T72-19413
CR-128551

S4M-2-250
DECEMBER 1971

FINAL STRESS ANALYSIS REPORT

ULTRAVIOLET SPECTROMETER S169

Prepared for
NASA Manned Spacecraft Center
under Contract NAS9-11528

The Johns Hopkins University
Applied Physics Laboratory
8621 Georgia Avenue
Silver Spring, Maryland 20910

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I. INTRODUCTION.
A. GENERAL INFORMATION.
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a. NASA/MSC PD5/M500 DATED 2-18-71 "REVISED VIBRATION & SHOCK CRITERIA FOR THE FAR UV SPECTROMETER LUNAR ORBITAL EXPERIMENT."

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r. GENERAL ELECTRIC PLASTICS DEPT. TECHNICAL BROCHURE - "LEXAN."

B. DISCUSSION.

1. SUMMARY OF RESULTS.

THE STRESS ANALYSIS REPORT CONTAINED IN THESE PAGES VERIFIES THE STRUCTURAL INTEGRITY OF THE APOLLO S-169 UV-SPECTROMETER EXPERIMENT. SECTION B.2 DESCRIBES THE METHODS BY WHICH THE VARIOUS MEMBERS WERE ANALYZED. A DETAILED SUMMARY OF RESULTS FOR THE INDIVIDUAL STRUCTURAL ELEMENTS APPEARS IN THE FORM OF A TABLE OF MINIMUM MARGINS OF SAFETY (SEE PAGES 23-24). NO NEGATIVE MARGINS OF SAFETY WERE EXPERIENCED. THE DETAILED ANALYSES ARE PRESENTED IN SECTION II OF THIS REPORT. IT IS THE CONCLUSION OF THE WRITER THAT THE COMPONENT STRUCTURE IS MORE THAN ADEQUATE TO WITHSTAND THE ENVIRONMENTAL LOAD CONDITIONS GIVEN IN THE DESIGN CRITERIA SECTION ON PAGE 18.

B. DISCUSSION.

2. METHOD OF ANALYSIS.

SECTIONS II 1.0 AND II 2.0 OF THIS REPORT PRESENT IN DETAIL THE DERIVATION OF CRITICAL ACCELERATIONS USED IN THE SUBSEQUENT ANALYSES. COVERED SPECIFICALLY ARE THE RESPONSES TO THE PYROTECHNIC SHOCK AND RANDOM VIBRATION INPUT SPECTRA GIVEN IN THE DESIGN CRITERIA (PAGE 18). BECAUSE THE ISOLATOR MOUNTS HAVE BEEN DESIGNED TO GIVE THE SPECTROMETER A RESONANT FREQUENCY OF 90 ± 15 HZ. IN THREE MUTUALLY PERPENDICULAR AXES AND SINUSOIDAL VIBRATION IS TERMINATED AT 35 HZ, THIS VIBRATORY CONDITION IS NOT A CONSIDERATION.

MATHEMATICAL MODELS WERE FORMULATED FOR LUMPED-MASS SYSTEMS (25 MASSES FOR EXCITATION ALONG THE SPECTROMETER OPTICAL AXIS AND 16 MASSES FOR EXCITATION IN THE REMAINING TWO DIRECTIONS) AND EFFECTIVE SPRING CONSTANTS WERE CALCULATED. THE BASIC ASSUMPTION MADE WAS THAT FOR A GIVEN RESONANT FREQUENCY AND DAMPING RATIO, THE RATIO OF THE PEAK GENERALIZED RESPONSE OF A MULTI-DEGREE OF FREEDOM SYSTEM TO THAT OF A SINGLE DEGREE OF FREEDOM SYSTEM FOR BOTH SHOCK AND RANDOM VIBRATION WILL BE THE SAME AS THE RATIO OF THE STEADY STATE GENERALIZED RESPONSES OF THE TWO SYSTEMS WHEN EXCITED BY A UNIT SINUSOIDAL BASE MOTION INPUT.

THE PEAK RESPONSES TO SHOCK AND RANDOM VIBRATION WERE DETERMINED BY A COMPUTER PROGRAM AND ARE TABULATED FOR THE VARIOUS SPECTROMETER COMPONENTS ON PAGE 41. FOR EACH COMPONENT IN EACH DIRECTION OF LOADING THE HIGHER OF THE TWO ACCELERATIONS WAS USED DURING ANALYSIS. SINCE THE 3σ RANDOM RESPONSE WAS BASED ON AN INPUT OF 10 SECONDS DURATION, THE NUMBER OF STRESS CYCLES AT THIS LEVEL WAS DEEMED NEGLIGIBLE. THEREFORE, FATIGUE WAS NOT A PROBLEM AND MARGINS OF SAFETY WERE BASED UPON YIELD OR ULTIMATE STRENGTHS FOR BOTH SHOCK AND RANDOM VIBRATION.

AFTER ESTABLISHING THE ABILITY OF THE BAFFLE ASSEMBLY CROSS-SECTION TO CARRY AND TRANSMIT THE APPLIED VERTICAL SHEAR LOADING, THE ASSEMBLY WAS ANALYZED AS A CANTILEVER BEAM BUILT IN AT ITS ATTACHMENT TO THE FRONT PLATE. THE BAFFLE ASSEMBLY IS A "THREE-STAGE" MEMBER AND THE STRUCTURAL INTEGRITY OF THE ATTACHMENTS AT THE VARIOUS INTERFACES WAS ESTABLISHED. THE OVERALL ASSEMBLY WAS THEN CHECKED FOR THE MAXIMUM BENDING STRESS CONDITION. THE CRITICALLY LOADED CORNER GUSSETS WERE CHECKED FOR COMPRESSIVE BUCKLING AND LOCALIZED BENDING STRESSES. SATISFACTORY ATTACHMENTS WERE FOUND TO EXIST BETWEEN THE GUSSETS AND THE BAFFLE BASE AND BETWEEN THE BAFFLE BASE AND THE FRONT PLATE.

Analysis of the detector electronics included determination of the maximum bending stress on all the printed circuit boards. The boards were taken to be uniformly loaded square plates supported at the four corners only. The accelerations resulting from the computer runs were taken as inputs to the boards. Transmissibilities were assumed across the boards and the peak dynamic stresses were calculated. Those boards mounted to the right gusset of the electronics package were conservatively assumed to be cantilevered from the gusset. Consequently the attaching bolts were also assumed to carry bending loads. Localized bending conditions were then considered on the gusset.

Analyses were made for the attachment of the P/M tube to the converter housing and the attachment of the entire electronics package to the front plate. In all attachment analyses it was assumed that the fasteners used were torqued up to 80% of the bolt material yield strength. Margins of safety were based upon the ratio of this preload to the maximum applied load. A coefficient of friction was assumed between the mating surfaces and the friction force at each of the fastener locations was determined. The friction capacity of the entire joint was then established and margins of safety were calculated based upon the ratio of this friction capacity to the applied shear force tending to separate the joint. The smaller of the two margins of safety was reported as being critical for the attachment.

The front plate was initially analyzed as a solid rectangular flat plate of constant thickness (the minimum plate thickness was assumed to be constant over the entire plate surface) with all edges simply supported. The weights of all the elements mounted to the plate were assumed to be uniformly distributed. This was a conservative approach since most of the equipment attachment points are located at or near to the supported edges. The front plate was then analyzed as a series of localized beams and plates under concentrated loads at the attachment points of the Baffle Assembly, the Detector Electronics and the Motor Mount plus Grating Support. The Motor Mount Housing and the fiducial mark detector bracket were also analyzed by conservatively assuming sections effective for bending stresses. A similar analysis was performed for the Tripling Mirror Housing.

The loads applied to the Wavelength Cam Assembly result from the Motor output torque. The Camshaft, keyed to the Motor shaft, and the dowel, pinned to the cam were checked for shear stresses as they transmitted the torque. Assuming this twisting moment to be resisted by couple forces, the Grating Arm and Collar were found to be capable of withstanding the applied bending stresses resulting from the couple forces.

The UVS housing was analyzed by assuming a model consisting of a beam (constant EI) uniformly loaded between supports and overhung at both ends. A concentrated load and moment at the forward end represent the effects of the front plate and its mounted components (baffle, grating, electronics, etc.) and a similar concentrated load and moment at the aft end represent the effects of the Mirror Cell Assembly. The Housing is weight relieved and the minimum pocket skin thickness was used in a conservative determination of section properties. Ample margins of safety were found for the bending stresses resulting from this loading condition.

The attachments at the forward and aft ends were examined and shown to be adequate. Localized bending stresses produced by the attachment loads were calculated at the housing flanges and found to be of no significance. A detailed loads analysis through the isolator mounting feet was performed, the result of which was a determination of the loads dumped into the housing at the points of attachment with the mounts. Under this condition the hardware was checked and the bearing stresses in the housing were computed. In addition, the housing was checked out for handling loads. In all cases the margins of safety determined were ample.

The analyses of the optical elements, the Ebert Mirror and the Grating, were similar in most respects. For excitation in the plane of the mirrors, the plastic buttons supporting the mirrors were assumed to be springs. The effective spring constants were calculated and the expected dynamic displacements of the mirrors were determined. The "springs" were assumed to be preloaded (to a deflection 1.25 times the dynamic displacement) in order to prevent chattering during vibration. The combined effects of preload and dynamic loading were then considered and the compressive stresses in the buttons and the mirrors were evaluated. Similar assumptions and analyses were made for loading normal to the mirror surfaces. For this mode of excitation the mirrors are supported by three pairs of parallel, preloaded "springs". (This assumption was modified slightly in the Grating analysis and an impact loading condition was hypothesized). The Grating was analyzed as a plate and the ribbed Ebert mirror as three simply supported beams for normal loading. Conservative, simplifying assumptions were made for the clamps which hold down the mirrors. The reactions to the mirrors dynamic loads were in turn applied to the clamps which were analyzed as simply supported or cantilevered beams.

Analyses were performed on the structures which house the mirrors (the Mirror Cell in the case of the Ebert Mirror and the Grating Box in the case of the Grating). Again conservative assumptions were made in modeling the structures into beams with various end conditions. The margins of safety experienced were all large.

The Mirror Cell rear cover was analyzed as a rectangular flat plate simply supported on all four edges under a uniformly distributed pressure. The stiffening effect of the underside reinforcing ribs was neglected. It was conservatively assumed that at the time of SIM door jettison an internal pressure of 1.0 psi existed on the cover due to differential rates of pressure loss within and without the spectrometer. The calculated margin of safety was satisfactory.

C. DESCRIPTION OF STRUCTURE.

FROM A STRUCTURAL STANDPOINT, THE UV-SPECTROMETER MAY BE THOUGHT TO CONSIST OF FIVE MAIN ASSEMBLIES OR ELEMENTS:

1. BAFFLE ASSEMBLY (7232-0300).
2. DETECTOR ELECTRONICS.
3. FRONT PLATE ASSEMBLY (7232-0200).
4. HOUSING (7232-0107).
5. EBERT MIRROR ASSEMBLY (7232-0150).

1. BAFFLE ASSEMBLY.

THE BAFFLE ASSEMBLY PROVIDES FOR THE ENTRANCE OF LIGHT TO THE SPECTROMETER OPTICS. IT IS ESSENTIALLY A THREE-STAGE MEMBER OF RECTANGULAR CROSS-SECTION ATTACHED TO A BASE WHICH IN TURN IS CANTILEVERED OFF THE FRONT PLATE BY SIX PRELOADED FASTENERS AND THERMAL STANDOFFS. THE RECTANGULAR CROSS-SECTION OF THE FIRST STAGE IS MADE UP OF FOUR BAFFLE PLATES (7232-0308), FOUR CORNER GUSSETS (7232-0306, 0307) AND FOUR INTERMEDIATE RIVET STRIPS (7232-0315). THE CROSS-SECTION VARIES FROM APPROXIMATELY 1.00 INCHES x 3.00 INCHES AT THE BASE TO APPROXIMATELY 3.88 INCHES x 6.41 INCHES AT THE INTERFACE WITH THE INTERMEDIATE BAFFLE (7232-0309) OR SECOND STAGE. THE LENGTH OF THE ENTIRE ASSEMBLY IS ABOUT 20 INCHES.

THE BAFFLE PLATES, .05 INCHES THICK, HAVE FINS OR RIBS NORMAL TO THE DIRECTION OF THE ENTERING LIGHT FOR THE PURPOSE OF DEFLECTING THE LIGHT IN A PRESCRIBED MANNER. THE FINS ARE .031 INCHES THICK AND ARE EQUALLY SPACED .50 INCHES APART.

THE CORNER GUSSETS WHICH TIE THE PLATES TOGETHER STIFFEN THE ENTIRE ASSEMBLY. THE GUSSET CROSS-SECTION IS APPROXIMATELY THAT OF AN UNEQUAL-LEGGED ANGLE. THE MEMBER IS TAPERED AND AS SUCH PROVIDES LESS AREA AS THE DISTANCE FROM THE BASE INCREASES.

THE BAFFLE BASE (7232-0305) IS A MACHINED PLATE WITH A CENTRAL RECTANGULAR SLOT TO ALLOW FOR THE PASSAGE OF THE DEFLECTED LIGHT. THERE ARE THREE HELI-COIL INSERTS IN EACH OF THE FOUR CORNERS TO ACCOMMODATE THE ATTACHMENT OF THE GUSSETS.

THE INTERMEDIATE BAFFLE IS 1.47 INCHES LONG, OF VARYING RECTANGULAR CROSS-SECTION AND FLANGED AT ITS INTERFACE WITH THE BAFFLE TOP (7232-0310). ATTACHMENT TO THE BAFFLE TOP IS BY 24 RIVETS AND TO THE FIRST STAGE ASSEMBLY BY 20 RIVETS.

THE BAFFLE TOP IS THE THIRD AND FINAL STAGE OF THE OVERALL ASSEMBLY. AT ITS ATTACHMENT TO THE INTERMEDIATE STAGE, THE CROSS-SECTION IS RECTANGULAR. AT ITS FREE END THE TOP HAS AN IRREGULAR (TRAPEZOIDAL) OPENING.

ALL THE MEMBERS OF THE BAFFLE ASSEMBLY ARE FABRICATED FROM 6061-T6 ALUMINUM ALLOY.

2. DETECTOR ELECTRONICS.

THE ELECTRONICS ASSOCIATED WITH THE SPECTROMETER EXPERIMENT ARE PACKAGED IN A BOX WHICH IS ALSO CANTILEVERED OFF THE FRONT PLATE. THREE SIDES OF THE BOX ARE PROVIDED BY THE MOUNTING PLATE (7232-2016), THE LEFT GUSSET (7232-2015) AND THE CONVERTER HOUSING (7232-2013). A THREE-SIDED COVER OR WRAPAROUND (7232-2017) DEFINES THE REMAINDER OF THE PACKAGE.

THE CONVERTER HOUSING IS A MACHINED 2024-T351 ALUMINUM ALLOY MEMBER TO WHICH THE P/M TUBE AND THE LEFT AND RIGHT GUSSETS ARE MOUNTED. THE HOUSING ALSO HAS A RECESS INTO WHICH THE CONVERTER AND REGULATOR BOARD ASSEMBLIES ARE PLACED IN ADDITION TO TWO DC-DC CONVERTER HEAT SINKS. A THIN (.040 INCH) 2024-T3 ALUMINUM ALLOY COVER ENCLOSES THE RECESS. RF SHIELDING FOR THE P/M TUBE IS ALSO ATTACHED TO THE CONVERTER HOUSING.

THE LEFT GUSSET PROVIDES A MOUNTING SURFACE FOR THE VARIOUS ELECTRICAL CONNECTORS USED IN THE UNIT.

THE RIGHT GUSSET (7232-2014) PROVIDES FOR THE MOUNTING OF SIX LEXAN STANDOFFS (7232-2046) WHICH ARE THE SUPPORTS FOR THE THREE ELECTRONICS PC BOARDS. A BRACKET OR CLIP (7232-2039) IS UTILIZED ON THE OUTERMOST BOARD SO THAT THE ASSEMBLY OF BOARDS AND STANDOFFS IS NOT CANTILEVERED OFF THE RIGHT GUSSET. THE #6-32 BOLTS (7232-2048) THROUGH THE STANDOFFS ARE HIGH-STRENGTH, HEAT TREATED BERYLLIUM COPPER. BOTH GUSSETS ARE 2024-T351 ALUMINUM ALLOY.

THE FLANGED MOUNTING PLATE (2024-T351 ALUMINUM ALLOY) IS ATTACHED TO THE FRONT PLATE IN FOUR PLACES. BOTH LEFT AND RIGHT GUSSETS TIE INTO THE VERTICAL WEBS OF THE MOUNTING PLATE. ATTACHMENT OF THE CONVERTER HOUSING THROUGH A HORIZONTAL WEB IS ALSO PROVIDED FOR.

THE WRAPAROUND IS AN ALUMINUM ALLOY (6061-T6) SHEET, .032 INCHES THICK, THAT IS FASTENED TO THE GUSSETS, THE CONVERTER HOUSING, AND THE MOUNTING PLATE.

3. FRONT PLATE.

THE FRONT PLATE (7232-0201) IS A MACHINED ALUMINUM (6061-T6) ALLOY MEMBER FASTENED TO THE FLANGE OF THE UVS HOUSING AT THE FORWARD END BY 14 - #8-32 SCREWS. IN ADDITION TO PROVIDING THE SURFACE FOR THE ATTACHMENT OF THE BAFFLE ASSEMBLY AND THE DETECTOR ELECTRONICS PACKAGE, THE FRONT PLATE SUPPORTS THE MOTOR HOUSING (7232-0230), GRATING BOX ASSEMBLY (7232-0215) AND THE TRIPLING MIRROR HOUSINGS (7232-0271, 0272).

THE PLATE IS RECTANGULAR IN SHAPE (APPROXIMATELY 8.9 INCHES BY 7.3 INCHES) AND NOMINALLY .12 INCHES THICK WITH REINFORCING RIBS ON ITS UNDERSURFACE. IN THE AREAS OF LOCALIZED HIGH LOADING (ATTACHMENT POINTS) THE PLATE THICKNESS IS INCREASED AND "PADS" ARE PROVIDED. THERE ARE HELI-COIL INSERTS IN THE PLATE AT THESE ATTACHMENT POINTS.

TWO SUPPORT FLANGES NEAR THE CENTER OF THE LONG EDGES ARE CANTILEVERED FROM THE PLATE. IN TURN, ONE OF THESE FLANGES HAS THE MOTOR MOUNT CANTILEVERED FROM IT AND BOTH FLANGES SUPPORT THE GRATING BOX ASSEMBLY.

THE GRATING BOX ASSEMBLY CONSISTS ESSENTIALLY OF A RECTANGULAR MIRROR (GRATING BLANK, 7232-0001) MADE OF A GLASS-LIKE MATERIAL CER-VIT 100 SUPPORTED ON KEL-F-81 PLASTIC BUTTONS OR PADS WITHIN THE CONFINES OF THE RECTANGULAR GRATING BOX (7232-0220). THE PADS ARE INTEGRAL PARTS OF THE GRATING CLAMP ASSEMBLIES (7232-0217, 0218). THE GRATING BOX IS AN ALUMINUM ALLOY (6061-T6) MACHINING WHICH IS SUPPORTED BY BEARINGS AT ITS ENDS. LIKE THE FRONT PLATE, THE GRATING BOX IS MADE WITH UNDERSIDE STIFFENING RIBS. ITS POSITION CAN BE ADJUSTED BY MEANS OF AN ATTACHED CAM ASSEMBLY.

4. UVS HOUSING.

THE UVS HOUSING (7232-0107) IS MACHINED FROM A CAST ALUMINUM (356-ALLOY) MEMBER OF VARYING RECTANGULAR CROSS-SECTION. THE HOUSING IS 6.500 INCHES BY 8.125 INCHES (INSIDE DIMENSIONS) AT THE FORWARD END AND 6.313 INCHES BY 9.438 INCHES AT THE AFT END AND APPROXIMATELY 19 INCHES LONG. THE NOMINAL THICKNESS IS .19 INCHES BUT THERE ARE A NUMBER OF WEIGHT RELIEVING POCKETS IN WHICH THE SKIN THICKNESS IS REDUCED TO .04 INCHES. THE HOUSING IS FLANGED AT ITS FORWARD AND AFT ENDS TO PROVIDE MOUNTING SURFACES FOR THE FRONT PLATE AND MIRROR CELL ASSEMBLY RESPECTIVELY.

THERE ARE MACHINED FLAT AREAS AT BOTH ENDS OF THE HOUSING WHERE THE ISOLATOR SHOCK MOUNTS ARE TO BE ATTACHED. ADDITIONAL FLATS OR PAD AREAS ARE PROVIDED FOR THE UTILIZATION OF HANDLING FIXTURES.

AN ACCESS DOOR PERMITS INTERNAL ADJUSTMENTS.

5. MIRROR CELL ASSEMBLY.

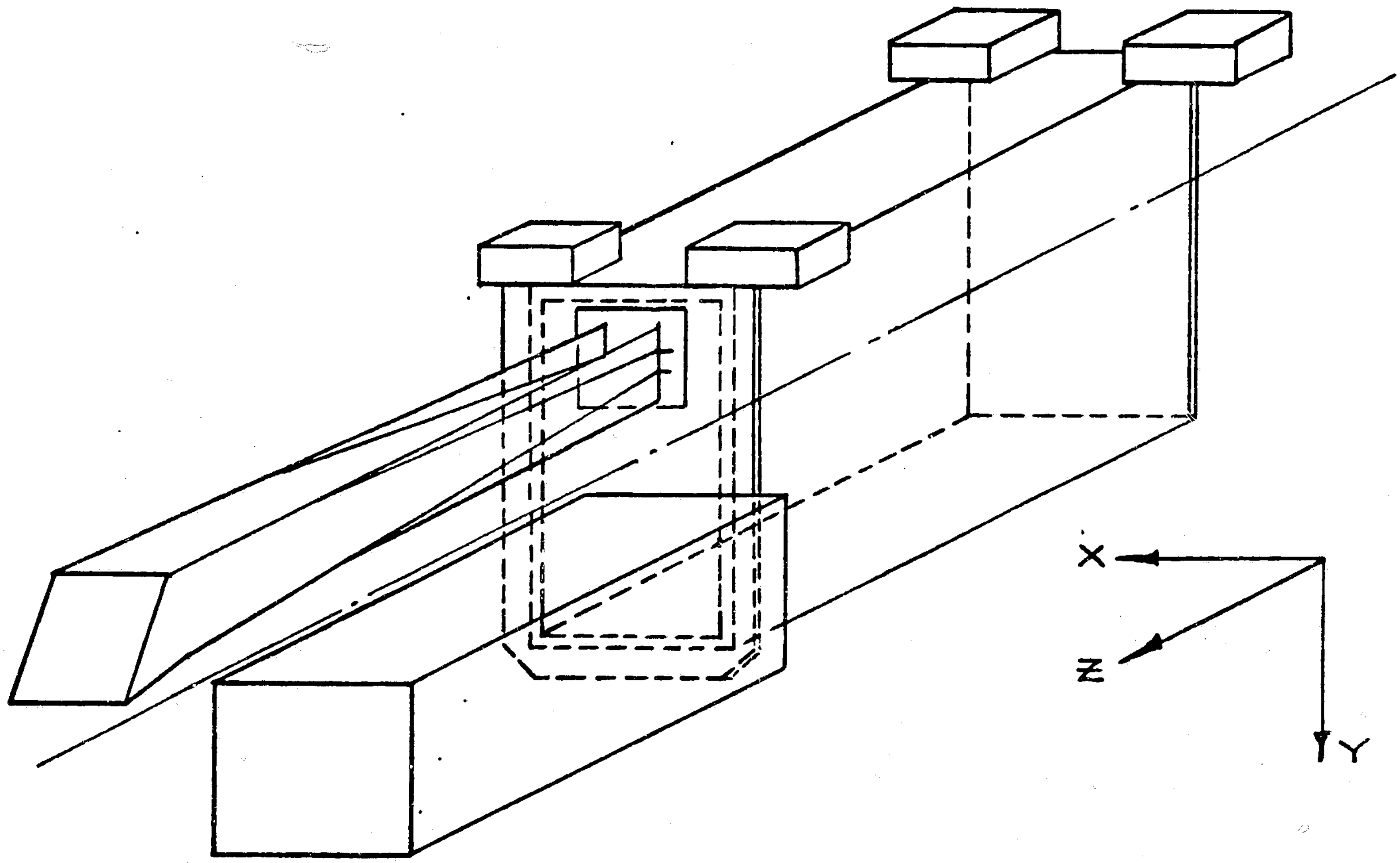
THE ESSENTIAL ELEMENTS OF THE MIRROR CELL ASSEMBLY (7232-0150) ARE THE MIRROR CELL (7232-0151), THE EBERT MIRROR (7232-0002) AND THE MIRROR CELL COVER OR AFT END CAP (7232-0152).

THE MIRROR CELL IS AN ALUMINUM ALLOY (6061-T6) MACHINING WHICH SUPPORTS THE EBERT MIRROR BY MEANS OF SEVERAL KEL-F-81 PLASTIC BUTTONS OR "SPRINGS" PRELOADED TO PREVENT CHATTERING OF THE MIRROR DURING VIBRATION. THE ENTIRE ASSEMBLY BOLTS TO THE AFT END OF THE UVS HOUSING.

THE EBERT MIRROR, LIKE THE GRATING BLANK, IS MADE FROM CER-VIT 100. ITS MIRRORED SURFACE IS CONCAVE AND ITS BACK FACE IS RIBBED FOR STIFFNESS PURPOSES. CLAMP ASSEMBLIES (7232-0153, 0154) ARE BOLTED TO THE MIRROR CELL STRUCTURE AND HOLD THE MIRROR IN PLACE AGAINST MOTION NORMAL TO THE OPTICAL SURFACE. THE MIRROR CELL COVER IS A THIN (.03 INCH) ALUMINUM ALLOY (6061-T6) PLATE, APPROXIMATELY 6.87 INCHES BY 10.00 INCHES, USED TO CLOSE IN THE AFT END OF THE ASSEMBLY. THIS PLATE IS ALSO PROVIDED WITH REINFORCING RIBS.

D. DESIGN CRITERIA (REF. a)

UVS DESIGNATED AXES



1. PYROTECHNIC DETONATION INDUCED SHOCK SPECTRA.

A. CSM/SLA SEPARATION.

SHOCK RESPONSE SPECTRA BASED ON A 1/6 OCTAVE BAND ANALYSIS AND Q=10 AS FOLLOWS:

20 TO 800	CPS	+12 db/OCT. INCREASE
800 TO 10000	CPS	1700 G. PK.

B. SIM DOOR JETTISON.

SHOCK RESPONSE SPECTRA BASED ON A 1/6 OCTAVE BAND ANALYSIS AND Q=10 AS FOLLOWS:

20 TO 900	CPS	+12 db/OCT. INCREASE
900 TO 10000	CPS	2400 G. PK.

2. LIFT-OFF AND MAX. Q FLIGHT.

A. RANDOM VIBRATION. (1.5 MIN. PER AXIS DURATION)

Y-AXIS

20 TO 200	CPS	+ 6 db/OCT. INCREASE.
200 TO 1000	CPS	0.07 G^2/CPS
1000 TO 2000	CPS	- 9 db/OCT. DECREASE.

X AND Z- AXES

20 TO 60	CPS	+ 3 db/OCT. INCREASE
60 TO 160	CPS	0.03 G^2/CPS
160 TO 240	CPS	+ 12 db/OCT. INCREASE
240 TO 1000	CPS	0.15 G^2/CPS
1000 TO 2000	CPS	- 9 db/OCT. DECREASE

IN ADDITION, TO DEMONSTRATE CAPABILITY TO WITHSTAND THE TRANSONIC FLIGHT (MACH 1) CONDITION, THE LIFTOFF AND MAX Q RANDOM SPECTRA SHOULD BE INCREASED BY A FACTOR OF 2.5 FOR 10 SEC. IN EACH AXIS.

B. SINUSOIDAL VIBRATION. (3 OCT./MIN. SWEEP RISE AND FALL EACH AXIS)

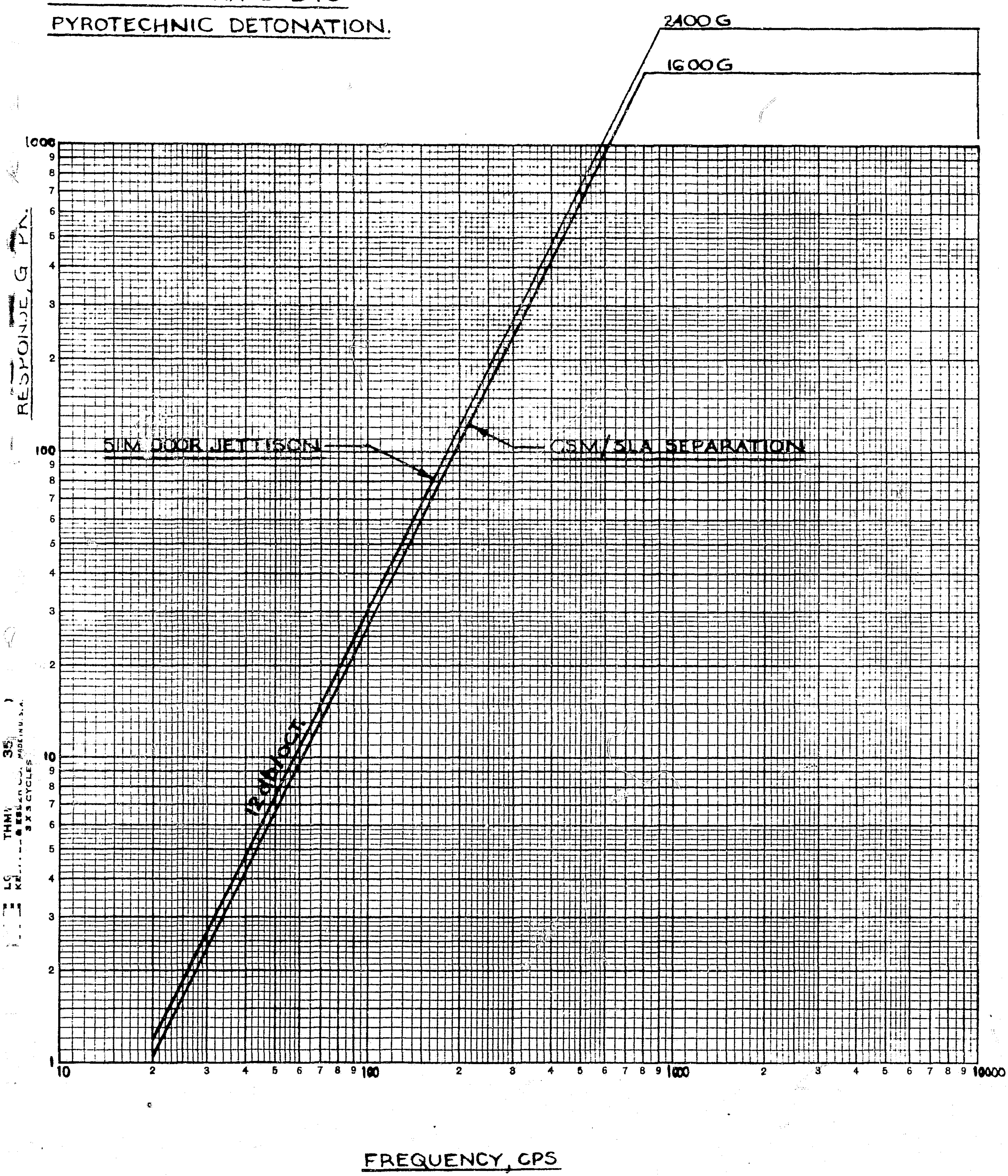
X AND Y- AXES

5 TO 35	CPS	0.25 G. PK.

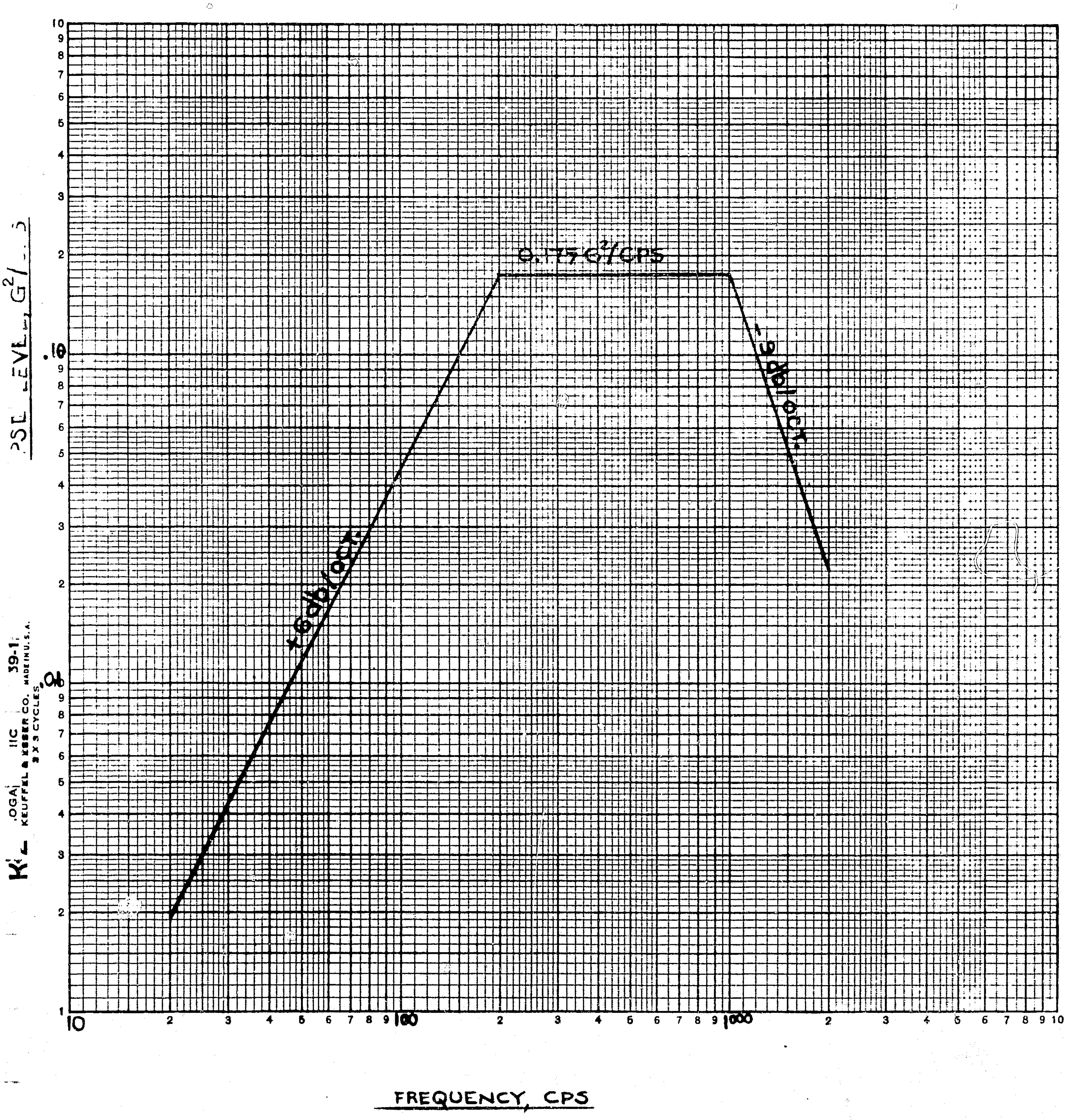
Z- AXIS

5 TO 14	CPS	0.25 G. PK.
14 TO 18	CPS	0.025 IN. D.A.
18 TO 20	CPS	0.40 G. PK.
20 TO 35	CPS	0.25 G. PK.

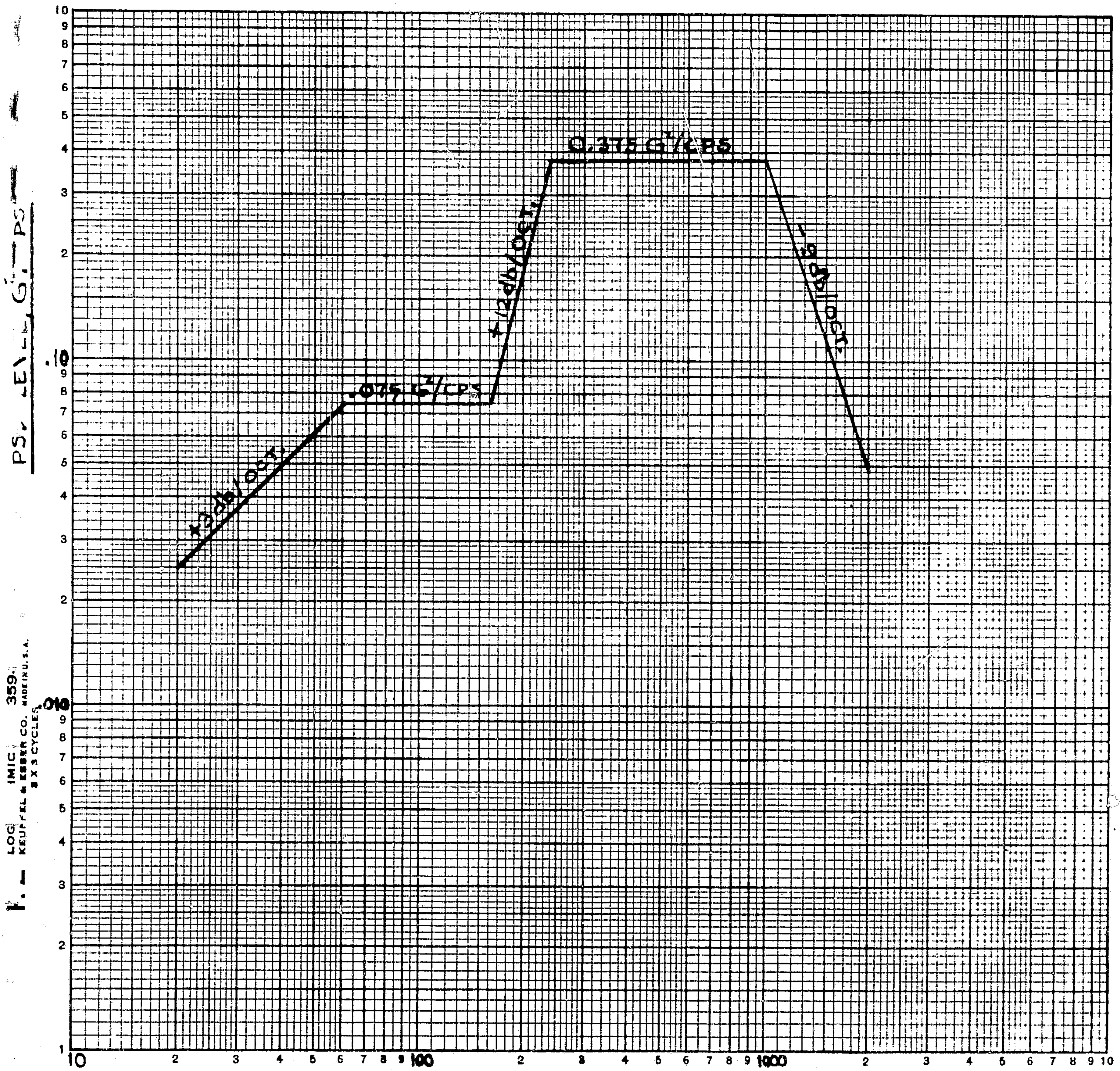
SHOCK SPECTRA DUE TO PYROTECHNIC DETONATION.



Y-AXIS RANDOM VIBRATION (MAX. Q LEVEL)



X&Z - AXES RANDOM VIBRATION (MAX. Q LEVEL)



FREQUENCY, CPS

E. MARGINS OF SAFETY

THE MINIMUM MARGINS OF SAFETY (M.S.) ARE CALCULATED FROM THE FOLLOWING RELATIONSHIP :

$$M.S. = \frac{\text{ALLOWABLE STRESS}}{\text{LIMIT LOAD STRESS} \times F.S.} - 1$$

WHERE F.S. = FACTOR OF SAFETY

WHEN THE ALLOWABLE STRESS IS OF THE NATURE OF AN ULTIMATE OR FRACTURE STRENGTH, THE FACTOR OF SAFETY USED IS 1.50. WHEN THE ALLOWABLE STRESS IS THE YIELD STRENGTH OF THE MATERIAL, THE FACTOR OF SAFETY USED IS 1.15. SINCE THE USE OF THE 1.50 AND 1.15 FACTORS OF SAFETY ARE PARALLEL REQUIREMENTS, THE PROCEDURE IS TO ESTABLISH WHICH CASE RESULTS IN THE LOWER MARGIN AND REPORT THAT ONE ONLY. MARGINS OF SAFETY IN EXCESS OF 1.00 WILL BE REPORTED AS LARGE.

TABLE OF MINIMUM MARGINS OF SAFETY

ITEM	CRITICAL LOAD (STRESS)	ALLOWABLE LOAD (STRESS)	REF. PG.	M. S.
BAFFLE ASS'Y. GUSSET.	COMPRESSION	INTERIVET BUCKLING	44	LARGE
—— RIVET SHEAR.	SHEAR	SHEAR (ULTIMATE)	45	LARGE
BAFFLE PLATE SHEET.	SHEAR	BEARING (YIELD)	45	LARGE
LOWER BAFFLE.	BENDING	TENSILE (ULTIMATE)	46	LARGE
ATTACH. - BAFFLE TOP TO INTERMEDIATE BAFFLE.	COMBINED LOADING	SHEAR & TENSILE	51	LARGE
ATTACH. - INTERMEDIATE BAFFLE TO LOWER BAFFLE.	SHEAR	BEARING (YIELD)	55	LARGE
ATTACH. - GUSSET TO BAFFLE BASE.	SHEAR	FRICTION CAPACITY	56	LARGE
BAFFLE ASS'Y. GUSSET LEG.	BENDING	TENSILE (YIELD)	57	LARGE
ATTACH. - BAFFLE BASE TO FRONT PLATE.	TENSION	TENSILE (YIELD)	60	+0.60
STANDOFF.	COMPRESSION	COMPRESSIVE	61	LARGE
A3 TIMING CIRCUIT PC BOARD.	BENDING	FLEXURE	65	+0.45
BOARD BOLTS.	COMBINED LOADING	TENSILE (ULTIMATE)	74	+0.39
BOARD STANDOFFS.	COMPRESSION	COMPRESSIVE	66	LARGE
RIGHT GUSSET.	BENDING	TENSILE (YIELD)	69	LARGE
SUPPORT BRACKET (PC BDS.).	FASTENER TENSION	TENSILE (YIELD)	76	+0.39
ATTACH. - P/M TUBE TO CONV. HSG.	SHEAR	FRICTION CAPACITY	79	LARGE
ATTACH. - ELECT. TO FRONT PLATE.	SHEAR	FRICTION CAPACITY	84	+0.84
MOUNTING PLATE FLANGE.	BENDING	TENSILE (YIELD)	86	+0.75
DC-DC CONV. HEAT SINK # 1.	BENDING OF SCREW	TENSILE (YIELD)	88	LARGE
DC-DC CONV. HEAT SINK # 2.	BENDING OF SCREW	TENSILE (YIELD)	91	LARGE
ATTACH. - GUSSETS TO MTG. PLATE.	SHEAR	SHEAR (ULT.)	72	+0.71
ATTACH. - TRANS. T2 TO HOUSING.	BENDING OF SCREW	TENSILE (YIELD)	92	+0.47
REGULAR BOARD.	BENDING	FLEXURE	93	LARGE
CONVERTER BOARD.	BENDING	FLEXURE	94	+0.59
CONVERTER HOUSING.	BENDING	TENSILE (ULTIMATE)	95	+0.98
FRONT PLATE.	BENDING	TENSILE (ULTIMATE)	96	LARGE
FRONT PLATE "BAFFLE-BEAM".	BENDING	TENSILE (ULTIMATE)	97	LARGE
FRONT PLATE LOCAL LUG.	BENDING	TENSILE (ULTIMATE)	99	LARGE
LUG UNDER ELECTRONICS.	BENDING	TENSILE (ULTIMATE)	100	LARGE
GRATING SUPPORT BEAM.	BENDING	TENSILE (ULTIMATE)	102	LARGE

M.S. TABLE (CONTINUED)

ITEM	CRITICAL LOAD (STRESS)	ALLOWABLE LOAD (STRESS)	REF. PG.	M.S.
ATTACH.—MOTOR MOUNT TO GRATING SUPPORT.	SHEAR	FRICTION CAPACITY	104	LARGE
MOTOR MOUNT STRUCTURE.	BENDING	TENSILE (ULTIMATE)	105	LARGE
FIDUCIAL MARK DETECT. BRACKET.	COMBINED LOADING	COMPRESSIVE (YIELD)	107	LARGE
GRATING PAD.	COMPRESSION	COMPRESSIVE (YIELD)	111	LARGE
GRATING BLANK.	COMPRESSION	COMPRESSIVE (YIELD)	112	+0.24
GRATING BOX.	BENDING	TENSILE (ULTIMATE)	114	LARGE
GRATING CLAMP.	BENDING	TENSILE (YIELD)	115	+0.55
TRIPLING MIRROR HOUSING.	BENDING	TENSILE (ULTIMATE)	116	LARGE
ATTACH.—HOUSING TO FRONT PLATE.	SCREW TENSION	TENSILE (YIELD)	117	LARGE
CLAMP.	BENDING	TENSILE (ULTIMATE)	118	LARGE
MIRROR PAD.	COMPRESSION	COMPRESSIVE	119	LARGE
TRIPLING MIRROR.	COMPRESSION	COMPRESSIVE	119	LARGE
WAVELENGTH CAMSHAFT.	SHEAR (TORSION)	SHEAR (ULTIMATE)	121	LARGE
CAMSHAFT DOWEL PIN.	SHEAR	SHEAR (ULTIMATE)	121	LARGE
CAMSHAFT KEY.	BENDING	TENSILE (YIELD)	123	LARGE
GRATING ARM AXIAL PIN.	SHEAR	SHEAR (ULTIMATE)	124	LARGE
GRATING ARM.	BENDING	TENSILE (ULTIMATE)	125	LARGE
GRATING ARM COLLAR.	BENDING	TENSILE (ULTIMATE)	126	LARGE
UVS HOUSING.	BENDING	TENSILE (ULTIMATE)	128	LARGE
ATTACH.— FRONT PLATE TO HOUSING.	SHEAR	FRICTION CAPACITY	130	+0.45
HOUSING FLANGE TAB.	BENDING	TENSILE (ULTIMATE)	131	LARGE
ATTACH.— ISOLATOR MOUNTS TO HSG.	COMBINED LOADING	SHEAR AND TENSILE	145	LARGE
ATTACH.— MOUNTS TO NR BKT.	COMBINED LOADING	TENSILE (YIELD)	149	+0.08
ATTACH.— AFT PLATE TO HOUSING.	SHEAR	FRICTION CAPACITY	150	LARGE
HOUSING HANDLING PADS.	BENDING	TENSILE (ULTIMATE)	152	LARGE
EBERT MIRROR.	BENDING	TENSILE	158	+0.60
EBERT MIRROR.	COMPRESSION	COMPRESSIVE	160	+0.32
MIRROR PAD.	COMPRESSION	COMPRESSIVE	159	LARGE
MIRROR CLAMP (SIDE).	BENDING	TENSILE (ULTIMATE)	161	LARGE
MIRROR CLAMP (CENTER).	BENDING	TENSILE (ULTIMATE)	162	LARGE
MIRROR CELL.	BENDING	TENSILE (ULTIMATE)	163	LARGE
MIRROR CELL COVER.	BENDING	TENSILE (YIELD)	164	+0.45
ATTACH.— CELL COVER TO CELL.	TENSION	TENSILE (YIELD)	165	LARGE

F. MATERIAL MECHANICAL PROPERTIES. (REF. C)

	ALUM. ALLOY 6061-T6 SHEET	ALUM. ALLOY 2024-T4 (*)	ALUM. ALLOY 356-T6 CAST.
F_{TU}	42000 psi	64000 psi	30000 psi
F_{Ty}	36000 psi	42000 psi	20000 psi
F_{cy}	35000 psi	40000 psi	20000 psi
F_{su}	27000 psi	40000 psi	25000 psi
$F_{bru\,(e/D=1.5)}$	67000 psi	98000 psi	48000 psi
$F_{bru\,(e/D=2.0)}$	88000 psi	124000 psi	63000 psi
$F_{bry\,(e/D=1.5)}$	50000 psi	69000 psi	34000 psi
$F_{bry\,(e/D=2.0)}$	58000 psi	79000 psi	40000 psi

* VALUES GIVEN FOR SHEET & PLATE (t = .010 - .249 IN.). ALL VALUES ESSENTIALLY THE SAME FOR T351 CONDITION

II. DETAILED ANALYSES.

GENERAL.

