56 \text{ lb}$$

Part of this external interface load in taken by the flange. The geometery of the flange is closest to case 3 of reference 2.3. The calculation of the effective flange area is

$$A_j = PI * (D_w^2 - D_h^2) / 4$$
$$+ PI * (D_j/D_w - 1) * (D_w * L_j/5 + L_j^2/100) / 8$$
$$= 3.14 * (0.375^2 - 0.25^2) / 4$$
$$+ 3.14*(0.75/0.375 - 1)*(0.375 * 0.15/5 + 0.15^2/100) / 8$$
$$= 0.07 \text{ in}^2$$

The fraction of the external interface load taken by the adapter plate is

$$FRAC = \frac{A_j * E_j}{A_B * E_B + A_j * E_j}$$
$$= \frac{0.07 * 1 E+07}{0.02 * 2.91 E+07 + 0.07 * 1 E+07} = 0.54$$

A joint diagram will thus show that 0.54 of the external interface force is taken by the flange and 0.46 of the interface force is taken by the bolt which adds to the preload of the bolt to produce the maximum bolt stress.

The expected tightening torque is 30 to 34 in-lb. Thus for minimum friction (u =0.0784) and maximum torque (using the "nut factors" from reference 2.6)

$$F_{maxPRE} = 44.4 * 34 = 1510 \text{ lb}$$

For an average friction value of u = 0.1,

$$F_{avgPRE} = 36.5 * 34 = 1241 \text{ lb}$$

and for the maximum friction value and minimum torque,

$$F_{minPRE} = 28.8 * 30 = 864 \text{ lb}$$

This minimum preload will be reduced an additional 20 % for the gapping analysis which will follow

The margin of safety can then be calculated using a safety factor of two on the external interface load.

$$MS_t = (2800)/(2 * (1-0.54) * 56 + 1510) - 1 = 0.79$$

Using an average value for the coefficent of friction, and a safety factor of 1.4 on the resulting pretension.

$$MS_t = (2800)/(2 * (1-0.54) * 56 + 1.4 * 1241) - 1 = 0.57$$

Gapping of the joint under minimum mounting torque and maximum torque friction must be considered. After the 20 % relaxation from the initial preload, a preload of 0.8 * 864 = 691 lb. If the adapter plate fraction of the external load were to exceed this minimum preload, gapping would occur.

$$MS_{gap} = 691 / (2 * 0.54 * 56) - 1 = 10.3$$

The shear loading comes from the X-axis and Y-axis acceleration.

$$F_{sx} = (W_{synsub}) * XG / 12 \text{ bolts}$$

$$= (18) * 37.1 / 12 = 55.6 \text{ lb}$$

$$F_{sy} = (W_{synsub}) * YG / 12 \text{ bolts}$$

$$= (18) * 4.0 / 12 = 6.0 \text{ lb}$$

and $$F_{sxy} = ((F_{sx})^2 + (F_{sy})^2)^{1/2} = 55.9 \text{ lb}$$

To avoid slipping between the adapter and the FEB baseplate, the maximum possible shear force is calculated using a friction value of 0.15 between the members and the relaxed minimum bolt preload.

$$F_{fric} = 0.15 * 691 \text{ lb} = 104 \text{ lb}$$

Thus $$MS_{slip} = 104 / (2 * 55.9) - 1 = -0.07$$

Since the holding friction is not sufficient, the mounting bolts must absorb the shear load.

$$MS_s = (91E+03 * 0.02) / (2 * 55.9) - 1 = 15.3$$

Since these mounting bolts sustain both tension and shear, the effect of both stress are combined through calculation of the corresponding "factor of limit load capabilities" which are then combined.

$$FLC_t = MS_t + 1 = 0.79 + 1 = 1.79$$

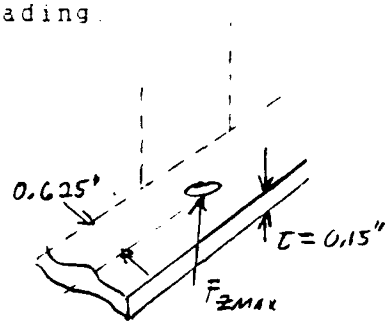
and $$FLC_s = MS_s + 1 = 15.3 + 1 = 16.3$$