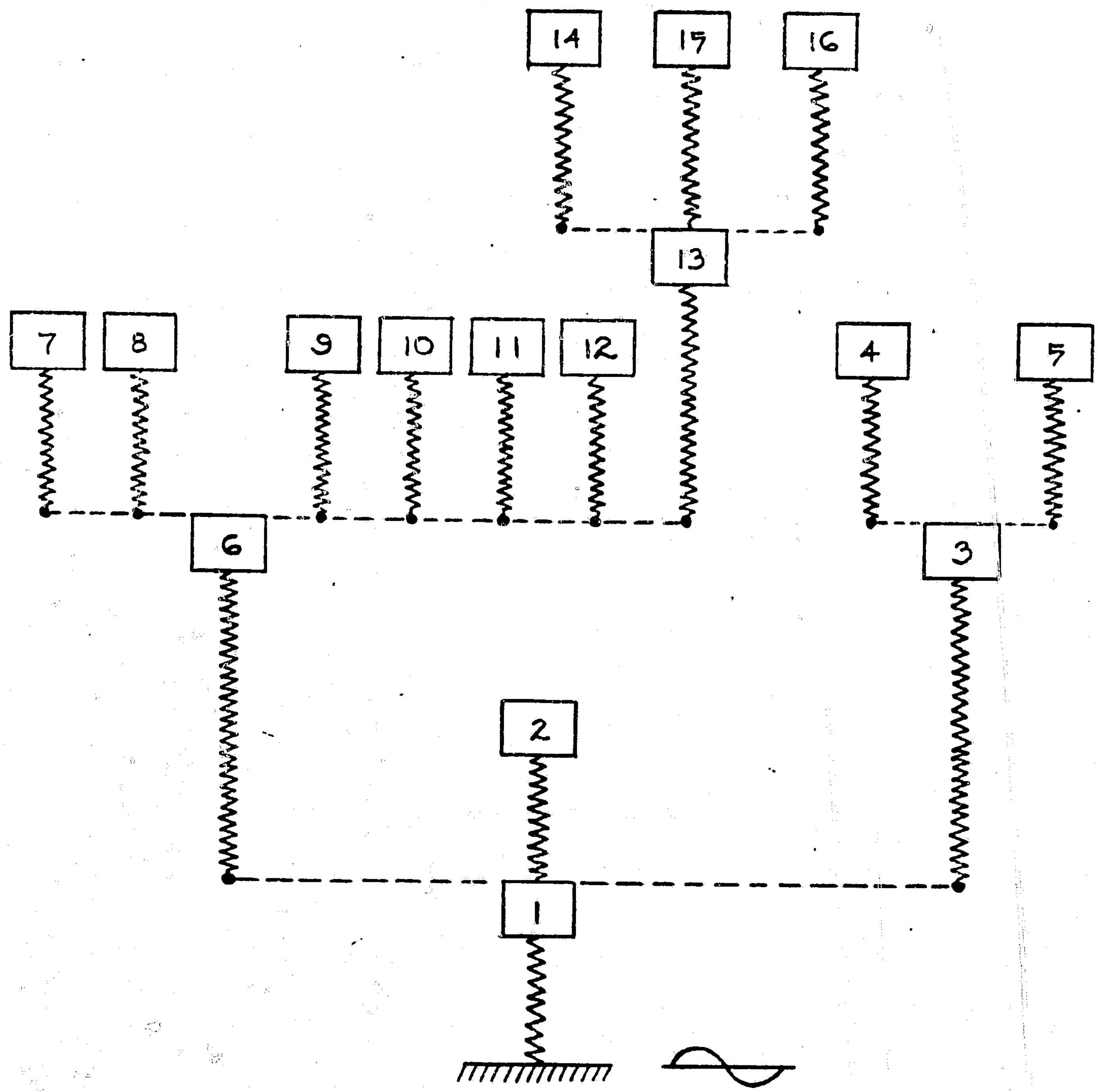
THE DYNAMIC MODELS USED TO DESCRIBE THE SPECTROMETER ARE SHOWN ON PAGES 27 AND 29. THE INDIVIDUAL SPRINGS AND MASSES ARE DEFINED IN TABULAR FORM ON PAGES 28 AND 30. THE MODAL DISPLACEMENTS OF ALL MASS POINTS FOR EACH MODE OF VIBRATION ARE TABULATED ON PAGES 31–33 FOR EACH OF THE THREE MUTUALLY PERPENDICULAR AXES OF EXCITATION. FOR THE SPECIFIC MODELS SELECTED THERE WERE SIX SIGNIFICANT MODES FOR X AND Z-AXIS EXCITATION AND TEN SIGNIFICANT MODES FOR Y-AXIS EXCITATION.

CONSIDERATION OF THE SPECTROMETER AS A SINGLE DEGREE OF FREEDOM SYSTEM AT THE FUNDAMENTAL FREQUENCY LEADS TO THE COMPARISON OF SHOCK AND 3σ RANDOM VIBRATION ACCELERATION RESPONSES TABULATED ON PAGE 40. THE RESPONSE TO THE INPUT SHOCK SPECTRUM, ASSUMED TO BE AT THE SPECTROMETER C.G., IS THE WEIGHTED-AVERAGE RESPONSE OF ALL THE INDIVIDUAL MASSES (I.E., $G = \Sigma W_i G_i / \Sigma W_i$ WHERE W_i IS THE WEIGHT OF AN ITEM AND G_i IS THE ACCELERATION OF THAT PARTICULAR ITEM). THE 3σ RANDOM VIBRATION RESPONSES, ALSO AT THE C.G., ARE DETERMINED BY THE CLASSICAL FORMULA ($G_{3\sigma} = 3\sqrt{0.5\pi f_n Q S_o}$) AS A FUNCTION OF FREQUENCY, TRANSMISSIBILITY AND POWER SPECTRAL DENSITY.

THE CRITICAL RESPONSES TO BOTH SHOCK AND RANDOM VIBRATION INPUTS ARE TABULATED ON PAGE 41 FOR THE COMPONENT PARTS OF THE SPECTROMETER. FOR A GIVEN ITEM AND AXIS OF VIBRATION THE GREATER OF THE TWO ACCELERATIONS IS USED IN THE ANALYSES THAT FOLLOW.

SECTIONS 1.0 AND 2.0 PRESENT THE THEORY, EQUATIONS AND BASIC ASSUMPTIONS USED IN DETERMINING THE RESPONSE ACCELERATIONS FOR SHOCK AND RANDOM VIBRATORY INPUTS TO A MULTI-DEGREE OF FREEDOM SYSTEM.

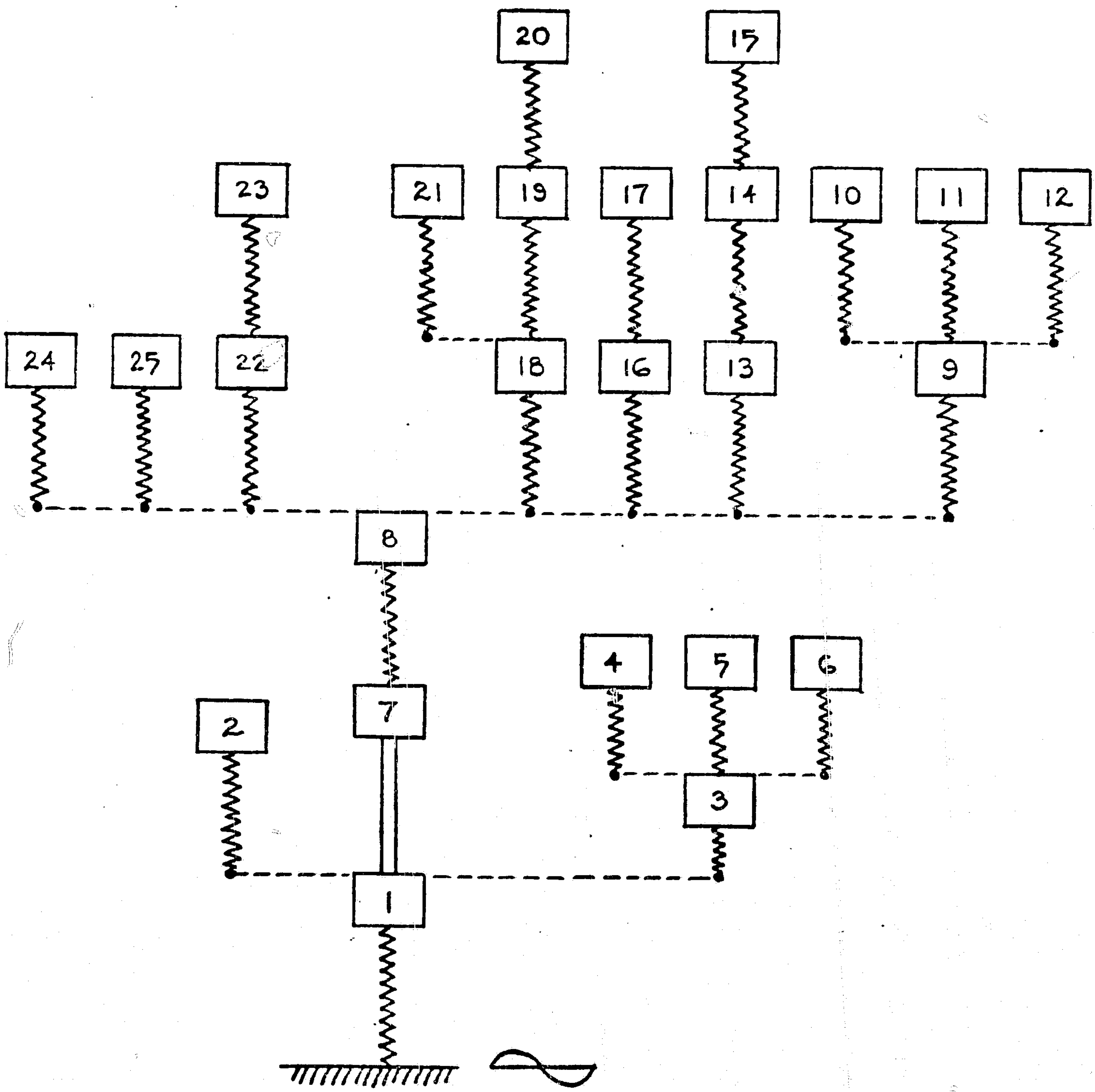
X&Y AXES EXCITATION



X&Y AXES EXCITATION

MASS PT.	COMPONENT	WEIGHT, LBS.	SPRING CONST. LBS/IN. K_X	K_Y
1	MOUNTING FEET	0.88	3.35×10^4	3.35×10^4
2	UVS HOUSING	10.80	1.51×10^5	2.43×10^5
3	AFT PLATE ASSEMBLY	3.00	3.26×10^5	5.25×10^5
4	AFT END CAP	0.35	1.74×10^5	2.80×10^5
5	EBERT MIRROR	3.20	2.36×10^4	2.36×10^4
6	FWD. PLATE ASS'Y.	3.87	7.66×10^5	1.23×10^6
7	MOTOR & HOUSING	1.07	3.25×10^4	2.72×10^5
8	GRATING	1.46	6.38×10^3	2.33×10^5
9	SLIT MIRROR	0.16	3.07×10^3	4.40×10^4
10	SLIT MIRROR	0.16	3.07×10^3	4.40×10^4
11	ENTRANCE SHUTTER	0.04	4.20×10^5	2.79×10^5
12	BAFFLE	3.21	1.20×10^5	3.37×10^4
13	DETECT. ELECTRONICS	4.48	9.28×10^4	9.28×10^4
14	PC BOARDS	0.75	3.27×10^5	2.43×10^3
15	CONVERTER	1.50	2.69×10^5	1.86×10^3
16	PM TUBE	1.38	5.38×10^5	7.45×10^3
		36.31		

Z - AXIS EXCITATION



Z-AXIS EXCITATION

MASS PT.	COMPONENT	WEIGHT, LBS.	K_z (LBS./IN.)
1	UVS HOUSING	8.85	3.35×10^4
2	AFT END CAP	0.35	2.62×10^3
3	MIRROR MTG. PADS	0.20	3.54×10^4
4	EBERT MIRROR	1.25	1.70×10^5
5	EBERT MIRROR	0.50	0.91×10^5
6	EBERT MIRROR	1.25	1.70×10^5
7	UVS HOUSING	5.83	∞
8	FWD. PLATE ASS'Y.	3.57	4.60×10^5
9	DETECT. ELECTRONICS	4.48	2.00×10^6
10	PM TUBE	1.38	5.38×10^5
11	CONVERTER	1.50	2.69×10^5
12	PC BOARDS	0.75	2.43×10^3
13	BAFFLE	0.53	∞
14	BAFFLE	2.36	0.92×10^6
15	SUN SENSOR	0.32	8.82×10^6
16	SUPPORT	0.16	1.08×10^6
17	GRATING	0.65	4.28×10^4
18	SUPPORT	0.16	1.08×10^6
19	SUPPORT	0.00	∞
20	GRATING	0.65	4.28×10^4
21	MOTOR	0.91	3.52×10^5
22	ENT. SHUTTER HSG.	0.30	∞
23	ENT. SHUTTER	0.04	2.79×10^5
24	SLIT MIRROR	0.16	4.75×10^5
25	SLIT MIRROR	0.16	4.75×10^5
		36.31	

TABULATION OF MODAL DISPLACEMENTS.

MASS NO. ↓	MODE 1			MODE 2			MODE 3		
	X-AXIS	Y-AXIS	Z-AXIS	X-AXIS	Y-AXIS	Z-AXIS	X-AXIS	Y-AXIS	Z-AXIS
f_n, HZ. →	91.535	88.205	93.266	209.839	115.887	180.370	266.085	178.422	271.706
1	0.903	0.759	0.964	0.691	0.813	1.076	0.252	3.090	0.714
2	0.962	0.787	1.094	1.018	0.866	1.936	0.522	3.612	-94.575
3	0.920	0.767	1.049	0.822	0.829	1.554	-0.091	3.274	2.529
4	0.922	0.768	1.056	0.829	0.830	1.593	-0.092	3.287	2.678
5	1.041	0.860	1.054	2.107	1.018	1.582	-4.726	5.857	2.639
6	0.923	0.770	1.056	0.597	0.817	1.593	0.353	2.985	2.678
7	0.950	0.773	0.964	0.701	0.821	1.076	0.464	3.024	0.714
8	1.148	0.774	1.000	-20.238	0.824	0.999	-0.539	3.047	0.997
9	0.966	0.773	1.003	0.780	0.821	0.964	0.567	3.021	1.024
10	0.966	0.773	1.006	0.780	0.821	0.973	0.567	3.021	1.044
11	0.923	0.771	1.008	0.597	0.817	0.983	0.353	2.987	1.069
12	0.945	0.834	1.382	0.678	0.940	-37.228	0.438	4.325	-0.771
13	0.998	0.853	1.000	0.988	0.746	0.999	0.981	0.398	0.997
14	1.000	1.130	1.002	0.999	1.294	1.009	0.998	-102.165	1.020
15	1.003	2.376	1.000	1.014	-7.000	1.000	1.023	-0.245	1.000
16	1.000	1.000	1.000	1.000	1.000	1.001	1.000	1.000	1.004
17			1.014			1.055			1.134
18			1.001			1.004			1.010
19			1.001			1.004			1.010
20			1.015			1.058			1.141
21			1.003			1.013			1.031
22			1.000			0.999			0.997
23			1.000			0.999			0.999
24			1.000			1.000			1.000
25			1.000			1.000			1.000

MASS NO. ↓	MODE 4			MODE 5			MODE 6		
	X-AXIS	Y-AXIS	Z-AXIS	X-AXIS	Y-AXIS	Z-AXIS	X-AXIS	Y-AXIS	Z-AXIS
f_n, HZ. →	335.128	224.475	329.380	429.458	274.370	711.790	485.072	335.906	839.298
1	-0.139	-0.041	0.567	-0.382	-0.361	-2.035	-0.793	-0.891	-0.862
2	-0.777	-0.054	-1.179	1.100	-0.549	0.344	1.102	-1.828	0.100
3	-0.129	-0.048	-6.950	-0.414	-0.177	0.327	-0.927	-0.848	0.073
4	-0.132	-0.048	-7.567	-0.430	-0.178	0.528	-0.974	-0.860	0.154
5	0.231	-0.159	-7.401	0.266	4.121	0.457	0.410	1.506	0.120
6	-0.022	-0.037	-7.567	-0.670	-0.411	0.528	-1.123	-0.742	0.154
7	-0.036	-0.038	0.567	-1.771	-0.424	-2.035	-5.386	-0.777	-0.862
8	0.014	-0.038	0.996	0.202	-0.432	0.983	0.249	-0.800	0.976
9	-0.056	-0.038	1.038	-34.694	-0.423	1.244	4.519	-0.774	1.417
10	-0.056	-0.038	1.068	-34.694	-0.423	1.435	4.461	-0.774	1.737
11	-0.023	-0.037	1.106	-0.671	-0.411	1.750	-1.125	-0.743	2.367
12	-0.032	-0.073	-0.429	-1.353	-1.536	-0.083	-3.146	7.564	-0.067
13	0.971	0.046	0.996	0.952	-0.425	0.983	0.938	-1.135	0.976
14	0.999	-0.079	1.029	0.995	0.309	1.154	0.993	0.444	1.228
15	1.037	-0.015	1.001	1.063	0.082	1.003	1.084	0.137	1.003
16	1.000	1.000	1.006	1.000	1.000	1.159	1.000	1.000	1.086
17			1.210			5.418			3.614
18			1.016			1.237			0.300
19			1.016			1.237			0.300
20			1.222			5.784			-17.365
21			1.046			1.428			0.382
22			0.996			0.983			0.976
23			0.998			0.990			0.986
24			1.000			1.000			1.000
25			1.000			1.000			1.000

MASS NO. ↓	MODE 7			MODE 8			MODE 9			MODE 10		
	X	Y-AXIS	Z	X	Y-AXIS	Z	X	Y-AXIS	Z	X	Y-AXIS	Z
fn, HZ. →	—	468.585	—	—	705.823	—	—	1159.577	—	—	1405.216	—
1		0.004			9.394			-110.527			103.789	
2		1.212			-7.442			21.644			-13.031	
3		0.004			12.948			-727.577			-362.873	
4		0.004			13.828			-878.425			-455.242	
5		-0.002			-2.194			41.267			13.766	
6		-0.235			11.116			134.595			313.895	
7		-0.258			13.899			292.859			1520.999	
8		-0.274			16.322			967.148			-1183.249	
9		-0.256			13.641			268.950			1178.525	
10		-0.256			13.641			268.950			1178.525	
11		-0.236			11.198			137.295			323.243	
12		0.207			-2.889			-11.138			-17.229	
13		-3.155			-8.428			-24.469			-36.459	
14		0.533			0.573			0.583			0.588	
15		0.185			0.210			0.231			0.220	
16		1.000			1.000			1.000			1.000	
17												
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II. DETAILED ANALYSES.

1.0 SHOCK SPECTRUM RESPONSE.

A COMPUTER PROGRAM WAS SET UP FOR THE CALCULATION OF MODAL RESPONSES OF A MULTI-DEGREE-OF-FREEDOM SYSTEM TO A SHOCK SPECTRUM. THE PROGRAM WAS BASED ON THE NASA/MSC SOLUTION FOR SYSTEM RESPONSES. BASIC TO THE ANALYSIS IS THE ASSUMPTION THAT FOR A GIVEN NATURAL FREQUENCY AND DAMPING RATIO, THE RATIO OF THE PEAK GENERALIZED RESPONSE ($\hat{\ddot{\xi}}_K$) OF A MULTI-DEGREE-OF-FREEDOM (MDOF) SYSTEM TO THAT OF A SINGLE-DEGREE-OF-FREEDOM (SDOF) SYSTEM WITH THE SAME NATURAL FREQUENCY WILL BE THE SAME AS THE RATIO OF THE STEADY-STATE GENERALIZED RESPONSES OF THE TWO SYSTEMS WHEN EXCITED BY A UNIT SINUSOIDAL BASE MOTION INPUT.

SUBSCRIPTS:

- i = GRID POINT NUMBER.
- j = EXCITATION AXIS OR DEFLECTION AXIS; EITHER X, Y, OR Z.
- K = MODE NUMBER.
- n = TOTAL NUMBER OF GRID POINTS.

SYMBOLS:

- m_{ij} = MASS AT THE i^{TH} GRID POINT IN THE j^{TH} DIRECTION.
- ϕ_{ijK} = MODE SHAPE FOR THE i^{TH} GRID POINT IN THE j^{TH} DIRECTION FOR THE K^{TH} MODE.
- M_K = GENERALIZED MASS FOR THE K^{TH} MODE ($\{\phi\}^T[m]\{\phi\}$).
- ζ_K = PERCENT OF CRITICAL DAMPING FOR THE K^{TH} MODE (C/C_c).
- $\ddot{A}_{jK}$ = ACCELERATION INPUT AT THE BASE IN THE j^{TH} DIRECTION FOR THE K^{TH} MODE.
- $\ddot{\xi}_{jK}$ = GENERALIZED RESPONSE OF THE K^{TH} MODE FOR AN EXCITATION IN THE j^{TH} AXIS.
- $\{\ddot{R}_{jK}\}$ = SYSTEM RESPONSE OF THE K^{TH} MODE FOR AN EXCITATION IN THE j^{TH} AXIS, I.E., $\ddot{\xi}_{jK}\{\phi\}_K$.
- $\{\ \}$ = COLUMN MATRIX.
- $[\ ]$ = SQUARE MATRIX.

THE GENERALIZED RESPONSE OF A MDOF SYSTEM TO A BASE MOTION INPUT IS AS FOLLOWS:

$$\ddot{\xi}_{jK} = \frac{\ddot{A}_{jK}}{2M_K\zeta_K}\sum_{i=1}^{n} m_{ij}\phi_{ijK} \tag{1}$$

FOR A SDOF SYSTEM THIS REDUCES TO

$$\ddot{\xi}_K = \frac{\ddot{A}_K}{2\zeta_K} \quad \text{IF } \phi = 1 \tag{2}$$

FOR RATIOING THE GENERALIZED RESPONSES, THE MODE SHAPES FOR THE TWO SYSTEMS MUST BE NORMALIZED THE SAME. IT IS CONVENIENT FOR THIS DISCUSSION TO

ASSUME THAT ALL MODE SHAPES ARE NORMALIZED TO A MAXIMUM OF 1, ALTHOUGH LATER IT WILL BE SHOWN THAT THIS IS NOT NECESSARY IF WE ONLY WANT THE SYSTEM RESPONSES $\{\ddot{R}_{jK}\}$.

FOR A UNIT INPUT, EQUATION (1) BECOMES

$$\ddot{\xi}'_{jK} = \frac{1}{2M_K\zeta_K} \sum_{i=1}^{n} m_{ij}\phi_{ijK} \qquad (3)$$

AND EQUATION (2) BECOMES

$$\ddot{\xi}'_K = \frac{1}{2\zeta_K} \qquad (4)$$

NOW, THE SHOCK SPECTRUM VALUE AT A GIVEN FREQUENCY IS THE PEAK RESPONSE OF A SDOF, WITH THE INDICATED VALUE OF ζ, SUBJECTED TO A SPECIFIED AMPLITUDE-TIME HISTORY. LET $\hat{\ddot{\xi}}_{jK}$ INDICATE PEAK GENERALIZED SHOCK RESPONSE. FROM THE BASIC ASSUMPTION

$$\frac{\hat{\ddot{\xi}}_{jK}}{\hat{\ddot{\xi}}_K} = \frac{\ddot{\xi}'_{JK}}{\ddot{\xi}'_K}$$

$$\therefore \ \hat{\ddot{\xi}}_{jK} = \frac{\ddot{\xi}'_{jK}}{\ddot{\xi}'_K}\left(\hat{\ddot{\xi}}_K\right) \qquad (5)$$

FOR A SDOF WITH $\phi = 1$,

$$\ddot{R}_K = \ddot{\xi}_K \quad \text{AND} \quad \hat{\ddot{R}}_K = \hat{\ddot{\xi}}_K$$

$$\therefore \ \hat{\ddot{\xi}}_{JK} = \frac{\ddot{\xi}'_{JK}}{\ddot{\xi}'_K}\left(\hat{\ddot{R}}_K\right) \qquad (6)$$

SINCE THE SHOCK RESPONSE SPECTRUM IS THE TRANSIENT RESPONSE OF A SDOF, $\hat{\ddot{R}}_K$ IS DIRECTLY OBTAINABLE FROM THE SHOCK SPECTRUM PLOT.

IF ζ IS THE SAME FOR THE SHOCK SPECTRUM AND THE SYSTEM UNDER CONSIDERATION,

$$\frac{\ddot{\xi}'_{JK}}{\ddot{\xi}'_K} = \frac{\dfrac{\sum_{i=1}^{n} m_{ij}\phi_{ijK}}{2M_K\zeta_K}}{\dfrac{1}{2\zeta_K}} = \frac{\sum_i m_{ij}\phi_{ijK}}{M_K} \qquad (7)$$

FROM EQUATIONS (6) AND (7)

$$\hat{\ddot{\xi}}_{JK} = \frac{\sum_i m_{ij}\phi_{ijK}}{M_K}\left(\hat{\ddot{R}}_K\right) \qquad (8)$$

THEN

$$\{\hat{\ddot{R}}_{jK}\} = (\hat{\ddot{\xi}}_{jK})\{\phi\}_K \qquad (9)$$

$\{\hat{\ddot{R}}_{jK}\}$ WILL HAVE THE UNITS OF $\hat{\ddot{R}}_K$

For a specific grid point, $\hat{R}_{jk}$ may be plotted against the modal frequen[illegible] to produce a response spectrum for that grid point in the j-axis, althou[illegible] this will not be a shock response spectrum of SDOF's as discussed earli[illegible]. For the case where ϕ_{MAX} not equal to 1, but some value λ:

$$\phi_{MAX} = \lambda$$

$$\therefore \{\phi\} = \frac{1}{\lambda}\{\Phi\} \quad \text{where } \Phi \text{ is the unnormalized mode shape}$$

$$M_K = \{\phi\}_K^T[m]\{\phi\}_K = \frac{1}{\lambda^2}\{\Phi\}_K^T[m]\{\Phi\}_K = \frac{1}{\lambda^2}\overline{M}_K$$

where $[m]$ is mass matrix

Now equation (9) becomes

$$\boxed{\{\hat{R}_{jK}\} = \frac{\sum_i m_{ij}\frac{1}{\lambda}\Phi_{ijK}}{\frac{1}{\lambda^2}\overline{M}_K}(\hat{R}_K)\left(\frac{1}{\lambda}\right)\{\Phi\}_K} \qquad (1[illegible])$$

Since the λ's cancel, the system response will be the same regardless o[illegible] normalization method used if the mode shapes and generalized masses are [illegible] consistent.

II. DETAILED ANALYSES.

2.0 RANDOM VIBRATION RESPONSE.

A COMPUTER PROGRAM WAS SET UP TO SIMILARLY DETERMINE RESPONSES OF THE SYSTEM AS A RESULT OF THE IMPOSED RANDOM VIBRATION SPECTRUM. THE LUMPED-MASS MATHEMATICAL MODELS USED IN THE SHOCK SPECTRUM ANALYSIS WERE ALSO USED FOR RANDOM VIBRATION.

THE BASIC ASSUMPTION MADE IN THE RANDOM VIBRATION ANALYSIS IS THAT THE RATIO OF THE GENERALIZED 3σ- RESPONSE FOR A MULTI-DEGREE-OF-FREEDOM SYSTEM (MDOF) TO THAT OF THE 3σ- RESPONSE FOR A SINGLE-DEGREE-OF-FREEDOM SYSTEM (SDOF), BOTH DETERMINED FOR THE r^{TH} MODE, IS EQUAL TO THE RATIO OF THE STEADY-STATE GENERALIZED RESPONSES OF THE TWO SYSTEMS WHEN EXCITED BY A UNIT SINUSOIDAL BASE MOTION INPUT. THAT IS

$$\frac{[(\text{GENERALIZED } 3\sigma\text{- RESPONSE})_{MDOF} = (GA3)_r]}{[(3\sigma\text{- RESPONSE})_{SDOF} = 3\sqrt{(\pi/2) fn_r Q_r So_r}\,]} = \frac{\ddot{X}_{MDOF}}{\ddot{X}_{SDOF}} \qquad (1)$$

WHERE fn_r = FREQUENCY OF r^{TH} MODE, HZ.

Q_r = $(1/2\zeta_r)$ = MAGNIFICATION FACTOR.

So_r = POWER SPECTRAL DENSITY FOR THE r^{TH} MODE.

THE RIGHT-HAND SIDE OF EQUATION (1) HAS ALREADY BEEN EVALUATED (SEE SECTION II 1.0) AND SHOWN TO BE

$$\frac{\ddot{X}_{MDOF}}{\ddot{X}_{SDOF}} = \frac{\sum_i m_i \phi_{ir}}{\{\phi_{ir}\}^T [m_i] \{\phi_{ir}\}} \qquad (2)$$

JUST AS IT WAS CONVENIENT TO DENOTE THE DENOMINATOR $\{\phi\}^T[m]\{\phi\}$ BY M_r AND DEFINE IT AS THE GENERALIZED MASS ASSOCIATED WITH THE r^{TH} MODE, THE NUMERATOR $\sum m_i \phi_{ir}$ CAN BE DEFINED AS THE PARTICIPATION FACTOR FOR THE r^{TH} MODE AND REPRESENTED BY

$$\Gamma_r = \sum_i m_i \phi_{ir} \qquad (3)$$

EQUATION (1) MAY BE WRITTEN AS

$$(GA3)_r = \frac{\Gamma_r}{M_r}\left\{3\sqrt{\frac{\pi}{2} fn_r Q_r So_r}\right\} = \frac{\Gamma_r}{M_r}\left[(\ddot{X})_{3\sigma}\right]_r \qquad (4)$$

WHERE $\left[(\ddot{X})_{3\sigma}\right]_r = 3\sqrt{\frac{\pi}{2} fn_r Q_r So_r}$

THE 3σ RESPONSES OF THE INDIVIDUAL MASSES FOR THE r<u>TH</u> MODE CAN NOW BE DETERMINED FROM THE PRODUCT OF THE GENERALIZED 3σ ACCELERATION AND THE MODAL DISPLACEMENTS OF THE MASSES

$$_{3\sigma}(\ddot{X}_i)_r = (GA3)_r \times (\phi ir) \qquad (5)$$

WHERE ϕir = NORMALIZED DISPLACEMENT OF MASS i FOR THE r<u>TH</u> MODE.

IN THE SOLUTION OF THE RANDOM VIBRATION OF MULTI-DEGREE-OF-FREEDOM SYSTEMS, IT IS OBSERVED THAT THE MAGNIFICATION FACTORS HAVE REGIONS OF PRONOUNCED PEAKS IN THE NEIGHBORHOODS OF THE CORRESPONDING FREQUENCIES. ADDITIONALLY, THE FREQUENCIES WERE FOUND TO BE FAR ENOUGH APART TO JUSTIFY ASSUMING NO CROSS-PRODUCT CONTRIBUTION TO THE TOTAL RESPONSE. THE COMPUTER SOLUTION THUS GIVES THE 3σ-RESPONSE OF THE j<u>TH</u> MASS SUMMED OVER THE K MODES OF VIBRATION AS

$$\boxed{_{3\sigma}(\ddot{\eta}_j) = \sqrt{\sum_{r=1}^{K} \left[_{3\sigma}(\ddot{X}_j)_r \right]^2}} \qquad (6)$$

THE 3σ RESPONSES FOR THE MASSES ARE TABULATED ON PAGE 41.

TABULATION OF RANDOM VIBRATION PARAMETERS.

AXIS	MODE	NATURAL FREQ. f_n, HZ.	PSD*, S_0 G^2/HZ INPUT	** $[(\ddot{X})_{3\sigma}]$, G	PARTICIPATION FACTOR, Γ_r	GENERALIZED MASS, M_r
X	1	91.535	0.075	31.153	0.091476	0.089287
X	2	209.839	0.220	80.786	0.013308	1.650350
X	3	266.085	0.375	118.770	0.003018	0.218358
X	4	335.128	0.375	133.291	-0.001053	0.033136
X	5	429.458	0.375	150.889	-0.001758	1.082031
X	6	485.072	0.375	160.361	-0.002860	0.256783
Y	1	88.205	0.032	19.976	0.082797	0.082429
Y	2	115.887	0.060	31.353	0.051354	0.259553
Y	3	178.442	0.135	58.354	0.082364	21.342285
Y	4	224.475	0.175	74.522	-0.000698	0.003993
Y	5	274.370	0.175	82.389	-0.004072	0.178178
Y	6	335.906	0.175	91.161	-0.006704	0.624622
Y	7	468.585	0.175	107.670	0.000014	0.162185
Y	8	705.823	0.175	132.144	0.016003	7.107428
Y	9	1159.577	0.115	137.303	-0.069893	8886.992183
Y	10	1405.216	0.063	111.872	0.044432	15173.433594
Z	1	93.266	0.075	31.447	0.094051	0.094499
Z	2	180.370	0.120	55.316	0.028073	2.802640
Z	3	271.706	0.375	120.018	0.008210	8.228740
Z	4	329.380	0.375	132.143	0.004435	0.529402
Z	5	711.790	0.375	194.255	-0.003425	0.329245
Z	6	839.298	0.375	210.938	-0.001072	0.636150

* REF. DESIGN CRITERIA, PAGES 20, 21.
** Q = 10

FOR THE ENTIRE SPECTROMETER AS A SINGLE DEGREE OF FREEDOM SYSTEM, THE 3σ RANDOM VIBRATION ACCELERATIONS ARE CALCULATED FROM THE FOLLOWING RELATIONSHIP :

$$\eta_{3\sigma} = 3\sqrt{\frac{\pi}{2} f_n Q S_o^{*}} \qquad \text{(REF. 9)}$$

* THE POWER SPECTRAL DENSITY, So, WILL INCLUDE A FACTOR OF 2.5 (REF. PG. 18) TO ACCOUNT FOR TRANSONIC FLIGHT CONDITIONS. THE ASSUMED TRANSMISSIBILITY Q = 10.

$$(\eta_{3\sigma})_X = 3\sqrt{\frac{\pi}{2}\{f_{n_X} = 91.5\}(Q=10)\{2.5(0.030)\}} = \underline{31.1\,g}$$

$$(\eta_{3\sigma})_Y = 3\sqrt{\frac{\pi}{2}\{f_{n_Y} = 88.2\}(Q=10)\{2.5(0.014)\}} = \underline{20.9\,g}$$

$$(\eta_{3\sigma})_Z = 3\sqrt{\frac{\pi}{2}\{f_{n_Z} = 93.3\}(Q=10)\{2.5(0.030)\}} = \underline{31.4\,g}$$

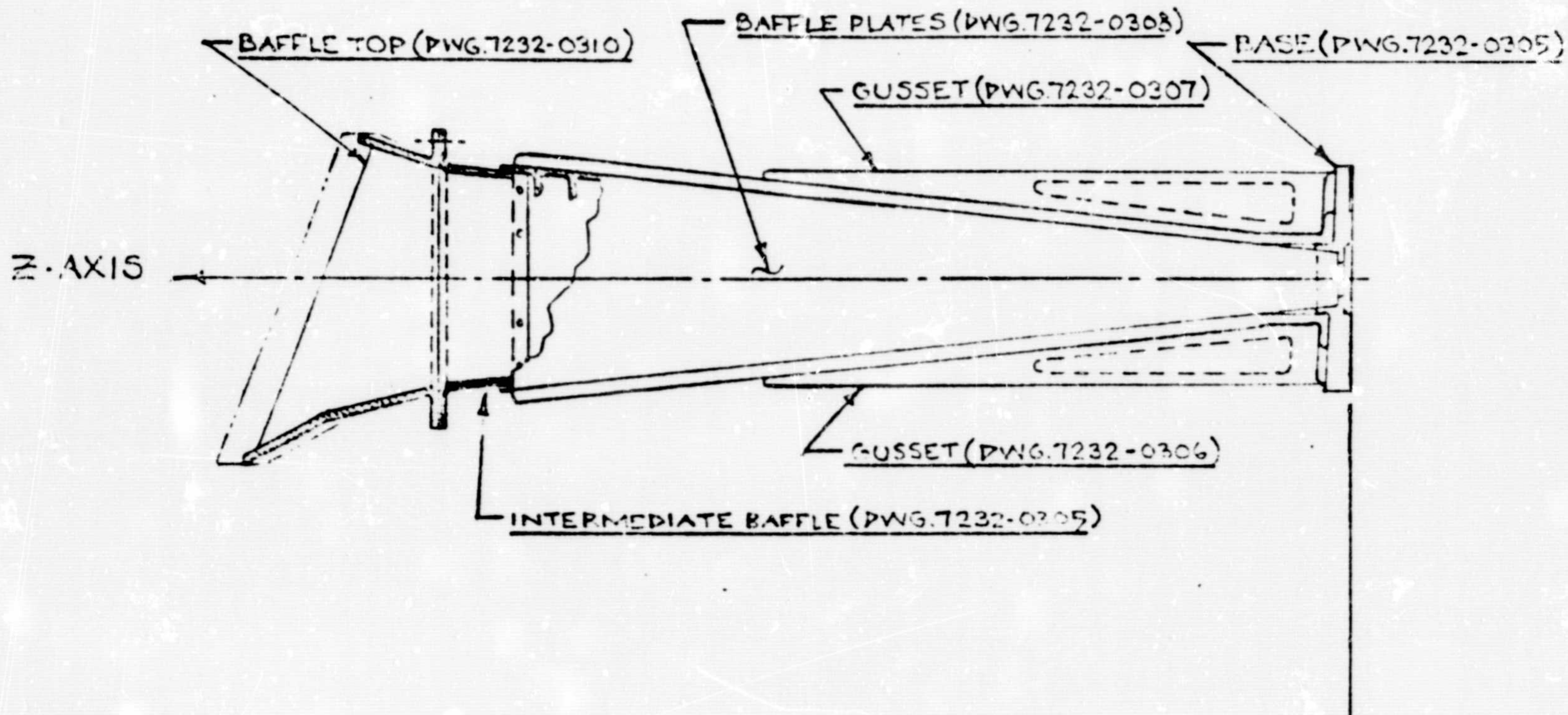
<u>COMPARISON OF SHOCK AND 3σ RANDOM VIBRATION ACCELERATION RESPONSES FOR UVS UNIT AT FUNDAMENTAL FREQUENCY.</u>

UVS AXIS	SHOCK	RANDOM
X	24.4 g	31.1 g
Y	20.4 g	20.9 g
Z	25.7 g	31.4 g

COMPARISON OF SHOCK AND 3σ RANDOM VIBRATION ACCELERATION RESPONSES FOR UVS COMPONENTS

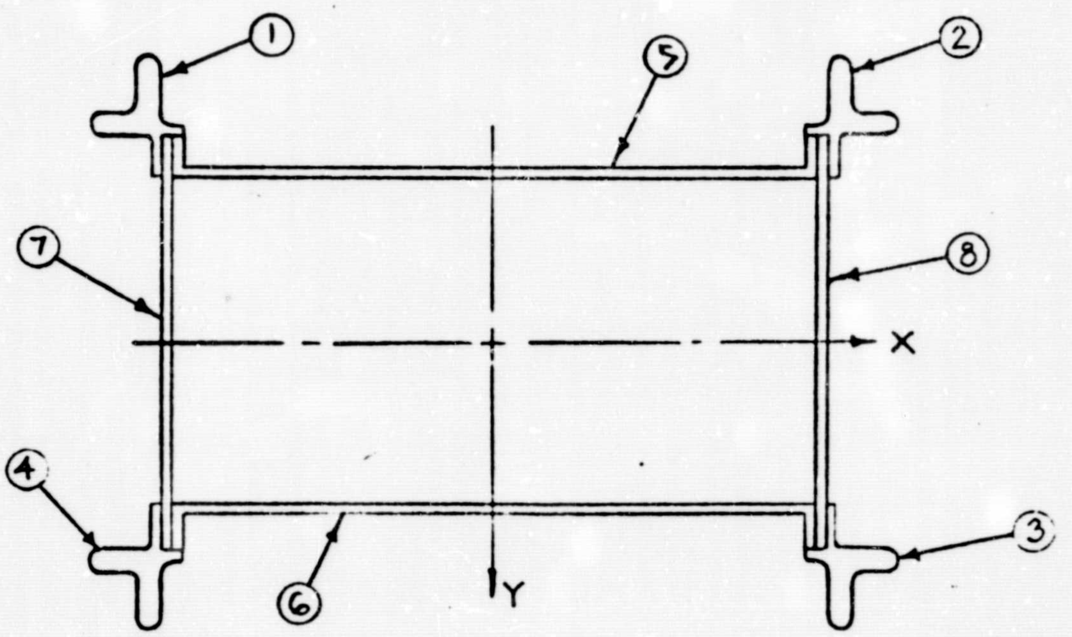
	X-AXIS		Y-AXIS		Z-AXIS	
COMPONENT	SHOCK	RANDOM	SHOCK	RANDOM	SHOCK	RANDOM
DETECT. ELECT.	25.4 g	32.1 g	27.4 g	18.0 g	28.2 g	35.7 g
PC BOARDS	25.4	32.2	37.2	33.3	36.0	47.9
CONVERTER	25.5	32.3	55.1	64.5	27.3	31.8
PM TUBE	25.4	32.2	26.1	24.8	25.8	31.6
BAFFLE	24.4	30.7	27.1	19.5	25.7	31.5
MIRROR	26.5	34.2	21.0	20.1	27.1	34.1
MOTOR	41.8	31.8	46.2	16.9	25.7	31.6
AFT END CAP	23.4	29.5	46.0	16.8	28.1	36.1
SLIT MIRROR	34.9	33.0	45.3	16.9	25.7	31.4
GRATING	29.2	39.0	54.3	17.2	87.5	34.0
FWD. PLATE	23.5	29.5	37.0	16.7	25.6	31.4
SHUTTER	23.5	29.5	37.2	16.7	25.7	31.4

3.0 ANALYSIS OF BAFFLE ASSEMBLY.
(REF. RAY LEE DWG. 7232-0300 "BAFFLE ASSEMBLY")



3.1 SECTION PROPERTIES

FOR ANALYTICAL PURPOSES, ASSUME BAFFLE TO BE UNIFORMLY LOADED CANTILEVER BEAM WHOSE AVERAGE CROSS-SECTION IS CONSTANT OVER ITS ENTIRE LENGTH.



AVG. CROSS-SECTION OF ASSEMBLY

ITEM	DESCRIPTION	AREA, A	X	Y	AX	AX²	Io Y-Y	AY	AY²	Io X-X
1	GUSSET	0.1100	-2.640	-1.690	-0.2904	0.7667	0.0000	-0.1859	0.3142	0.0000
2	GUSSET	0.1100	2.640	-1.690	0.2904	0.7667	0.0000	-0.1859	0.3142	0.0000
3	GUSSET	0.1100	2.640	1.690	0.2904	0.7667	0.0000	0.1859	0.3142	0.0000
4	GUSSET	0.1100	-2.640	1.690	-0.2904	0.7667	0.0000	0.1859	0.3142	0.0000
5	0.05 × 5.05	0.2525	0.000	1.400	0.0000	0.0000	0.5366	-0.3535	0.4949	0.0000
6	0.05 × 5.05	0.2525	0.000	1.400	0.0000	0.0000	0.5366	0.3535	0.4949	0.0000
7	3.25 × 0.05	0.1625	-2.550	0.000	-0.4144	1.0567	0.0000	0.0000	0.0000	0.1430
8	3.25 × 0.05	0.1625	2.550	0.000	0.4144	1.0567	0.0000	0.0000	0.0000	0.1430
Σ		1.2700			0.0000	5.1802	1.0732	0.0000	2.2466	0.2860

$$\bar{X} = (\Sigma AX)/(\Sigma A) = 0$$
$$\bar{Y} = (\Sigma AY)/(\Sigma A) = 0$$

$$I_{X-X} = \Sigma I o_{X-X} + \Sigma AY^2 - \bar{Y} \cdot \Sigma AY = \underline{2.533 \text{ IN.}^4}$$
$$I_{Y-Y} = \Sigma I o_{Y-Y} + \Sigma AX^2 - \bar{X} \cdot \Sigma AX = \underline{6.253 \text{ IN.}^4}$$

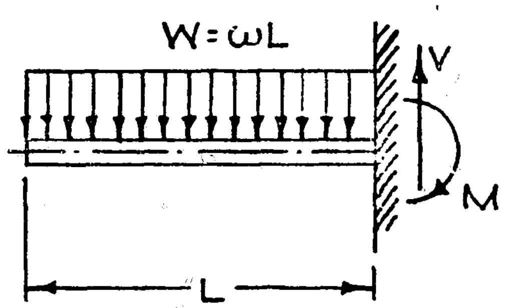
3.2 DYNAMIC LOADING CONDITION.

FROM THE SHOCK SPECTRUM AND RANDOM VIBRATION ANALYSES, THE PEAK BAFFLE ACCELERATIONS WERE FOUND TO BE :

$$\left.\begin{array}{l} G_X = 30.7\ g \\ G_Y = 27.1\ g \\ G_Z = 31.5\ g \end{array}\right\} \quad \text{(REF. PG. 41)}$$

NOTE : RANDOM VIBRATION IS SEEN TO BE MORE CRITICAL THAN SHOCK FOR THE BAFFLE ASSEMBLY IN BOTH THE X & Z – DIRECTIONS. THE HIGH INPUT POWER SPECTRAL DENSITIES ARE APPLIED FOR ONLY 10 SECONDS DURATION. THEREFORE, THE NUMBER OF CYCLES AT THE 3σ- LEVEL IS CONSIDERED NEGLIGIBLE. IF THE 3σ PEAKS ARE CALCULATED BASED ON 90 SECONDS DURATION WITHOUT THE 2.50 TRANSONIC LOAD FACTOR, THE X-AND Z-AXIS ACCELERATION LEVELS WOULD BE REDUCED BY A FACTOR OF $(1/\sqrt{2.5})$. THE NUMBER OF CYCLES AT THE 3σ- LEVEL, WHILE NO LONGER NEGLIGIBLE, WOULD STILL BE SMALL. IT IS THUS ASSUMED THAT THE MARGINS OF SAFETY MAY BE BASED UPON TENSILE YIELD OR ULTIMATE STRENGTH RATHER THAN UPON A FATIGUE ALLOWABLE.

3.3 SPACING OF RIVETS ATTACHING GUSSETS



$V_{MAX} = W = 3.4$ LBS/g

THE DYNAMIC LOADING CONDITION IS

$_x(V_{MAX})_{DYN} = 3.4(30.7)$

$_x(V_{MAX})_{DYN} = \underline{104.4 \text{ LBS. (LIMIT)}}$

$_Y(V_{MAX})_{DYN} = 3.4(27.1)$

$_Y(V_{MAX})_{DYN} = \underline{92.1 \text{ LBS. (LIMIT)}}$

LET $Q \equiv$ STATICAL MOMENT OF AREA OF 2 GUSSETS AROUND N.A.

$Q_X = A\bar{X} = 0.220(2.640) = 0.581 \text{ IN}^3$

$Q_Y = A\bar{Y} = 0.220(1.690) = 0.372 \text{ IN}^3$

SHEAR FLOW

$$q_X = \frac{_x(V_{MAX})_{DYN}\, Q_X}{I_{YY}} = \frac{104.4(0.581)}{6.253} = \underline{9.70 \text{ LBS/IN (LIMIT)}}$$

$$q_Y = \frac{_Y(V_{MAX})_{DYN}\, Q_Y}{I_{XX}} = \frac{92.1(0.372)}{2.533} = \underline{13.52 \text{ LBS/IN (LIMIT)}}$$

RIVET SPACING, $d = 1.62$ IN.

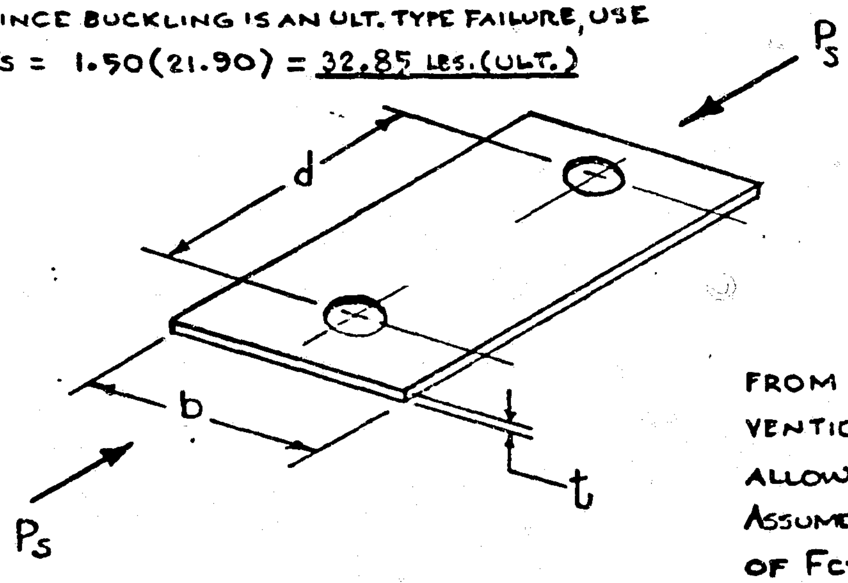
LET P_S = MAX. SHEAR FORCE ON RIVET

$$P_S = 1.62\, q_{MAX} = 1.62(13.52) = \underline{21.9 \text{ LBS. (LIMIT)}}$$

3.4 INTERIVET SHEET BUCKLING

SINCE BUCKLING IS AN ULT. TYPE FAILURE, USE

$P_S = 1.50(21.90) = \underline{32.85 \text{ LBS. (ULT.)}}$



$A = bt = 0.20(0.05)$

$A = 0.01 \text{ IN}^2$

SKIN THICKNESS, t, IS THAT OF BAFFLE PLATE.

$f_c = (P_S/A) = (32.85/.01)$

$f_c = \underline{3285 \text{ PSI (ULT.)}}$

FROM REF. p, "MAX. RIVET SPACING FOR PREVENTION OF SHEET INTERIVET BUCKLING", THE ALLOW. STRESS FOR 2024-T4 AL. ALLOY IS 31000 PSI. ASSUME FOR 6061-T6, ALLOW. IS REDUCED BY RATIO OF F_{CY} FOR BOTH MATERIALS.

$$\therefore F_c = 31000 \frac{(F_{CY})_{6061\text{-}T6 \text{ SHEET}}}{(F_{CY})_{2024\text{-}T4 \text{ SHEET}}} = 31000 \frac{35000}{40000} = 27125 \text{ PSI}$$

$$M.S. = \frac{F_c}{f_c} - 1 = \frac{27125}{3285} - 1 = \underline{\text{LARGE}}$$

3.5 SHEAR OF RIVETS

FOR AD-2 RIVETS, $P_{S_{ALLOW}}$ = 106 LBS. (REF. C)

APPLIED LOAD, P_S = 32.85 LBS.

$$M.S. = \frac{P_{S\,ALLOW}}{P_S} - 1 = \frac{106}{32.85} - 1 = \underline{LARGE}$$

3.6 BEARING IN SHEET

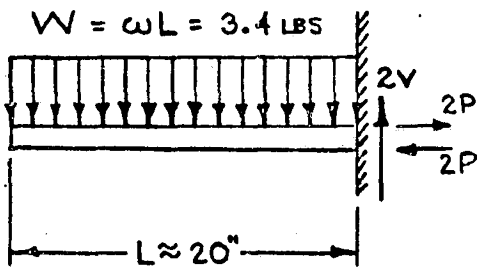
BEARING AREA IN BAFFLE PLATE, $A_{br} = dt = 0.062(0.05) = 0.0031 \text{ IN.}^2$

$$f_{br} = \frac{P_S}{A_{br}} = \frac{32.85}{0.0031} = \underline{10600 \text{ PSI}}$$

FOR 6061-T6 AL. ALLOY PLATE, $F_{bry\,(e/d=2.0)}$ = 58000 PSI (REF. C)

$$M.S. = \frac{F_{bry}}{f_{br}} - 1 = \frac{58000}{10600} - 1 = \underline{LARGE}$$

3.7 X-AXIS EXCITATION



FOR THE OVERALL SECTION

$$M_{MAX_{STATIC}} = \frac{1}{2} WL = \frac{1}{2}(3.4)(20) = 34 \text{ IN.-LBS.}$$

$$_x(M_{MAX})_{DYN} = 1.5(34)(30.7g)$$

$$_x(M_{MAX})_{DYN} = 1566 \text{ IN.-LBS.(ULT)}$$

$$I_{YY} = 6.253 \text{ IN}^4 \quad \text{(REF. PG. 43)}$$

$$C_Y = 2.750 \text{ IN.}$$

$$Z_Y = \frac{I_{YY}}{C_Y} = 2.274 \text{ IN}^3$$

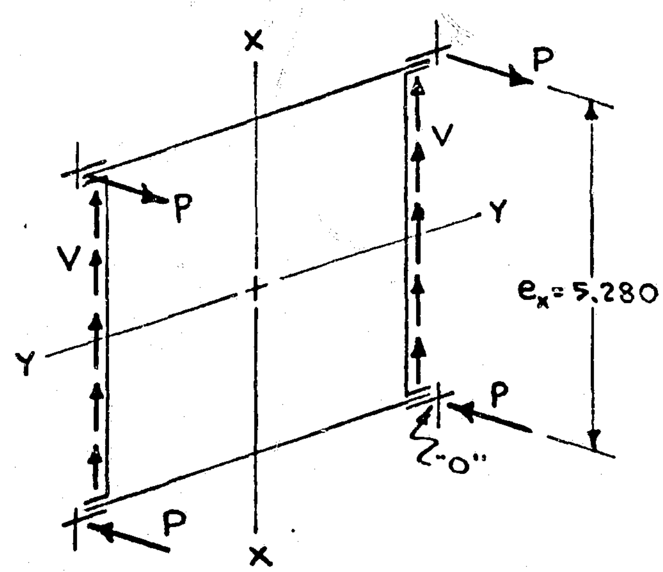
3.7.1 MAX. BENDING STRESS

$$f_{b_x} = \frac{_x(M_{MAX})_{DYN}}{Z_Y} = \frac{1566}{2.274} = \underline{690 \text{ PSI (ULT.)}}$$

FOR 6061-T6 ALUM. ALLOY SHEET, $F_{TU} = 42000$ PSI

M.S. LARGE BY INSPECTION

NOW ASSUME VERTICAL SHEAR CARRIED BY WEBS ONLY AND BENDING MOMENT REACTED BY AXIAL COUPLE FORCES ON THE 4 GUSSETS. A ONE INCH WIDTH OF WEB WILL BE CHECKED FOR SHEAR BUCKLING AND AFTER SOME CONSERVATIVE ASSUMPTIONS AS TO SIZE AND MEANS OF SUPPORT, THE GUSSETS WILL BE CHECKED FOR COMPRESSIVE BUCKLING.



FROM EQS. OF EQUILIBRIUM

$$\Sigma M_o = 0 = (WL/2) - 2Pe_x$$

$$2P(5.280) = 10.000W$$

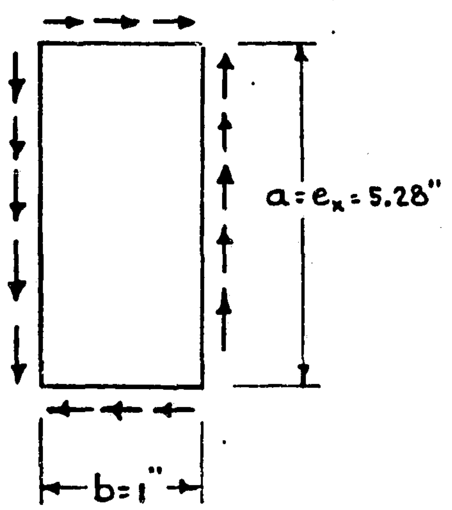
$$P = 0.95W$$

$$_xW_{DYN} = 1.5(3.4 \text{ LBS})(30.7g) = 156.6 \text{ LBS. (ULT.)}$$

$$\therefore \quad P_{DYN} = 0.95\,_xW_{DYN} = \underline{148.8 \text{ LBS. (ULT.)}}$$

$$V_{DYN} = 0.50\,_xW_{DYN} = \underline{78.3 \text{ LBS. (ULT.)}}$$

3.7.2 SHEAR BUCKLING OF BAFFLE PLATE



FROM REF. d, FIG. 15.1

FOR (a/b) = 5.28
K = EDGE COEFFICIENT = 4.9
(SIMPLY SUPPORTED EDGES)

SHEAR BUCKLING ALLOWABLE IS

$$Fs_{cr} = KE\left(\frac{t}{b}\right)^2$$

$$Fs_{cr} = 4.9(10\times10^6)\left[\frac{0.05}{1.00}\right]^2$$

$$Fs_{cr} = 122{,}500 \text{ psi}$$

FOR 6061-T6 AL. ALLOY PLATE, $Fs_u = 27000$ psi

∴ SHEAR BUCKLING NOT A PROBLEM.

FOR A RECTANGULAR SECTION

$$f_s = \frac{3}{2}\frac{V}{A} = \frac{3}{2}\,\frac{78.3}{(.05)(5.280)} = \underline{445 \text{ psi (ULT.)}}$$

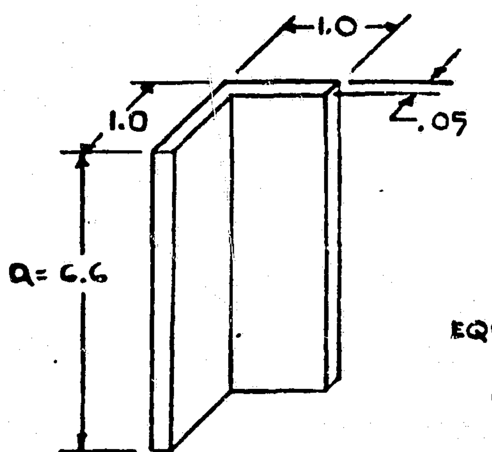
M.S. LARGE BY INSPECTION

3.7.3 COMPRESSIVE BUCKLING OF GUSSET

CONSERVATIVELY ASSUME $P_{DYN.}$ LOADS GUSSET IN COMPRESSION OVER ITS ENTIRE LENGTH. ASSUME GUSSET AREA AS GIVEN ON PAGE 43 IS 0.110 IN.^{2}

$$\therefore f_c = \frac{P_{DYN}}{A} = \frac{148.8}{0.110} = \underline{1350 \text{ psi (ULT.)}}$$

ASSUME GUSSET IS EQUAL LEGGED ANGLE CONSTANT OVER ITS LENGTH WITH ONE SIDE FREE (OTHER SIDE AND ENDS SIMPLY SUPPORTED.).



b = 1.0 IN.
a = 6.6 IN.
(a/b) = 6.6
t = .05 IN.

FROM FIG. 14.28 OF REF. d, FOR (a/b) = 6.6
K = 0.385

EQUIVALENT SLENDERNESS RATIO

$$(L'/\rho)_{EQUIV.} = \frac{\pi}{\sqrt{K}}\frac{b}{t} \approx 101$$

FROM FIG. 14.32, $Fc_{cr} = 10000$ psi

$$M.S. = \frac{Fc_{cr}}{f_c} - 1 = \frac{10000}{1350} - 1 = \underline{\text{LARGE}}$$

3.8 Y-AXIS EXCITATION

FOR THE OVERALL SECTION.

$$M_{MAX_{STATIC}} = \frac{1}{2}WL = \frac{1}{2}(3.4)(20) = 34 \text{ IN-LBS.}$$

$$_Y(M_{MAX})_{DYN.} = 1.5(34)(27.1) = 1382 \text{ IN-LBS. (ULT.)}$$

$$I_{XX} = 2.533 \text{ IN.}^4 \quad \text{(REF. PG. 43)}$$

$$C_X = 1.750 \text{ IN.}$$

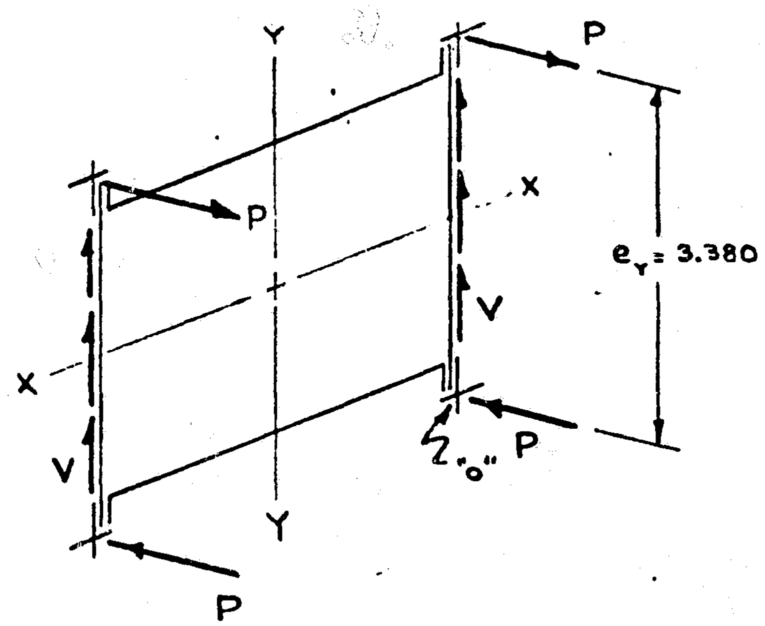
$$Z_X = \frac{I_{XX}}{C_X} = 1.447 \text{ IN.}^3$$

3.8.1 MAX. BENDING STRESS

$$f_{b_Y} = \frac{_Y(M_{MAX})_{DYN}}{Z_X} = \frac{1382}{1.447} = \underline{955 \text{ PSI (ULT.)}}$$

FOR 6061-T6 AL. ALLOY SHEET, $F_{TU} = 42000$ PSI

∴ <u>M.S. LARGE BY INSPECTION</u>



FROM EQS. OF EQUILIBRIUM

$$\Sigma M_o = 0 = (WL/2) - 2Pe_Y$$

$$2P(3.380) = 10.000W$$

$$P = 1.48W$$

$$_YW_{DYN} = 1.5(3.4 \text{ LBS.})(27.1g)$$

$$_YW_{DYN} = 138.2 \text{ LBS. (ULT.)}$$

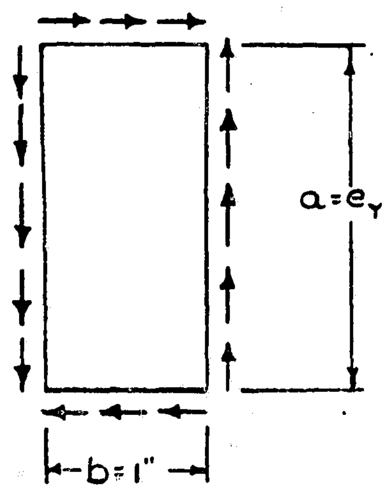
$$\therefore {_YP_{DYN}} = 1.48\, {_YW_{DYN}}$$

$$_YP_{DYN} = \underline{204.5 \text{ LBS. (ULT.)}}$$

$$_YV_{DYN} = 0.50\, {_YW_{DYN}}.$$

$$_YV_{DYN} = \underline{69.1 \text{ LBS. (ULT.)}}$$

3.8.2 SHEAR BUCKLING OF BAFFLE PLATE



FROM REF. d, FIG. 15.1

FOR (a/b) = 3.38

K = EDGE COEFFICIENT = 5.2

(SIMPLY SUPPORTED EDGES)

SHEAR BUCKLING ALLOWABLE IS

$$Fs_{cr} = KE\left(\frac{t}{b}\right)^2$$

$$Fs_{cr} = 5.2(10\times10^6)\left[\frac{0.05}{1.00}\right]^2$$

$$Fs_{cr} = 130{,}000 \text{ psi}$$

FOR 6061-T6 AL. ALLOY SHEET, F_{su} = 27000 psi

∴ SHEAR BUCKLING NOT A PROBLEM

FOR A RECTANGULAR SECTION

$$f_s = \frac{3V}{2A} = \frac{3}{2}\,\frac{69.1}{(.05)(3.38)} = \underline{613 \text{ psi (ULT.)}}$$

M.S. LARGE BY INSPECTION

3.8.3 COMPRESSIVE BUCKLING OF GUSSET

(REFER TO PAGE 47 FOR DETERMINATION OF BUCKLING ALLOWABLE)

$$f_c = \frac{P_{DYN}}{A} = \frac{204.5}{0.110} = \underline{1860 \text{ psi (ULT.)}}$$

$$Fc_{cr} = 10{,}000 \text{ psi}$$

$$M.S. = \frac{F_{ccr}}{f_c} - 1 = \frac{10000}{1860} - 1 = \underline{\text{LARGE}}$$

3.9 Z-AXIS LOADING

$$_{Z}W_{DYN.} = 1.5(3.4 \text{ LBS.})(31.5g) = 160.7 \text{ LBS. (ULT.)}$$

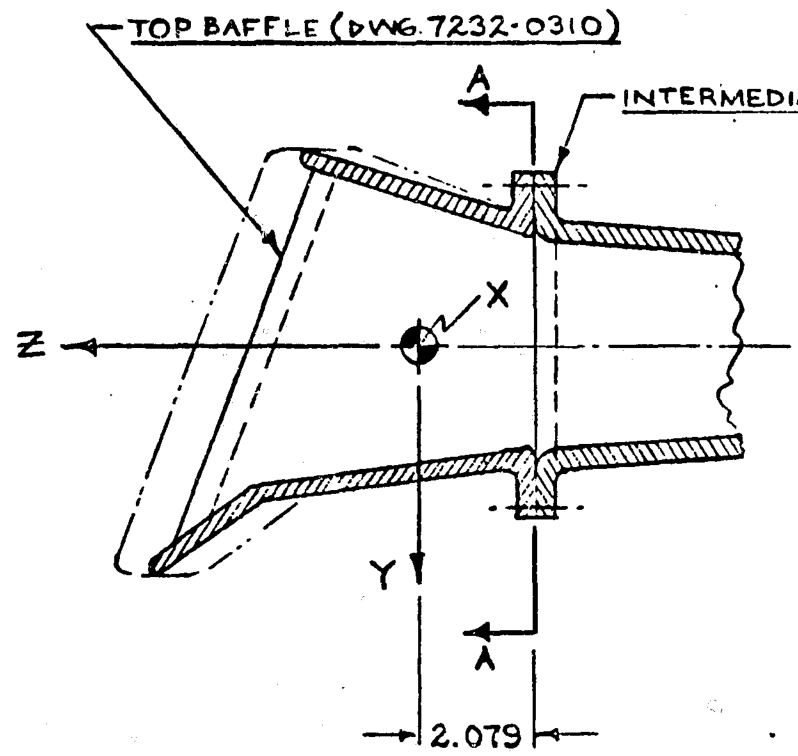
ASSUME ENTIRE LOAD CARRIED BY GUSSETS ONLY.

$$_{Z}P_{DYN.} \text{ PER GUSSET} = \frac{1}{4}\,_{Z}W_{DYN} = \frac{160.7}{4} = 40.2 \text{ LBS. (ULT.)}$$

THIS LOAD IS CONSIDERABLY LESS THAN THE CORRESPONDING GUSSET AXIAL LOAD FOR BOTH THE X AND Y AXES. THEREFORE, Z-AXIS LOADING IS NOT CRITICAL.

M.S. LARGE BY INSPECTION

3.10 ATTACHMENT OF BAFFLE TOP TO INTERMEDIATE BAFFLE



ATTACHMENT OF TOP BAFFLE AND INTERMEDIATE BAFFLE IS MADE THROUGH 24 MS20470-AD2-5 RIVETS. SHEAR ALLOWABLE IS

Ps_{ALLOW} = 106 LBS. (REF. C)

ASSUME TENSILE ALLOWABLE

$P_{T_{ALLOW}} = 0.5\, P_{S_{ALLOW}} \approx 50$ LBS.

ASSUME WEIGHT OF TOP BAFFLE

$W_{TB} = 0.94$ LBS.

X-AXIS EXCITATION

$_x(W_{TB})_{DYN} = 1.5(0.94)(30.7)$

$_x(W_{TB})_{DYN} = 43.3$ LBS. (ULT.)

SHEAR LOAD PER RIVET

$$_xP_s = \frac{_x(W_{TB})_{DYN}}{24}$$

$_xP_s = (43.3/24) =$ 1.8 LBS. (ULT.)

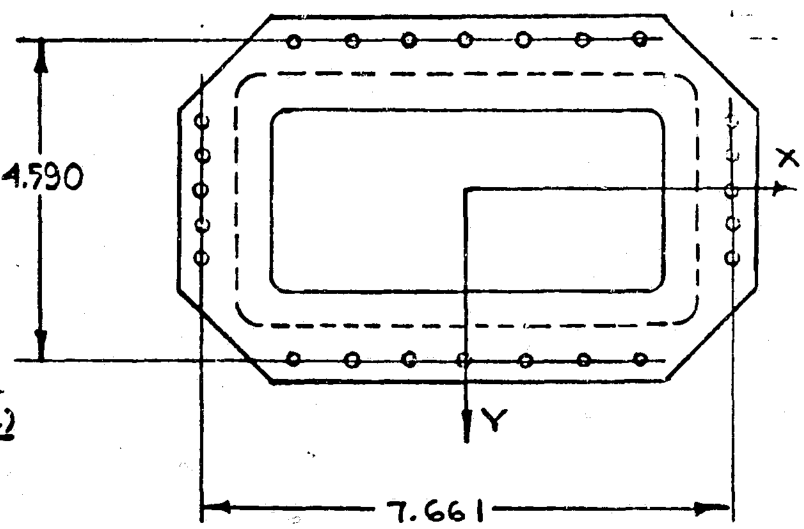
TENSILE LOAD RIVET

$M_x = (2.079)\left[_x(W_{TB})_{DYN}\right]$

$M_x = 2.079(43.3) = 90.0$ IN-LBS. (ULT.)

$$\Sigma\, _xP_T = \frac{M_x}{7.661} = 11.75 \text{ LBS}$$

$_xP_T = (\Sigma\, _xP_T / 5) =$ 2.4 LBS. (ULT.)



SECTION A-A

M.S. LARGE BY INSPECTION

Y-AXIS EXCITATION

$$_Y(W_{TB})_{DYN} = 1.5(0.94)(27.1) = 38.2 \text{ LBS (ULT.)}$$

SHEAR LOAD PER RIVET

$$_YP_S = \frac{_Y(W_{TB})_{DYN}}{24} = \frac{38.2}{24} = \underline{1.6 \text{ LBS.(ULT.)}}$$

TENSILE LOAD PER RIVET

$$M_Y = (2.079)\left[_Y(W_{TB})_{DYN}\right]$$

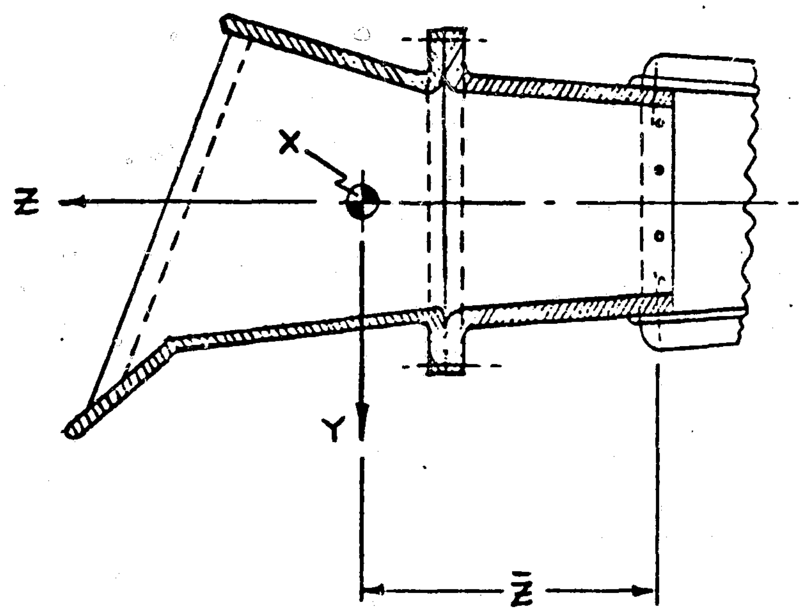
$$M_Y = 2.079(38.2) = 79.4 \text{ IN-LBS (ULT.)}$$

$$\Sigma_Y P_T = \frac{M_Y}{4.590} = \frac{79.4}{4.590} = 17.30 \text{ LBS.}$$

$$_YP_T = (\Sigma_Y P_T / 7) = \underline{2.5 \text{ LBS.(ULT.)}}$$

M.S. LARGE BY INSPECTION

3.11 ATTACHMENT OF INTERMEDIATE TO LOWER BAFFLE



W = WEIGHT OF TOP BAFFLE + INTERMEDIATE BAFFLE

W = 0.94 + 0.27 = 1.21 LBS.

$$\bar{Z} = \frac{0.94(2.079+1.470) + 0.27(1.470/2)}{(0.94+0.27)} = 2.921 \text{ IN.}$$

$$_x(W_{DYN}) = 1.5(1.21)(30.7) = \underline{55.7 \text{ LBS. (ULT.)}}$$

$$_y(W_{DYN}) = 1.5(1.21)(27.1) = \underline{49.2 \text{ LBS. (ULT.)}}$$

$$_z(W_{DYN}) = 1.5(1.21)(31.5) = \underline{57.2 \text{ LBS. (ULT.)}}$$

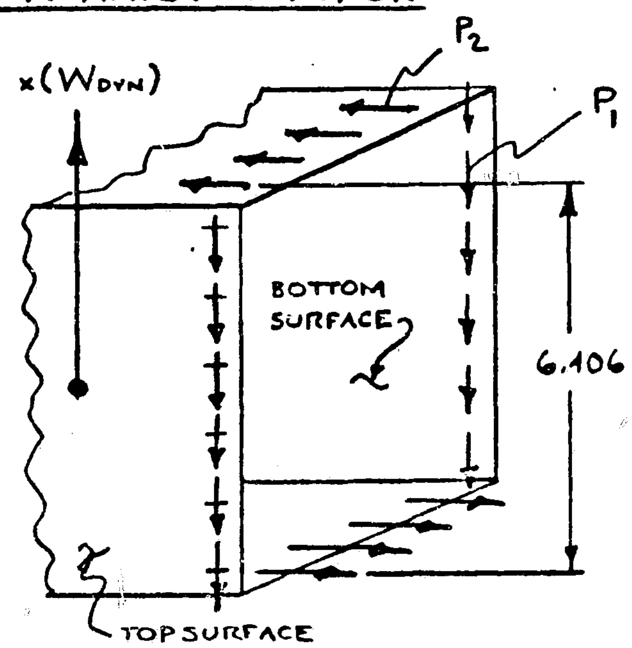
3.11.1 Z - AXIS EXCITATION

$_z(W_{DYN})$ IS RESISTED BY EQUAL SHEAR FORCES IN 20 PLACES. THE APPLIED SHEAR LOAD PER RIVET IS

$$_zP_s = \frac{_z(W_{DYN})}{20} = \frac{57.2}{20} = \underline{2.9 \text{ LBS (ULT.)}}$$

<u>M.S. LARGE BY INSPECTION</u>

3.11.2 X-AXIS EXCITATION



X(WDYN) REACTED IN SHEAR BY P_1 FORCES ON TOP AND BOTTOM SURFACES AS SHOWN. MOMENT $M = x(W_{DYN}) \cdot \bar{Z}$ REACTED BY COUPLE SHEAR FORCES P_2 ON SIDES.

$P_1 = x(W_{DYN})/12 = (55.7/12)$

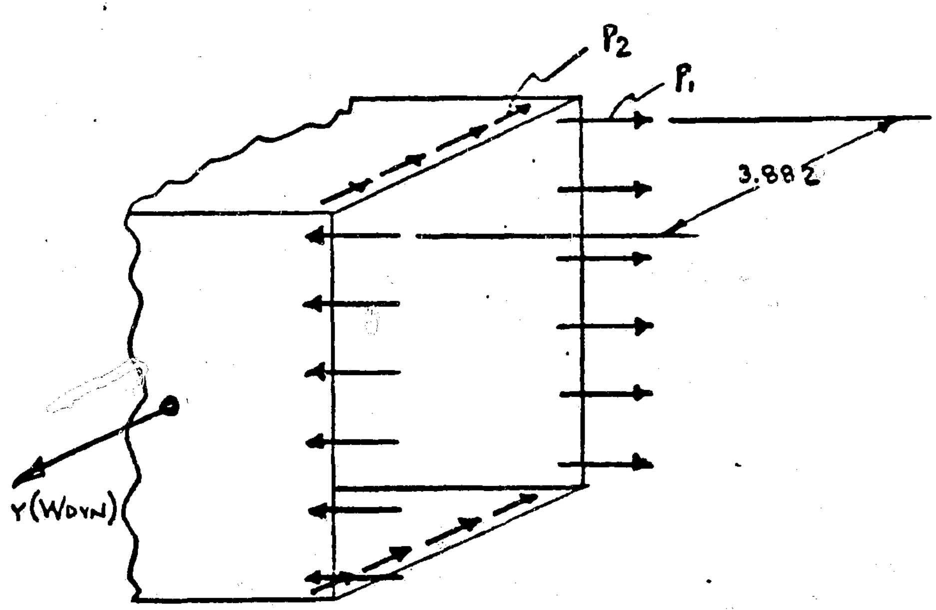
$P_1 =$ 4.6 LBS. (ULT.)

$\Sigma M = 0 = 4P_2(6.406) - x(W_{DYN}) \cdot \bar{Z}$

$4P_2(6.406) = 55.7(2.921)$

$P_2 =$ 6.3 LBS. (ULT.)

3.11.3 Y-AXIS EXCITATION



Y(WDYN) REACTED IN SHEAR BY P_2 FORCES ON THE SIDES. THE MOMENT $M = y(W_{DYN}) \cdot \bar{Z}$ IS REACTED BY COUPLE SHEAR FORCES P_1 ON THE TOP AND BOTTOM SURFACES AS SHOWN

$P_2 = \frac{y(W_{DYN})}{8} = \frac{49.2}{8}$

$P_2 =$ 6.2 LBS. (ULT.)

$\Sigma M = 0 = 6P_1(3.882) - y(W_{DYN}) \cdot \bar{Z}$

$23.292P_1 = 49.2(2.921)$

$P_1 =$ 6.2 LBS. (ULT.)

THE MAXIMUM SHEAR LOAD ON ANY GIVEN RIVET OCCURS DURING X-AXIS EXCITATION AND IS EQUAL TO

$P_{MAX} =$ 6.3 LBS. (ULT.)

3.11.4 FASTENER SHEAR

P_S = MAX. SHEAR LOAD = 6.3 LBS (ULT.)

P_{S_A} = ALLOWABLE SHEAR LOAD = 106.0 LBS. (REF. PG. 51)

$$M.S. = \frac{P_{S_A}}{P_S} - 1 = \frac{106.0}{6.30} - 1 = \underline{LARGE}$$

3.11.5 BEARING ON SHEET (BAFFLE PLATE)

A_{br} = BEARING AREA = dt

$A_{br} = 0.062(0.05) = 0.0031 \text{ in}^2$

$$f_{br} = \frac{P_s}{A_{br}} = \frac{6.30}{0.0031} = \underline{2030 \text{ psi (ULT.)}}$$

FROM REF. C FOR 6061-T6 AL. ALLOY

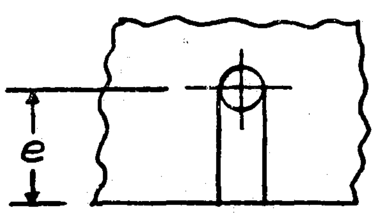
$F_{bry_{(e/d=2)}} = 88{,}000$ psi

$$M.S. = \frac{F_{bry}}{f_{br}} - 1 = \frac{88000}{2030} - 1 = \underline{LARGE}$$

3.11.6 SHEAR TEAROUT

A_S = SHEAR TEAROUT AREA = $2et$ = $2(.150)(.05)$

$A_S = 0.015 \text{ in}^2$



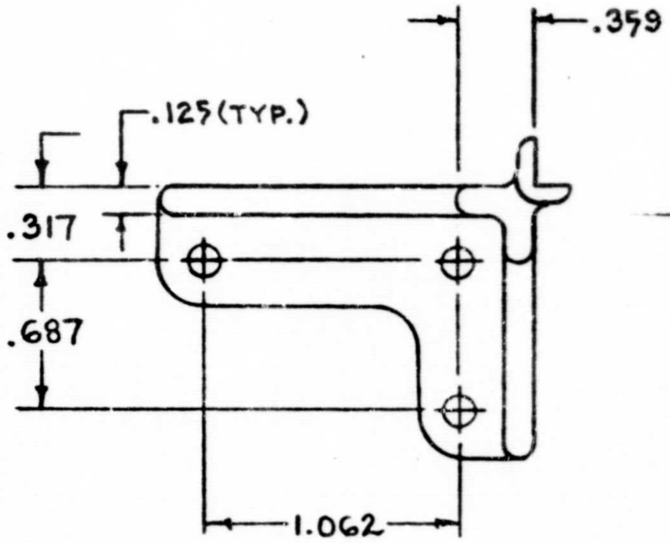
$$f_S = \frac{P_s}{A_s} = \frac{6.30}{0.015} = \underline{420 \text{ psi (ULT.)}}$$

FROM REF. C FOR 6061-T6 AL. ALLOY

F_{su} = 27000 psi

$$M.S. = \frac{F_{su}}{f_s} - 1 = \frac{27000}{420} - 1 = \underline{LARGE}$$

3.12 ATTACHMENT OF GUSSET TO BASE



GUSSET (DWG. 7232-0307)
MAT'L. 6061-T6 AL. ALLOY.

THE CRITICAL LOADING CONDITION IS DUE TO PYROTECHNIC SHOCK (Y-AXIS EXCITATION). FROM PAGE 48 IT IS SEEN THAT EACH FASTENER MUST CARRY AN AXIAL LOAD OF $(P_{DYN}/3)$ AND THE GUSSET TO BASE JOINT MUST HAVE A FRICTIONAL CAPACITY $> (V_{DYN}/2)$.

AXIAL LOAD PER SCREW

$$P_A = \frac{P_{DYN}}{3} = \frac{204.5}{3} = \underline{68.2 \text{ LBS.}}$$

FRICTIONAL FORCE PER JOINT

$$F_F = \frac{V_{DYN}}{2} = \frac{69.1}{2} = \underline{34.6 \text{ LBS.}}$$

3.12.1 ATTACHMENT LOADS

THE SCREWS CALLED OUT ARE NO. 8-32 UNC 300 SERIES CRES. FROM REF. e, THE MINIMUM YIELD LOAD CAPACITY OF THE FASTENER IS

$$P'_{TY} = 420 \text{ LBS.}$$

ASSUME SCREW TORQUED TO 90% OF YIELD

$$P_{TY} = 0.9(420) = 378.0 \text{ LBS.}$$

$$M.S. = \frac{P_{TY}}{P_A} - 1 = \frac{378.0}{68.2} - 1 = \underline{\text{LARGE}}$$

ASSUME A COEFFICIENT OF FRICTION $\mu = 0.20$ BETWEEN GUSSET AND BASE.
FRICTION CAPACITY PER SCREW

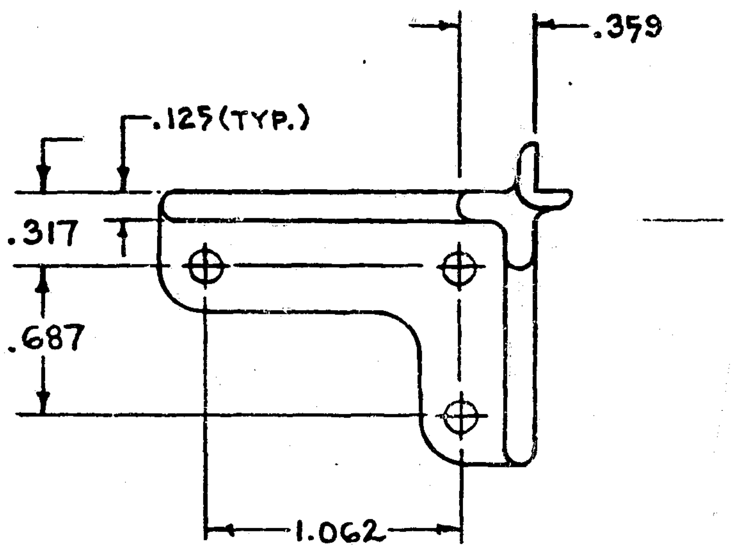
$$F'_{FJ} = \mu \times P_{TY} = 0.2(378.0) = 75.6 \text{ LBS}$$

FRICTION CAPACITY OF JOINT IS

$$F_{FJ} = 3F'_{FJ} = 3(75.6) = 226.8 \text{ LBS.}$$

$$M.S. = \frac{F_{FJ}}{F_F} - 1 = \frac{226.8}{34.6} - 1 = \underline{\text{LARGE}}$$

3.12 ATTACHMENT OF GUSSET TO BASE



GUSSET (DWG. 7232-0307)
MAT'L. 6061-T6 AL. ALLOY.

THE CRITICAL LOADING CONDITION IS DUE TO PYROTECHNIC SHOCK (Y-AXIS EXCITATION). FROM PAGE 48 IT IS SEEN THAT EACH FASTENER MUST CARRY AN AXIAL LOAD OF ($P_{DYN}/3$) AND THE GUSSET TO BASE JOINT MUST HAVE A FRICTIONAL CAPACITY > ($V_{DYN}/2$).

AXIAL LOAD PER SCREW

$$P_A = \frac{P_{DYN}}{3} = \frac{204.5}{3} = \underline{68.2 \text{ LBS.}}$$

FRICTIONAL FORCE PER JOINT

$$F_F = \frac{V_{DYN}}{2} = \frac{69.1}{2} = \underline{34.6 \text{ LBS.}}$$

3.12.1 ATTACHMENT LOADS

THE SCREWS CALLED OUT ARE NO. 8-32 UNC 300 SERIES CRES. FROM REF. e, THE MINIMUM YIELD LOAD CAPACITY OF THE FASTENER IS

$$P'_{TY} = 420 \text{ LBS.}$$

ASSUME SCREW TORQUED TO 90% OF YIELD

$$P_{TY} = 0.9(420) = 378.0 \text{ LBS.}$$

$$M.S. = \frac{P_{TY}}{P_A} - 1 = \frac{378.0}{68.2} - 1 = \underline{\text{LARGE}}$$

ASSUME A COEFFICIENT OF FRICTION $\mu = 0.20$ BETWEEN GUSSET AND BASE.
FRICTION CAPACITY PER SCREW

$$F'_{FJ} = \mu \times P_{TY} = 0.2(378.0) = 75.6 \text{ LBS}$$

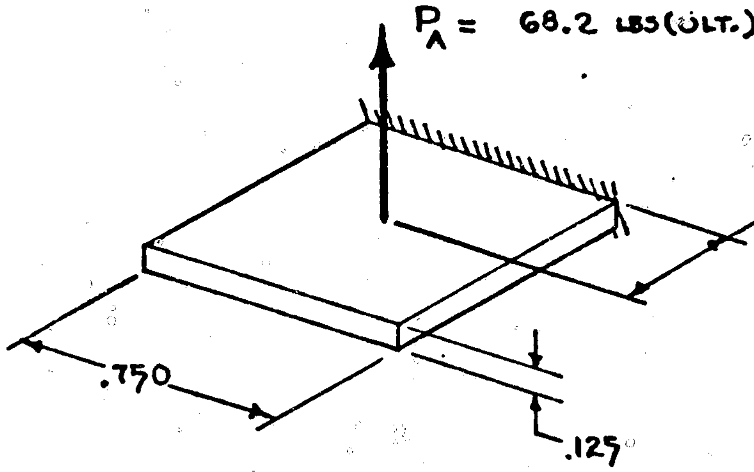
FRICTION CAPACITY OF JOINT IS

$$F_{FJ} = 3F'_{FJ} = 3(75.6) = 226.8 \text{ LBS.}$$

$$M.S. = \frac{F_{FJ}}{F_F} - 1 = \frac{226.8}{34.6} - 1 = \underline{\text{LARGE}}$$

3.12.2 BENDING STRESS IN GUSSET LEG

CONSERVATIVELY ASSUME ONE OF THE P_A LOADS TO STRESS A PORTION OF THE GUSSET LEG AS A CANTILEVER BEAM. ASSUME AN EFFECTIVE WIDTH OF LEG EQUAL TO 0.750 IN.



$$f_b = \frac{6M}{bt^2} = \frac{6\{0.234(68.2)\}}{0.75(0.125)^2}$$

$$f_b = 8170 \text{ psi}$$

FROM REF. C, FOR 6061-T6 AL. ALLOY

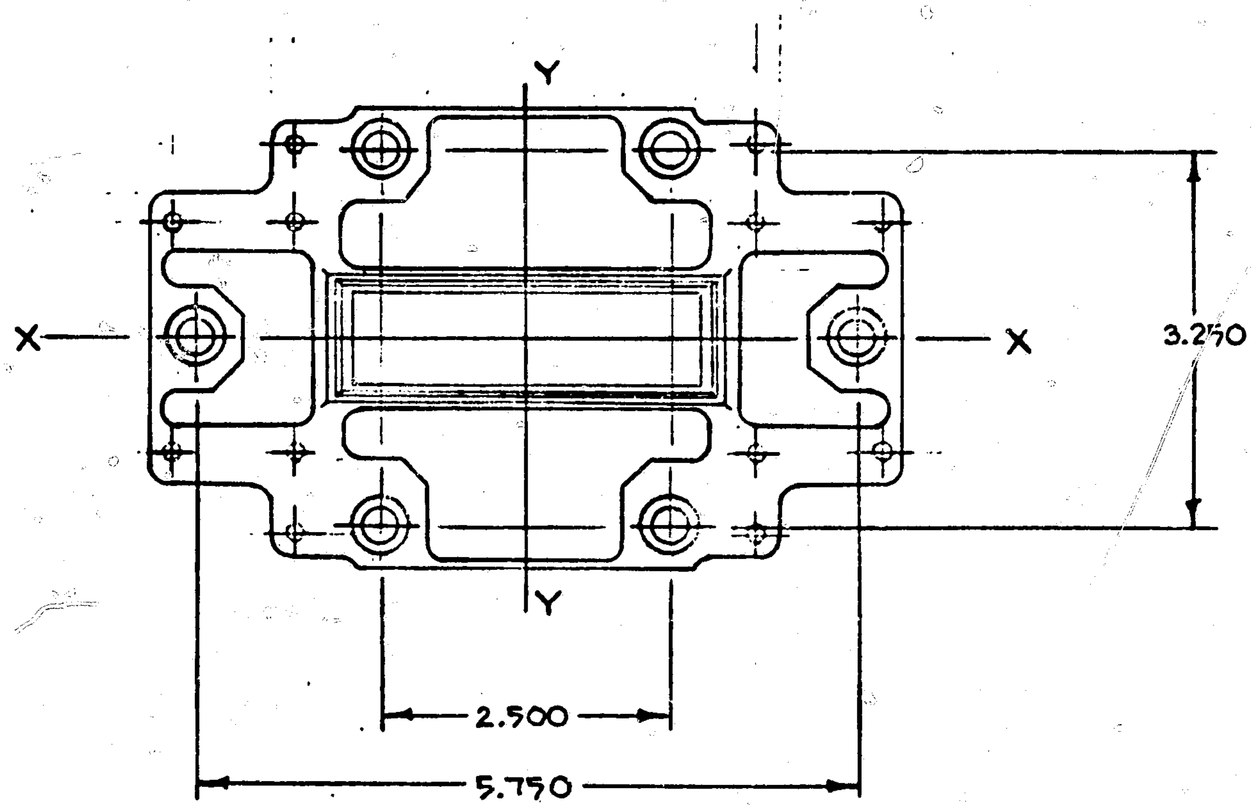
$$F_{TY} = 35000 \text{ psi}$$

$$M.S. = \frac{F_{TY}}{f_b} - 1 = \frac{35000}{8170} - 1 = \underline{\text{LARGE}}$$

3.12.3 INSERTS IN BAFFLE BASE (REF. f)

KNOWING THE BOLT MATERIAL ULTIMATE TENSILE STRENGTH (FTU = 85,000 PSI FOR 300 SERIES CRES) AND THE SHEAR STRENGTH OF THE BASE PARENT MATERIAL (Fsu ≅ 25000 PSI FOR 6061-T6 AL. ALLOY PLATE), THE RECOMMENDED MINIMUM INSERT LENGTH IS BETWEEN 1 AND 1½ TIMES THE NOMINAL BOLT DIAMETER (.138-.207 IN FOR A #8 SCREW). THIS LENGTH OF THREAD ENGAGEMENT PROVIDES SUFFICIENT STRENGTH TO BREAK THE BOLT WITHOUT DAMAGING THE INSERT OR THE PARENT MATERIAL. ACTUAL ENGAGEMENT IS .250 IN. SO THAT NO PROBLEMS ARE ANTICIPATED WITH THE SELECTED INSERT.

3.13 ATTACHMENT OF BAFFLE (BASE) TO FWD. PLATE



LET W_B = TOTAL BAFFLE WEIGHT SUPPORTED BY FWD. PLATE = 4.07 LBS.
UTILIZING A 1.5 LOAD FACTOR AND THE CRITICAL ACCELERATIONS GIVEN ON PAGE 41, THE ULTIMATE DYNAMIC LOADS ARE:

$(W_{DYN})_X = 1.5(4.07)(30.7) =$ 187.4 LBS. (ULT.)

$(W_{DYN})_Y = 1.5(4.07)(27.1) =$ 165.4 LBS. (ULT.)

$(W_{DYN})_Z = 1.5(4.07)(31.5) =$ 192.3 LBS. (ULT.)