Then $$FLC_{comb} = \frac{1}{[(1/1.79)^2 + (1/15.3)^2]^{1/2}} = 1.78$$

Margin of safety summary, synthesizer subassembly adapter plate bolts

LOAD CASE	BOLT 2.0 SF	ADAPT<->BASE 2.0 & 1.4	GAP	SLIP	BOLT SHEAR	MS COMB
1	0.79	0.57	11.4	-0.07	15.3	0.78
2	0.75	0.53	6.6	0.18	19.7	0.74
3	0.75	0.54	5.9	3.79	83.1	0.75
4	0.32	0.59	24.4	2.36	58.0	0.32

The adapter plate must withstand the bending produced on its flange by the Y-axis loading.



$b = 6.25/3 = 2.08$

$$M_o = 0.625 \text{ inch} * F_{zmax} = 0.625 * 51 = 31.9 \text{ in-lb}$$

then the stress

$$\sigma_{bend} = 6 * M_o / b\, t^2$$
$$= 6 * 31.9 / (2.08 * (0.15)^2)$$
$$= 4087 \text{ psi}$$

Shear stress is also present,

$$\sigma_{shear} = F_{zmax} / A_{shear}$$
$$= 51 / (2.08 * 0.15) = 163 \text{ psi}$$

The combined stress is then

$$\sigma_{max} = ((\sigma_{bend})^2 + 3 * (\sigma_{shear})^2)^{1/2}$$
$$= ((4087)^2 + 3 * (163)^2)^{1/2} = 4100 \text{ psi}$$

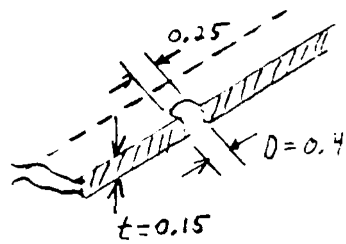
The margin of safety is

$$MS_{ubend} = 42 \text{ E+03} / (2 * 4100) - 1 = 4.1$$

Checking for yield,

$$MS_{ybend} = 35 \text{ E+03} / (1.25 * 4100) - 1 = 5.8$$

The flange geometery for tension failure is



$b = 6.25/3 = 2.08$

The ultimate strength for tension tear out is

$$P_u = \sigma_{tu} * A_t$$

where $A_t = (b - (D_1 + D_2)/2) * t$

$$= (2.08 - (0.25 + 0.4)/2) * 0.15 = 0.263 \text{ in}^2$$

$$P_u = 42 \text{ E+03} * 0.263 = 1.1 \text{ E+04 lb}$$

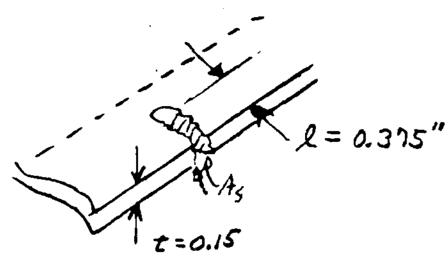
The maximum shear load was 11.5 lb, thus

$$MS_{utear} = 1.1\text{E+04} / (2 * 56) - 1 = 98.$$

Checking for yield by using as safety factor of 1.25,

$$MS_{ytear} = 35 \text{ E+03} * 0.263 / (1.25 * 56) - 1 = 131$$

The geometry for shear tear out is



$$A_s = 2 * (0.375 * 0.15) = 0.113 \text{ in}^2$$

Then $MS_{shear} = 27 \text{ E+03} * 0.113 / (2 * 56) - 1 = 26$

Checking for the bearing strength:

$$e/D = 0.375 / 0.25 = 1.5$$

$$D/t = 0.25 / 0.15 = 1.7$$

yields $k_{bru} = 1.4$

$$A_{br} = D * t = 0.25 * 0.15 = 3.75 E-02 \text{ in2}$$

Then $P_{bru} = k_{bru} * \sigma_t * A_{bru}$

$$= 1.4 * 42 E+03 * 3.75 E-02 = 2205$$

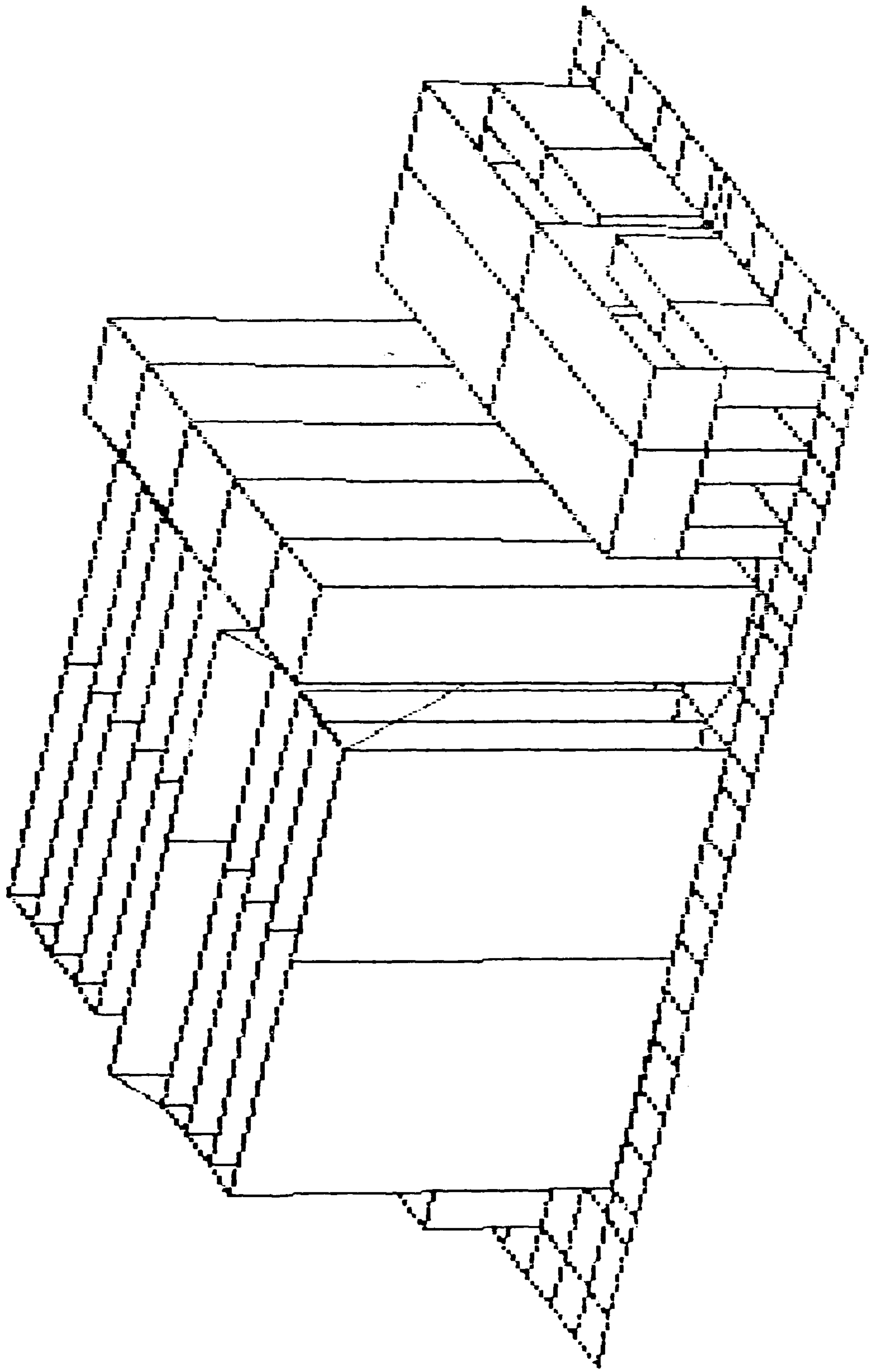
$$MS_{bru} = 2205 / (2 * 56) - 1 = 19$$

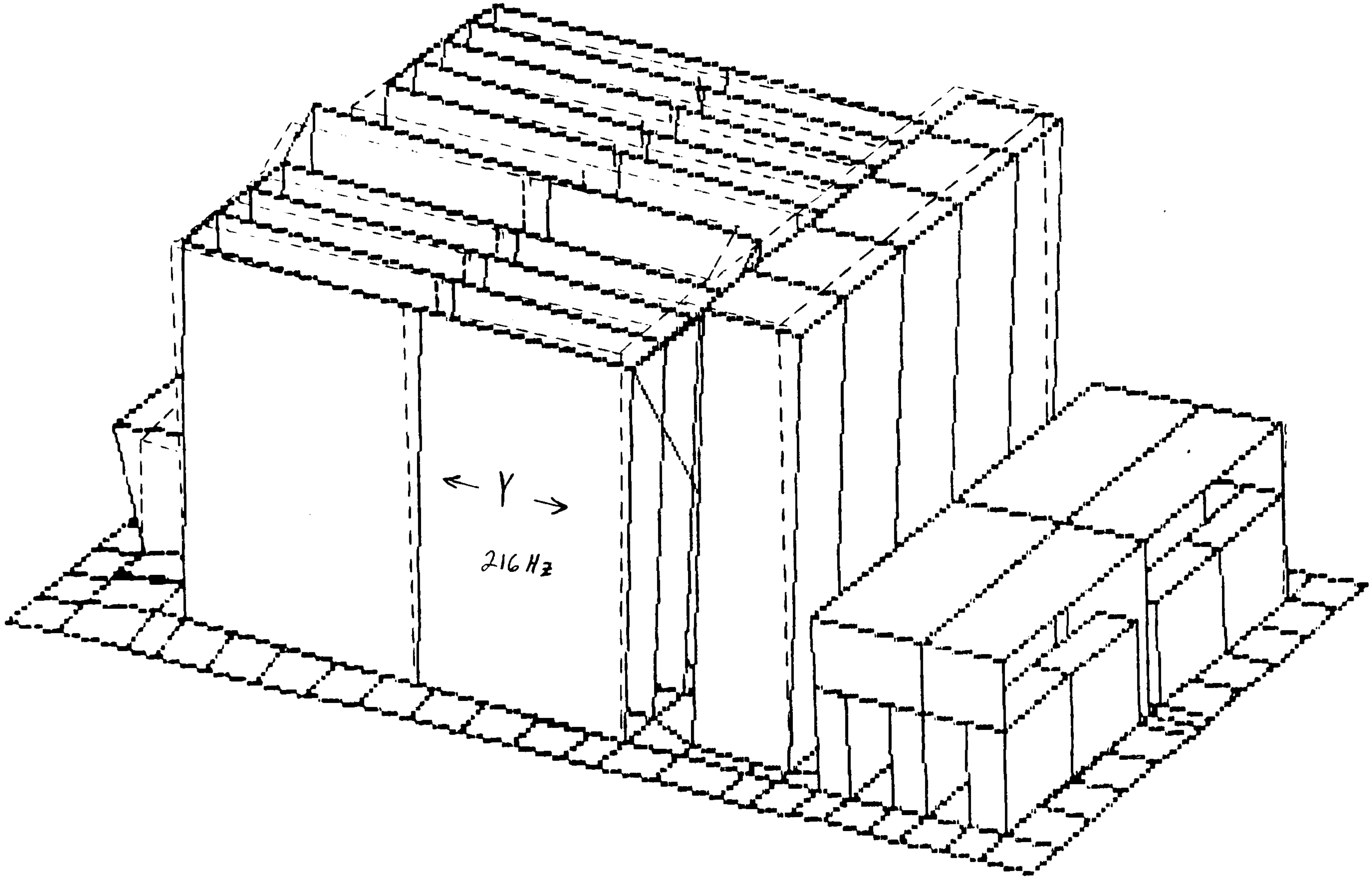
Margin of safety summary, synthesizer adapter plate flange