THE C.G. OF THE TOTAL BAFFLE ASSEMBLY IS LOCATED AT A DISTANCE $Z = 9.3$ IN. ABOVE THE MOUNTING SURFACE. AT THIS FWD. PLATE INTERFACE, THE FOLLOWING BENDING MOMENTS ARE PRODUCED:

$(M_{DYN})_X = 9.30\,(W_{DYN})_X =$ 1743 IN-LBS. (ULT.)

$(M_{DYN})_Y = 9.30\,(W_{DYN})_Y =$ 1538 IN-LBS. (ULT.)

3.13.1 X-AXIS LOADING

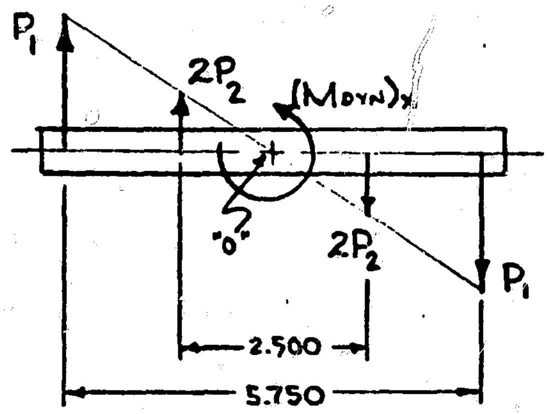
THE SHEAR LOAD PER FASTENER IS

$$P_S = \frac{(W_{DYN})_X}{6} = \frac{187.4}{6} = \underline{31.2 \text{ LBS (ULT.)}}$$

THE FASTENERS ARE #8 – 32 UNC CRES (300 SERIES). THE MINIMUM YIELD LOAD CAPACITY FROM REF. e IS 420 LBS. ASSUME PRELOADING UP TO 90% OF YIELD AND A COEFFICIENT OF FRICTION BETWEEN THE FWD. PLATE AND THE STANDOFF OF 0.15. THEREFORE, THE FRICTIONAL CAPACITY AT EACH PICK-UP POINT IS

$$F_F = 0.15\left[0.90(420)\right] = \underline{56.7 \text{ LBS.}}$$

$$M.S. = \frac{F_F}{P_S} - 1 = \frac{56.7}{31.2} - 1 = \underline{+0.82}$$



$$\Sigma M_O = 0 = 5.750 P_1 + 2.500(2P_2) - (M_{DYN})_X$$

$$5.750 P_1 + 5.000 P_2 = 1743$$

FROM GEOMETRY

$$\frac{P_1}{P_2} = \frac{2.875}{1.250}$$

$$P_1 = 2.300 P_2$$

$$\therefore\ 5.750(2.300 P_2) + 5.000 P_2 = 1743$$

$$P_2 = \underline{95.6 \text{ LBS.}}$$

$$P_1 = \underline{220.0 \text{ LBS.}} = P_{T_X}$$

MAX. TENSILE LOAD IS P_1 = 220 LBS. BASED ON A PRELOAD OF 90% YIELD

$$M.S. = \frac{0.90(420)}{220} - 1 = \underline{+0.72}$$

3.13.2 Y-AXIS LOADING

THE SHEAR LOAD PER FASTENER IS

$$P_S = \frac{(W_{DYN})_Y}{6} = \frac{165.4}{6} = \underline{27.6 \text{ LBS. (ULT.)}}$$

THIS IS LESS CRITICAL THAN FOR X-AXIS LOADING. THUS, FOR FRICTION CAPACITY,

M.S. LARGE BY INSPECTION

THE MOMENT $(M_{DYN})_Y$ WILL BE REACTED ONLY BY THE FOUR SCREWS 3.25" APART.

$$\therefore \quad \Sigma M = 0 = 2P_{T_Y}(3.250) - (M_{DYN})_Y$$

$$6.500 P_{T_Y} = 1538$$

$$P_{T_Y} = \underline{236.6 \text{ LBS (ULT.)}}$$

$$M.S. = \frac{0.90(420)}{236.6} - 1 = \underline{+0.60}$$

3.13.3 Z-AXIS LOADING

FOR Z-AXIS INPUTS, THE TENSILE LOAD PER FASTENER IS

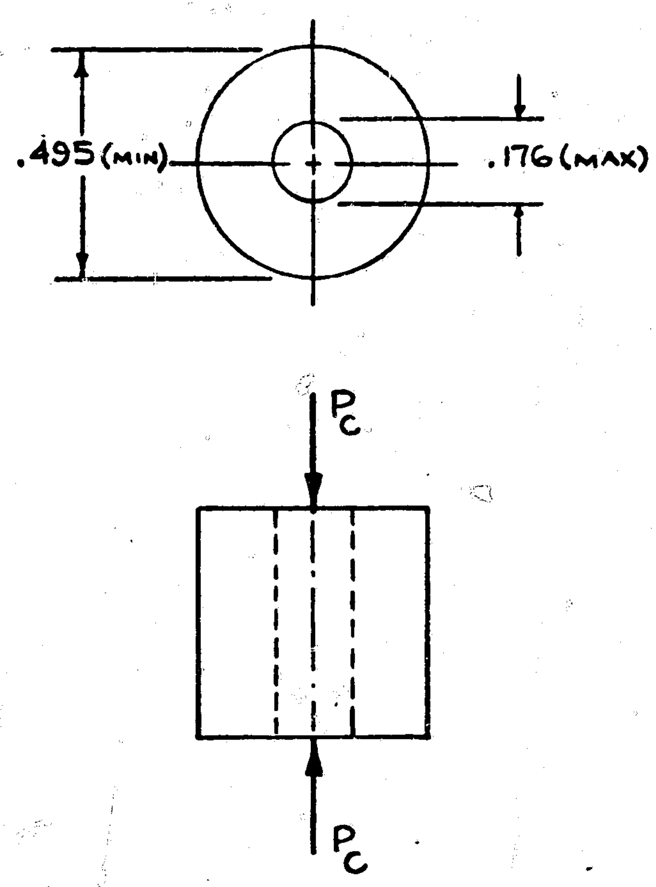
$$P_{T_Z} = \frac{(W_{DYN})_Z}{6} = \frac{192.3}{6} = \underline{32.1 \text{ LBS. (ULT.)}}$$

$\therefore$ Z-AXIS LOADING NOT CRITICAL.

3.14 COMPRESSIVE STRESS IN STANDOFF (REF. RAY LEE DWG. 7232-0313)

THE STANDOFF IS LOADED IN COMPRESSION BY A FORCE EQUAL TO THE PRELOAD IN THE #8-32 UNC SCREW WHICH IS TAKEN AS 90% OF THE MINIMUM YIELD LOAD.

$$P_c = 0.90(420) = 378 \text{ LBS.}$$



$$A = \frac{\pi}{4}\left[(0.495)^2 - (0.176)^2\right]$$

$$A = 0.168 \text{ IN}^2$$

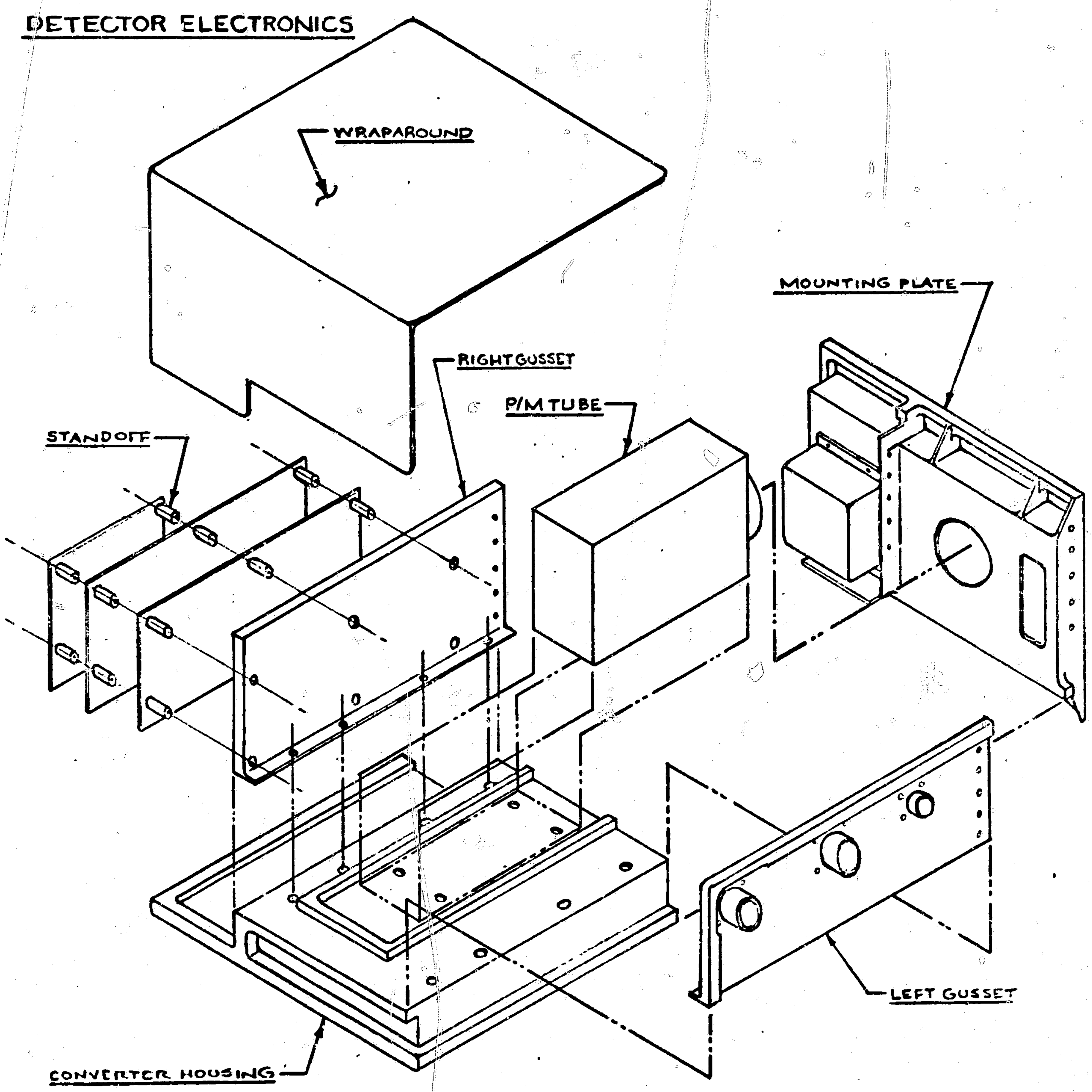
$$f_c = \frac{P_c}{A} = \frac{378}{0.168} = \underline{2250 \text{ psi (ULT.)}}$$

FOR 2024-T4 AL. ALLOY

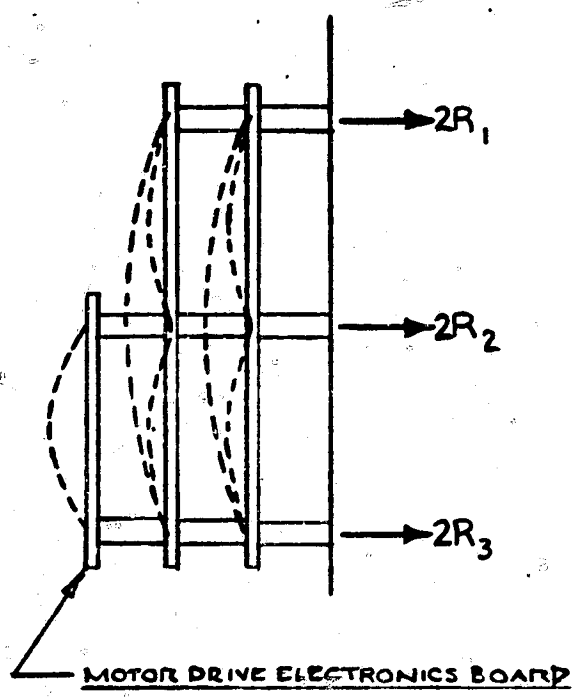
$F_{cy} = 40000$ psi (REF. C)

$$\text{M.S.} = \frac{F_{cy}}{f_c} - 1 = \frac{40000}{2250} - 1 = \underline{\text{LARGE}}$$

4.0 ANALYSIS OF DETECTOR ELECTRONICS



4.1 X-AXIS EXCITATION



4.1.1 PC BOARD BENDING STRESS

FOR THE MOTOR DRIVE ELECTRONICS BOARD, THE PEAK, THE PEAK ACCELERATION FOR X-AXIS LOADING IS

$$G_x = 32.2\,g \quad \text{(REF. PG. 41)}$$

IT IS ASSUMED THAT THIS IS THE INPUT TO THE BOARD. IT IS FURTHER ASSUMED THAT THE TRANSMISSIBILITY ACROSS THE BOARD IS Q = 5. FOR A 0.25 LB. BOARD, THE DYNAMIC LOAD IS

$$W_{DYN} = 1.5(0.25\text{ LB})(32.2\,g)(Q=5)$$

$$W_{DYN} = 60.38 \text{ LBS. (ULT.)}$$

BOARD AREA IS APPROXIMATELY

$$A = 3.25(3.45) = 11.21 \text{ IN.}^2$$

$$q \equiv (W_{DYN}/A) \cong 5.4 \text{ PSI}$$

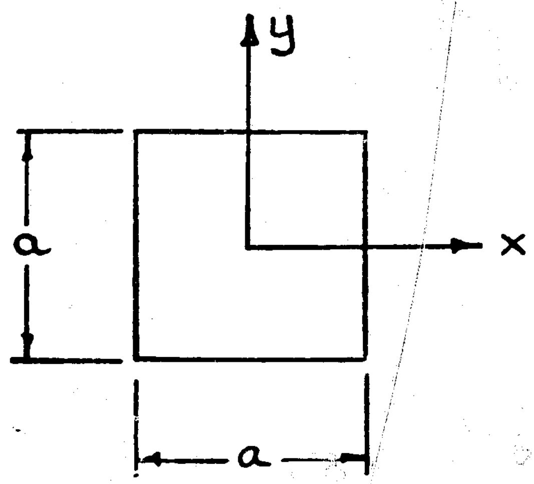
FROM REF. g THE MAXIMUM BENDING MOMENT OCCURS AT X=0, y=(a/2)

$$M_{MAX} = 0.1527\,q a^2$$

USING $a \approx 3.25$ IN.

$$M_{MAX} = 0.1527(5.4)(3.25)^2$$

$$M_{MAX} = 8.7 \text{ IN-LB/IN}$$



$$f_b = \frac{6M_{MAX}}{t^2} = \frac{6(8.7)}{(0.072)^2}$$

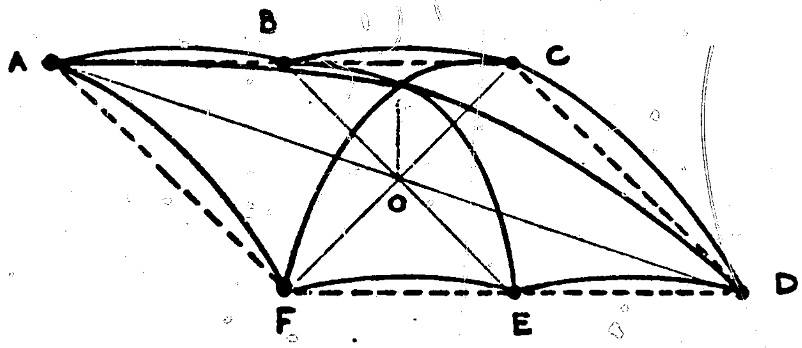
$$f_b = \underline{10700 \text{ PSI (ULT.)}}$$

FROM REF. h, THE BOARD FLEXURAL STRENGTH IS

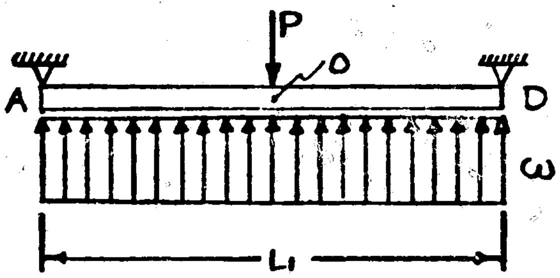
$$F_F = 40{,}000 \text{ PSI}$$

$$\text{M.S.} = \frac{F_F}{f_b} - 1 = \underline{\text{LARGE}}$$

4.1.2 STRESS IN A3 TIMING CIRCUIT BOARD



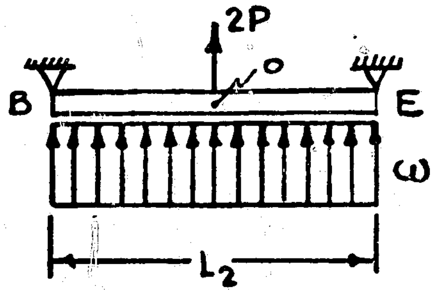
FOR "BEAM" AD (AND "BEAM" FC)



$$y_{o_{AD}} = \frac{5\omega L_1^4}{384EI} - \frac{PL_1^3}{48EI}$$

$$y_{o_{AD}} = \frac{5\omega L_1^4 - 8PL_1^3}{384EI}$$

FOR "BEAM" BE



$$y_{o_{BE}} = \frac{5\omega L_2^4}{384EI} + \frac{2PL_2^3}{48EI}$$

$$y_{o_{BE}} = \frac{5\omega L_2^4 + 16PL_2^3}{384EI}$$

$$y_{o_{AD}} = y_{o_{BE}}$$

$$5\omega L_1^4 - 8PL_1^3 = 5\omega L_2^4 + 16PL_2^3$$

$$\boxed{P = \frac{5\omega(L_1^4 - L_2^4)}{8(L_1^3 + 2L_2^3)}}$$

ASSUMING A BOARD WEIGHT W = 0.500 LBS

$$\omega = \frac{W}{(2L_1 + L_2)}$$

L_2 = 2.125 IN.

L_1 = 6.100 IN.

$$\omega = \frac{0.500}{2(6.100)+2.125} = 0.0349 \text{ LBS/IN.}$$

$$P = \frac{5(0.0349)\{(6.100)^4-(2.125)^4\}}{8\{(6.100)^3+2(2.125)^3\}} = \underline{0.1208 \text{ LBS.}}$$

FROM EQS. OF EQUILIBRIUM

$$R_A = R_D = \frac{1}{2}(\omega L_1 - P) = \underline{0.0460 \text{ LBS.}} = R_F = R_C$$

$$R_B = R_E = \frac{1}{2}(\omega L_2 + 2P) = \underline{0.1579 \text{ LBS.}}$$

THE MOTOR DRIVE ELECTRONICS BOARD WHICH WEIGHS 0.250 LBS. IS SUPPORTED AT POINTS A, B, E AND F. EACH REACTION WILL THEN BE (0.250/4) = 0.0625 LBS. THEREFORE, THE TOTAL REACTIONS WILL BE

$$\Sigma R_A = \Sigma R_F = 0.0625 + 2(0.0460) = \underline{0.1545 \text{ LBS.}} = R_3$$
$$\Sigma R_B = \Sigma R_E = 0.0625 + 2(0.1579) = \underline{0.3783 \text{ LBS.}} = R_2$$
$$\Sigma R_C = \Sigma R_D = 2(0.0460) = \underline{0.0920 \text{ LBS.}} = R_1$$

MAX. APPLIED DYNAMIC BOLT LOAD

$$P_d = (0.3783 \text{ LBS})(32.2g)(Q=5) = 60.9 \text{ LBS. (LIMIT LOAD)}$$
$$P_Y = 1.15(60.9) = \underline{70.0 \text{ LBS. (YIELD)}}$$
$$P_U = 1.50(60.9) = \underline{91.4 \text{ LBS. (ULT.)}}$$

CONSERVATIVELY ASSUME THAT THE MAXIMUM STRESS IN THE A3 TIMING CIRCUIT BOARD IS THAT STRESS DETERMINED FROM THE BENDING OF "BEAM" BE.

$$M_{MAX.} = \frac{1}{8}\omega L_2^2 + \frac{1}{4}(2PL_2)$$

$$M_{MAX.} = \frac{1}{8}(0.0349)(2.125)^2 + \frac{1}{4}(0.2416)(2.125)$$

$$M_{MAX.} = 0.020 + 0.128 = 0.148 \text{ IN-LBS}/g$$

THE DYNAMIC MOMENT IS

$$M_{MAX_d} = 1.5(32.2g)(Q=5)(0.148) = 35.7 \text{ IN-LBS (ULT.)}$$

ASSUME "BEAM" BE IS .072" THICK × 1.50" WIDE

$$f_b = \frac{6M_{MAX_d}}{bt^2} = \frac{6(35.7)}{1.5(0.072)^2} = \underline{27550 \text{ PSI (ULT.)}}$$

$$F_F = 40{,}000 \text{ PSI (REF. h)}$$

$$\text{M.S.} = \frac{F_F}{f_b} - 1 = \frac{40000}{27550} - 1 = \underline{+0.45}$$

4.1.3 BOLT LOADING

ASSUME A NO. 6 SCREW (BERYLLIUM-COPPER)
STRESS AREA = 0.009 $IN.^2$

FOR BE-CU @ 160,000 PSI YIELD

$$P_{ALLOW} = F_{TY} \times A$$

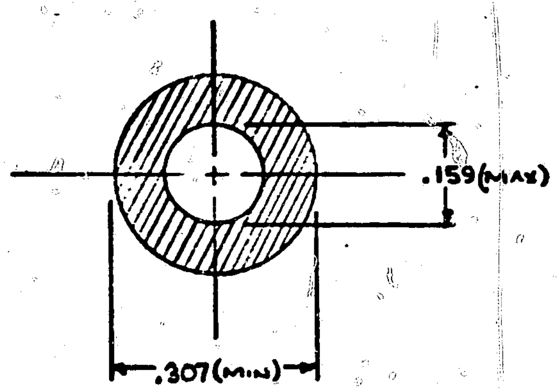
$$P_{ALLOW} = 160{,}000(0.009) = \underline{1440 \text{ LBS. (YIELD)}}$$

MAX. APPLIED LOAD, $P = R_2$ (REF. PG. 65)

$$P_Y = \underline{70.0 \text{ LBS (YIELD)}}$$

$$M.S. = \frac{P_{ALLOW}}{P_Y} - 1 = \underline{\text{LARGE}}$$

4.1.4 COMPRESSIVE LOADING IN STANDOFF



$$AREA,\ A = \frac{\pi}{4}\left\{(0.307)^2 - (0.159)^2\right\}$$

$$A = 0.054 \text{ IN}^2$$

ALLOWABLE BOLT LOAD, P_{ALLOW} = 1440 LBS.
ASSUME BOLT TORQUED TO APPROXIMATELY 300 LBS. THEREFORE, STANDOFF COMPRESSIVE STRESS IS

$$f_c = \frac{300}{0.054} = \underline{5555 \text{ PSI (YIELD)}}$$

STANDOFF MAT'L IS LEXAN FOR WHICH

$$F_c = 12500 \text{ PSI} \qquad \text{(REF. F)}$$

$$M.S. = \frac{F_c}{f_c} - 1 = \frac{12500}{5555} - 1 = \underline{\text{LARGE}}$$

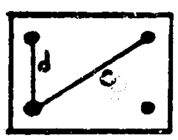
4.1.5 BOARD NATURAL FREQUENCIES

4.1.5.1 MOTOR DRIVE ELECTRONICS BOARD

THE NATURAL FREQUENCY IS GIVEN BY

$$f_n = \beta\sqrt{\frac{Et^3A}{(1-\nu^2)W}}\left\{\frac{m^2}{C^2}+\frac{n^2}{d^2}\right\} \qquad \text{(REF. } i \text{)}$$

WHERE
- β = CONSTANT DEPENDING UPON MODE NUMBER AND MOUNTING CONFIGURATION
- E = MODULUS OF ELASTICITY, PSI = 2.5×10^6 PSI
- t = BOARD THICKNESS = 0.072 IN.
- A = TOTAL BOARD AREA = 11.21 IN.^{2}
- ν = POISSON'S RATIO = 0.125
- W = TOTAL WEIGHT OF BOARD & COMPONENTS = 0.25 LBS.
- n, m = MODE NUMBERS = 1
- C = LONGEST UNSUPPORTED DIMENSION = 3.717 IN.
- d = SHORTEST UNSUPPORTED DIMENSION = 2.125 IN.

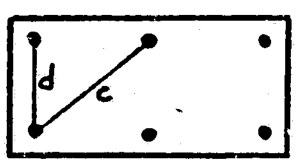


FOR (C/d) = 1.75, $\beta \approx 5.80$ (MTG. CONFIGURATION 1)

$$f_n = 5.80\sqrt{\frac{(2.5\times10^6)(7.2\times10^{-2})^3(11.21)}{[1-(0.125)^2](0.25)}}\left\{\frac{1}{(3.717)^2}+\frac{1}{(2.125)^2}\right\}$$

$f_n \cong$ 350 CPS

4.1.5.2 A3 TIMING CIRCUITS BOARD



A = TOTAL BOARD AREA = 3.00 × 6.50 = 19.50 IN^2

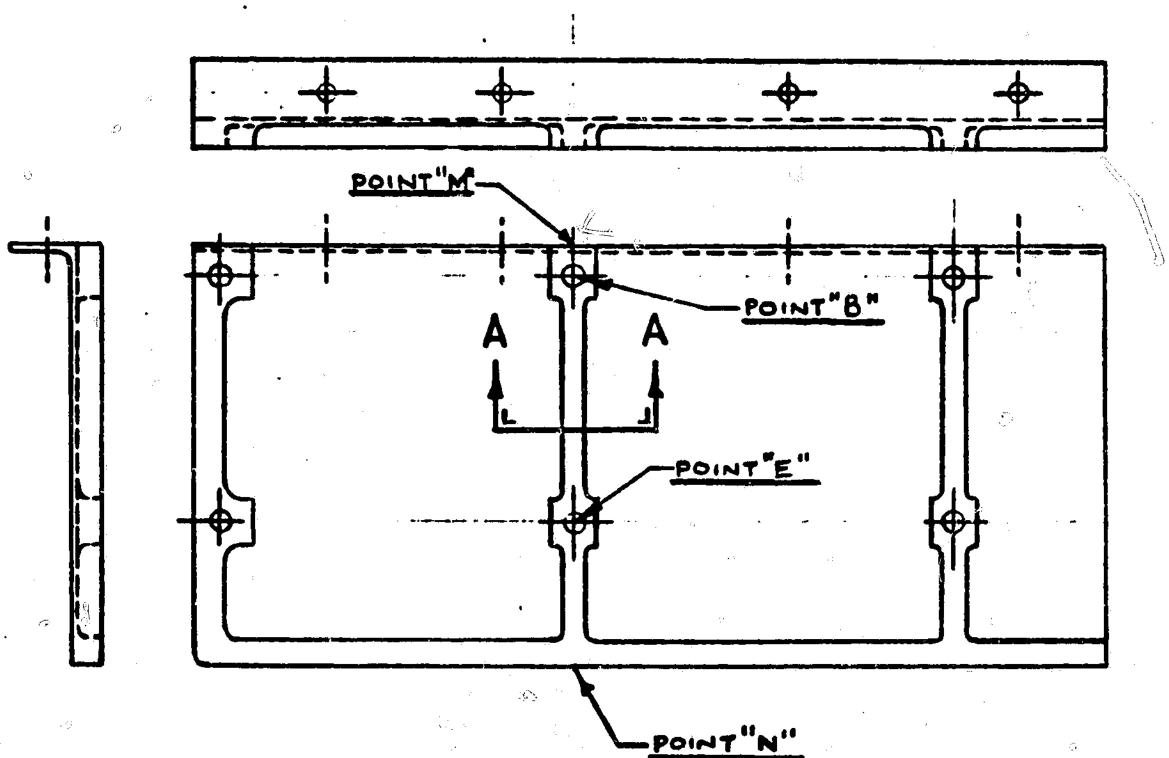
W = TOTAL BOARD WEIGHT ≈ 0.50 LBS.

FOR (C/d) = 1.75, $\beta = 5.71$ (MTG. CONFIGURATION 2)

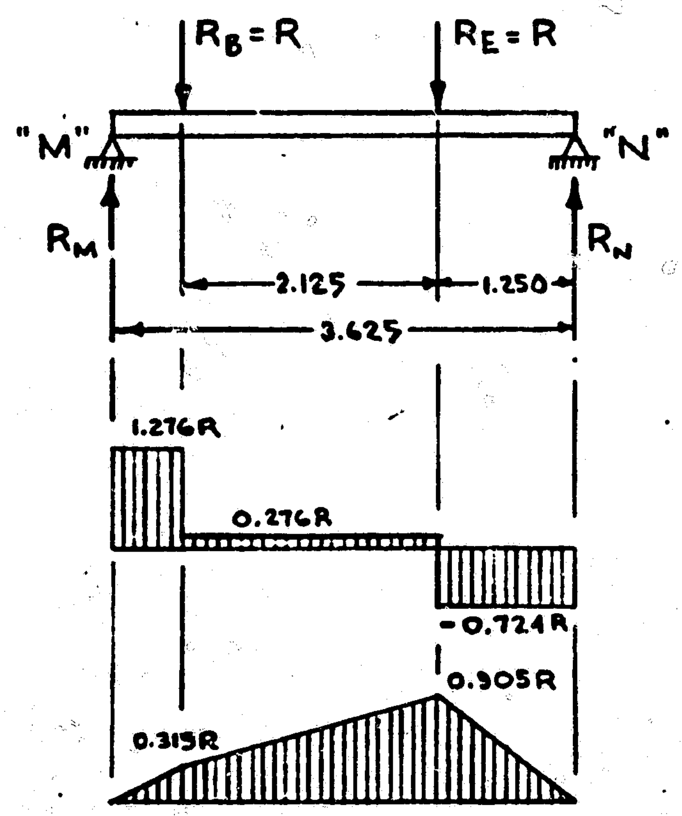
$$f_n = 5.71\sqrt{\frac{(2.5\times10^6)(7.2\times10^{-2})^3(19.50)}{[1-(0.125)^2](0.50)}}\left\{\frac{1}{(3.717)^2}+\frac{1}{(2.125)^2}\right\}$$

$f_n \cong$ 322 CPS

4.1.6 BENDING STRESSES - RIGHT GUSSET



THE MAXIMUM LOADS ON THE GUSSET ARE DUE TO THE X-AXIS EXCITATION OF THE PC BOARDS (REF. PG. 63). CONSIDER SECTION A-A TO BE A SIMPLY SUPPORTED BEAM BETWEEN POINTS "M" AND "N". THE LOADS AT POINTS "B" AND "E" WERE FOUND TO BE 0.3783 LBS/g. (REF. PG. 65)



$$R_M = \frac{3.375}{3.625} R_B + \frac{1.250}{3.625} R_E$$

$$R_M = 0.931 R_B + 0.345 R_E$$

$$R_M = 1.276\,R$$

$$R_N = 0.724\,R$$

$$M_{MAX} = 1.250 R_N = 0.905\,R$$

$$M_{MAX} = 0.905(0.3783)$$

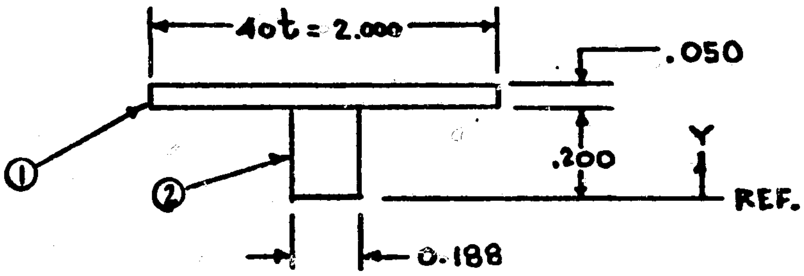
$$M_{MAX} = 0.342 \text{ IN-LBS.}/g$$

DYNAMIC LOAD IS

$$M_d = 1.5(32.2g)(Q=5)(0.342)$$

$$M_d = \underline{82.6 \text{ IN-LBS. (ULT.)}}$$

SECTION A-A



ITEM	DESCRIPTION	AREA, A	Y	AY	AY^2	I_{oxx}
1	2.000 x 0.050	0.1000	0.225	0.02250	0.00506	0.000021
2	0.188 x 0.200	0.0375	0.100	0.00375	0.00038	0.000125
	Σ	0.1375		0.02625	0.00544	0.000146

$\bar{Y} = (\Sigma AY)/(\Sigma A) = 0.191 \text{ in.} = C$

$I_x = \Sigma I_{o_{xx}} + \Sigma AY^2 - \bar{Y} \cdot \Sigma AY$

$I_x = 0.00015 + 0.00544 - 0.00501 = 0.00058 \text{ in}^4$

$(I_x/C) = 0.0030 \text{ in.}^3$

$$f_{b_u} = \frac{Md}{(I_x/C)} = \frac{82.6}{0.0030} = \underline{27530 \text{ psi (ULT)}}$$

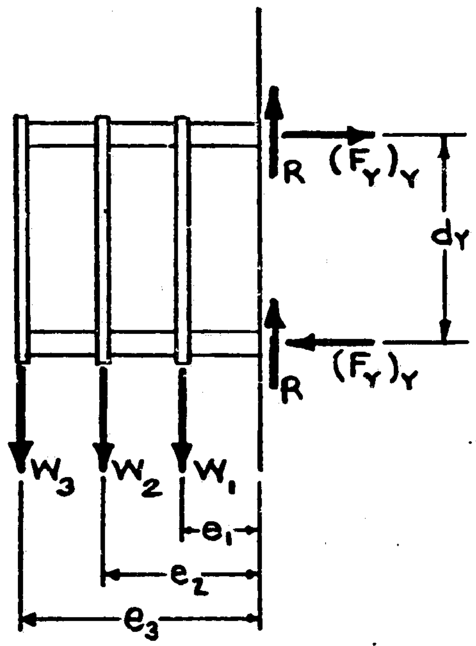
$$f_{b_y} = 27530(1.15/1.50) = \underline{21120 \text{ psi (YIELD)}}$$

FOR 2024-T351 AL. ALLOY PLATE

$F_{TU} = 65000$ psi
$F_{TY} = 46000$ psi } (REF. C)

M.S. LARGE BY INSPECTION

4.2 Y-AXIS EXCITATION

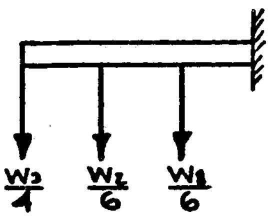


FROM PAGE 41, THE MAXIMUM Y-AXIS ACCELERATION IS

$$G_Y = 37.2\,g$$

4.2.1 BOLT BENDING STRESS (REF. DWG. 7232-2048)

THE LOADING ON THE CRITICAL BOLT IS SHOWN BELOW:



THE MAXIMUM BENDING MOMENT ON THE BOLT IS

$$M = (W_1/6)e_1 + (W_2/6)e_2 + (W_3/4)e_3$$

$e_1 = 0.636$ IN

$e_2 = 1.308$ IN

$e_3 = 1.980$ IN

$$M = (0.50/6)(0.636) + (0.50/6)(1.308) + (0.25/4)(1.980)$$

$$M = 0.286 \text{ IN-LBS}/g$$

$$M = 1.15(37.2)(0.286) = 12.2 \text{ IN-LBS. (YIELD)}$$

$$M = 1.50(37.2)(0.286) = 16.0 \text{ IN-LBS. (ULT.)}$$

FOR A #6 BOLT

$$I = 17.81 \times 10^{-6} \text{ IN}^4$$

$$C = 0.069 \text{ IN}$$

$$(I/C) = 2.58 \times 10^{-4} \text{ IN}^3$$

BENDING STRESS

$$f_{b_1} = \frac{M}{(I/C)}$$

$$f_{b_1} = (12.2 / 2.58 \times 10^{-4}) = 47300 \text{ psi (YIELD)}$$

$$f_{b_1} = (16.0 / 2.58 \times 10^{-4}) = 62000 \text{ psi (ULT.)}$$

AXIAL STRESS

FOR THE AXIAL STRESS DUE TO THE 300 LB. PRELOAD

$$f_{b_2} = 1.15(300/0.009) = 38300 \text{ psi (YIELD)}$$
$$f_{b_2} = 1.50(300/0.009) = 50000 \text{ psi (ULT)}$$

COMBINED STRESS

$$f_b = f_{b_1} + f_{b_2} = 47300 + 38300 = \underline{85600 \text{ psi (YIELD)}}$$
$$f_b = f_{b_1} + f_{b_2} = 62000 + 50000 = \underline{112000 \text{ psi (ULT.)}}$$

FOR BE-CU SCREW, $F_{TY}(\text{MIN}) = 160000$ psi, $F_{TU}(\text{MIN}) = 180000$ psi } REF. j

$$(\text{M.S.})_{\text{MIN.}} = \frac{F_{TU}}{f_b} - 1 = \frac{180000}{112000} - 1 = \underline{+0.61}$$

4.2.2 FRICTION CAPACITY AT CONVERTER HSG. INTERFACE

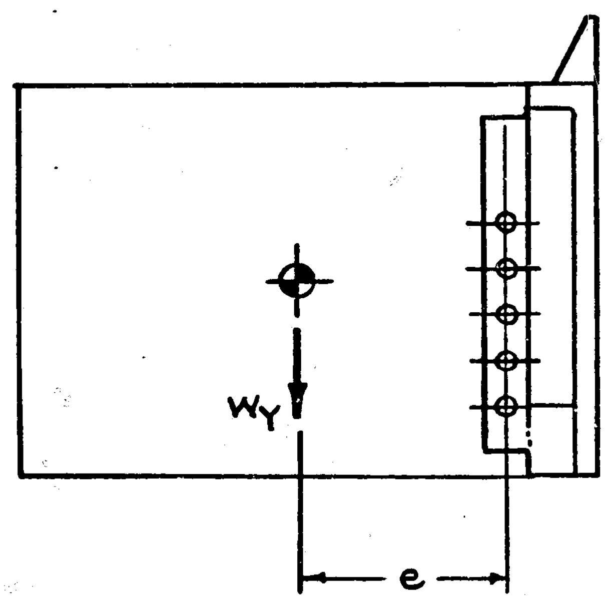
TOTAL LATERAL LOAD $(W_{DYN})_Y = 1.5\{37.2g(W_1+W_2+W_3)\}$
$$(W_{DYN})_Y = 1.5\{37.2(0.50+0.50+0.25)\}$$
$$(W_{DYN})_Y = \underline{69.8 \text{ LBS (ULT.)}}$$

ASSUMING BOLTS TORQUED TO 300 LB AXIAL LOAD AND A COEFFICIENT OF FRICTION $\mu = 0.2$, TOTAL CAPACITY (6 BOLTS) IS

$$F_F = 6(0.2)(300) = 360.0 \text{ LBS}$$

$$\text{M.S.} = \frac{F_F}{(W_{DYN})_Y} - 1 = \underline{\text{LARGE}}$$

4.2.3 ATTACHMENT OF GUSSET TO MOUNTING PLATE.



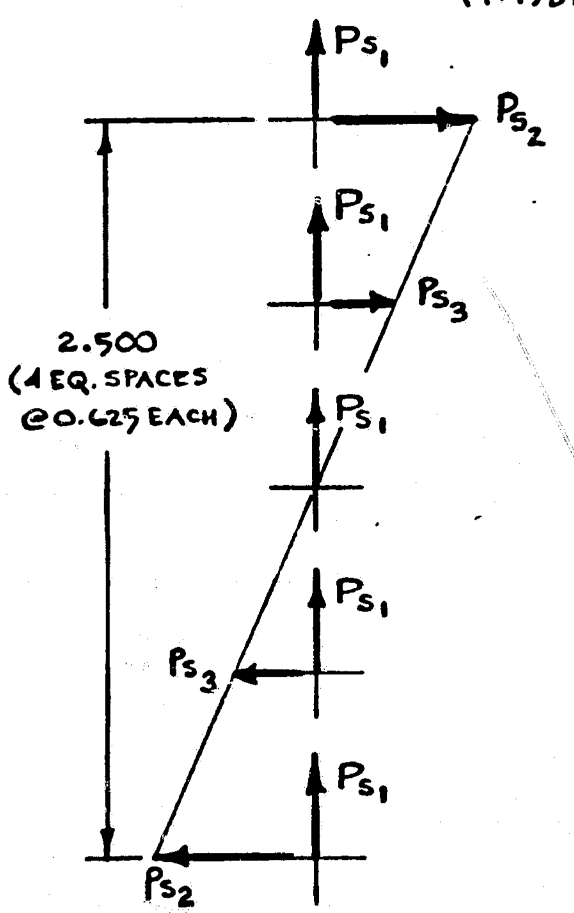
FOR THE ATTACHMENT OF THE GUSSETS TO THE FRONT MOUNTING PLATE, THE CRITICAL LOADING CONDITION IS IN THE Y-AXIS DIRECTION. SEVERAL CONSERVATIVE ASSUMPTIONS WILL BE MADE FOR THIS ANALYSIS. FIRST, IT WILL BE ASSUMED THAT THE MAGNITUDE OF THE LOAD TO BE TRANSMITTED THROUGH THE ATTACHMENT IS 7.50 LBS. SECOND, IT IS ASSUMED THAT ALL OF THIS LOAD IS CARRIED BY THE TEN RIVETS (NO LOAD IS ASSUMED TO BE TRANSMITTED THROUGH THE ATTACHMENTS OF THE WRAPAROUND AND THE CONVERTER HOUSING TO THE MOUNTING PLATE).

THE MAXIMUM Y-AXIS ACCELERATION WILL BE SHOWN ON PG. 80 TO BE $G_Y = 33.2g$

$$\therefore (W_Y)_{DYN.} = 1.50(7.50)(33.2) = \underline{373.50 \text{ LBS. (ULT.)}}$$

FOR AN ECCENTRICITY $e \approx 3.75$ IN., THE MOMENT ACROSS THE RIVETS IS

$$(M)_{DYN} = 3.75 (W_Y)_{DYN.} \approx \underline{1400 \text{ IN-LBS. (ULT.)}}$$



THE REACTIONS AT EACH ATTACHMENT POINT ARE SHOWN AT THE LEFT. FROM STATICS

$$\Sigma F = 0 = 5P_{s_1} - [(W_Y)_{DYN.}/2]$$

$$5P_{s_1} = 186.75$$

$$P_{s_1} = \underline{37.35 \text{ LBS. (ULT.)}}$$

$$\Sigma M = 0 = 2P_{s_2}(1.250) + 2P_{s_3}(0.625) - [(M)_{DYN}/2]$$

$$2.500 P_{s_2} + 1.250 P_{s_3} = 700$$

FROM SIMILAR TRIANGLES $P_{s_3} = 0.5 P_{s_2}$

$$\therefore \quad 2.500 P_{s_2} + 0.625 P_{s_2} = 700$$

$$P_{s_2} = \underline{224.00 \text{ LBS. (ULT.)}}$$

$$P_R = \sqrt{P_{s_1}^2 + P_{s_2}^2} = \underline{227 \text{ LBS. (ULT.)}}$$

RIVETS ARE 1/8" DIA. AD DESIGNATION. ALLOWABLE SHEAR PER RIVET IS 388 LBS (MIN.) — REF. C

$$\therefore \text{M.S.} = \frac{P}{P_R} - 1 = \frac{388}{227} - 1 = \underline{+0.71}$$

BEARING IN MOUNTING FLANGE WALL.

LET A_{br} = BEARING AREA = RIVET DIAM. x FLANGE THICK.

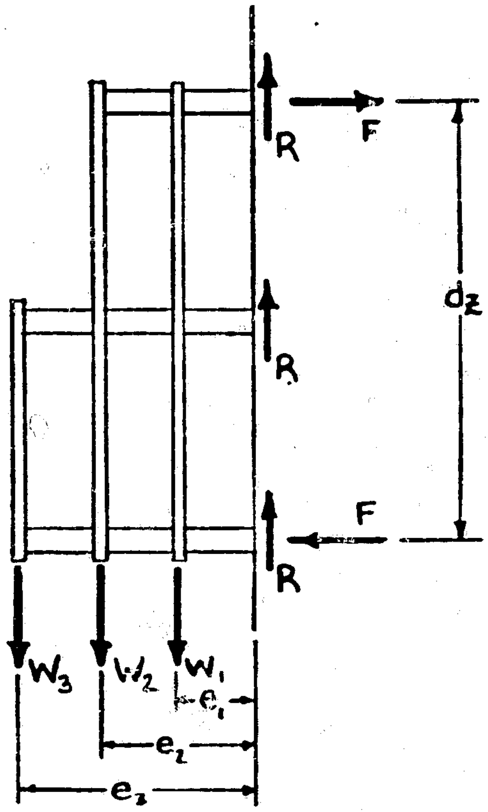
$$A_{br} = 0.125(0.050) = 0.00625 \text{ IN.}^2$$

$$f_{br} = \frac{P_R}{A_{br}} = \frac{227}{0.00625} = 36{,}320 \text{ PSI (ULT.)}$$

FROM REF. C FOR 2024-T351 PLATE, $F_{bru_{(e/D=2.0)}} = 124{,}000$ PSI

$$M.S. = \frac{F_{bru}}{f_{br}} - 1 = \underline{\text{LARGE}}$$

4.3 Z-AXIS EXCITATION



4.3.1 BOLT STRESSES.

FROM PAGE 41, THE MAXIMUM Z-AXIS ACCELERATION IS

$$G_Z = 47.9\,g$$

BOLT BENDING STRESS

THE BOLT BENDING STRESS CAN BE FOUND BY UTILIZING THE FOLLOWING RELATIONSHIP SINCE ACCELERATION LEVEL IS THE ONLY VARIABLE:

$$(f_b)_{Z\text{-AXIS}} = (f_b)_{Y\text{-AXIS}} \frac{G_Z}{G_Y}$$

$$\therefore\ f_{b_1} = 47300 \frac{47.9}{37.2} \cong \underline{60900 \text{ psi (YIELD)}}$$

$$f_{b_1} = 62000 \frac{47.9}{37.2} = \underline{79860 \text{ psi (ULT.)}}$$

BOLT AXIAL STRESS

THE AXIAL STRESS RESULTS FROM PRELOADING AND IS AS PREVIOUSLY CALCULATED

$$f_{b_2} = \underline{38300 \text{ psi (YIELD)}}$$
$$f_{b_2} = \underline{50000 \text{ psi (ULT.)}}$$

BOLT COMBINED STRESS

$$f_b = f_{b_1} + f_{b_2} = 60900 + 38300 = \underline{99200 \text{ psi (YIELD)}}$$
$$f_b = f_{b_1} + f_{b_2} = 79860 + 50000 = \underline{129860 \text{ psi (ULT.)}}$$

$$(M.S.)_{MIN.} = \frac{F_{TU}}{f_b} - 1 = \frac{180000}{129860} - 1 = \underline{+0.39}$$

4.3.2 FRICTION CAPACITY AT CONVERTER HSG. INTERFACE

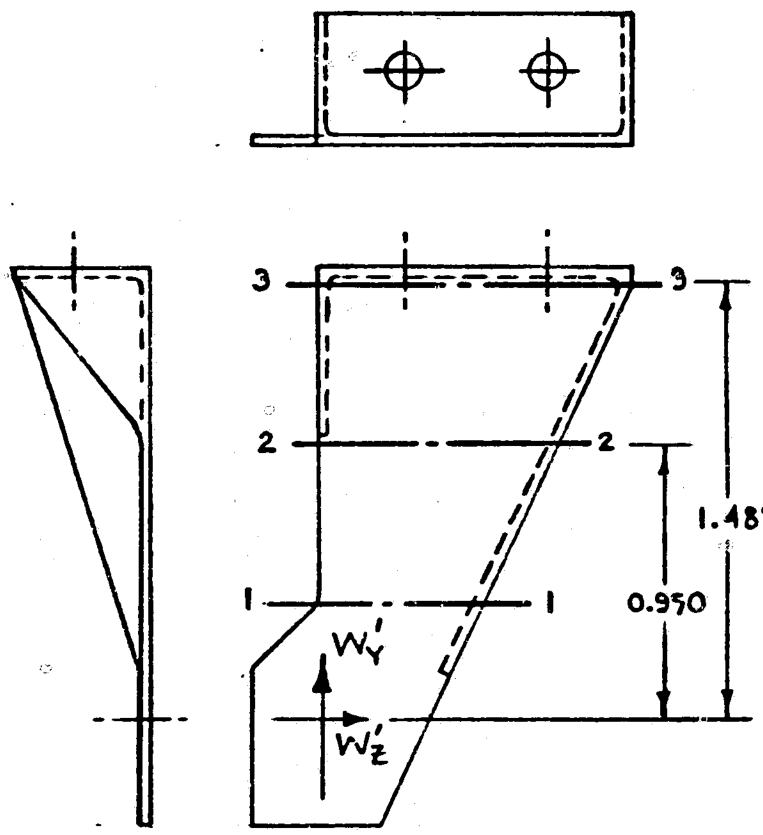
TOTAL LATERAL LOAD $(W_{DYN})_Z = (W_{DYN})_Y \times (G_Z/G_Y)$

$(W_{DYN})_Z = 69.8\,(47.9/37.2) = \underline{89.9 \text{ LBS. (ULT.)}}$

FRICTION FORCE, F_F = 360.0 LBS. (REF. PG. 71)

$$M.S. = \frac{F_F}{(W_{DYN})_Z} - 1 = \underline{\text{LARGE}}$$

4.4 STRESS ANALYSIS OF BRACKET.



MAT'L. 2024-T351 AL. AL.

IN THE PREVIOUS ANALYSIS OF THE PC BOARDS FOR BOTH Y & Z-AXIS LOADING, IT WAS ASSUMED THAT THE PC BOARDS WERE CANTILEVERED OFF THE RIGHT GUSSET PLATE. ACTUALLY, ONE END OF THE PC BOARD ARRANGEMENT IS ATTACHED TO SUPPORTING STRUCTURE BY MEANS OF THE BRACKET SHOWN AT THE LEFT. TO ANALYZE THE BKT. IT WILL CONSERVATIVELY BE ASSUMED THAT THE BRACKET CARRIES HALF THE TOTAL DYNAMIC LOAD. FROM PGS 71 & 74.

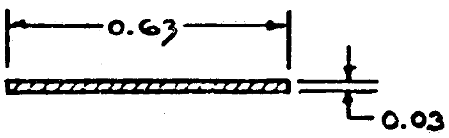
$W_Y' = 0.5(69.80) = \underline{34.90 \text{ LBS.}}$

$W_Z' = 0.5(89.90) = \underline{44.95 \text{ LBS.}}$

BOTH OF THE ABOVE LOADS ARE ULTIMATE LOADS.

4.4.1 Y-AXIS LOADING.

ASSUME THE BRACKET CROSS-SECTION 1-1 TO BE TYPICAL OVER THE ENTIRE LENGTH OF THE BRACKET (L ≈ 1.50 IN.). THE BRACKET WILL BE ANALYZED AS A PLATE SIMPLY SUPPORTED AT ITS ENDS, ITS EDGES FREE, UNDER A COMPRESSIVE LOAD W_Y'.



SECTION 1-1

$A = \text{AREA} = 0.63(0.03) = 0.0189 \text{ IN.}^2$

$$f_c = \frac{W_Y'}{A} = \frac{34.9}{0.0189} \cong \underline{1850 \text{ psi (ULT.)}}$$

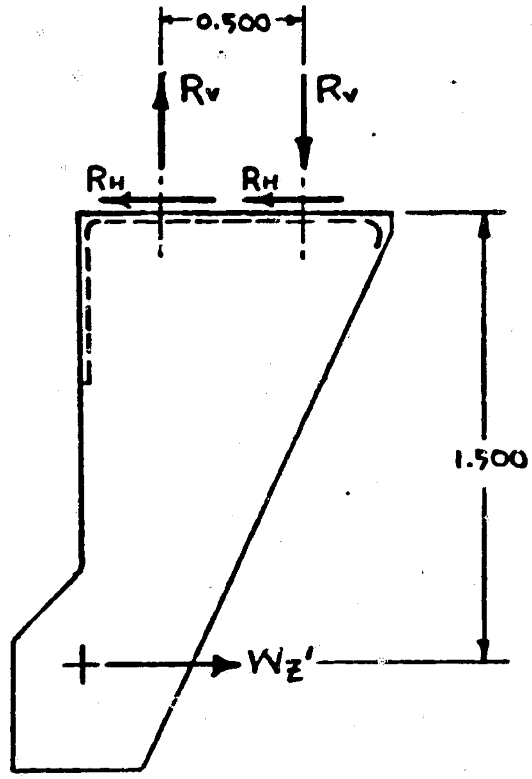
THE BUCKLING ALLOWABLE IS GIVEN BY

$$F_{c_{cr}} = \frac{\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{L}\right)^2 \qquad \text{(REF. d)}$$

$$F_{c_{cr}} = \frac{\pi^2(10.5\times 10^6)}{12(1-\nu^2)}\left\{\frac{0.03}{1.50}\right\}^2 = \underline{3800 \text{ psi}}$$

$$M.S. = \frac{F_{c_{cr}}}{f_c} - 1 = \frac{3800}{1850} - 1 = \underline{\text{LARGE}}$$

4.4.2 Z-AXIS LOADING.



ATTACHMENT LOADS.

$\Sigma F_H = 0 = W_Z' - 2R_H$

$R_H = 0.5W_Z' = 0.5(44.95)$

$R_H \approx$ 22.5 LBS. (ULT.)

$\Sigma M = 0 = 1.500W_Z' - 0.500R_V$

$R_V = 3W_Z' = 3(44.95)$

$R_V \approx$ 135 LBS. (ULT.)

FASTENERS ARE #4-40 UNC. CRES. FOR WHICH

$(P_Y)_{MIN} = 180$ LBS (REF. e)

ASSUME FASTENERS TORQUED TO 80% OF YIELD

$(P_Y)' = 0.8(180) = 144$ LBS.

$(R_V)_{YIELD} = (1.15/1.50)(R_V)_{ULT.}$

$(R_V)_{YIELD} = 0.767(135) \approx 103.5$ LBS. (YIELD)

$$M.S. = \frac{(P_Y)'}{(R_V)_Y} - 1 = \frac{144.0}{103.5} - 1 = +0.39$$

ASSUMING A COEFFICIENT OF FRICTION $\mu = 0.2$ THE FRICTION CAPACITY OF THE ATTACHMENT IS

$F_F = 2\{0.2(144)\} = 57.6$ LBS.

$(W_Z')_{YIELD} = (1.15/1.50)(44.95) = 34.5$ LBS.

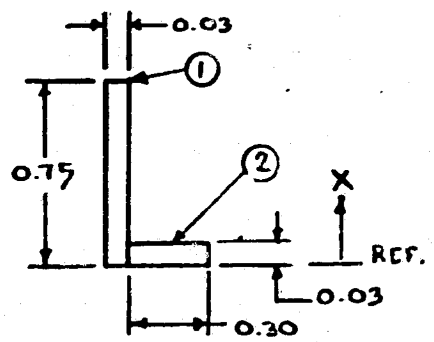
$$M.S. = \frac{F_F}{(W_Z')_Y} - 1 = \frac{57.6}{34.5} - 1 = +0.67$$

BENDING STRESSES.

For section 2-2, $M_{2-2} = 0.9W_Z' = 0.9(44.95) = 40.5$ IN-LBS (ULT.)

For section 3-3, $M_{3-3} = 1.5W_Z' = 1.5(44.95) = 67.4$ IN-LBS. (ULT.)

SECTION 2-2



ITEM	AREA, A	X	AX	AX²	Io
1	0.0225	0.375	0.00844	0.003164	0.001054
2	0.0090	0.015	0.00013	0.000002	0.000001
Σ	0.0315		0.00857	0.003166	0.001055

$\bar{X} = (\Sigma AX/\Sigma A) = 0.272$ IN

$C = 0.750 - 0.272 = 0.478$ IN.

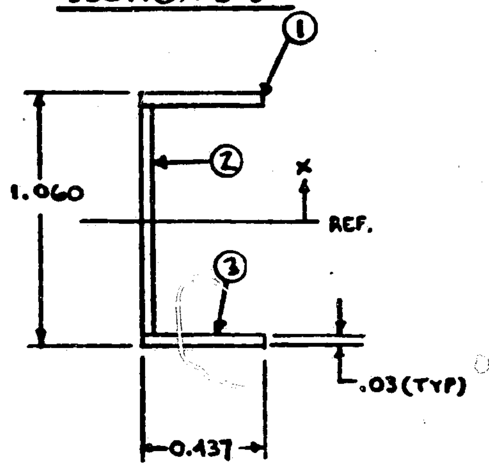
$I = \Sigma I_o + \Sigma AX^2 - \bar{X} \cdot \Sigma AX$

$I = 0.001889$ IN^4

$Z_{2-2} = (I/C) = 0.00395$ IN^3

$$f_{b_{2-2}} = \frac{M_{2-2}}{Z_{2-2}} = \frac{40.5}{0.00395} \approx \underline{10,250 \text{ psi (ULT.)}}$$

SECTION 3-3



ITEM	AREA A	X	AX	AX^2	I_o
1	0.01311	0.515	0.00675	0.00348	0.00000
2	0.03000	0	0	0	0.00250
3	0.01311	-0.515	-0.00675	0.00348	0.00000
Σ	0.05622		0	0.00696	0.00250

$C = 0.530$ in.

$I = \Sigma I_o + \Sigma AX^2 = 0.00946 \text{ in}^4$

$Z_{3-3} = (I/C) = 0.01785 \text{ in}^3$

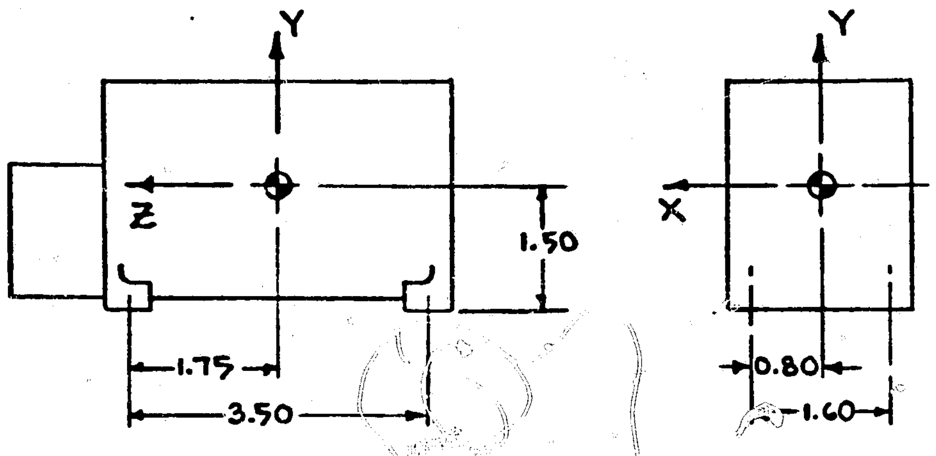
$$f_{b_{3-3}} = \frac{M_{3-3}}{Z_{3-3}} = \frac{67.4}{0.01785} = \underline{3775 \text{ psi (ULT.)}}$$

$f_{b\ MAX.} = f_{b_{2-2}} = 10,250$ psi (ULT.) $\therefore$

M.S. LARGE BY INSPECTION.

4.5 ATTACHMENT - P/M TUBE TO CONVERTER HOUSING

FROM PG. 28 THE WEIGHT OF THE P/M TUBE IS TAKEN AS 1.38 LBS.

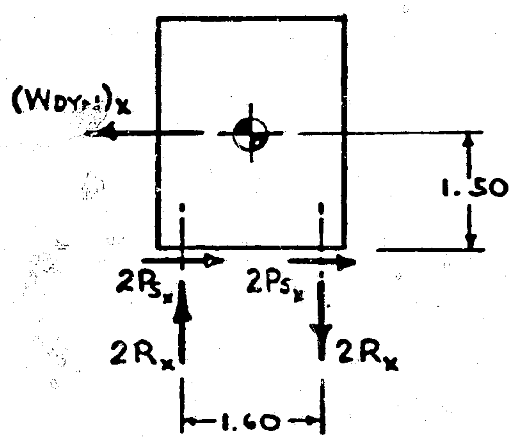


BASED ON THE PEAK ACCELERATIONS TABULATED ON PG. 41 THE P/M TUBE DYNAMIC LOADS ARE

$$(W_{DYN})_X = 1.15(1.38)(32.2) = \underline{51.1 \text{ LBS. (YIELD)}}$$
$$(W_{DYN})_Y = 1.15(1.38)(26.1) = \underline{41.4 \text{ LBS. (YIELD)}}$$
$$(W_{DYN})_Z = 1.15(1.38)(31.6) = \underline{50.1 \text{ LBS. (YIELD)}}$$

$$(W_{DYN})_X = (1.50/1.15)(51.1) = \underline{66.4 \text{ LBS. (ULT.)}}$$
$$(W_{DYN})_Y = (1.50/1.15)(41.4) = \underline{53.8 \text{ LBS. (ULT.)}}$$
$$(W_{DYN})_Z = (1.50/1.15)(50.1) = \underline{65.1 \text{ LBS. (ULT.)}}$$

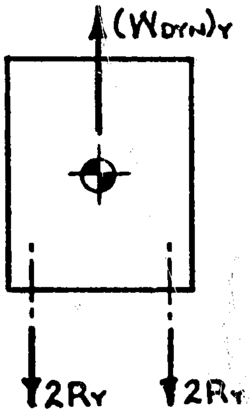
4.5.1 X-AXIS LOADING.



$$\Sigma F_x = 0 = 4Ps_x - (W_{DYN})_x$$
$$4Ps_x = (W_{DYN})_x$$
$$Ps_x = 0.25(W_{DYN})_x$$
$$Ps_x = \underline{12.8 \text{ LBS. (YIELD)}}$$
$$Ps_x = \underline{16.6 \text{ LBS. (ULT.)}}$$

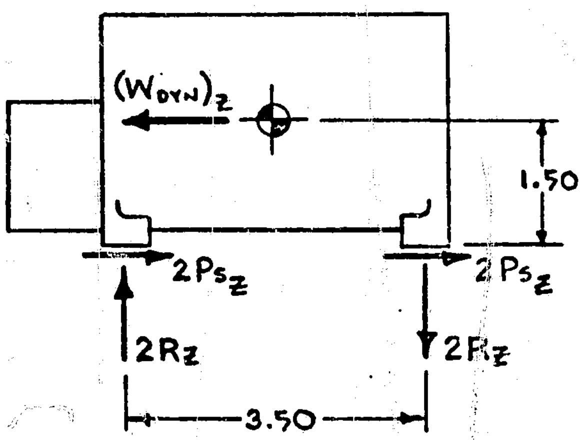
$$\Sigma M = 0 = 1.50(W_{DYN})_x - 1.60(2R_x)$$
$$3.20R_x = 1.50(W_{DYN})_x$$
$$R_x = 0.47(W_{DYN})_x$$
$$R_x = \underline{24.0 \text{ LBS. (YIELD)}}$$
$$R_x = \underline{31.2 \text{ LBS. (ULT.)}}$$

4.5.2 Y-AXIS LOADING.



$$\Sigma F_Y = 0 = (W_{DYN})_Y - 4R_Y$$
$$4R_Y = (W_{DYN})_Y$$
$$R_Y = 0.25(W_{DYN})_Y$$
$$R_Y = \underline{10.4 \text{ LBS. (YIELD)}}$$
$$R_Y = \underline{13.5 \text{ LBS. (ULT.)}}$$

4.5.3 Z-AXIS LOADING.



$$\Sigma F_Z = 0 = 4P_{s_Z} - (W_{DYN})_Z$$
$$4P_{s_Z} = (W_{DYN})_Z$$
$$P_{s_Z} = 0.25(W_{DYN})_Z$$
$$P_{s_Z} = \underline{12.5 \text{ LBS. (YIELD)}}$$
$$P_{s_Z} = \underline{16.3 \text{ LBS. (ULT.)}}$$

$$\Sigma M = 0 = 1.50(W_{DYN})_Z - 3.50(2R_Z)$$
$$7.00R_Z = 1.50(W_{DYN})_Z$$
$$R_Z = 0.21(W_{DYN})_Z$$
$$R_Z = \underline{10.5 \text{ LBS. (YIELD)}}$$
$$R_Z = \underline{13.7 \text{ LBS. (ULT.)}}$$

ATTACHMENT IS THRU FOUR #8-32 UNC CRES. FASTENERS. FROM REF. e, THE MINIMUM YIELD LOAD IS

$$(P_T)'_Y = 420 \text{ LBS.}$$

ASSUME FASTENERS TORQUED TO 80% OF YIELD

$$(P_Y)_Y = 0.80(420) = 336 \text{ LBS.}$$

THE MAXIMUM APPLIED BOLT LOAD IS $R_X = 24.0$ LBS (YIELD)

$$M.S. = \frac{(P_{TY})}{R_X} - 1 = \frac{336}{24} - 1 = \underline{\text{LARGE}}$$

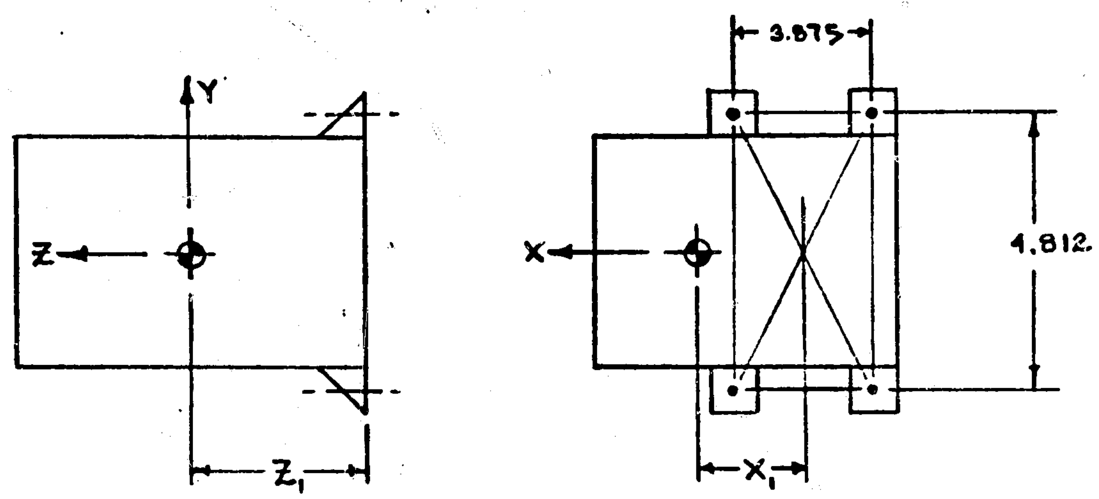
ASSUMING A COEFFICIENT OF FRICTION $\mu = 0.2$ BETWEEN THE P/M TUBE AND THE CONVERTER HOUSING, THE FRICTION CAPACITY OF THE JOINT IS

$$(P_F) = 4(0.2)(336) = 268.8 \text{ LBS.}$$

THE MAXIMUM SHEAR FORCE AT THE INTERFACE IS $(W_{DYN})_X = 51.1$ LBS. (YIELD)

$$M.S. = \frac{(P_F)}{(W_{DYN})_X} - 1 = \frac{268.8}{51.1} - 1 = \underline{\text{LARGE}}$$

4.6 ATTACHMENT - DETECT. ELECTRONICS TO FORWARD PLATE



ASSUME
$X_1 \approx 2.00$ IN.
$Z_1 \approx 4.50$ IN.

SCHEMATIC REPRESENTATION OF C.G. LOCATION AND INTERFACE MTG. POINTS

FROM PG. 28, W = 8.11 LBS

THE FOLLOWING TABLE IS USED TO DETERMINE THE OVERALL ACCELERATIONS AT THE CENTER OF GRAVITY.

ITEM	W, LBS.	X-AXIS				Y-AXIS				Z-AXIS			
		G_S	G_R	WG_S	WG_R	G_S	G_R	WG_S	WG_R	G_S	G_R	WG_S	WG_R
PC BOARDS	0.75	25.4	32.2	19.05	24.15	37.2	33.3	27.90	24.98	36.0	47.9	27.00	35.93
CONVERTER	1.50	25.5	32.3	38.25	48.45	55.1	64.5	82.65	96.75	27.3	31.8	40.95	47.70
P/M TUBE	1.38	25.4	32.2	35.05	44.44	26.1	24.8	36.02	34.22	25.8	31.6	35.60	43.61
REMAINDER	4.48	25.4	32.1	113.79	143.81	27.4	18.0	122.75	80.64	28.2	35.7	126.34	159.94
	8.11			206.14	260.85			269.32	236.59			229.89	287.18

AT THE PACKAGE C.G.

$$G_X = \frac{\Sigma WG_R}{\Sigma W} = \frac{260.85}{8.11} = \underline{32.2\ g}$$

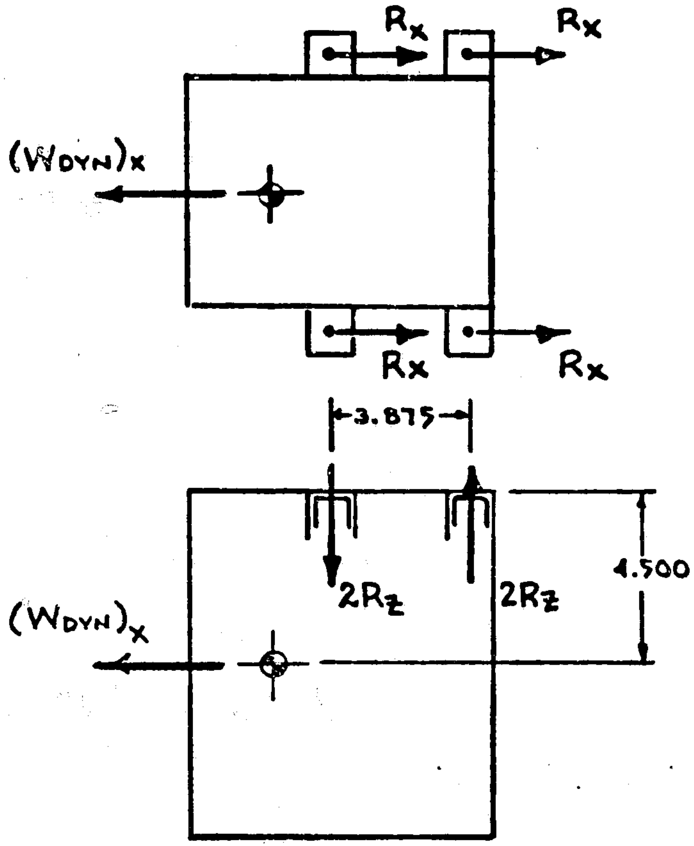
$$G_Y = \frac{\Sigma WG_S}{\Sigma W} = \frac{269.32}{8.11} = \underline{33.2\ g}$$

$$G_Z = \frac{\Sigma WG_R}{\Sigma W} = \frac{287.18}{8.11} = \underline{35.4\ g}$$

$(W_{DYN})_X = 1.15(8.11)(32.2) = \underline{300.3 \text{ LBS. (YIELD)}}$
$(W_{DYN})_Y = 1.15(8.11)(33.2) = \underline{309.6 \text{ LBS. (YIELD)}}$
$(W_{DYN})_Z = 1.15(8.11)(35.4) = \underline{330.2 \text{ LBS. (YIELD)}}$

$(W_{DYN})_X = 1.50(8.11)(32.2) = \underline{391.7 \text{ LBS. (ULT.)}}$
$(W_{DYN})_Y = 1.50(8.11)(33.2) = \underline{403.9 \text{ LBS. (ULT.)}}$
$(W_{DYN})_Z = 1.50(8.11)(35.4) = \underline{430.6 \text{ LBS. (ULT.)}}$

4.6.1 X-AXIS LOADING



$\Sigma F_x = 0 = (W_{DYN})_x - 4R_x$

$R_x = 0.250(W_{DYN})_x$

$R_x = \underline{75.1 \text{ LBS. (YIELD)}}$

$R_x = \underline{97.9 \text{ LBS. (ULT.)}}$

$\Sigma M = 0 = 4.500(W_{DYN})_x - 3.875(2R_z)$

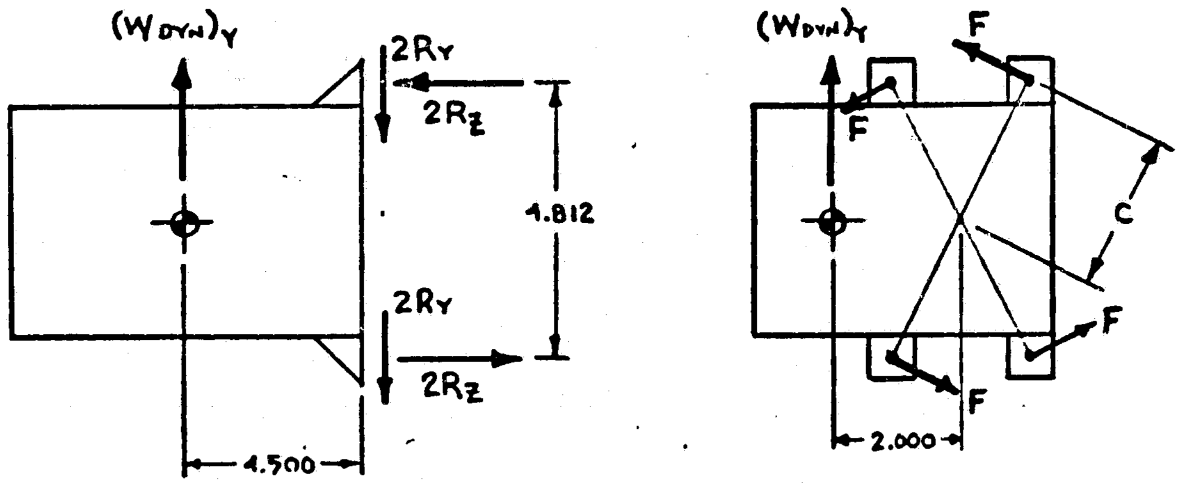
$7.750 R_z = 4.500(W_{DYN})_x$

$R_z = 0.581(W_{DYN})_x$

$R_z = \underline{174.2 \text{ LBS. (YIELD)}}$

$R_z = \underline{227.2 \text{ LBS. (ULT.)}}$

4.6.2 Y-AXIS LOADING



$\Sigma F_Y = 0 = (W_{DYN})_Y - 4R_Y$

$R_Y = 0.250(W_{DYN})_Y$

$R_Y = \underline{77.4 \text{ LBS. (YIELD)}}$

$R_Y = \underline{101.0 \text{ LBS. (ULT.)}}$

$\Sigma M = 0 = 4.500(W_{DYN})_Y - 4.812(2R_Z)$

$9.624 R_Z = 4.500(W_{DYN})_Y$

$R_Z = 0.468(W_{DYN})_Y$

$R_Z = \underline{145.5 \text{ LBS. (YIELD)}}$

$R_Z = \underline{189.8 \text{ LBS. (ULT.)}}$

$$C = \sqrt{\left\{\frac{3.875}{2}\right\}^2 + \left\{\frac{4.812}{2}\right\}^2} = 3.089 \text{ IN.}$$

$\Sigma T = 0 = 2.000(W_{DYN})_Y - 3.089(4F)$

$12.356F = 2.000(W_{DYN})_Y$

$F = 0.162(W_{DYN})_Y$

$F = \underline{49.5 \text{ LBS. (YIELD)}}$

$F = \underline{64.6 \text{ LBS. (ULT.)}}$

RESULTANT FORCE, $R_S = \sqrt{R_Y^2 + F^2}$

$$R_S = \sqrt{(77.4)^2 + (49.5)^2} = \underline{91.9 \text{ LBS. (YIELD)}}$$

$$R_S = \sqrt{(101.0)^2 + (64.6)^2} = \underline{119.9 \text{ LBS. (ULT.)}}$$

4.6.2.1 BOLT SHEAR

THE PREVIOUS ANALYSIS ASSUMED THAT THE BOLTS ATTACHING THE DETECTOR ELECTRONICS MOUNTING PLATE TO THE FORWARD PLATE WERE STANDARD BOLTS, PRELOADED SO THAT LATERAL LOADS ARE TRANSMITTED BY FRICTION. NOW, ASSUME A SHEAR OR SHOULDER BOLT IS BEING USED.

BOLT SHEAR AREA, $A_S = \frac{\pi}{4} d^2$

$A_S = 0.7854 (0.250)^2$

$A_S = 0.049 \text{ in}^2$

MAXIMUM SHEAR FORCE, $P_S = R_S = 91.9$ LBS (YIELD)

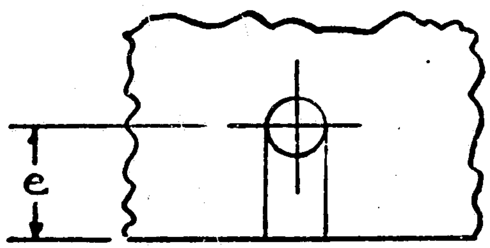
$= 119.9$ LBS (ULT.)

$$f_{s_Y} = \frac{91.9}{0.049} = \underline{1875 \text{ psi (YIELD)}}$$

$$f_{s_U} = \frac{119.9}{0.049} = \underline{2450 \text{ psi (ULT.)}}$$

<u>M.S. LARGE BY INSPECTION</u>

4.6.2.2 SHEAR TEAROUT - MOUNTING FLANGE.



$e = 0.343$ IN.

$t = 0.125$ IN.

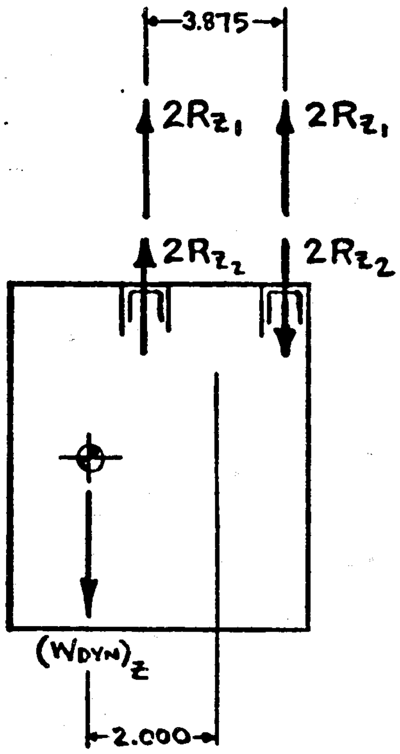
A_S = SHEAR TEAROUT AREA

$A_S = 2et = 0.086 \text{ in}^2$

$$f_{s_Y} = \frac{91.9}{0.086} = 1{,}070 \text{ psi (YIELD)}$$

<u>M.S. LARGE BY INSPECTION</u>

4.6.3 Z-AXIS LOADING



$\Sigma F_Z = 0 = (W_{DYN})_Z - 4R_{Z_1}$

$R_{Z_1} = 0.250(W_{DYN})_Z$

$R_{Z_1} = 82.6$ LBS. (YIELD)

$R_{Z_1} = 107.7$ LBS. (ULT.)

$\Sigma M = 0 = 2.000(W_{DYN})_Z - 3.875(2R_{Z_2})$

$7.750 R_{Z_2} = 2.000(W_{DYN})_Z$

$R_{Z_2} = 0.258(W_{DYN})_Z$

$R_{Z_2} = 85.2$ LBS. (YIELD)

$R_{Z_2} = 111.1$ LBS. (ULT.)

$\Sigma R_Z = R_{Z_1} + R_{Z_2} = 167.8$ LBS. (YIELD)

$\Sigma R_Z = R_{Z_1} + R_{Z_2} = 218.8$ LBS. (ULT.)

4.6.3.1 BOLT LOADING

MAXIMUM APPLIED AXIAL LOAD = R_Z (DUE TO X-AXIS LOADING)

$R_Z = 174.2$ LBS. (YIELD)

$R_Z = 227.2$ LBS. (ULT.)

BOLT IS 1/4"-20 UNC CRES.

$(P_Y)_{MIN.} = 950$ LBS. (REF. e)

ASSUME FASTENER TORQUED TO 80% OF YIELD

$(P_Y) = 0.80(950) = 760.0$ LBS.

$$M.S. = \frac{(P_Y)}{R_Z} - 1 = \frac{760.0}{174.2} - 1 = \text{LARGE}$$

4.6.3.2 JOINT FRICTION CAPACITY

ASSUMING $\mu = 0.2$, THE JOINT FRICTION CAPACITY IS

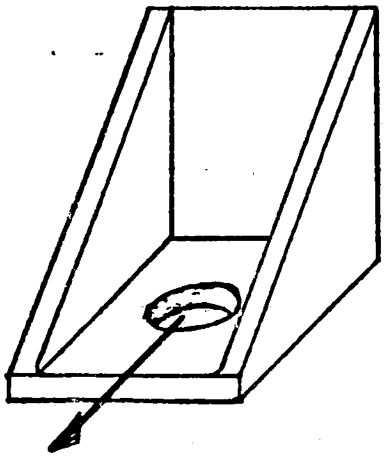
$F_F = 4(0.2)(760) = 608.0$ LBS.

MAX. LATERAL LOAD TO BE TRANSMITTED ACROSS JOINT IS

$(W_{DYN})_Z = 330.2$ LBS (YIELD)

$$M.S. = \frac{F_F}{(W_{DYN})_Z} - 1 = \frac{608.0}{330.2} - 1 = +0.84$$

4.6.4 BEARING STRESS - MOUNTING PLATE FLANGE



CONSERVATIVELY ASSUME $P = R_S$

$P = 91.9$ LBS. (YIELD)
$P = 119.9$ LBS. (ULT.)
} REF. PG. 82

A_{br} = BEARING AREA = dt

d = BOLT DIAMETER = 0.250 IN.

t = PLATE THICKNESS = 0.125 IN.

$A_{br} = 0.250(0.125) = 0.031 \text{ IN}^2$

$$f_{br_Y} = \frac{91.9}{0.031} = \underline{2960 \text{ psi (YIELD)}}$$

$$f_{br_U} = \frac{119.9}{0.031} = \underline{3870 \text{ psi (ULT.)}}$$

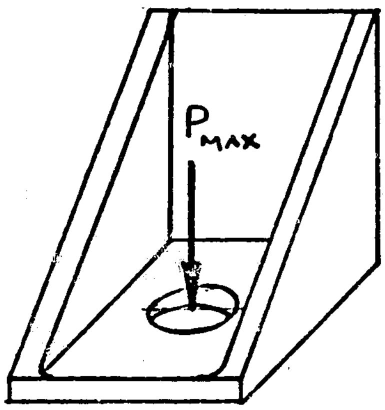
FROM REF. C FOR 2024-T351 AL ALLOY WITH ($e/D = 1.5$)

$F_{bry} = 69000$ psi

$F_{bru} = 98000$ psi

M.S. LARGE BY INSPECTION

4.65 BENDING STRESS - MOUNTING PLATE FLANGE



ASSUME THE MOUNTING PLATE FLANGE TO BE A RECTANGULAR FLAT PLATE WITH ONE LONG EDGE CLAMPED, THE OTHER LONG EDGE FREE AND BOTH SHORT EDGES SIMPLY SUPPORTED. THE LOAD IS ASSUMED TO BE UNIFORMLY DISTRIBUTED BY A WASHER. IN ACTUALITY, THE WEBS OF THE "BATHTUB FITTING" MAKE THE SHORT EDGES SOMEWHAT BETTER THAN SIMPLY SUPPORTED. ON THE OTHER HAND, ASSUMING A DISTRIBUTED LOAD IS SLIGHTLY UNCONSERVATIVE. TO COMPENSATE FOR THIS UNCERTAINTY, THE STRESS WILL BE MULTIPLIED BY A FACTOR OF 1.50 (ON TOP OF DESIGN LOAD FACTOR)

AT CENTER OF CLAMPED EDGE

$$f_b' = \frac{3\omega b^2}{t^2(1+3.2\alpha^3)} \qquad \text{(REF. K)}$$

a = LONG DIMENSION = 1.025 IN.
b = SHORT DIMENSION = 0.668 IN.
$\alpha = (b/a) = 0.652$
$\alpha^3 = 0.277$

$$f_b = 1.5 f_b' = \frac{1.5(3\omega b^2)}{1.886 t^2} = 2.386 \frac{\omega b^2}{t^2}$$

ASSUME $\omega = P_{MAX}/b^2$

$$\therefore \; f_b = 2.386 \frac{P_{MAX}}{t^2} = 2.386 \frac{P_{MAX}}{(0.125)^2} = 152.7 P_{MAX}$$

FROM PG. 81, P_{MAX} = 174.2 LBS (YIELD)
= 227.2 LBS (ULT)

$$f_b = 157.2(174.2) = 27380 \text{ PSI (YIELD)}$$
$$f_b = 157.2(227.2) = 35720 \text{ PSI (ULT.)}$$

FROM REF. G, FOR 2024-T3 AL, ALLOY: F_{TY} = 48000 PSI, F_{TU} = 65000 PSI

$$(M.S.)_{MIN.} = \frac{F_{TY}}{f_b} - 1 = \frac{48000}{27380} - 1 = +0.75$$

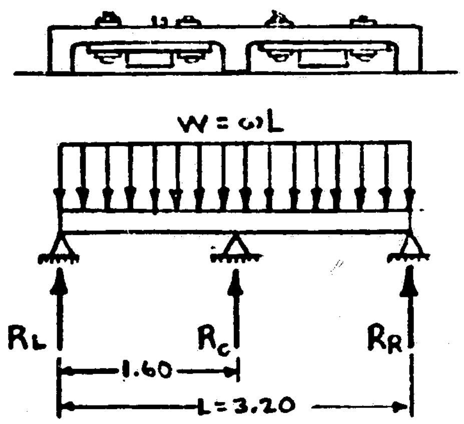
4.7 STRESS ANALYSIS OF DC-DC CONVERTER.

THE VARIOUS ELEMENTS OF THE DC-DC CONVERTER WILL BE ANALYZED FOR THE FOLLOWING ACCELERATIONS:

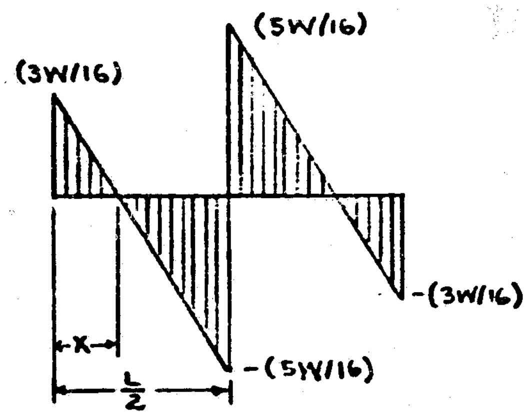
$$\left.\begin{aligned} G_x &= 32.3\ g \\ G_Y &= 64.5\ g \\ G_z &= 31.8\ g \end{aligned}\right\} \quad \text{(REF. PG. 41)}$$

4.7.1 ANALYSIS OF HEAT SINK # 1.

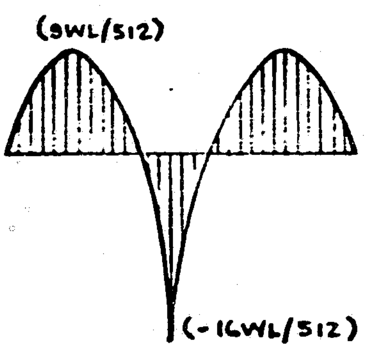
4.7.1.1 X-AXIS LOADING.



SHEAR DIAGRAM



MOMENT DIAGRAM



DETERMINATION OF REDUNDANT REACTION

FIRST REMOVE SUPPORT Rc AND ALLOW BEAM TO DEFLECT FREELY.

$$y_{c_1} = \frac{5WL^3}{384EI} \quad \downarrow$$

REAPPLYING Rc AS A CONCENTRATED UPWARDS LOAD

$$y_{c_2} = \frac{R_c L^3}{48EI} \quad \uparrow$$

THE NET DEFLECTION AT POINT C = 0. THEREFORE

$$\frac{5WL^3}{384EI} = \frac{R_c L^3}{48EI}$$

$$R_c = (5W/8)$$

$$\& \quad R_L = R_R = (3W/16)$$

FROM SHEAR DIAGRAM

$$\frac{x}{(L/2)} = \frac{(3W/16)}{(8W/16)}$$

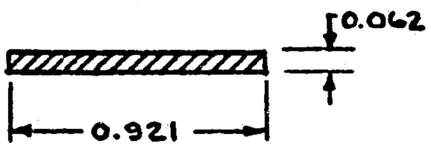
$$x = (3L/16)$$

$$\therefore M_1 = \frac{1}{2}(3L/16)(3W/16) = (9WL/512)$$

$$M_2 = \frac{1}{2}(5L/16)(-5W/16) = -(25WL/512)$$

$$\therefore M_{MAX} = (-16WL/512) = (-WL/32)$$

BEAM CROSS-SECTION



CONSERVATIVELY ASSUME $W = 0.125$ LBS.

$$\therefore f_b = \frac{6M_{MAX}}{bt^2} = \frac{6(WL/32)}{0.921(0.062)^2} = \frac{6W(3.20)/32}{0.921(0.003844}$$

$$f_b = 169.5\,W$$

FOR DYNAMIC LOADING

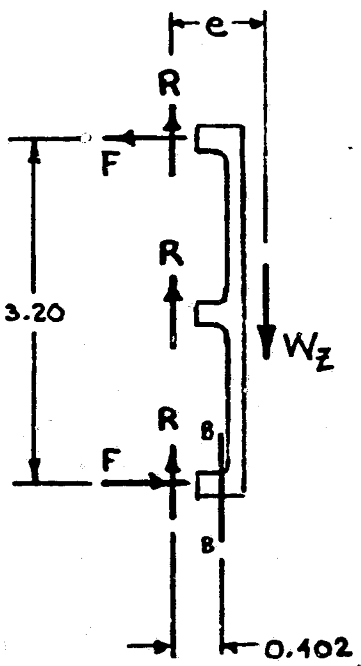
$$W = 1.5(0.125\text{ LBS})(32.3g) \cong 6 \text{ LBS. (ULT.)}$$

$$f_b = 169.5(6) = \underline{1017 \text{ psi (ULT.)}}$$

MAT'L. IS 2024-T351 ALUMINUM ALLOY PLATE.

∴ M.S. LARGE BY INSPECTION.

4.7.1.2 Z-AXIS LOADING.



ASSUME C.G. AT A DISTANCE $e = 0.40$ IN. ABOVE INTERFACE. THE DYNAMIC LOAD IS

$$W_z = 1.5(0.125\text{ LBS})(31.8g) \approx 6 \text{ LBS. (ULT)}$$

$$\Sigma M = 0 = W_z \cdot e - F(3.20)$$

$$3.20F = 6(0.40) = 2.40$$

$$F = 0.75 \text{ LBS. (ULT.)}$$

BENDING MOMENT ON SECTION B-B IS

$$M_{B-B} = 0.402R$$

WHERE $R = (W_z/3) = 2$ LBS (ULT.)

$$\therefore M_{B-B} = 0.402(2) = 0.80 \text{ IN-LBS (ULT)}$$

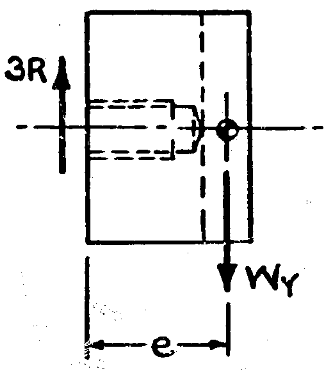
BENDING STRESS

$$f_b = \frac{6M_{B-B}}{bt^2} = \frac{6(0.80)}{0.921(0.25)^2} \approx \underline{84 \text{ psi (ULT)}}$$

M.S. LARGE BY INSPECTION.

4.7.1.3 Y-AXIS LOADING.

DYNAMIC LOADING



$W_Y = 1.5(0.125\text{ LBS.})(64.5g) = 12.1\text{ LBS. (ULT.)}$

$\Sigma F_Y = 0 = W_Y - 3R$

$3R = 12.1$

$R \approx 4\text{ LBS. (ULT.)}$

ASSUMING $e = 0.40$ IN., THE TOTAL MOMENT TO BE CARRIED BY THE THREE FASTENERS IS

$M' = W_Y \cdot e = 4.8\text{ IN-LBS (ULT.)}$

FOR EACH FASTENER

$M = (M'/3) = 1.6\text{ IN-LBS. (ULT.)}$

SCREWS ARE #4-40 UNC CRES. ($F_{TY} = 30000$ PSI)

FOR YIELD CONDITION

$M_Y = (1.15/1.50)(1.6) = 1.23\text{ IN-LBS (YIELD)}$

SCREW BENDING STRESS.

SCREW STRESS AREA = 0.006 IN^2

SCREW SECTION MODULUS, $Z = (\pi d^3/32) = \{(\pi)(0.112)^3/32\} = 1.38 \times 10^{-4}\text{ IN}^3$

$$f_b = \frac{M_Y}{Z} = \underline{8910\text{ PSI (YIELD)}}$$

CONSERVATIVELY ASSUMING THAT THIS STRESS DOESN'T RELIEVE PRELOAD,

$$M.S. = \frac{F_{TY}}{f_b} - 1 = \frac{30000}{8910} - 1 = \underline{\text{LARGE}}$$

JOINT FRICTION CAPACITY.

ASSUME SCREW TORQUED TO 80% YIELD

$P = 0.8(180) = 144\text{ LBS.}$

ASSUME A COEFFICIENT OF FRICTION $\mu = 0.2$

$F_F = 0.2P = 28.8\text{ LBS}$

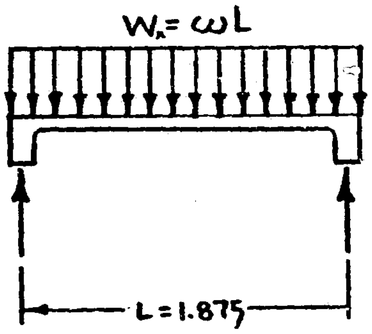
SINCE THERE ARE 3 SCREWS

$\Sigma F_F = 3(28.8) = 86.4\text{ LBS}$

$$M.S. = \frac{\Sigma F_F}{R} - 1 = \frac{86.4}{4.0} - 1 = \underline{\text{LARGE}}$$

4.7.2 ANALYSIS OF HEAT SINK #2.

4.7.2.1 X-AXIS LOADING.



ASSUME $W_X = 0.10$ LBS.

$(W_X)_{DYN.} = 1.5(0.10 \text{ LBS.})(32.3g)$

$(W_X)_{DYN.} \cong 4.8$ LBS. (ULT.)

$M_{MAX} = \frac{1}{8}WL = \frac{1}{8}(4.8)(1.875)$

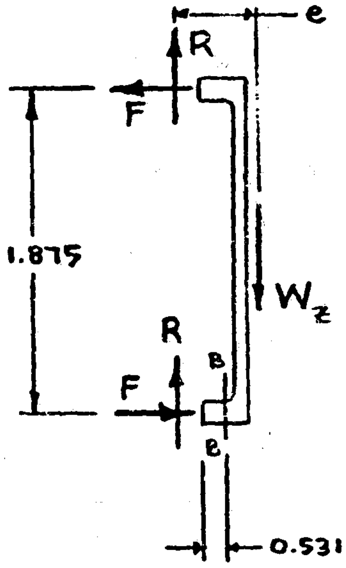
$M_{MAX} = 1.125$ IN-LBS. (ULT.)

$$f_b = \frac{6M_{MAX}}{bt^2} = \frac{6(1.125)}{1.125(.062)^2}$$

$f_b = 1560$ PSI (ULT.)