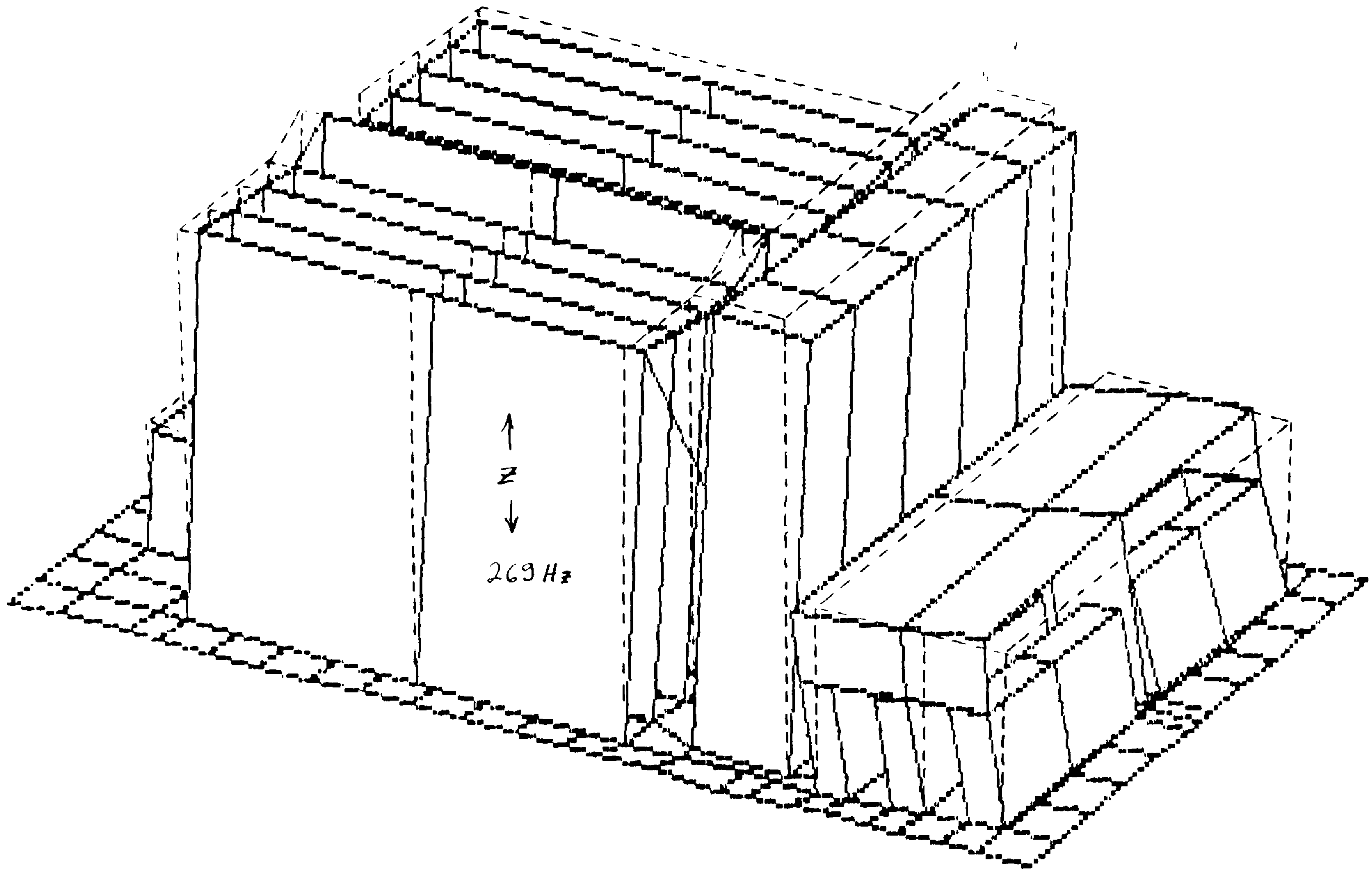
LOAD CASE	BENDING TENS	STRESS YIELD	TENSION TEAR TENS	YIELD	SHEAR OUT	BEARING MS
1	4.1	5.8	98	131	26	19
2	2.1	3.2	125	166	33	24
3	1.8	2.8	510	680	139	101
4	9.5	13.0	357	477	97	70

APPENDIX A

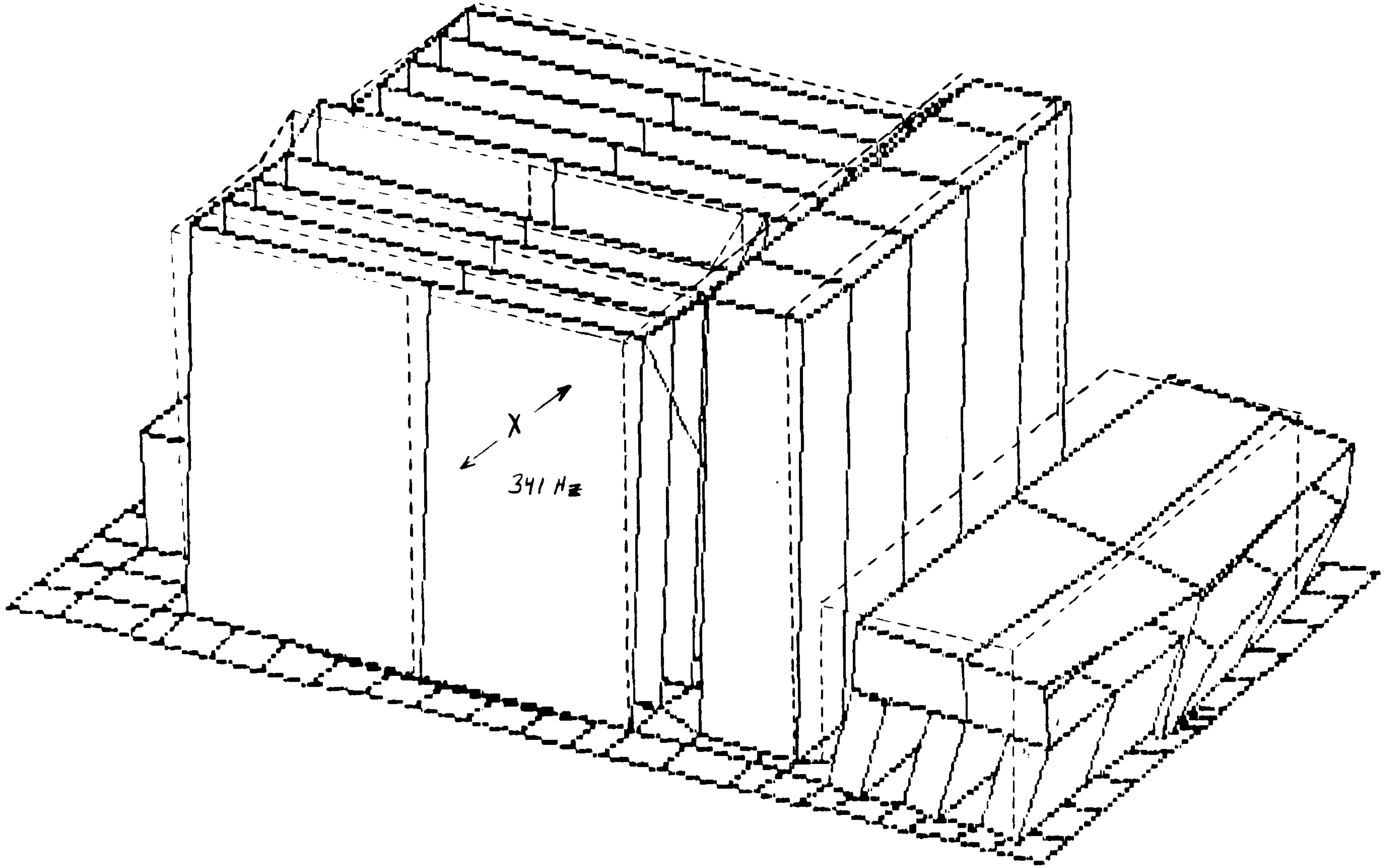
VIBRATION MODE PLOTS




← Y →
216 Hz



Z
269 Hz


X
341 Hz

END
DATE
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JULY 88