M.S. LARGE BY INSPECTION.

4.7.2.2 Z-AXIS LOADING.



$\Sigma F = 0 = W_Z - 2R$

$R = (W_Z/2)$

$(W_Z)_{DYN.} = 1.5(0.10 \text{ LBS.})(31.8g)$

$(W_Z)_{DYN.} \cong 4.8$ LBS. (ULT.)

$\therefore R = 2.4$ LBS. (ULT.)

FOR SECTION B-B

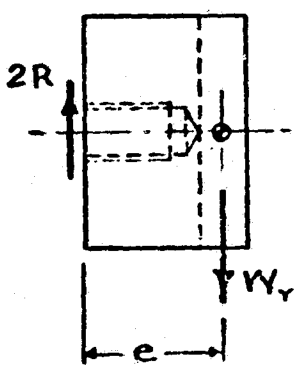
$M = 0.531R = 1.274$ IN-LBS. (ULT.)

$$f_b = \frac{6M}{bt^2} = \frac{6(1.274)}{0.66(0.250)^2}$$

$f_b = 185$ PSI (ULT.)

M.S. LARGE BY INSPECTION.

4.7.2.3 Y-AXIS LOADING.



DYNAMIC LOADING

$W_Y = 1.5(0.10 \text{ LBS.})(64.5g) \cong 9.7$ LBS. (ULT.)

$\Sigma F = 0 = W_Y - 2R$

$R = 0.5W_Y = 4.85$ LBS (ULT.)

ASSUMING $e = 0.40$ IN, THE BENDING MOMENT TO BE CARRIED BY EACH FASTENER IS

$M = R \cdot e = 4.85(0.40) = 1.94$ IN-LBS (ULT.)

FOR YIELD LOADING CONDITION

$M_Y = (1.15/1.50)(1.94) = 1.49$ IN-LBS. (YIELD)

SCREW BENDING STRESS.

SCREW SECTION MODULUS, $Z = 1.38 \times 10^{-4}$ IN.^{3}

$$f_b = \frac{M_Y}{Z} = \frac{1.49}{1.38 \times 10^{-4}} = \underline{10800 \text{ psi (YIELD)}}$$

$F_{TY} = 30000$ psi (REF. e)

$$\text{M.S.} = \frac{F_{TY}}{f_b} - 1 = \frac{30000}{10800} - 1 = \underline{\text{LARGE}}$$

JOINT FRICTION CAPACITY.

ASSUME SCREW TORQUED TO 80% YIELD

$P = 0.80(180) = 144$ LBS.

ASSUME A COEFFICIENT OF FRICTION $\mu = 0.2$

$F_F = 0.2P = 28.8$ LBS.

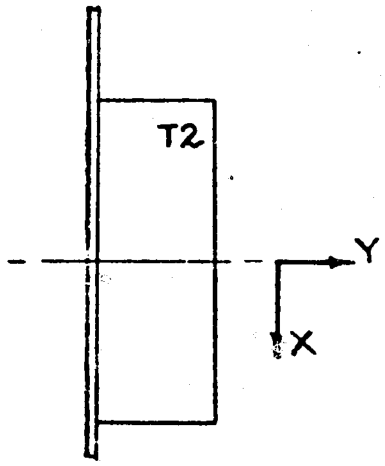
SINCE THERE ARE 2 SCREWS

$\Sigma F_F = 2(28.8) = 57.6$ LBS.

$$\text{M.S.} = \frac{\Sigma F_F}{R} - 1 = \frac{57.6}{4.85} - 1 = \underline{\text{LARGE}}$$

4.7.3 ATTACHMENT OF TRANSFORMER T2 TO CONVERTER HOUSING.

4.7.3.1 Y-AXIS LOADING.



ESTIMATED WEIGHT OF TRANSFORMER T2 = (1/3) LB.
T2 HELD DOWN BY ONE #6-32 UNC CRES. SCREW

$(P_Y)_{MIN}$ = 270 LBS. (REF. e)

ASSUME SCREW TORQUED TO 80% OF YIELD.

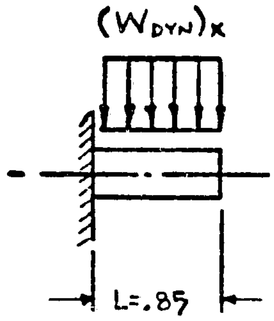
$(P_Y)' = 0.80(270) = 216.0$ LBS.

FOR ACCELERATION $G_Y = 64.5\,g$ (REF. PG. 41)

$(W_{DYN})_Y = 1.15(1/3)(64.5) = 24.7$ LBS. (YIELD)

$$M.S. = \frac{(P_Y)'}{(W_{DYN})_Y} - 1 = \frac{216.0}{24.7} - 1 = \underline{LARGE}$$

4.7.3.2 X-AXIS LOADING.



BENDING STRESS IN SCREW

$(W_{DYN.})_X = 1.15(1/3)(32.3g) = 12.4$ LBS. (YIELD)

$M_{MAX.} = \{(W_{DYN.})_X L/2\}$

$M_{MAX.} = (12.4)(0.85)/2 = 5.27$ IN-LBS. (YIELD)

Z = SCREW SECTION MODULUS = $(\pi d^3/32)$

$$Z = \frac{\pi(0.138)^3}{32} = 2.58 \times 10^{-4} \text{ IN.}^3$$

$$f_b = \frac{M_{MAX}}{Z} = \underline{20400 \text{ psi (YIELD)}}$$

$F_{TY} = 30000$ psi (REF. e)

$$M.S. = \frac{F_{TY}}{f_b} - 1 = \frac{30000}{20400} - 1 = \underline{+0.47}$$

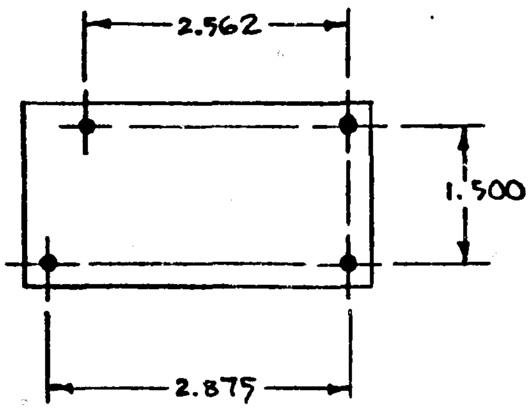
JOINT FRICTION CAPACITY

ASSUMING $\mu = 0.2$, THE FRICTION CAPACITY IS

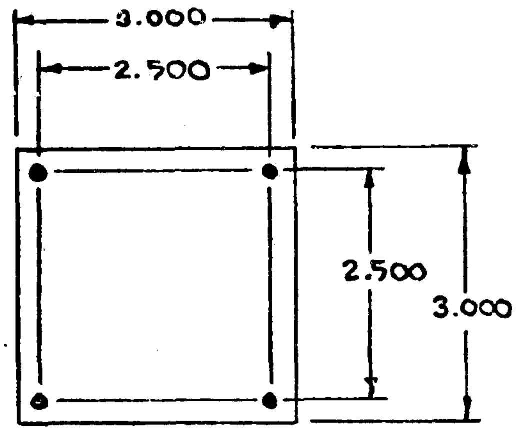
$F_F = \mu(P_Y)' = 0.2(216) = 43.2$ LBS.

$$M.S. = \frac{F_F}{(W_{DYN})_X} - 1 = \frac{43.2}{12.4} - 1 = \underline{LARGE}$$

4.7.4 STRESS ANALYSIS OF REGULATOR BOARD.



ACTUAL BOARD



ASSUMED BOARD

1. ASSUME LOADED BOARD WEIGHT 0.20 LBS.
2. ASSUME Q = 5 @ RESONANCE.
3. ASSUME GY = 64.5g IS BOARD INPUT.

$W_{DYN.} = 1.5[(0.20 \text{ LBS.})(64.5g)(Q=5)] = 96.75$ LBS. (ULT.)

LET ASSUMED BOARD AREA, $A = (3.00)^2 = 9 \text{ IN}^2$

$$q = \frac{W_{DYN}}{A} = \frac{96.75}{9} = 10.75 \text{ PSI (ULT.)}$$

$$M_{MAX.} = 0.1527\, q a^2 \qquad \text{(REF. } g\text{)}$$

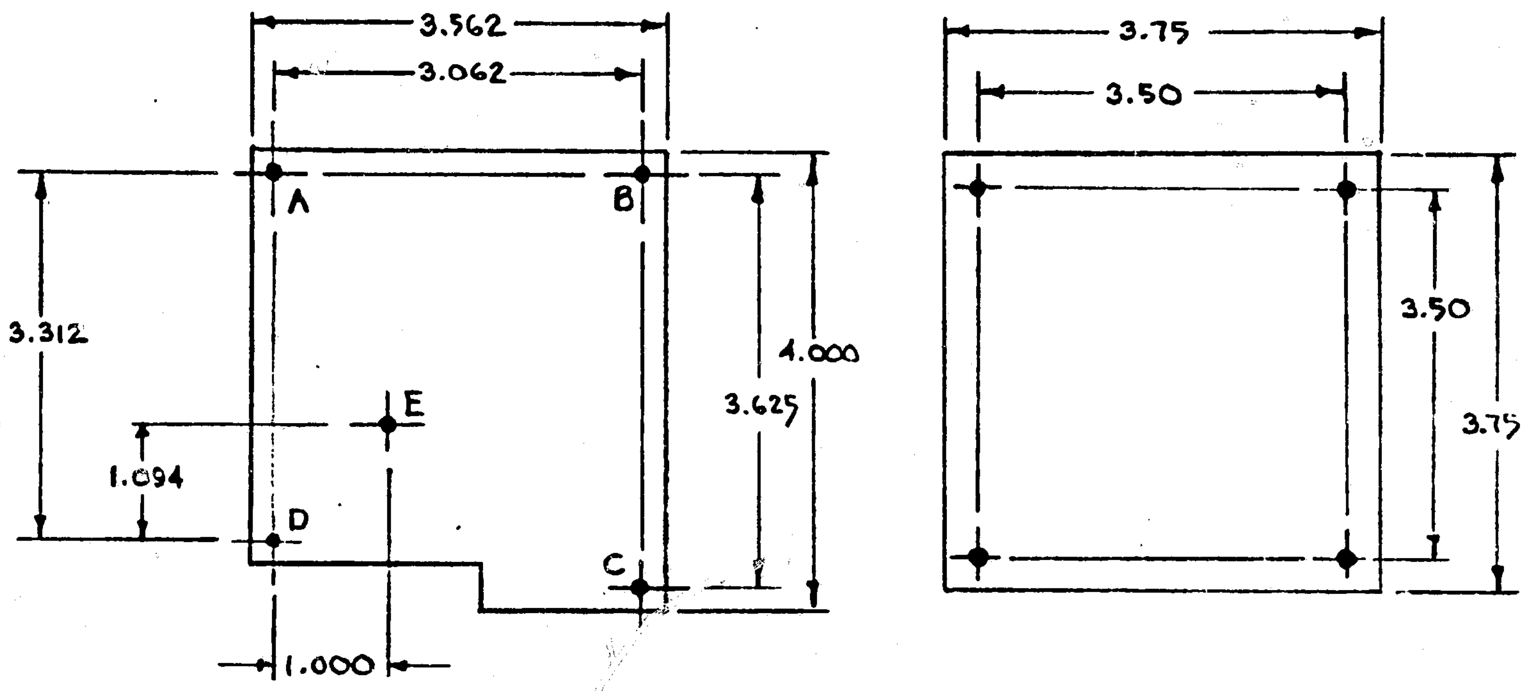
$$M_{MAX.} = 0.1527(10.75)(2.50)^2 = 10.26 \text{ IN-LBS/IN.}$$

BENDING STRESS, $f_b = \frac{6 M_{MAX}}{t^2} = \frac{6(10.26)}{(0.062)^2} \cong$ 16000 PSI (ULT.)

ALLOWABLE FLEXURAL STRENGTH, $F_{FL} = 40{,}000$ PSI (REF. h)

$$M.S. = \frac{F_{FL}}{f_b} - 1 = \frac{40000}{16000} - 1 = \text{LARGE}$$

4.7.5 STRESS ANALYSIS OF CONVERTER BOARD.



ACTUAL BOARD

ASSUMED BOARD

ASSUME THAT THE WEIGHT OF THE LOADED BOARD (LESS THE WEIGHT OF TRANSFORMER T2 WHICH MOUNTS DIRECTLY TO THE CONVERTER HOUSING AT POINT E) IS 0.25 LBS. IT IS ALSO ASSUMED THAT THE ACCELERATION G_Y = 64.5 g IS THE INPUT TO THE BOARD WHICH EXHIBITS A TRANSMISSIBILITY Q = 5.

$$W_{DYN} = 1.5[(0.25 \text{ LBS})(64.5g)(Q=5)] = 120.94 \text{ LBS. (ULT.)}$$

ASSUMED BOARD AREA, $A = (3.75)^2 = 14.06 \text{ IN}^2$

PRESSURE LOAD $q = (W_{DYN}/A) = 8.6$ PSI (ULT.)

MAX. BENDING MOMENT, $M = 0.1527\, qa^2$ (REF. g)

$$M = 0.1527(8.6)(3.50)^2$$

$$M = 16.1 \text{ IN-LBS/IN. (ULT.)}$$

BENDING STRESS, $f_b = \frac{6M}{t^2} = \frac{6(16.1)}{(0.062)^2} = \underline{25130 \text{ PSI (ULT.)}}$

ALLOWABLE FLEXURAL STRENGTH, $F_{FL} = 40000$ PSI (REF. h)

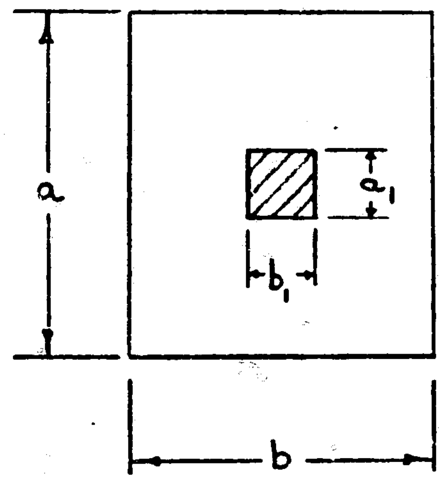
$$M.S. = \frac{F_{FL}}{f_b} - 1 = \frac{40000}{25130} - 1 = \underline{+0.59}$$

4.7.6 STRESS IN CONVERTER HOUSING DUE TO TRANSFORMER T2.

ASSUME LOAD DUE TO TRANSFORMER T2 (ESTIMATED WEIGHT = 1/3 LB.) IS

$$W = 1.5(1/3 \text{ LB.})(64.5g) = 32.3 \text{ LBS. (ULT.)}$$

ASSUME THE CONVERTER HOUSING TO BE A RECTANGULAR FLAT PLATE WITH SUPPORTED EDGES UNDER A LOAD DISTRIBUTED OVER A CENTRAL RECTANGULAR AREA. (REF. K - TABLE X - CASE 38)



MAX. BENDING STRESS

$$f_b = \beta \frac{W}{t^2}$$

FOR $a = b \approx 5.0$ IN.

$a_1 = b_1 \approx 0.3$ IN.

$(a_1/b) = (b_1/b) = 0.06$

AND $t = 0.050$ IN.

BY INTERPOLATION, $\beta \approx 2.5$

$$f_b = \frac{2.5(32.3)}{(0.050)^2} = 32300 \text{ PSI (ULT.)}$$

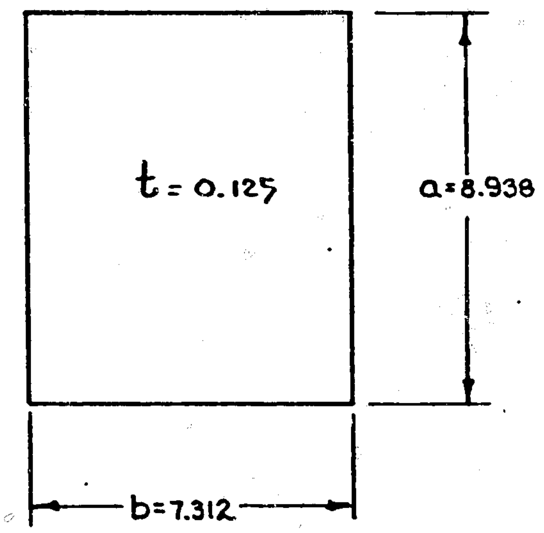
FOR 2024-T351 AL. AL. PLATE

$F_{TU} = 64000$ PSI (REF. C)

$$M.S. = \frac{F_{TU}}{f_b} - 1 = \frac{64000}{32300} - 1 = \underline{+0.98}$$

5.0 ANALYSIS OF FRONT PLATE.
REF. DWG. 7232-0201

5.1 BENDING STRESS.



INITIALLY, THE FORWARD OR FRONT PLATE WILL BE ANALYZED AS A SOLID RECTANGULAR FLAT PLATE OF CONSTANT THICKNESS WITH ALL EDGES SUPPORTED. THE LOAD IS ASSUMED TO BE UNIFORM OVER THE ENTIRE SURFACE. THE LOADS DUE TO THE WEIGHTS OF THE MOTOR AND GRATING BOX ASSEMBLY PLUS MOST OF THE BAFFLE ASSEMBLY AND DETECTOR ELECTRONICS WEIGHT ARE ACTUALLY APPLIED ON OR NEAR TO THE SUPPORTED EDGES OF THE PLATE. THEREFORE, CONSIDERING THE LOAD TO DISTRIBUTED OVER THE WHOLE SURFACE IS A CONSERVATIVE APPROACH.

W = LOAD ON FRONT PLATE = P_1 = 18.08 LBS. (REF. PG. 127)

THE PLATE IS CRITICAL FOR Z-AXIS LOADING

$G_Z = 31.4g$ (REF. PG. 41)

$\therefore W_{DYN.} = 1.50(18.08)(31.4) = 851.6$ LBS. (ULT.)

LET A = EFFECTIVE PLATE AREA

$A = ab = (8.938)(7.312) = 65.35$ IN.^{2}

$$\omega_{DYN} = \frac{W_{DYN}}{A} = \frac{851.6}{65.35} = 13.03 \text{ LB/IN}^2$$

THE MAXIMUM BENDING STRESS IS GIVEN BY (REF. K) AS

$$f_b = \frac{\beta \omega_{DYN} b^2}{t^2}$$

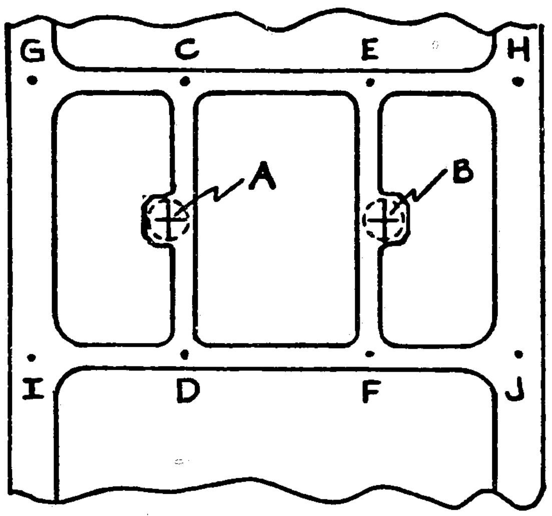
FOR $(a/b) = 1.22$, (REF. K) GIVES $\beta = 0.3844$

$$\therefore f_b = \frac{0.3844(13.03)(7.312)^2}{(0.125)^2} = \underline{17140 \text{ psi (ULT.)}}$$

FOR 6061-T6 ALUMINUM ALLOY, F_{TU} = 42000 PSI (REF. C)

$$M.S. = \frac{F_{TU}}{f_b} - 1 = \frac{42000}{17140} - 1 = \underline{\text{LARGE}}$$

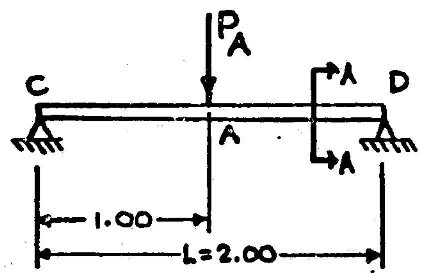
5.2 LOCAL BENDING STRESSES. DUE TO BAFFLE ASSEMBLY.

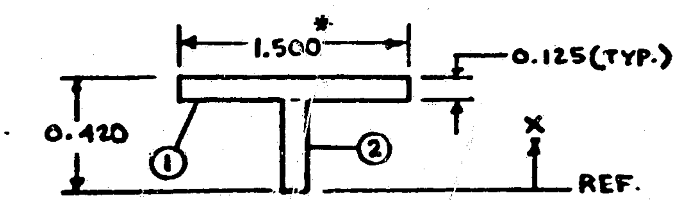


SCHEMATIC REPRESENTATION OF TWO LOAD POINTS ON FRONT PLATE DUE TO THE ATTACHMENT OF THE BAFFLE ASSEMBLY.

CONSERVATIVELY ASSUME THE LOADS AT POINTS A AND B (DUE TO THE ATTACHMENT OF THE BAFFLE ASS'Y.) LOAD THE SIMPLY SUPPORTED BEAMS CD AND EF RESPECTIVELY. THE REACTIONS AT C & E IN TURN LOAD THE SIMPLY SUPPORTED BEAM GH. (SIMILARLY, THE REACTIONS AT D & F IN TURN LOAD THE SIMPLY SUPPORTED BEAM IJ.)

BEAM CD (AND BEAM EF)





SECT. A-A

* ASSUMED EFFECTIVE

FROM PG. 60, $P_A = P_B = 236.6$ LBS. (ULT.)

$$M_{MAX} = \frac{P_A L}{4} = \frac{236.6(2.00)}{4}$$

$M_{MAX.} = 118.3$ IN-LBS. (ULT.)

SECTION PROPERTIES.

ITEM	AREA, A	x	AX	AX^2	I_o
1	0.1875	0.3575	0.06703	0.02396	0.00024
2	0.0369	0.1475	0.00544	0.00080	0.00027
Σ	0.2244		0.07247	0.02476	0.00051

$\bar{X} = C = (\Sigma AX)/(\Sigma A) = 0.323$ IN.

$I = \Sigma I_o + \Sigma AX^2 - \bar{X}\cdot(\Sigma AX) = 0.00187$ IN.^{4}

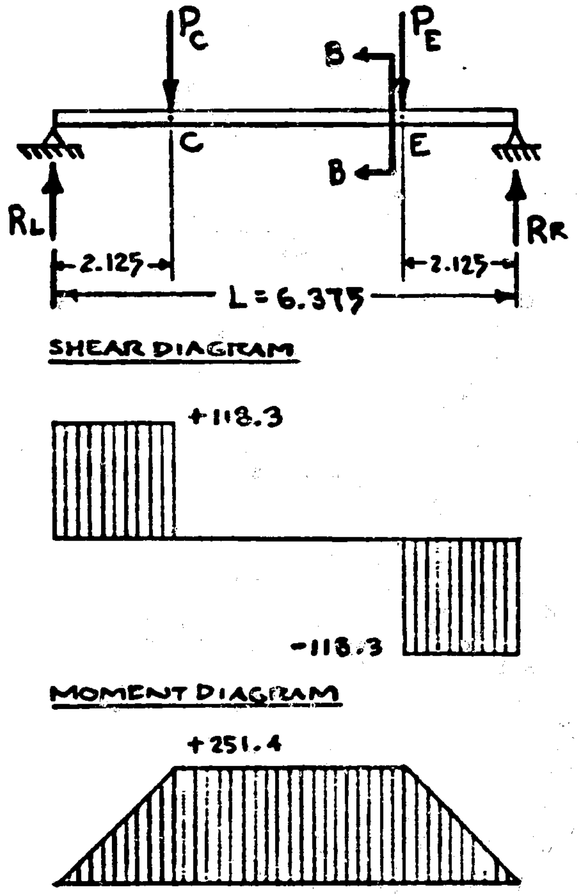
$Z = (I/C) = 0.0058$ IN.^{3}

$$f_b = \frac{M_{MAX}}{Z} = \frac{118.3}{0.0058} = \underline{20400 \text{ psi (ULT.)}}$$

FOR 6061-T6 AL. ALLOY, $F_{TU} = 42000$ psi (REF. C)

$$M.S. = \frac{F_{TU}}{f_b} - 1 = \frac{42000}{20400} - 1 = \underline{LARGE}$$

BEAM GH (AND BEAM IJ)

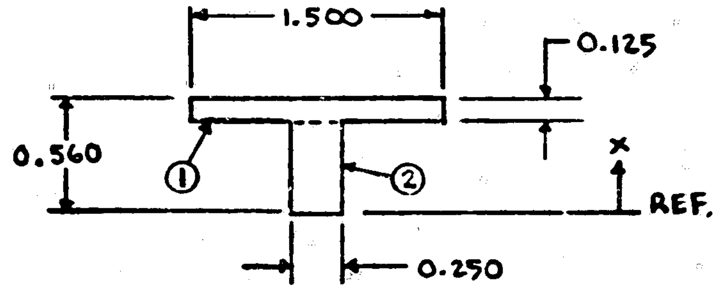


$P_C = P_E = (236.6)/2 = 118.3$ LBS.(ULT.)

$M_{MAX} = 2.125 R_L$

$R_L = R_R = 118.3$ LBS.(ULT.)

$M_{MAX} = 2.125(118.3) = 251.4$ IN-LBS.(ULT.)



SECTION B-B

SECTION PROPERTIES

ITEM	DESCRIPTION	AREA, A	X	AX	AX^2	I_o
1	1.500 x 0.125	0.1875	0.4975	0.09328	0.04641	0.00024
2	0.250 x 0.435	0.1088	0.2175	0.02366	0.00515	0.00171
Σ		0.2963		0.11694	0.05156	0.00195

$\bar{X} = C = (\Sigma AX)/(\Sigma A) = 0.395$ IN.

$I = \Sigma I_o + \Sigma AX^2 - \bar{X} \cdot \Sigma AX$

$I = 0.00195 + 0.05156 - 0.04614 = 0.00737$ IN^4

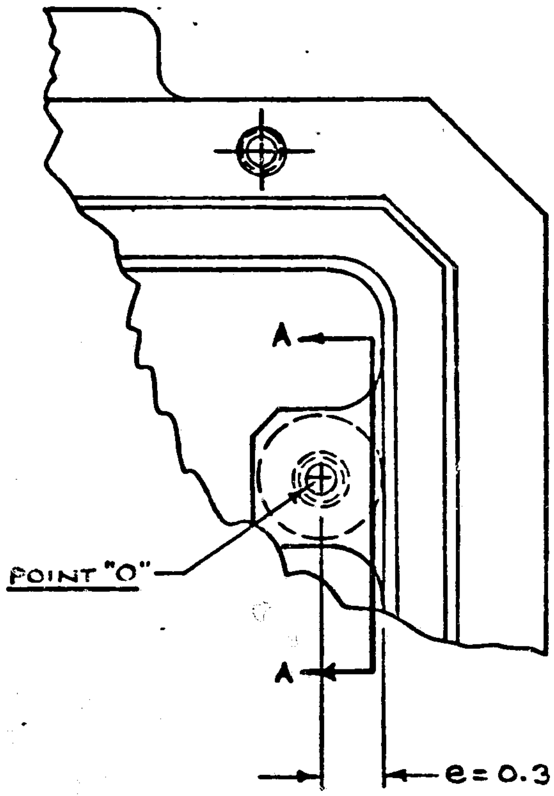
$Z = (I/C) = 0.0187$ IN^3

$$f_b = \frac{M_{MAX}}{Z} = \frac{251.4}{0.0187} = \underline{13450 \text{ PSI (ULT.)}}$$

FOR 6061-T6 AL. ALLOY, $F_{TU} = 42000$ PSI (REF. C)

$$M.S. = \frac{F_{TU}}{f_b} - 1 = \frac{42000}{13450} - 1 = \underline{\text{LARGE}}$$

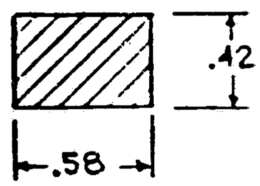
LOCAL LUG BENDING.



CONSERVATIVELY ASSUME LUG TO BE CANTILEVER BEAM WITH LOAD APPLIED AT PT. "O". THE MAGNITUDE OF THIS LOAD RESULTING FROM THE ATTACHMENT OF THE BAFFLE ASSEMBLY TO THE FRONT PLATE WAS FOUND TO BE

$$P_o = 220.0 \text{ LBS. (ULT.)} \quad \text{(REF. PG. 59)}$$

CONSERVATIVELY NEGLECTING THE FILLET RADIUS AT SECTION A-A, THE CROSS-SECTION IS TAKEN AS SHOWN BELOW:



SECT. A-A

BENDING MOMENT, $M = P_o e = 220.0(0.3)$

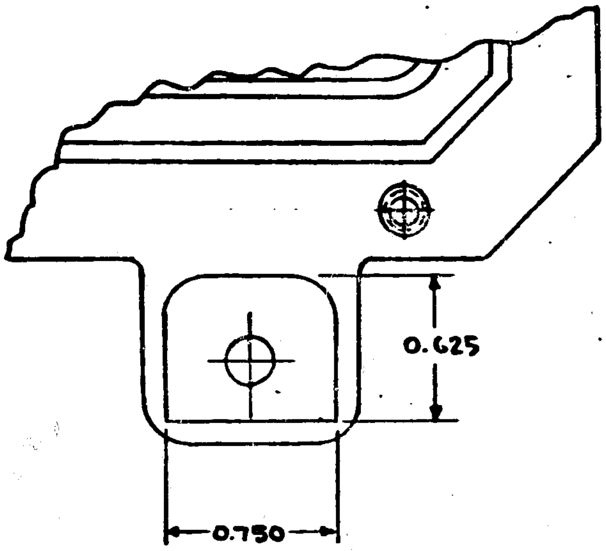
$$M = 66.0 \text{ IN-LBS. (ULT.)}$$

BENDING STRESS, $f_b = \dfrac{6M}{bt^2} = \dfrac{6(66.0)}{0.58(0.42)^2}$

$$f_b \approx \underline{4000 \text{ PSI (ULT.)}}$$

∴ <u>M.S. LARGE BY INSPECTION</u>

5.3 LOCAL BENDING STRESSES.
DUE TO DETECTOR ELECTRONICS.



THE LOWER LUG ON THE FRONT PLATE WILL BE ANALYZED IN MUCH THE SAME WAY AS ITS CORRESPONDING LUG OR "BATHTUB-TYPE" FITTING ON THE DETECTOR-ELECTRONICS MOUNTING PLATE (REF. PG. 86)

FROM REF. K, THE MAX. BENDING STRESS AT THE CENTER OF THE FIXED EDGE IS

$$f_b' = \frac{3\omega b^2}{t^2(1+3.2\alpha^3)}$$

FROM PG. 81 $P = 227.2$ LBS. (ULT.)

AREA $A = ab = (0.750)(0.625) = 0.47 \text{ IN}^2$

$\omega = (P/A) = (227.2)/(0.47) = 483.4 \text{ LB/IN}^2$

FOR $\alpha = (b/a) = (0.625)/(0.750) = 0.833$

$\alpha^3 = 0.58$

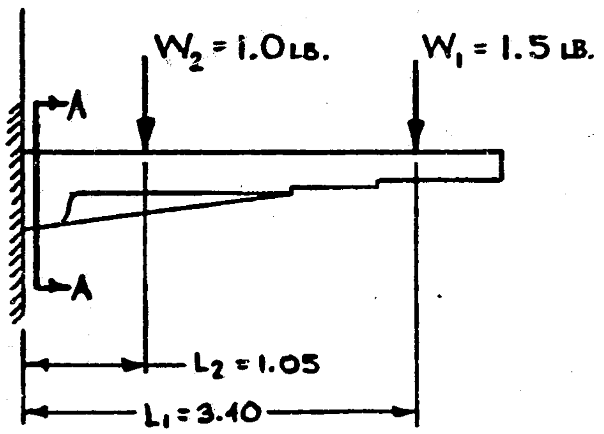
AGAIN ASSUMING $f_b = 1.5 f_b'$

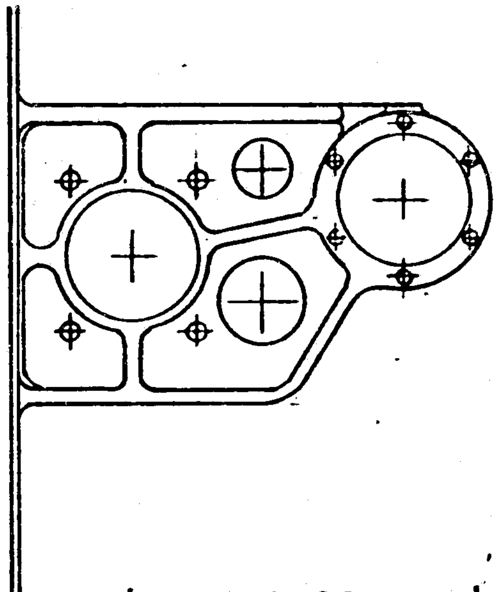
$$f_b = 1.5\left\{\frac{3(483.4)(0.625)^2}{(0.125)^2(1+3.2[0.58])}\right\} \approx \underline{19130 \text{ PSI (ULT.)}}$$

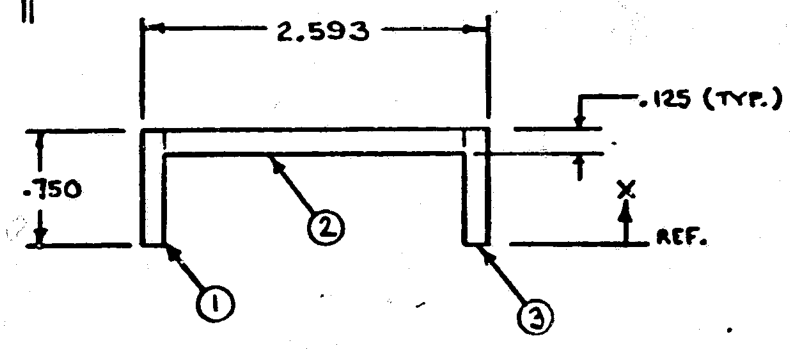
FROM REF. C FOR 6061-T6 AL. ALLOY, $F_{TU} = 42000$ PSI

$$M.S. = \frac{F_{TU}}{f_b} - 1 = \frac{42000}{19130} - 1 = \underline{\text{LARGE}}$$

5.4 LOCAL BENDING STRESSES. DUE TO MOTOR & GRATING.







SECTION A-A

SHOWN BELOW IS ONE OF TWO FLANGES THAT SUPPORTS THE GRATING BOX ASSEMBLY. THE PARTICULAR FLANGE ALSO SUPPORTS THE MOTOR HOUSING. IT IS ASSUMED THAT THE ENTIRE WEIGHT OF THE GRATING BOX ASS'Y. IS CARRIED BY THE ONE FLANGE ONLY.

W_1 = WEIGHT OF GRATING ASS'Y. = 1.5 LB.
W_2 = WEIGHT OF MOTOR & HSG. = 1.0 LB. } REF. PG. 28

THE CRITICAL ACCELERATIONS FOR THIS X-AXIS LOADING ARE

Gx_{MOTOR} = 41.8 g
$Gx_{GRATING}$ = 39.0 g } REF. PG. 41

THE DYNAMIC LOADS ARE

$(W_1)_{DYN} = 1.5(1.5\text{ LB.})(39.0\,g) = 87.75$ LB. (ULT.)

$(W_2)_{DYN} = 1.5(1.0\text{ LB.})(41.8\,g) = 62.70$ LB. (ULT.)

THE BENDING MOMENT AT SECT. A-A IS

$$M = \{(W_1)_{DYN} L_1 + (W_2)_{DYN} L_2\}$$

$$M = \{87.75(3.40) + 62.70(1.05)\}$$

$$M = 364.2 \text{ IN-LBS. (ULT.)}$$

SECTION PROPERTIES

ITEM	DESCRIPTION	AREA, A	X	AX	AX²	Io
1	0.125 × 0.750	0.093750	0.3750	0.035156	0.013184	0.004395
2	2.343 × 0.125	0.292875	0.6875	0.201352	0.138430	0.000381
3	0.125 × 0.750	0.093750	0.3750	0.035156	0.013184	0.004395
	Σ	0.480375		0.271664	0.164798	0.009171

$\bar{X} = C = (\Sigma AX)/(\Sigma A) = 0.5655$ IN.

$I = \Sigma I_o + \Sigma AX^2 - \bar{X} \cdot \Sigma AX = 0.0203$ IN.^{4}

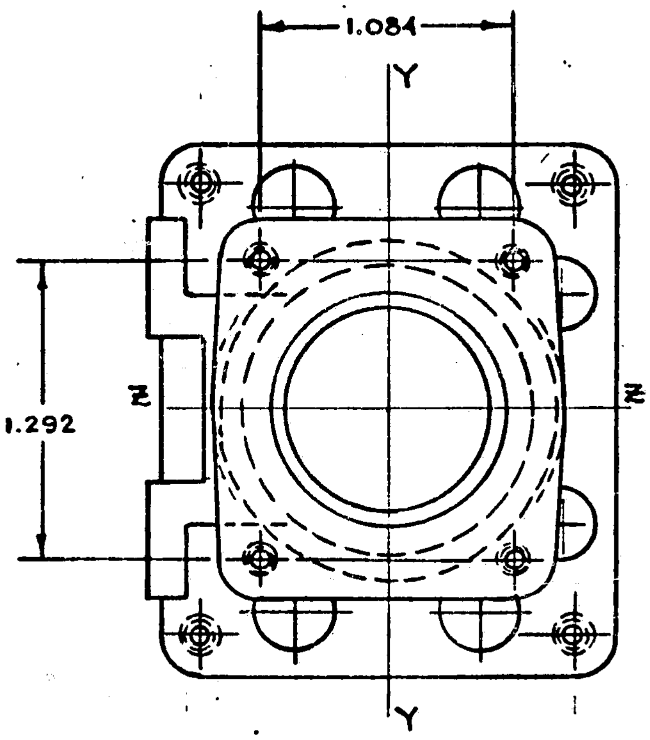
$Z = (I/C) = 0.036$ IN.^{3}

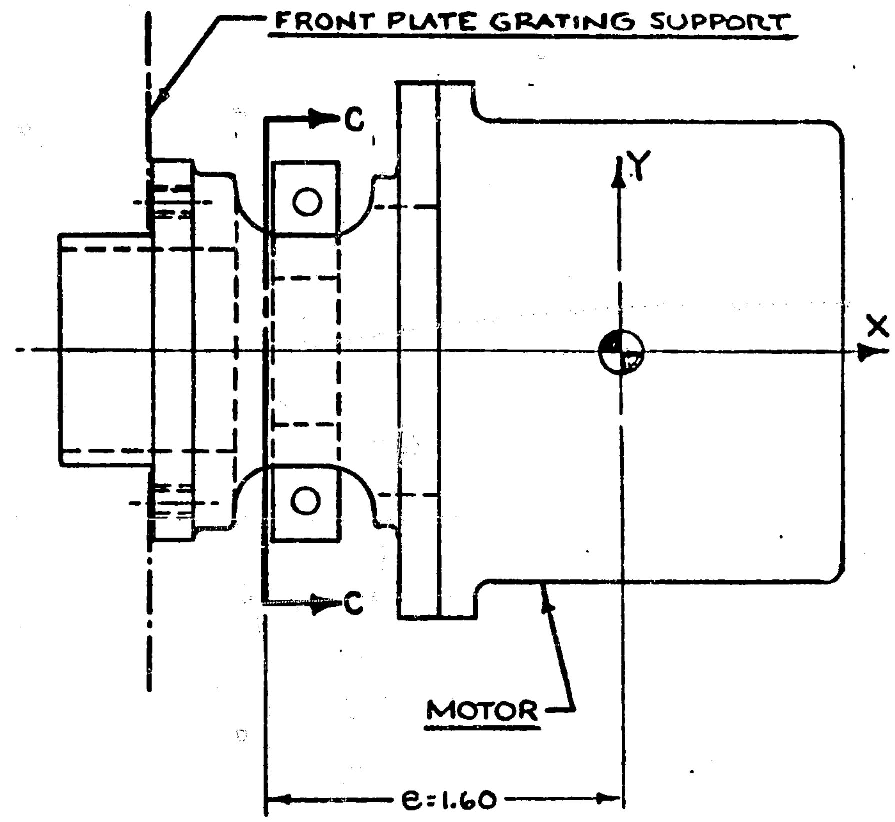
$$\therefore f_b = \frac{M}{Z} = \frac{364.2}{0.036} = 10125 \text{ psi (ULT.)}$$

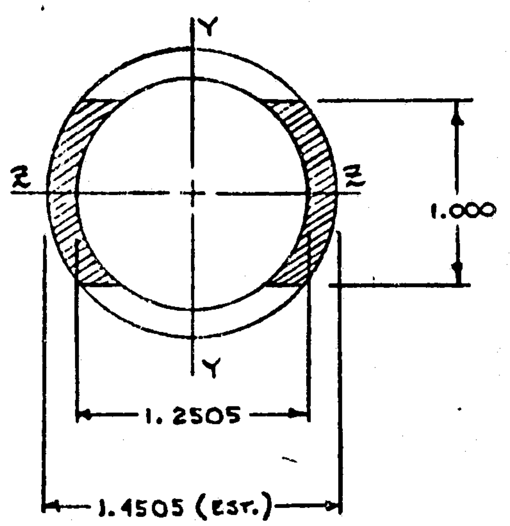
FOR 6061-T6 ALUMINUM ALLOY, F_{TU} = 42000 psi (REF. C)

$$M.S. = \frac{F_{TU}}{f_b} - 1 = \frac{42000}{10125} - 1 = \text{LARGE}$$

6.0 ANALYSIS OF MOTOR MOUNT.







ACTUAL SECTION C-C

FROM PAGE 41 THE CRITICAL ACCELERATIONS FOR THE MOTOR ARE:

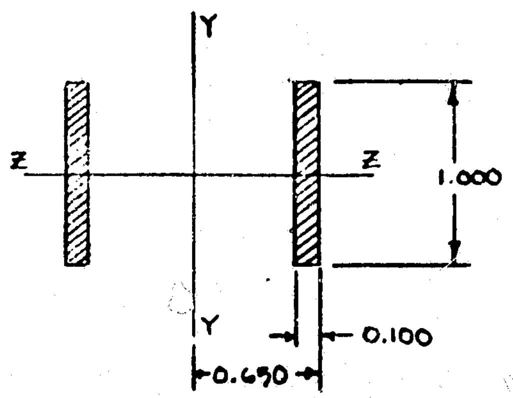
$G_X = 41.8\ g$
$G_Y = 46.2\ g$
$G_Z = 31.6\ g$

THE MOTOR WEIGHT IS $W = 0.91$ LBS. (REF. PG. 30)

$(W_{DYN})_X = 1.5(0.91)(41.8) = \underline{57.1}$ LBS.
$(W_{DYN})_Y = 1.5(0.91)(46.2) = \underline{63.1}$ LBS.
$(W_{DYN})_Z = 1.5(0.91)(31.6) = \underline{43.1}$ LBS.

ASSUME THE WEIGHT OF THE MOTOR PLUS THE HOUSING TO BE $W = 1.00$ LBS (MOUNTED TO THE FRONT PLATE GRATING SUPPORT)

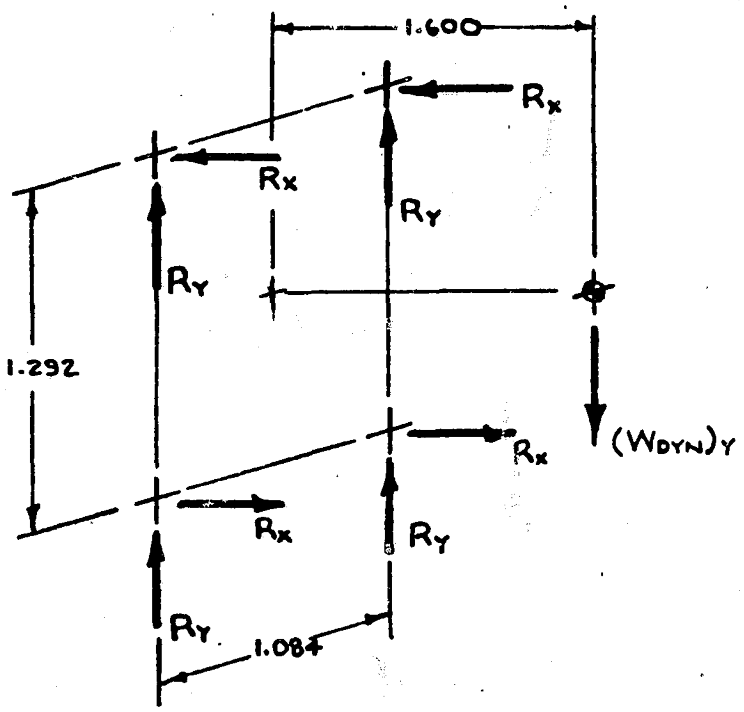
$(W_{DYN})_X = 1.5(1.00)(41.8) = \underline{62.7}$ LBS.
$(W_{DYN})_Y = 1.5(1.00)(46.2) = \underline{69.3}$ LBS.
$(W_{DYN})_Z = 1.5(1.00)(31.6) = \underline{47.4}$ LBS.



ASSUMED SECTION C-C

6.1 ATTACHMENT OF MOTOR PLUS MOUNT TO FRONT PLATE GRATING SUPPORT

MAXIMUM LOADING RESULTS FROM Y-AXIS INPUT.



$$\Sigma F_Y = 0 = (W_{DYN})_Y - 4R_Y$$
$$4R_Y = (W_{DYN})_Y = 69.3 \text{ LBS.}$$
$$R_Y = \underline{17.3 \text{ LBS. (ULT.)}}$$

$$\Sigma M = 0 = 1.600(W_{DYN})_Y - 2R_x(1.292)$$
$$2.584\, R_x = 1.600(69.3)$$
$$R_x = \underline{42.9 \text{ LBS. (ULT.)}}$$

FASTENERS ARE #6-32 UNC CRES.
MIN. YIELD LOAD FOR SCREW PRELOADED TO 80% YIELD IS

$$(P_Y) = 0.8(270) = \underline{216 \text{ LBS.}}$$

ASSUMING A COEFFICIENT OF FRICTION $\mu = 0.2$, THE FRICTION CAPACITY PER ATTACH. POINT IS

$$F_F = 0.2(216) = \underline{43.2 \text{ LBS.}}$$

BASED ON YIELD,

$$M.S. = \frac{(P_Y)}{(1.15/1.50)R_x} - 1 = \underline{\text{LARGE}}$$

$$M.S. = \frac{F_F}{(1.15/1.50)R_Y} - 1 = \underline{\text{LARGE}}$$

6.1.1 X-AXIS EXCITATION.

A = CROSS-SECTION AREA = $2(1.000)(0.10) = 0.20 \text{ in.}^2$
P = AXIAL LOAD = $(W_{DYN})_X$ = 57.1 LBS.
TAKING THE LOAD TO BE COMPRESSIVE

$$f_c = \frac{P}{A} = \frac{57.1}{0.200} \approx \underline{285 \text{ psi (ULT.)}}$$

FOR 6061-T6 AL. ALLOY, F_{CY} = 35,000 psi (REF. C)

M.S. LARGE BY INSPECTION.

6.1.2 Y-AXIS EXCITATION.

M_Y = BENDING MOMENT = $(W_{DYN})_Y \cdot e$
$M_Y = 63.1(1.60) = \underline{101 \text{ IN-LBS. (ULT.)}}$

$$I_{ZZ} = 2\left\{\frac{1}{12}(0.10)(1.00)^3\right\} = 0.0167 \text{ in.}^4$$

C_Z = 0.50 IN.
$(Z)_Z = (I_{ZZ}/C_Z) = 0.0333 \text{ in.}^3$

$$f_b = \frac{M_Y}{(Z)_Z} = \frac{101}{0.0333} \approx \underline{3030 \text{ psi (ULT.)}}$$

FOR 6061-T6 AL. ALLOY, F_{TU} = 42,000 psi (REF. C)

M.S. LARGE BY INSPECTION.

6.1.3 Z-AXIS EXCITATION.

M_Z = BENDING MOMENT = $(W_{DYN})_Z \cdot e$
$M_Z = 43.1(1.60) = \underline{69 \text{ IN-LBS. (ULT.)}}$

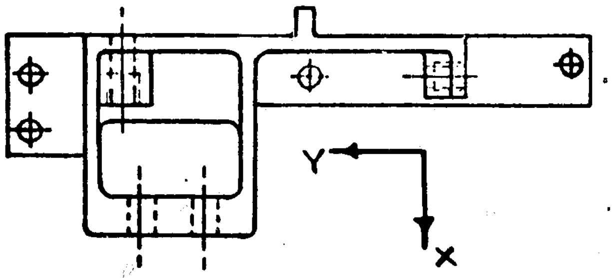
$$I_{YY} \approx 2\left\{(0.10)(1.00)(0.60)^2\right\} = 0.0720 \text{ in.}^4$$

C_Y = 0.65 IN.
$(Z)_Y = (I_{YY}/C_Y) = 0.111 \text{ in.}^3$

$$f_b = \frac{M_Z}{(Z_Y)} = \frac{69}{0.111} \approx \underline{620 \text{ psi (ULT.)}}$$

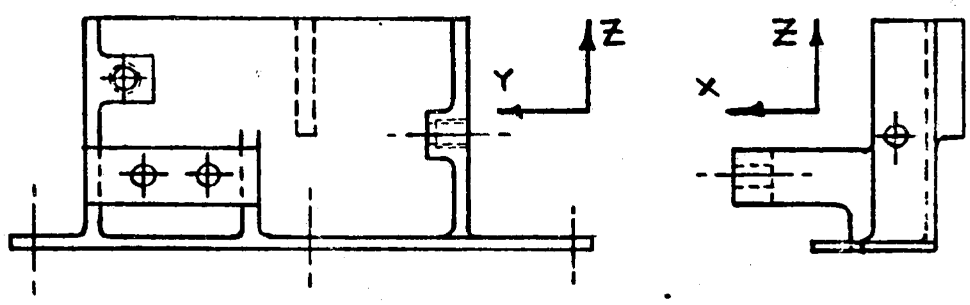
M.S. LARGE BY INSPECTION.

7.0 ANALYSIS OF FIDUCIAL MARK DETECTOR BRACKET.

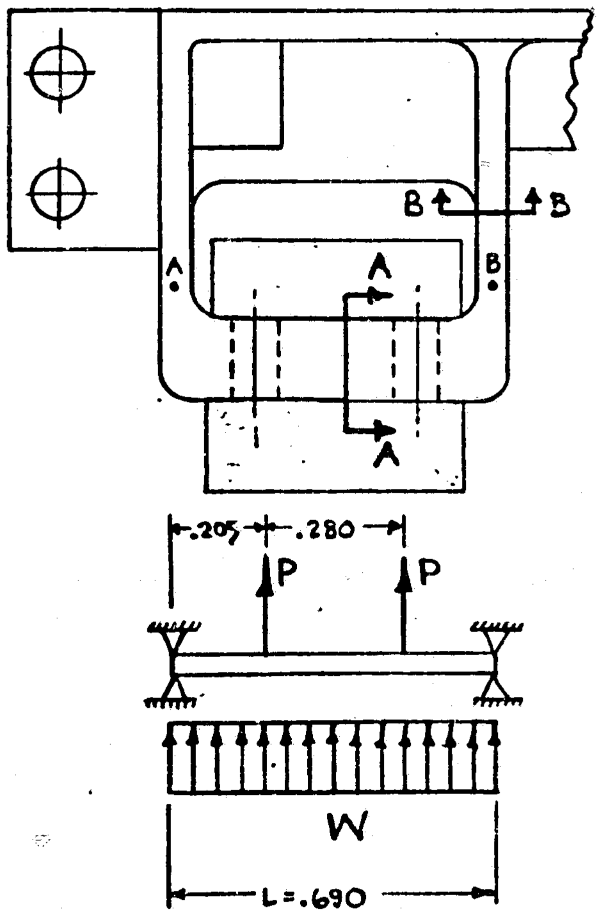


ASSUME TOTAL WEIGHT OF EQUIPMENT TO BE SUPPORTED IS 0.20 LBS. AND TOTAL WEIGHT INCLUDING BRACKET IS 0.25 LBS. ACCELERATIONS WILL BE TAKEN AS THOSE OF FORWARD PLATE.

$$G_X = 29.5\,g,\quad G_Y = 37.0\,g,\quad G_Z = 31.4\,g \quad \text{REF. PG. 41}$$



7.1 X-AXIS LOADING.



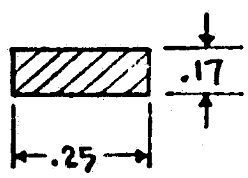
FOR UPWARD X-AXIS LOADING AS SHOWN AT THE LEFT, ASSUME HALF THE EQUIPMENT WEIGHT (W= 0.10 LBS) APPLIED UNIFORMLY ON ONE SIDE OF "BEAM A-B" AND THE OTHER HALF OF THE EQUIPMENT WEIGHT (P= 0.05 LBS.) APPLIED AT TWO DISCRETE POINTS.

7.1.1 SECTION A-A BENDING STRESS

$$\Sigma F = 0 = W + 2P - 2R$$
$$2R = W + 2P = 0.20$$
$$R = 0.10 \text{ LBS}/g$$

M = MAX. MOMENT @ L/2

$$M = R(L/2) - (W/2)(L/4) - P\{(L/2) - 0.205\}$$
$$M = 0.10(0.345) - 0.05(0.173) - 0.05(0.140)$$
$$M = 0.019 \text{ IN-LBS}/g$$
$$M_{DYN} = 1.5(0.019)(29.5) = \underline{0.84 \text{ IN-LBS (ULT.)}}$$



SECT. A-A

$$f_b = \frac{6M_{DYN}}{bt^2} = \frac{6(0.84)}{(.25)(.17)^2}$$
$$f_b = 700 \text{ PSI (ULT.)}$$

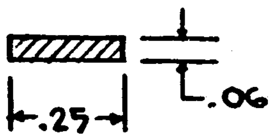
MAT'L. 2024-T351 ALUM. AL.

∴ M.S. LARGE BY INSPECTION

7.1.2 SECTION B-B COMPRESSIVE STRESS.

P_c = COMPRESSIVE FORCE = R = 0.10 LBS/g

$(P_c)_{DYN} = 1.5(0.10)(29.5) = 4.43$ LBS. (ULT.)



SECT. B-B

$A = (.25)(.06) = .015\ in^2$

$f_c = \frac{(P_c)_{DYN}}{A} = \frac{4.43}{.015} \approx$ 100 PSI (ULT.)

M.S. LARGE BY INSPECTION

7.1.3 ATTACHMENT TO FRONT PLATE.

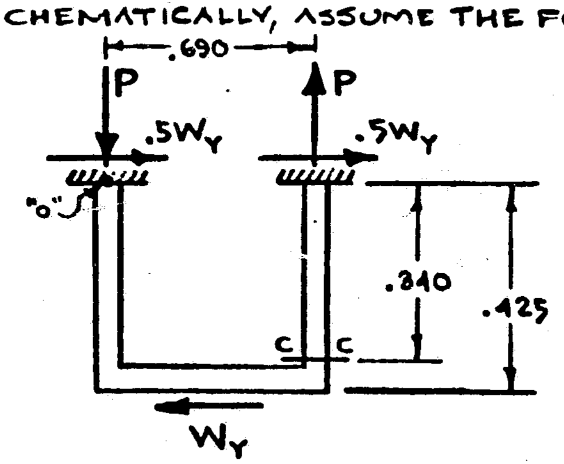
THE BRACKET IS ATTACHED TO THE FRONT PLATE BY 4 SCREWS ASSUMED TO BE #2-56 UNC CRES FOR WHICH $(P_Y)_{MIN.} = 110$ LBS. ASSUMING SCREWS TORQUED TO 80% YIELD, TOTAL ALLOWABLE TENSILE LOAD WOULD BE $4(0.8)(110) = 352$ LBS. AND JOINT FRICTION CAPACITY IS $(\mu = 0.2)(352) = 70.4$ LBS.. THUS, FOR ALL DIRECTIONS OF LOADING

∴ M.S. LARGE BY INSPECTION

7.2 Y-AXIS LOADING.

7.2.1 SECTION C-C BENDING STRESS

SCHEMATICALLY, ASSUME THE FOLLOWING FRAME



$(W_Y)_{DYN} = 1.5(0.20)(37.0g)$

$(W_Y)_{DYN} = 11.10$ LBS. (ULT.)

$\Sigma M_O = 0 = 0.425 W_Y - 0.690 P$

$0.690P = 0.425(11.10)$

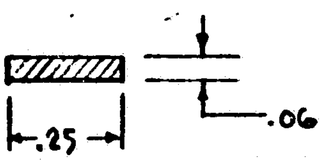
$P = 6.84$ LBS. (ULT.)

BENDING MOMENT ON SECTION C-C

$M_{C\text{-}C} = 0.340(0.50 W_Y)$

$M_{C\text{-}C} = 1.89$ IN-LBS. (ULT.)

ASSUME COMBINED COMPRESSIVE STRESS DUE TO BENDING AND AXIAL LOADING



SECT. C-C

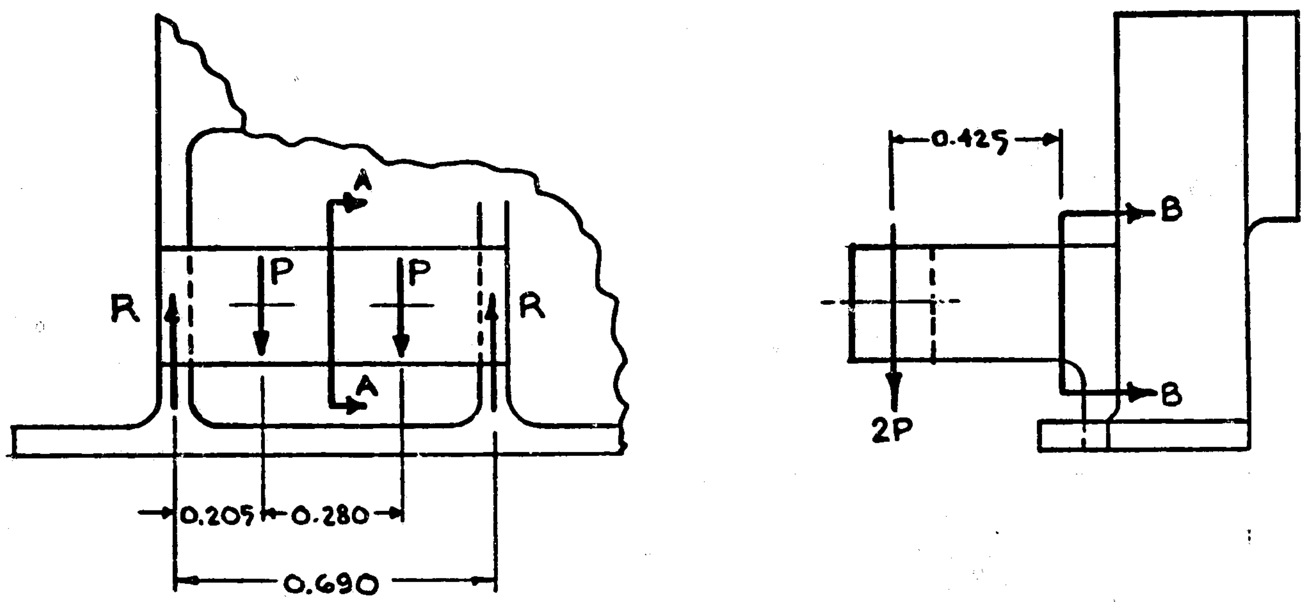
$$f_c = \frac{P}{A} + \frac{M_{C\text{-}C}\,C}{I} = \frac{P}{bt} + \frac{6M_{C\text{-}C}}{bt^2}$$

$$f_c = \frac{6.84}{(.25)(.06)} + \frac{6(1.89)}{(.25)(.06)^2} \approx 13000 \text{ PSI (ULT.)}$$

FOR 2024-T351 AL.AL, $F_{CY_{MIN}} \approx 37000$ PSI (REF. C)

∴ M.S. LARGE BY INSPECTION.

7.3 Z-AXIS LOADING.



7.3.1 SECTION A-A BENDING STRESS.

$$(W_Z)_{DYN} = 2P = 1.50(0.20)(31.4g) = \underline{9.42 \text{ LBS. (ULT.)}}$$

BENDING MOMENT ON SECTION A-A = $M_{A\text{-}A} = 0.205P = 0.205(9.42/2)$

$$M_{A\text{-}A} = 0.97 \text{ IN-LBS. (ULT.)}$$

SECTION A-A SHOWN ON PG. 106

$$\therefore f_b = \frac{6M_{A\text{-}A}}{tb^2} = \frac{6(0.97)}{0.17(0.25)^2} \approx \underline{550 \text{ psi (ULT.)}}$$

∴ M.S. LARGE BY INSPECTION.

7.3.2 SECTION B-B BENDING STRESS.

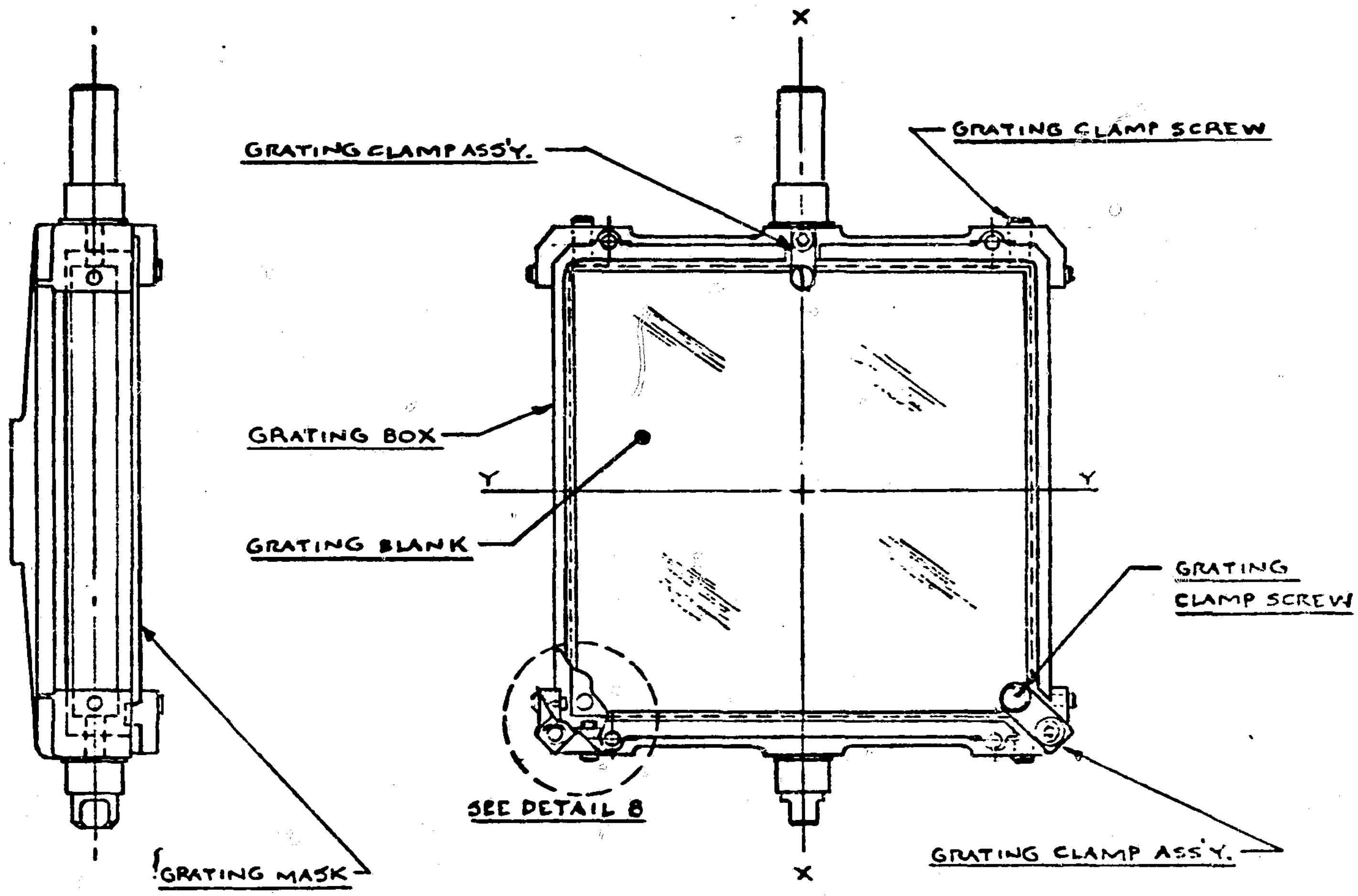
$$M_{B\text{-}B} = 0.425P = 0.425(9.42/2) = 2.00 \text{ IN-LBS (ULT.)}$$

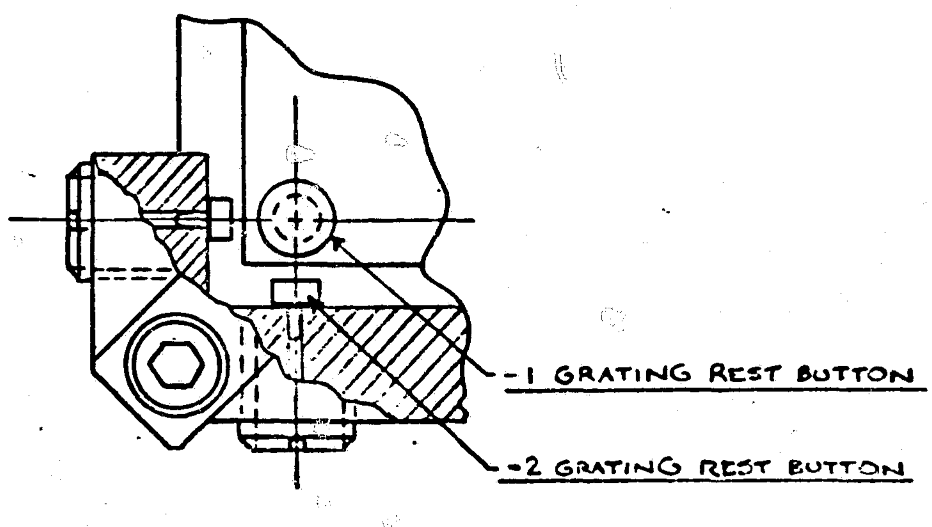
SECTION B-B SHOWN ON PG. 107

$$f_b = \frac{6M_{B\text{-}B}}{tb^2} = \frac{6(2.00)}{(.06)(.25)^2} = \underline{3200 \text{ psi (ULT.)}}$$

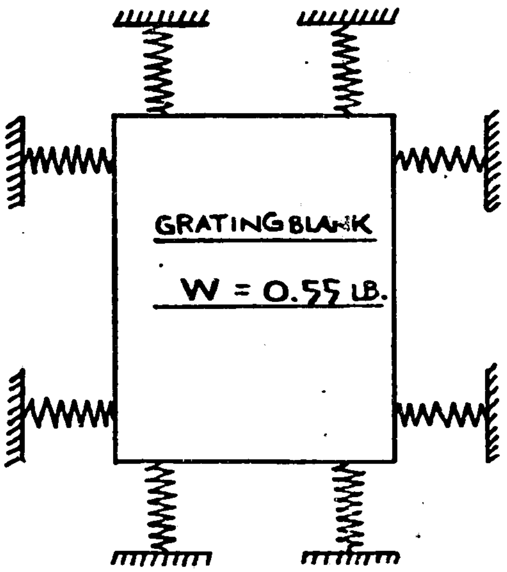
∴ M.S. LARGE BY INSPECTION.

8.0 ANALYSIS OF GRATING BOX ASSEMBLY.





DETAIL B



THE GRATING BLANK ACCELERATIONS ARE:

$G_X = 39.0\ g$
$G_Y = 54.3\ g$ } REF. PG. 41
$G_Z = 87.5\ g$

THE DYNAMIC LOADS ARE

$(W_{DYN})_x = 1.5(0.55)(39.0) = \underline{32.2\ \text{LBS (ULT)}}$

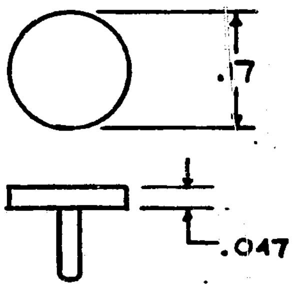
$(W_{DYN})_y = 1.5(0.55)(54.3) = \underline{44.8\ \text{LBS (ULT)}}$

$(W_{DYN})_z = 1.5(0.55)(87.5) = \underline{72.2\ \text{LBS (ULT)}}$

8.1 "SPRING" CONSTANT DETERMINATION.

GRATING REST BUTTONS ARE KEL-F-81 PLASTIC FOR WHICH E = 180,000 PSI (REF. 2)

① BUTTON



$A = \frac{\pi}{4} d^2 = 0.7854(0.17)^2$

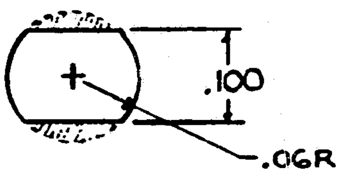
$A = 0.0227\ \text{in}^2$

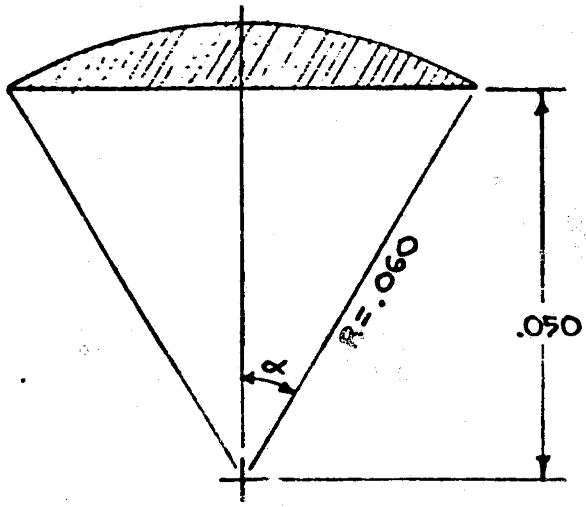
$L = 0.047\ \text{IN.}$

$E = 180{,}000\ \text{PSI}$

$K_1 = \frac{AE}{L} = \underline{8.69 \times 10^4\ \text{LBS/IN.}}$

② BUTTON





$A_{SHADED} = \frac{1}{2} R^2 (2\alpha - \sin 2\alpha)$

$\cos \alpha = (.050/.060) = 0.83333$

$\sin 2\alpha = 0.92132$

$\alpha = 1.17146\ \text{RAD.}$

$A = \pi R^2 - 2\left\{\frac{1}{2} R^2 (2\alpha - \sin 2\alpha)\right\}$

$A = 0.0104\ \text{in}^2$

$L = 0.047\ \text{IN}$

$E = 180{,}000\ \text{PSI}$

$K_2 = \frac{AE}{L} = \underline{3.98 \times 10^4\ \text{LBS/IN.}}$

8.2 DYNAMIC DISPLACEMENT OF GRATING BLANK

$$F = ma = (W_{DYN.}) = KX$$

OR $X = (W_{DYN})/K$

$$X_X = \frac{(W_{DYN})_X}{Ke_X} = \frac{32.2}{4(3.98 \times 10^4)} = 2.023 \times 10^{-4} \text{ IN.}$$

$$X_Y = \frac{(W_{DYN})_Y}{Ke_Y} = \frac{44.8}{4(3.98 \times 10^4)} = 2.814 \times 10^{-4} \text{ IN.}$$

ASSUME PRELOADING 25% ABOVE EXPECTED DYNAMIC DISPLACEMENT.

$$(X_X)_P = 1.25(2.023 \times 10^{-4}) = 2.53 \times 10^{-4} \text{ IN.}$$
$$(X_Y)_P = 1.25(2.814 \times 10^{-4}) = 3.52 \times 10^{-4} \text{ IN.}$$

THE CORRESPONDING PRELOAD PER "SPRING" IS

$$(P_X)_P = (3.98 \times 10^4)(2.53 \times 10^{-4}) = \underline{10.07 \text{ LBS.}}$$
$$(P_Y)_P = (3.98 \times 10^4)(3.52 \times 10^{-4}) = \underline{14.01 \text{ LBS.}}$$

8.2.1 FOR X-AXIS LOADING

THE LOAD PER SPRING IS

$$(P_X)_D = \left[(W_{DYN})_X / 4\right] = \underline{8.05 \text{ LBS.}}$$

THE TOTAL LOAD ON TWO OF THE SPRINGS IS

$$\Sigma P_{X_I} = (P_X)_P + (P_X)_D = 10.07 + 8.05 = \underline{18.12 \text{ LBS.}}$$

8.2.2 FOR Y-AXIS LOADING

THE LOAD PER SPRING IS

$$(P_Y)_D = \left[(W_{DYN})_Y / 4\right] = \underline{11.20 \text{ LBS.}}$$

THE TOTAL LOAD ON TWO OF THE SPRINGS IS

$$\Sigma P_{Y_I} = (P_Y)_P + (P_Y)_D = 14.01 + 11.20 = \underline{25.21 \text{ LBS.}}$$

THE Y-AXIS LOADING IS MORE CRITICAL.

8.3 COMPRESSIVE STRESS IN PAD.

$A =$ AREA OF ② BUTTON $= 0.0104 \text{ IN}^2$.
$P = \Sigma P_{Y_I} = 25.21$ LBS.

$$f_c = \frac{P}{A} = \frac{25.21}{0.0104} \approx \underline{2420 \text{ PSI (ULT)}}$$

FOR KEL-F-81, $F_{CY} = 5440$ PSI (REF. 2)

$$M.S. = \frac{F_{CY}}{f_c} - 1 = \frac{5440}{2420} - 1 = \underline{\text{LARGE}}$$

8.4 CONTACT STRESS IN GRATING BLANK.

BASED ON THE SAME CONSERVATIVE ASSUMPTIONS MADE IN THE ANALYSIS OF THE MIRROR, THE COMPRESSIVE STRESS IN THE GRATING IS GIVEN BY

$$f_c \cong \frac{P}{1.2A} \qquad \text{(REF. PG. 156)}$$

$$f_c = \frac{25.21}{1.2(0.0104)} = \underline{2015 \text{ psi (ULT.)}}$$

THE CONSERVATIVELY ASSUMED COMPRESSIVE ALLOWABLE IS $F_c = 2500$ psi

$$\text{M.S.} = \frac{F_c}{f_c} - 1 = \frac{2500}{2015} - 1 = \underline{+0.24}$$

8.5 FOR Z-AXIS LOADING.

8.5.1 BENDING STRESS IN GRATING.

ASSUME THE GRATING TO BE A SQUARE RECTANGULAR PLATE SUPPORTED AT THE CORNERS ONLY.

LET SIDE $a = 4.13$ in.

$\therefore \quad a^2 = 17.06$ in.^{2}

$(W_{DYN})_z = 72.2$ lbs. (ULT.) (REF. PG. 110)

$$\omega = \frac{(W_{DYN})_z}{a^2} = \frac{72.2}{17.06} = \underline{4.2 \text{ lbs./in.}^2 \text{ (ULT.)}}$$

MAX. BEND. MOMENT, $M = 0.1527\,\omega a^2$

$$M = 0.1527(4.2)(17.06) = \underline{11.0 \text{ in-lbs. (ULT.)}}$$

BENDING STRESS

$$f_b = \frac{6M}{t^2} = \frac{6(11.0)}{(0.394)^2} \doteq \underline{425 \text{ psi (ULT.)}}$$

ALLOWABLE $F_T \doteq 1000$ psi (REF. PG. 158)

$$\text{M.S.} = \frac{F_T}{f_b} - 1 = \frac{1000}{425} - 1 = \underline{\text{LARGE}}$$

GRATING ACTUALLY SUPPORTED AT 3 POINTS BY ① BUTTONS. IT WILL BE ASSUMED THAT THERE IS NO PRELOADING IN THIS DIRECTION SINCE THE BUTTONS ARE USED PRIMARILY FOR POSITIONING THE GRATING. BECAUSE OF THIS, AN IMPACT LOADING CONDITION IS ASSUMED TO EXIST AND THE DYNAMIC LOAD IS CONSERVATIVELY MULTIPLIED BY A FACTOR OF TWO.

$$W = 2(W_{DYN})_Z = 2(72.2) = 144.4 \text{ LBS.}$$

8.5.2 COMPRESSIVE STRESS IN PAD.

P = LOAD PER PAD = (144.4/3) ≈ 48.1 LBS.
A = AREA OF ① BUTTON = 0.0227 IN.^{2}

$$f_c = \frac{P}{A} = \frac{48.1}{0.0227} = \underline{2120 \text{ PSI (ULT.)}}$$

FOR KEL-F-81, F_c = 5440 psi (REF. 2)

$$M.S. = \frac{F_c}{f_c} - 1 = \frac{5440}{2120} - 1 = \underline{\text{LARGE}}$$

8.5.3 COMPRESSIVE STRESS (LOCAL) IN GRATING.

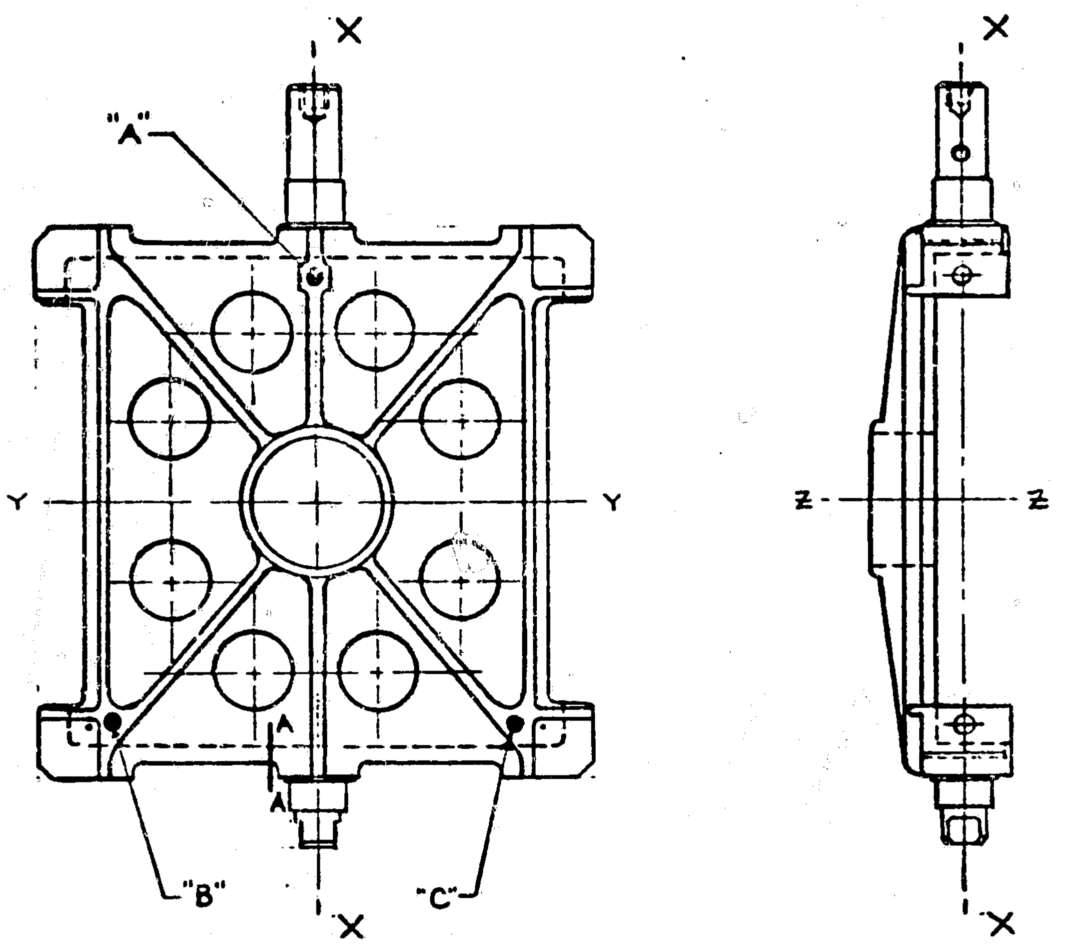
$$f_c = \frac{P}{1.2\pi R^2} \quad \text{(REF. PG. 156)}$$

$$f_c = \frac{48.1}{1.2\pi(0.085)^2} \approx \underline{1765 \text{ psi (ULT.)}}$$

F_c = 2500 psi (REF. PG. 156)

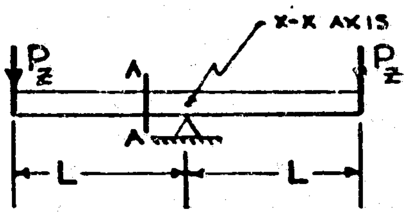
$$M.S. = \frac{F_c}{f_c} - 1 = \frac{2500}{1765} - 1 = \underline{+0.42}$$

8.6 ANALYSIS OF GRATING BOX.



8.6.1 Z-AXIS LOADING.

CONSERVATIVELY ASSUME THE FOLLOWING MODEL FOR Z-AXIS LOADING.



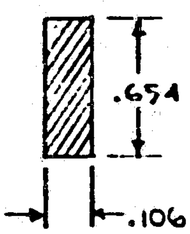
LET THE MAX. BENDING MOMENT ON SECTION A-A BE GIVEN BY

$$M = P_Z \cdot L = (48.1)(1.937)$$

$$M = 93.2 \text{ IN-LBS. (ULT.)}$$

BENDING STRESS

$$f_b = \frac{6M}{bt^2} = \frac{6(93.2)}{0.106(0.654)^2} = \underline{12320 \text{ psi (ULT.)}}$$



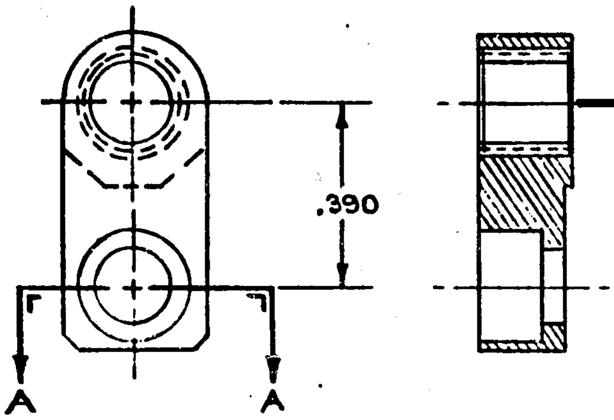
SECT. A-A

MATERIAL IS 6061-T6 ALUM. ALLOY

$$F_{TU} = 42000 \text{ psi (REF. C)}$$

$$M.S. = \frac{F_{TU}}{f_b} - 1 = \frac{42000}{12320} - 1 = \underline{\text{LARGE}}$$

8.7 GRATING CLAMP STRESSES.

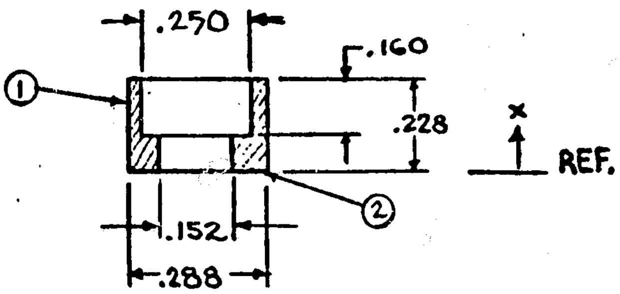


P_Z = 48.1 LBS.

$M_{A-A} = 0.390\,P_Z$

$M_{A-A} = 0.390(48.1)$

$M_{A-A} = \underline{18.76 \text{ IN-LBS. (ULT.)}}$



SECT. A-A

MAT'L. 2024-T4 ALUM. ALLOY

F_{TY} = 42000 PSI (REF. C)

ITEM	DESCRIPTION	AREA A	X	AX	AX²	Io
1	0.038 × 0.160	0.006080	0.148	0.000900	0.000148	0.000013
2	0.136 × 0.068	0.009248	0.034	0.000314	0.000011	0.000003
	Σ	0.015328		0.001214	0.000159	0.000016

$\bar{x} = (\Sigma AX)/(\Sigma A) = (0.001214)/(0.015328) = 0.079$ IN.

$C = 0.228 - 0.079 = 0.149$ IN.

$I = \Sigma I_o + \Sigma AX^2 - \bar{x}\cdot\Sigma AX$

$I = 0.000016 + 0.000159 - 0.000096$

$I = 0.000079$ IN.^{4}

$Z = (I/C) = 0.00053$ IN.^{3}

$$f_b = \frac{M_{A-A}}{Z} = \frac{18.76}{0.00053} = \underline{35400 \text{ PSI (ULT.)}}$$

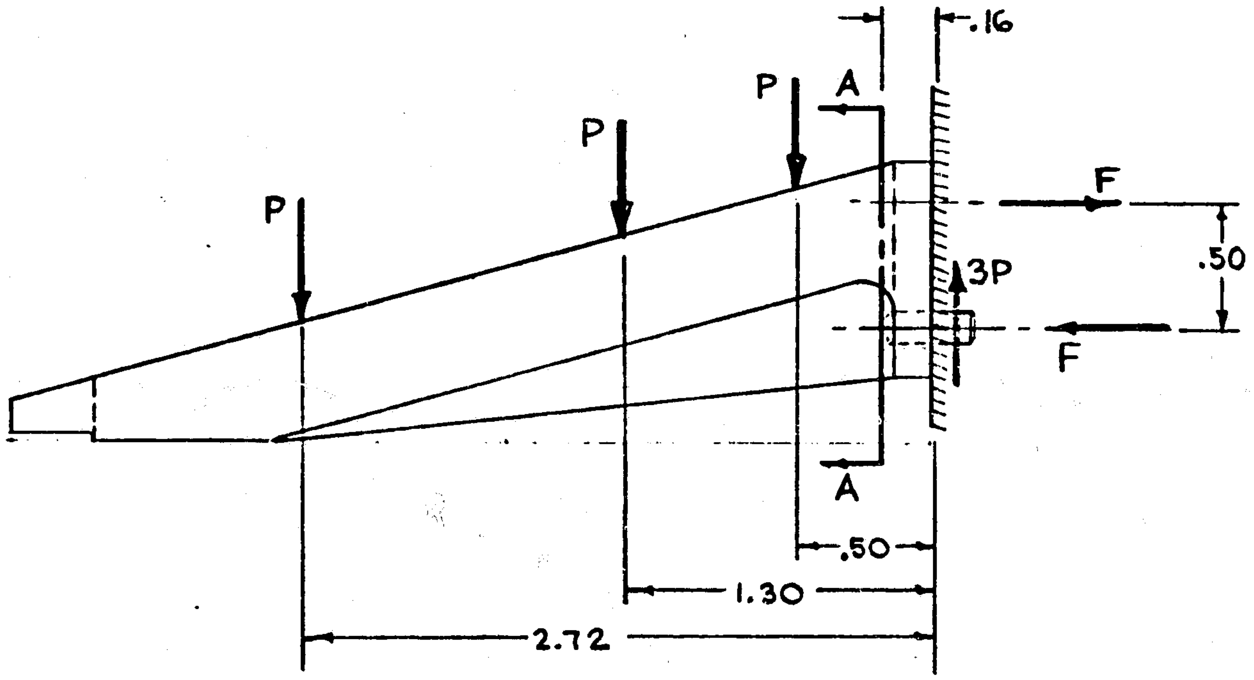
FOR YIELD CONDITION,

$$f_{bY} = (1.15/1.50)(35400) = \underline{27140 \text{ PSI (YIELD)}}$$

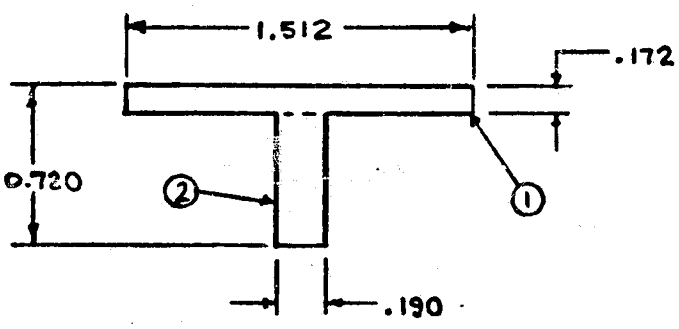
$$M.S. = \frac{F_{TY}}{f_{bY}} - 1 = \frac{42000}{27140} - 1 = \underline{+0.55}$$

9.0 ANALYSIS OF SLIT MIRROR BRACKET (TRIPLING MIRROR HOUSING)

9.1 X-AXIS LOADING.



9.1.1 BENDING STRESS IN HOUSING.



SECTION A-A

SECTION PROPERTIES

$C = 0.530$ IN.

$I = 0.015$ IN.^{4}

$Z = (I/C) = 0.028$ IN.^{3}

FROM PG. 28, $W_{SLIT\ MIRROR} = 0.16$ LBS.
(USE $P = W/3 \approx 0.05$ LBS./g)
FROM PG. 41, THE PEAK ACCELERATION FOR X-AXIS LOADING IS

$$G_x = 34.9\,g$$

$$\therefore P_{DYN.} = 1.50(0.05 \text{ LBS}/g)(34.9\,g)$$

$$P_{DYN.} = 2.62 \text{ LBS. (ULT.)}$$

BENDING MOMENT (SECT. A-A)

$$M_{A-A} = 0.34P + 1.14P + 2.56P$$

$$M_{A-A} = 4.04P = 4.04(2.62)$$

$$M_{A-A} = 10.58 \text{ IN-LBS. (ULT.)}$$

BENDING MOMENT (MAX.)

$$M_{MAX} = 0.50P + 1.30P + 2.72P$$

$$M_{MAX} = 4.52P = 4.52(2.62)$$

$$M_{MAX} = 11.84 \text{ IN-LBS. (ULT.)}$$

BENDING STRESS

$$f_b = \frac{M_{A-A}}{Z} = \frac{10.58}{0.028} \approx \underline{400 \text{ psi (ULT.)}}$$

MAT'L. IS 6061-T6 ALUMINUM ALLOY, $F_{TU} = 42000$ psi (REF. C)

$\therefore$ M.S. LARGE BY INSPECTION

9.1.2 ATTACHMENT TO FRONT PLATE.

ASSUME MOMENT, M_{MAX}, TO BE TAKEN OUT BY COUPLE FORCES ON THE SCREWS AND SHEAR TO BE TAKEN OUT BY DOWEL PINS.

9.1.2.1 SCREW TENSION.

$$\Sigma M = 0 = M_{MAX} - .50F$$

$$0.50F = 11.84$$

$$F = 23.7 \text{ LBS. (ULT.)}$$

ASSUMING A NO. 6-32 UNC CRES SCREW TORQUED TO 80% YIELD

$$P_Y = 216 \text{ LBS.} \quad (\text{REF. } e)$$

M.S. LARGE BY INSPECTION.

9.1.2.2 DOWEL PIN SHEAR.

P_S = SHEAR FORCE PER PIN = $(3P/2)$

$$P_S = \frac{3}{2}(2.62) = 3.93 \text{ LBS. (ULT.)}$$

A = PIN AREA (MS 16555-625, DIA. = .1251 IN.)

$$A = (\pi/4)(.1251)^2 = 0.0123 \text{ IN.}^2$$

$$f_s = \frac{P_s}{A} = \frac{3.93}{0.0123} \approx 320 \text{ PSI (ULT.)}$$

M.S. LARGE BY INSPECTION.

9.1.2.3 BEARING IN HOUSING WALL.

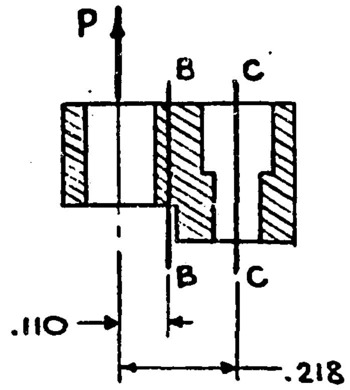
$$A_{br} = dt = .125(.16) = .02 \text{ IN}^2$$

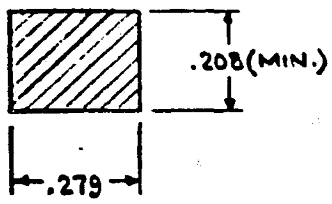
$$P_{MAX} = 3.93 \text{ LBS.}$$

$$f_{br} = \frac{P_{MAX}}{A_{br}} = \frac{3.93}{0.02} \approx 200 \text{ PSI (ULT.)}$$

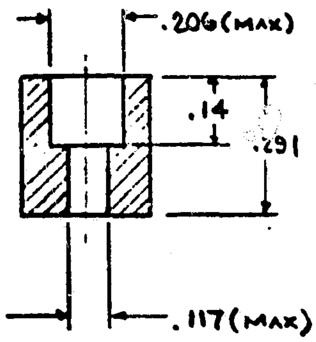
M.S. LARGE BY INSPECTION.

9.2 CLAMP STRESS.





SECT. B-B



SECT. C-C

SECTION B-B

$$M_{B\text{-}B} = 0.110P = 0.110(2.62)$$

$$M_{B\text{-}B} = 0.29 \text{ IN-LBS (ULT.)}$$

$$f_b = \frac{6 M_{B\text{-}B}}{bt^2} = \frac{6(0.29)}{(0.279)(0.208)^2}$$

$$f_b = 150 \text{ PSI (ULT.)}$$

MAT'L. IS 6061-T6 AL. ALLOY, F_{tu} = 42000 PSI

∴ <u>M.S. LARGE BY INSPECTION.</u>

SECTION C-C

SECTION PROPERTIES FOR SECT. C-C ARE

$$C = 0.173 \text{ IN.}$$

$$I = 0.000226 \text{ IN}^4$$

$$Z = (I/C) = 0.0013 \text{ IN}^3$$

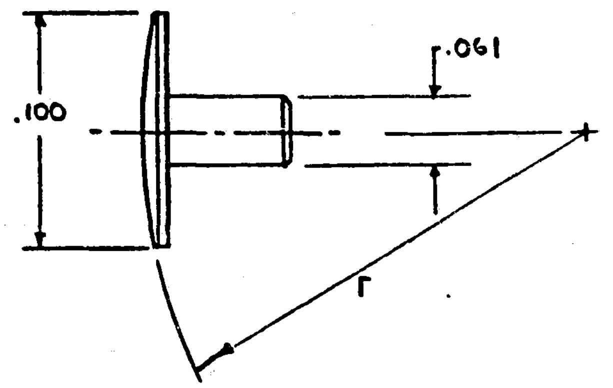
$$M_{CC} = 0.218P = 0.218(2.62)$$

$$M_{CC} = 0.57 \text{ IN-LBS. (ULT.)}$$

$$f_b = \frac{M_{C\text{-}C}}{Z} = \frac{0.57}{0.0013} \approx 440 \text{ PSI (ULT.)}$$

<u>M.S. LARGE BY INSPECTION.</u>

9.3 COMPRESSION IN PAD.



ASSUME MIN. DIAM. FOR − 4 PAD

$d' = .100$ IN.

ASSUME WHEN COMPRESSED, DIAMETER OF CONTACT AREA = .075 IN.

$$A = \frac{\pi}{4}(.075)^2 = 0.0044 \text{ IN.}^2$$

$$P_{MAX.} = 2.62 \text{ LBS.}$$

COMPRESSIVE STRESS

$$f_c = \frac{P_{MAX}}{A} = \frac{2.62}{0.0044} \approx \underline{600 \text{ PSI (ULT.)}}$$

Fc FOR KEL-F-81 PLASTIC = 5440 PSI (REF. 2)

$$M.S. = \frac{F_c}{f_c} - 1 = \frac{5440}{600} - 1 = \underline{\text{LARGE}}$$

9.4 COMPRESSION IN TRIPLING MIRROR.

AS IN THE CASE OF THE EBERT MIRROR AND THE GRATING BLANK, ASSUME THE STRESS GIVEN BY

$$f_c = \frac{P'}{1.2\pi R^2} \qquad \text{(REF. PG. 156)}$$

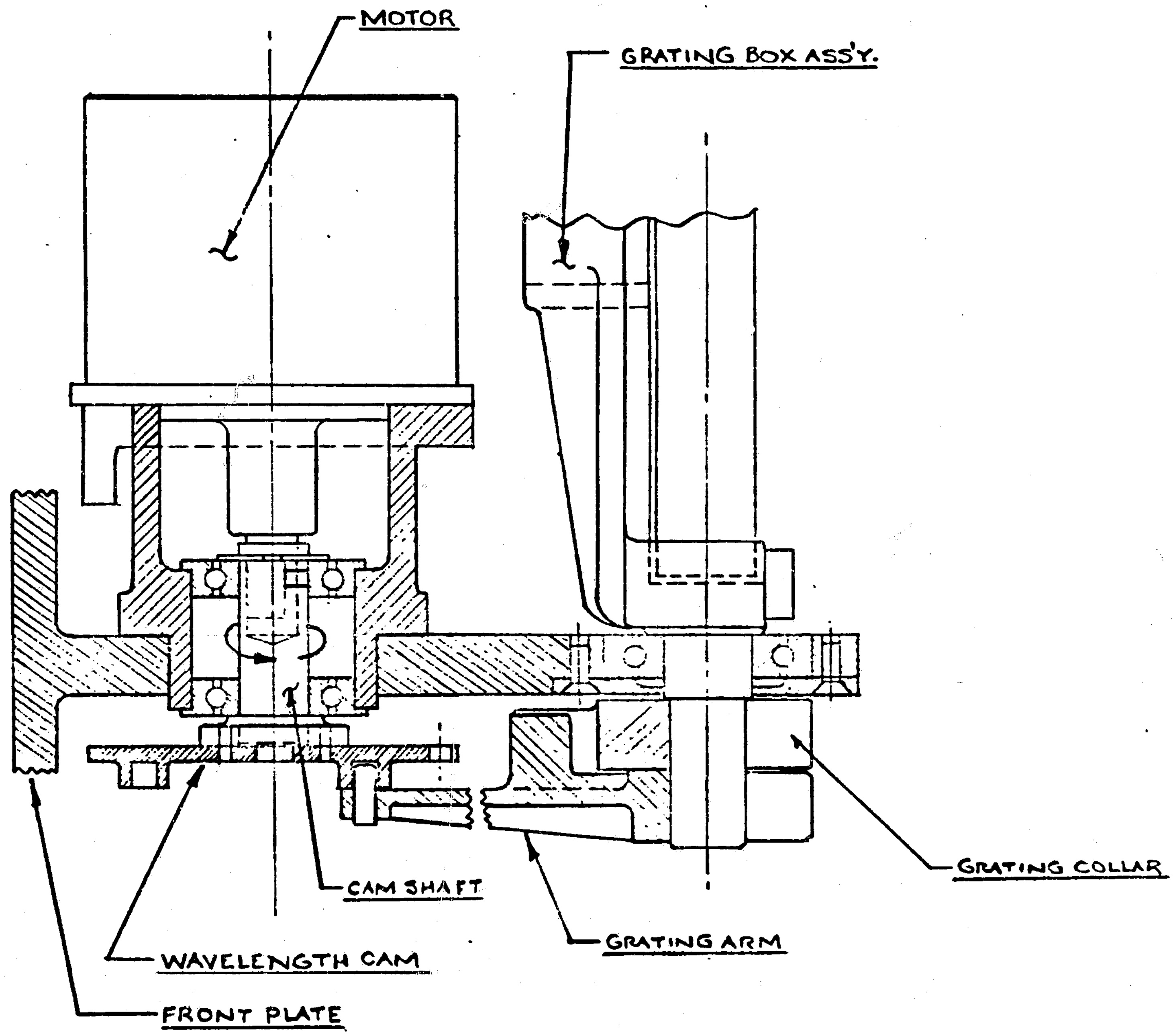
AGAIN ASSUMING $P' = 2P$ FOR IMPACT LOADING

$$f_c = \frac{2(2.62)}{1.2\pi(0.075/2)^2} \approx 1000 \text{ PSI (ULT.)}$$

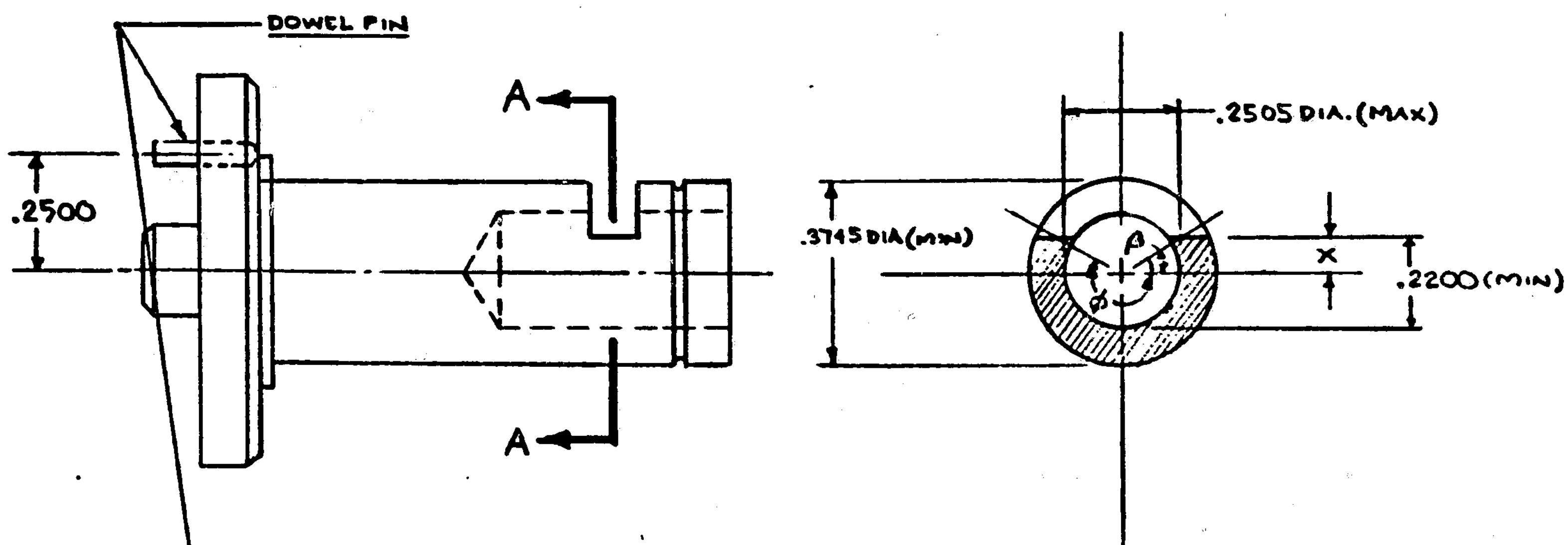
FOR CER-VIT MAT'L, F_c = 2500 PSI (REF. PG. 156)

$$M.S. = \frac{F_c}{f_c} - 1 = \frac{2500}{1000} - 1 = \underline{\text{LARGE}}$$

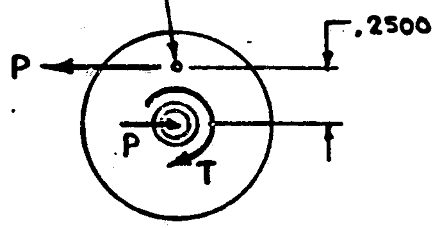
10.0 ANALYSIS OF WAVELENGTH CAM ASS'Y.



10.1 CAM SHAFT STRESSES.



SECTION A-A



10.1.1 CAM SHAFT SHEAR STRESS.

MAX. APPLIED TORQUE, T = 20 OZ-IN. = 1.25 IN-LB (REF. m)

FROM SECT. A-A, $r_{AVG.}$ = 0.15625 IN.

t = SECTION THICKNESS = 0.0620 IN.

$x = .22000 - (.25050/2) = 0.09475$

$\cos\alpha = (x/r_{AVG}) = (0.09475/0.15625) = 0.60640$

$\alpha = 52.67°$

$\beta = 90 - \alpha = 37.33°$

$\phi = 2\beta + 180° = 254.33° = 4.44$ RADIANS

$b = \phi r_{AVG} = 4.44(0.15625) = 0.694$ IN.

USING 1.50 LOAD FACTOR, $T = 1.50(1.25) = 1.875$ IN-LB (ULT)

$$f_s = \frac{3T}{bt^2} = \frac{3(1.875)}{0.694(0.062)^2} = 2100 \text{ psi (ULT.)} \quad \text{(REF. n)}$$

FOR 300 SERIES STAINLESS STEEL, F_{SU} = 40000 PSI (REF. c)

$$M.S. = \frac{F_{SU}}{f_s} - 1 = \frac{40000}{2100} - 1 = \underline{\text{LARGE}}$$

10.1.2 DOWEL PIN SHEAR STRESS.

$T = 1.50(1.25) = 1.875$ IN-LBS (ULT.)

$P = (T/.2500) = 4T = 4(1.875) = 7.50$ LBS. (ULT.)

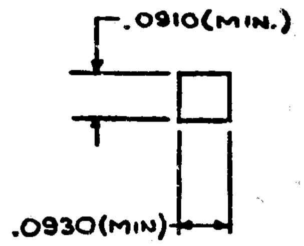
PIN IS MS16555-602, 0.0626 IN. DIAM.

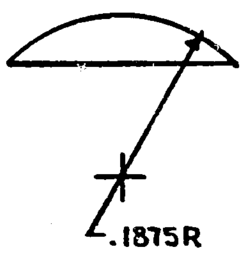
$A = (\pi/4)(0.0626)^2 = 0.0031$ IN^2

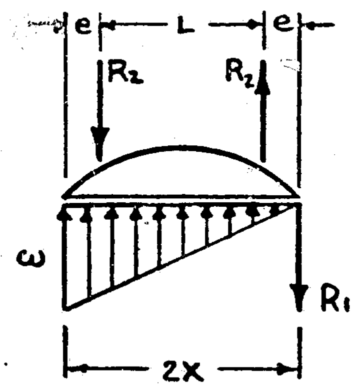
$$f_s = \frac{P}{A} = \frac{7.50}{0.0031} = \underline{2420 \text{ psi (ULT.)}}$$

M.S. LARGE BY INSPECTION.

10.2 CAM SHAFT KEY

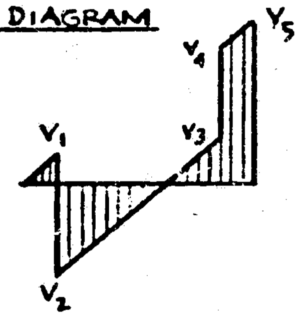




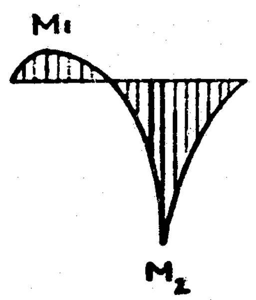


e = 0.0358 in.
L = 0.2500 in.
2X = 0.3216 in.

SHEAR DIAGRAM



MOMENT DIAGRAM



FROM STATICS

$$\Sigma M_0 = 0 = \frac{1}{2}\omega(2x)\left\{\frac{2}{3}(2x) - \frac{1}{2}(2x)\right\} + R_1 x - R_2 L$$

$$T = R_2 L = \omega x\left\{\frac{4}{3}x - \frac{3}{3}x\right\} + R_1 x$$

$$T = \frac{1}{3}\omega x^2 + R_1 x$$

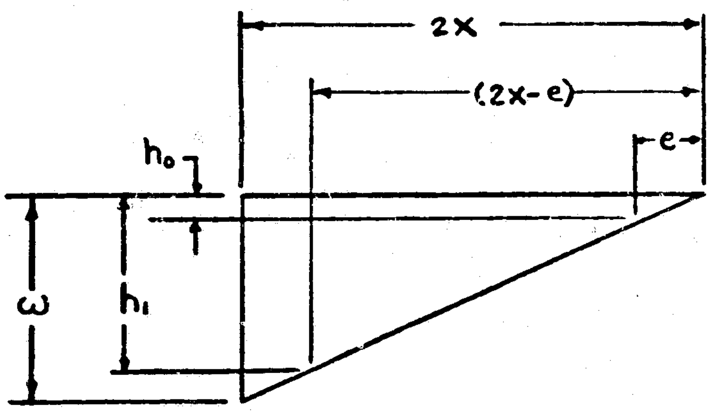
$$\Sigma F_v = 0 = \frac{1}{2}\omega(2x) + R_2 - R_1 - R_2$$

$$R_1 = \omega x$$

$$\therefore\ T = \frac{1}{3}\omega x^2 + (\omega x)x = \frac{4}{3}\omega x^2$$

$$\omega = \frac{3T}{4X^2} = \frac{3(1.25)}{4(0.1608)^2} = 36.26 \text{ lbs/in.}$$

R_1 = 5.83 lbs.
R_2 = (T/L) = 5.00 lbs.



BY SIMILAR TRIANGLES,

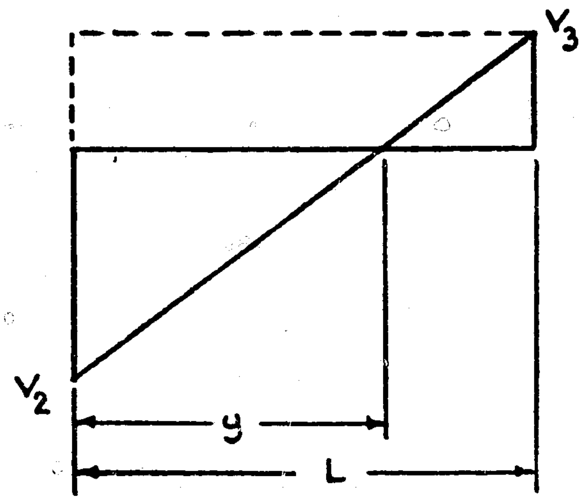
$$h_0 = \omega\left[e/2x\right] = 4.036 \text{ lbs/in.}$$

$$h_1 = \omega\left[(2x-e)/2x\right] = 32.224 \text{ lbs/in.}$$

$V_1 = h_1 e + \frac{1}{2}(\omega - h_1)e$ = **1.226 lbs.**
$V_2 = V_1 - 5.000$ = **-3.774 lbs.**
$\Delta V_1 = (L/2)(h_1 + h_0)$ = 4.533 lbs.
$V_3 = V_2 + \Delta V_1$ = **0.759 lbs.**
$V_4 = V_3 + 5.000$ = **5.759 lbs.**
$\Delta V_2 = \frac{1}{2}h_0 e$ = 0.071 lbs.
$V_5 = V_4 + \Delta V_2$ = **5.830 lbs.**

$$M_1 = \frac{1}{2} V_1 e = \frac{1}{2}(1.226)(.0358)$$

M_1 = 0.0219 IN-LBS.



$$\frac{V_2}{y} = \frac{(V_2 + V_3)}{L}$$

$$y = L \frac{V_2}{(V_2 + V_3)} = L \frac{3.774}{4.533}$$

$$y = 0.8326L = 0.208 \text{ IN.}$$

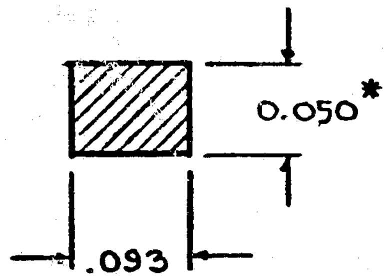
$$M_2 = M_1 - \frac{1}{2} V_2 y$$

$$M_2 = 0.0219 - \frac{3.774(0.208)}{2}$$

$$M_2 = -0.3706 \text{ IN-LBS.}$$

USING A 1.15 FACTOR FOR YIELD

$$M_{MAX} = 1.15(0.3706) = 0.4262 \text{ IN-LBS. (YIELD)}$$



KEY CROSS-SECTION

$$f_b = \frac{6 M_{MAX}}{bt^2} = \frac{6(0.4262)}{0.093(0.050)^2}$$

$$f_b \cong 11000 \text{ PSI (YIELD)}$$

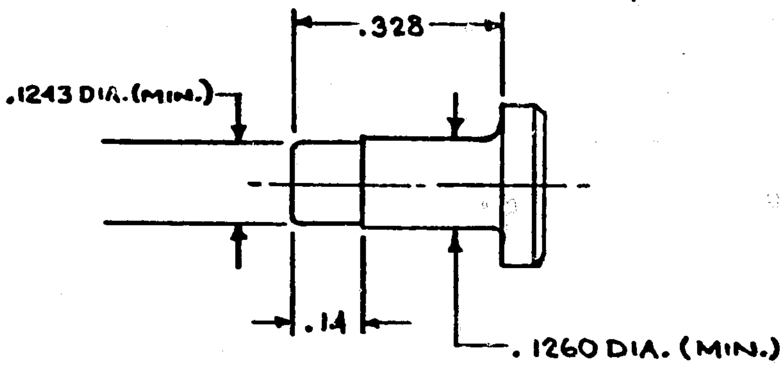
MAT'L. IS 300 SERIES STAINLESS STEEL.

F_{ty} = 35000 PSI (REF. C)

$$M.S. = \frac{F_{ty}}{f_b} - 1 = \frac{35000}{11000} - 1 = \text{LARGE}$$

* CONSERVATIVELY ASSUMED

10.3 GRATING ARM AXIAL PIN.



ASSUME MINIMUM CAM RADIUS, $r_{MIN} = 0.490$ IN.

ASSUME TORQUE $T' = N \times r_{MIN}$ CAN OVERCOME MAXIMUM APPLIED TORQUE, T.

N ≡ FRICTION FORCE

$\therefore \quad T' = T = 1.875$ IN-LBS (ULT.) $= (N)(.490)$

$N = 3.83$ LBS. (ULT.)

ASSUME A COEFFICIENT OF FRICTION, $\mu = 0.20$

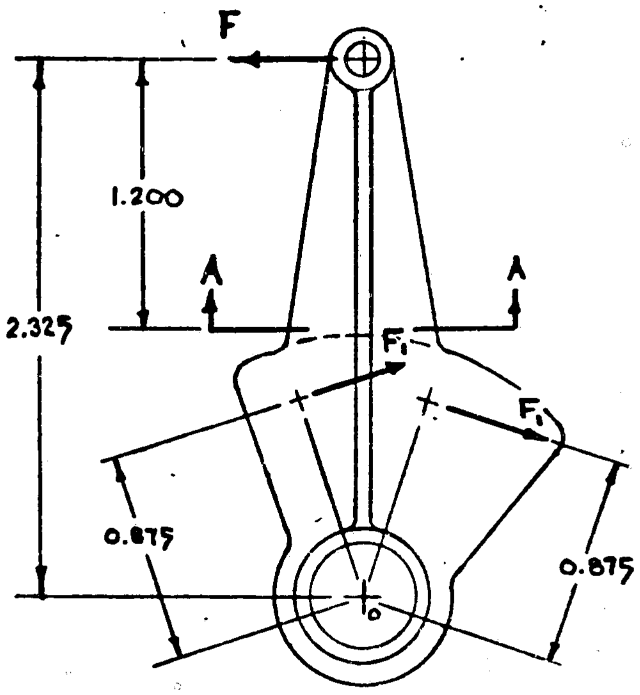
$F = \mu N = 0.765$ LBS. (ULT.)

PIN SHEAR AREA, $A = \frac{\pi}{4}(.1243)^2 = 0.0121$ IN^2

$$f_s = \frac{F}{A} = \frac{0.765}{0.0121} = \text{NEGLIGIBLE}$$

M.S. LARGE BY INSPECTION.

10.4 GRATING ARM STRESSES.



SCREW SHEAR

$F_{ULT.} = 1.5(0.51) = 0.765$ LBS. (ULT.)

$\Sigma M_o = 0 = 2F_1(0.875) - F(2.325)$

$1.750 F_1 = 2.325(0.765)$

$F_1 \approx 1.0$ LBS. (ULT.)

SHEAR IN SCREWS ATTACHING GRATING ARM TO GRATING COLLAR NEGLIGIBLE.

BENDING STRESS

$M = 1.200F = 1.200(0.51)$

$M = 0.918$ IN-LBS. (ULT.)

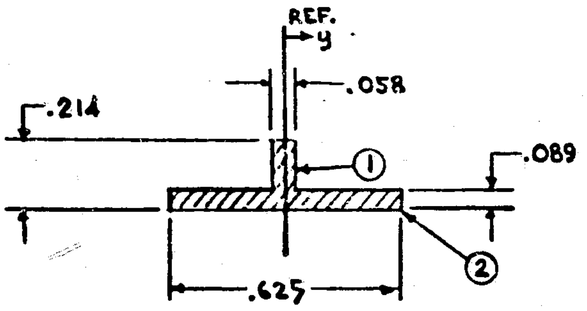
$C = (0.625/2) = 0.3125$ IN

$I = \frac{1}{12}\left\{.125(.058)^3 + .089(.625)^3\right\}$

$I = 0.0018 \text{ IN}^4$

$Z = (I/C) = 0.00576 \text{ IN}^3$

$f_b = \frac{M}{Z} = \frac{918}{5.76} \approx$ 160 PSI (ULT.)

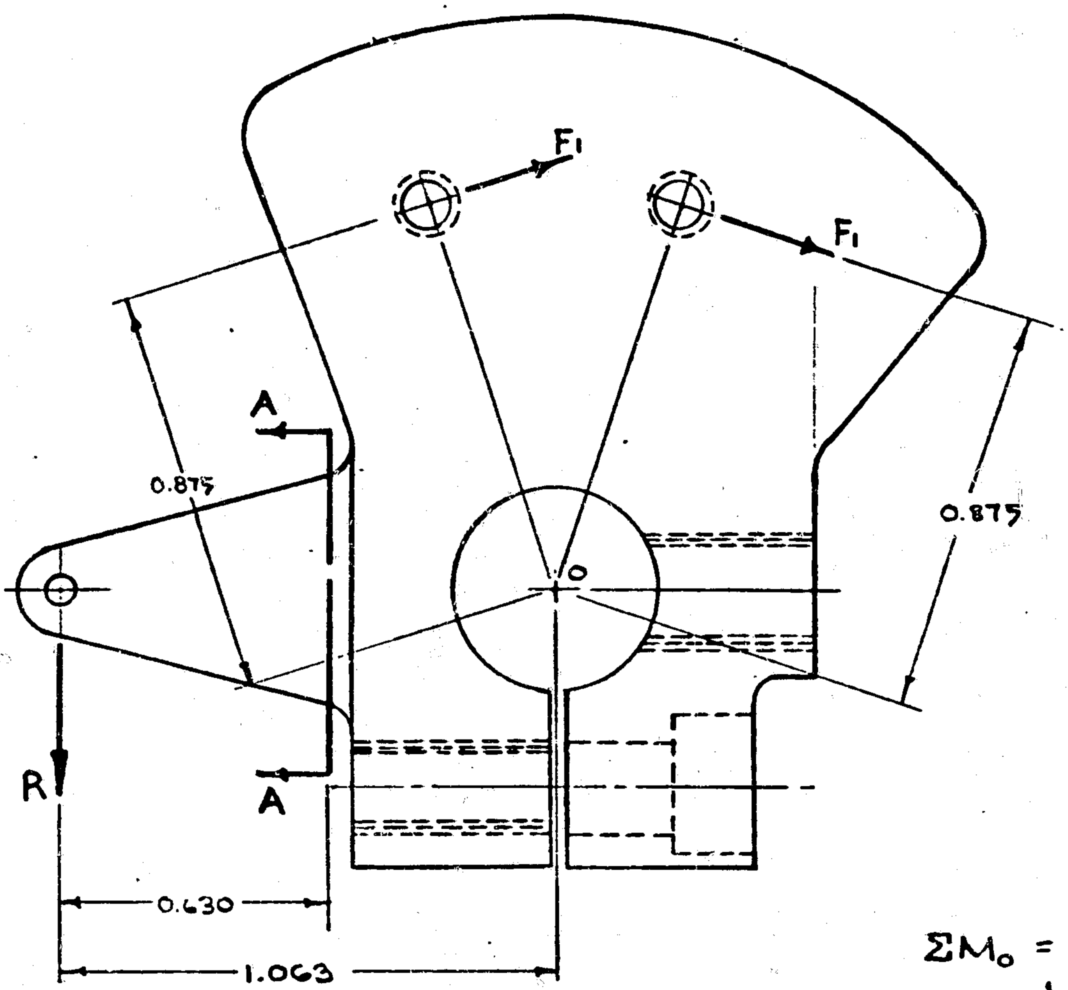


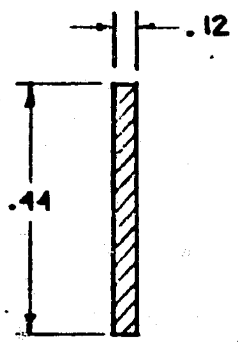
SECTION A-A

MAT'L. 6061-T6 ALUMINUM ALLOY.

M.S. LARGE BY INSPECTION.

10.5 GRATING ARM COLLAR STRESSES.





SECTION A-A

$$\Sigma M_o = 0 = 2F_1(0.875) - R(1.063)$$

$$1.063R = 1.778$$

$$R = 1.7 \text{ LBS. (ULT.)}$$

$$M_{AA} = 0.630R = 0.63(1.7)$$

$$M_{AA} = 1.07 \text{ IN-LBS (ULT.)}$$

$$f_b = \frac{6M_{AA}}{bt^2} = \frac{6(1.07)}{0.12(0.44)^2}$$

$$f_b \approx \underline{280 \text{ PSI (ULT.)}}$$

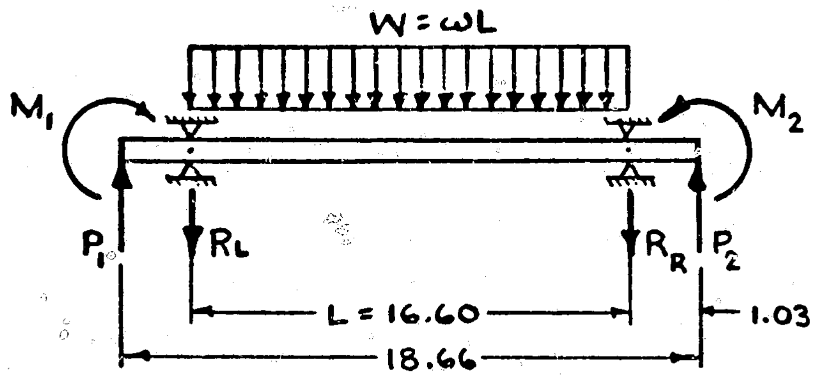
MAT'L. IS 6061-T6 ALUMINUM ALLOY.

M.S. LARGE BY INSPECTION.

11.0 ANALYSIS OF UVS HOUSING.

11.1 BENDING STRESSES, X & Y-AXES LOADING.

FOR LOADING IN THE X AND Y AXES DIRECTIONS, THE FOLLOWING MODEL WILL BE ASSUMED:



P_1 = WEIGHT OF FWD. PLATE, BAFFLE ASS'Y., DETECT. ELECTRONICS, MOTOR, HOUSING, SLIT MIRRORS, GRATING AND SUPPORT = 18.08 LBS.

P_2 = WEIGHT OF AFT PLATE, AFT END CAP, EBERT MIRROR = 6.55 LBS.

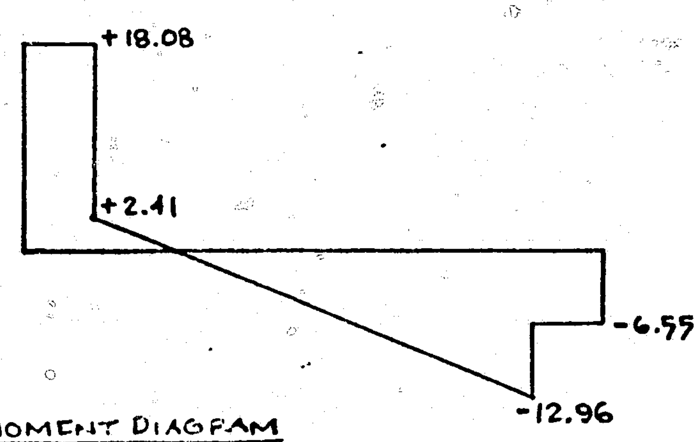
W = WEIGHT OF UVS HOUSING, MISC. HARDWARE, THERMAL INSUL. = 15.37 LBS.

FROM THE WEIGHTS OF THE INDIVIDUAL ITEMS AND THE DETAILED DRAWINGS, THE FOLLOWING MOMENTS ARE CALCULATED:

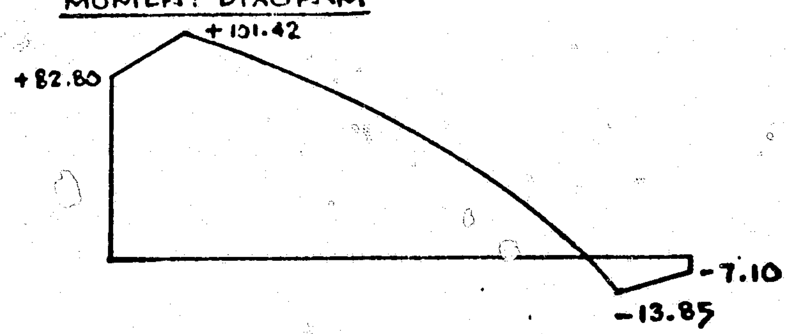
M_1 = 82.8 IN-LBS.

M_2 = 7.1 IN-LBS.

SHEAR DIAGRAM



MOMENT DIAGRAM



MOMENT AT L.H. REACTION, $M_L = M_1 + 1.03 P_1$

$M_L = 82.8 + 1.03(18.08) = 101.4$ IN-LBS.

$R_L' = (M_L/16.60) =$ 6.11 LBS ↓

$R_R' =$ 6.11 LBS ↑

MOMENT AT R.H. REACTION, $M_R = M_2 + 1.03 P_2$

$M_R = 7.1 + 1.03(6.55) = 13.8$ IN-LBS

$R_L'' = (M_R/16.60) =$ 0.83 LBS ↑

$R_R'' =$ 0.83 LBS ↓

DUE TO P_1, $R_L''' =$ 18.08 LBS ↑

DUE TO P_2, $R_R''' =$ 6.55 LBS ↓

DUE TO W, $R_L'''' = R_R'''' =$ 7.69 LBS ↑

THE TOTAL REACTIONS ARE

R_L = 15.67 LBS. ↓

R_R = 6.41 LBS. ↑

MAXIMUM BENDING MOMENT = M_{MAX} = + 101.42 IN-LBS./g

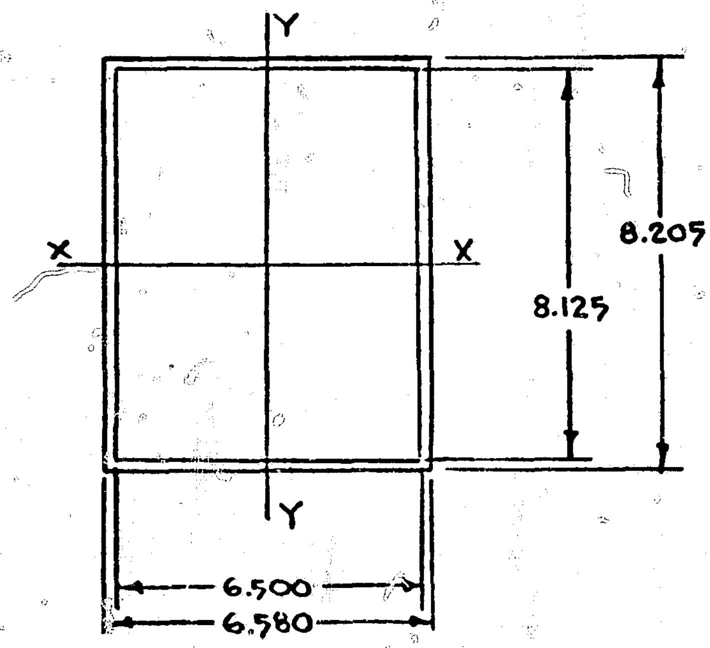
FROM PG. 40 THE CRITICAL ACCELERATIONS FOR X & Y LOADINGS ARE

$$G_X = 31.1 g$$
$$G_Y = 20.9 g$$

UTILIZING A 1.50 LOAD FACTOR, THE DYNAMIC BENDING MOMENTS ARE

$$x(M_{MAX})_{DYN.} = 1.5(101.42)(31.1) = \underline{4731 \text{ IN-LBS (ULT.)}}$$
$$y(M_{MAX})_{DYN.} = 1.5(101.42)(20.9) = \underline{3180 \text{ IN-LBS (ULT.)}}$$

FOR WEIGHT RELIEF, THE HOUSING HAS POCKETS IN WHICH THE NOMINAL THICKNESS IS REDUCED FROM .19 TO .04. TO BE ULTRA-CONSERVATIVE, THE FOLLOWING CROSS-SECTION ASSUMES A TYPICAL THICKNESS OF .04 OVER ENTIRE AREA.



HOUSING CROSS-SECTION

$$I_X = \frac{1}{12}\left\{6.580(8.205)^3 - 6.500(8.125)^3\right\}$$
$$I_X = 12.35 \text{ IN}^4$$
$$C = 4.10 \text{ IN.}$$
$$Z_X = (I_X/C) = 3.01 \text{ IN}^3$$

$$I_Y = \frac{1}{12}\left\{8.205(6.580)^3 - 8.125(6.500)^3\right\}$$
$$I_Y = 8.84 \text{ IN.}^4$$
$$C = 3.29 \text{ IN.}$$
$$Z_Y = (I_Y/C) = 2.69 \text{ IN}^3$$

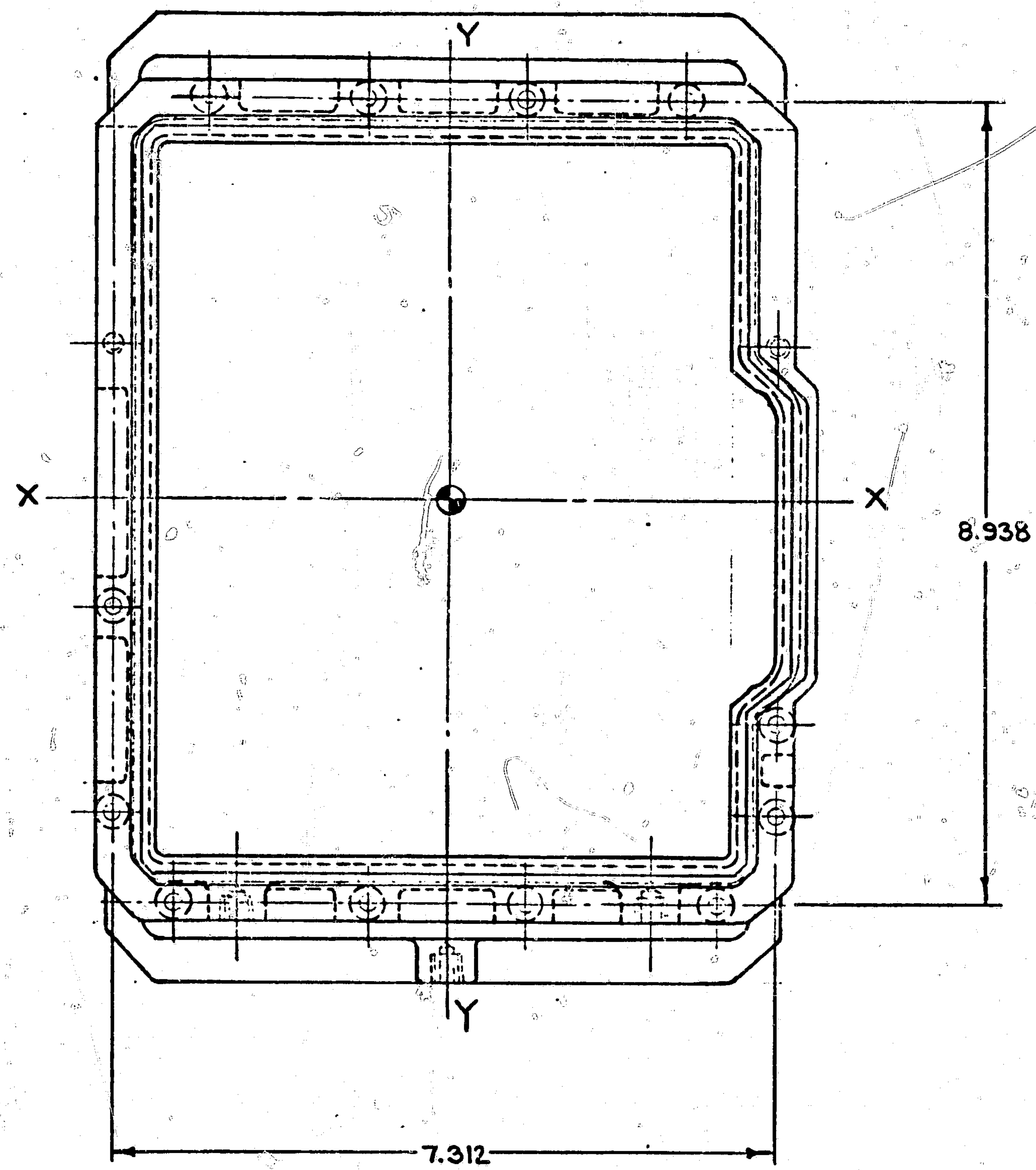
THE STRESSES ARE

$$f_{b_X} = \frac{x(M_{MAX})_{DYN}}{Z_Y} = \frac{4731}{2.69} = \underline{1760 \text{ PSI (ULT.)}}$$
$$f_{b_Y} = \frac{y(M_{MAX})_{DYN}}{Z_X} = \frac{3180}{3.01} = \underline{1060 \text{ PSI (ULT.)}}$$

FOR 356-T6 AL. ALLOY CASTING, F_{TU} = 30000 PSI (REF. C)

∴ M.S. LARGE BY INSPECTION

11.2 ATTACHMENT OF FWD. PLATE TO HOUSING.



11.2.1 X-AXIS LOADING.

THE PEAK DYNAMIC LOADS ARE

$$ {}_xP_{1Y} = 1.15(18.08 \text{ LBS.})(31.1g) = \underline{646.6 \text{ LBS.(YIELD)}} $$
$$ {}_xP_{1U} = 1.50(18.08 \text{ LBS.})(31.1g) = \underline{843.4 \text{ LBS.(ULT.)}} $$
$$ {}_xM_{1Y} = 1.15(82.8 \text{ IN-LBS.})(31.1g) = \underline{2961 \text{ IN-LBS.(YIELD)}} $$
$$ {}_xM_{1U} = 1.50(82.8 \text{ IN-LBS.})(31.1g) = \underline{3863 \text{ IN-LBS.(ULT.)}} $$

11.2.1.1 JOINT FRICTION CAPACITY

SCREWS ARE #8-32 UNC CRES.

MIN. YIELD LOAD IS 420 LBS. (REF. e)

ASSUME SCREWS TORQUED TO 80% OF YIELD

$$ (P_Y) = 0.80(420) = 336 \text{ LBS.} $$

ASSUME A COEFFICIENT OF FRICTION OF 0.2

FRICTION FORCE PER SCREW IS

$$ F_F = 0.2(336) = 67.2 \text{ LBS.} $$

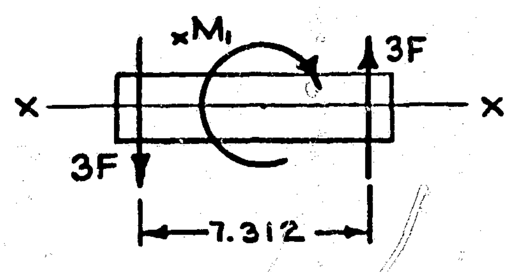
SINCE THERE ARE 14 SCREWS, THE FRICTION CAPACITY OF THE JOINT IS

$$ \Sigma F_F = 14(67.2) = 940.8 \text{ LBS.} $$

$$ \text{M.S.} = \frac{\Sigma F_F}{{}_xP_{1Y}} - 1 = \frac{940.8}{646.6} - 1 = \underline{+0.45} $$

11.2.1.2 FASTENER TENSION

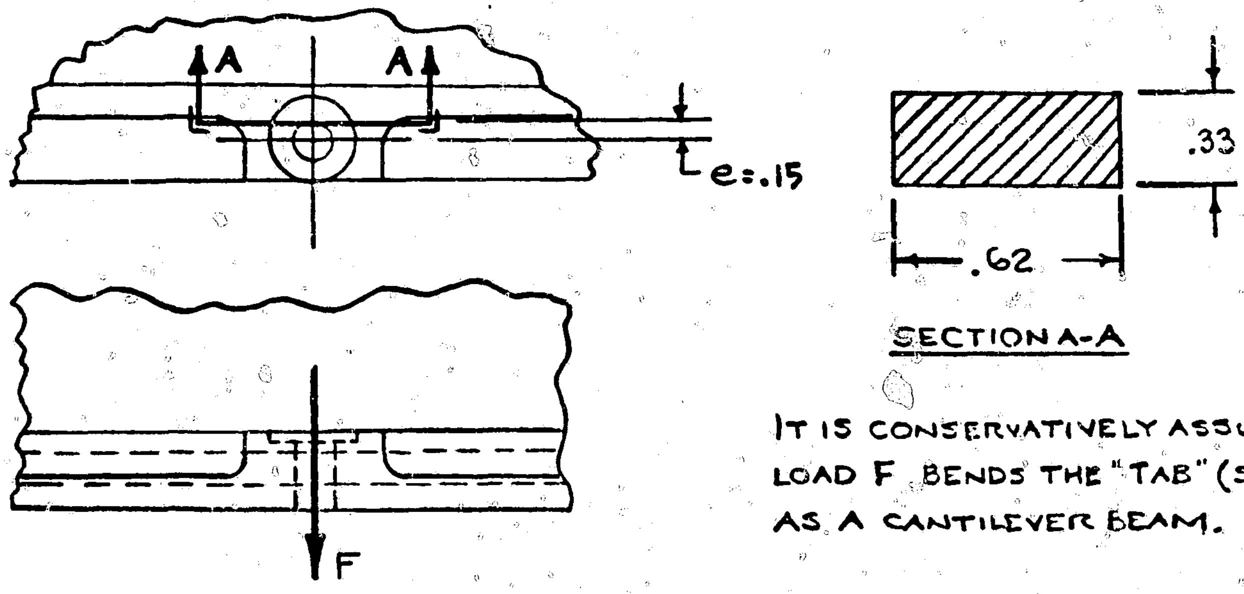
CONSERVATIVELY ASSUME ONLY 6 SCREWS EFFECTIVE IN TAKING OUT BENDING MOMENT ${}_xM_1$



$$ \Sigma M = 0 = {}_xM_1 - 3F(7.312) $$
$$ F = \frac{{}_xM_1}{21.936} $$
$$ F_Y = (2961/21.936) = \underline{135.0 \text{ LBS.(YIELD)}} $$
$$ F_U = (3863/21.936) = \underline{176.1 \text{ LBS.(ULT.)}} $$

$$ \text{M.S.} = \frac{P_Y}{F_Y} - 1 = \frac{336.0}{135.0} - 1 = \underline{\text{LARGE}} $$

11.2.1.3 LOCAL TAB BENDING.



SECTION A-A

IT IS CONSERVATIVELY ASSUMED THAT LOAD F BENDS THE "TAB" (SECTION A-A) AS A CANTILEVER BEAM.

BENDING MOMENT, $M = Fe = (176.1)(0.15) = 26.42$ IN-LBS (ULT.)

BENDING STRESS

$$f_b = \frac{6M}{bt^2} = \frac{6(26.42)}{0.62(0.33)^2} \approx 2380 \text{ PSI (ULT.)}$$

M.S. LARGE BY INSPECTION

11.2.2 Y-AXIS LOADING.

THE PEAK DYNAMIC LOADS ARE

$$_YP_{1_Y} = 1.15(18.08 \text{ LBS.})(20.9g) = 434.6 \text{ LBS. (YIELD)}$$

$$_YP_{1_U} = 1.50(18.08 \text{ LBS.})(20.9g) = 566.8 \text{ LBS. (ULT.)}$$

$$_YM_{1_Y} = 1.15(82.8 \text{ IN-LBS.})(20.9g) = 1990 \text{ IN-LBS. (YIELD)}$$

$$_YM_{1_U} = 1.50(82.8 \text{ IN-LBS.})(20.9g) = 2596 \text{ IN-LBS. (ULT.)}$$

11.2.2.1 JOINT FRICTION CAPACITY

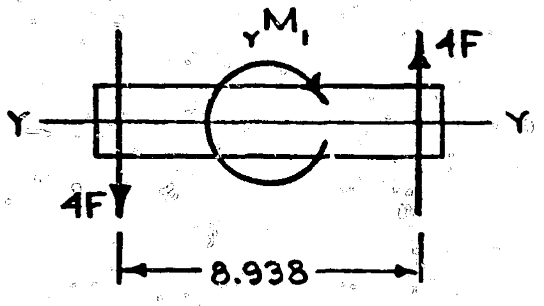
$\Sigma F_F = 940.8$ LBS. (REF. PG. 130)

$_YP_{1_Y} = 434.6$ LBS.

$$M.S. = \frac{\Sigma F_F}{_YP_{1_Y}} - 1 = \frac{940.8}{434.6} - 1 = \text{LARGE}$$

11.2.2.2 FASTENER TENSION

CONSERVATIVELY ASSUME ONLY 8 SCREWS EFFECTIVE IN REACTING BENDING MOMENT ${}_{Y}M_1$.



$$\Sigma M = 0 = {}_{Y}M_1 - 4F(8.938)$$

$$F = \frac{{}_{Y}M_1}{35.792}$$

$F_Y = (1990/35.792) = $ 55.7 LBS. (YIELD)

$F_U = (2596/35.792) = $ 72.6 LBS (ULT.)

$$M.S. = \frac{P_Y}{F_Y} - 1 = \frac{336.0}{55.7} - 1 = \text{LARGE}$$

11.2.2.3 LOCAL TAB BENDING

SINCE FORCES LOWER FOR Y-AXIS LOADING,

M.S. LARGE BY INSPECTION

11.2.3 Z-AXIS LOADING.

11.2.3.1 FASTENER TENSION.

FOR Z-AXIS LOADING, $G_Z = 31.4\ g$ (REF. PG. 40)

$\therefore\ {}_{Z}P_{1Y} = 1.15(18.08 \text{ LBS.})(31.4\ g) = $ 652.9 LBS. (YIELD)

${}_{Z}P_{1U} = 1.50(18.08 \text{ LBS.})(31.4\ g) = $ 851.6 LBS. (ULT.)

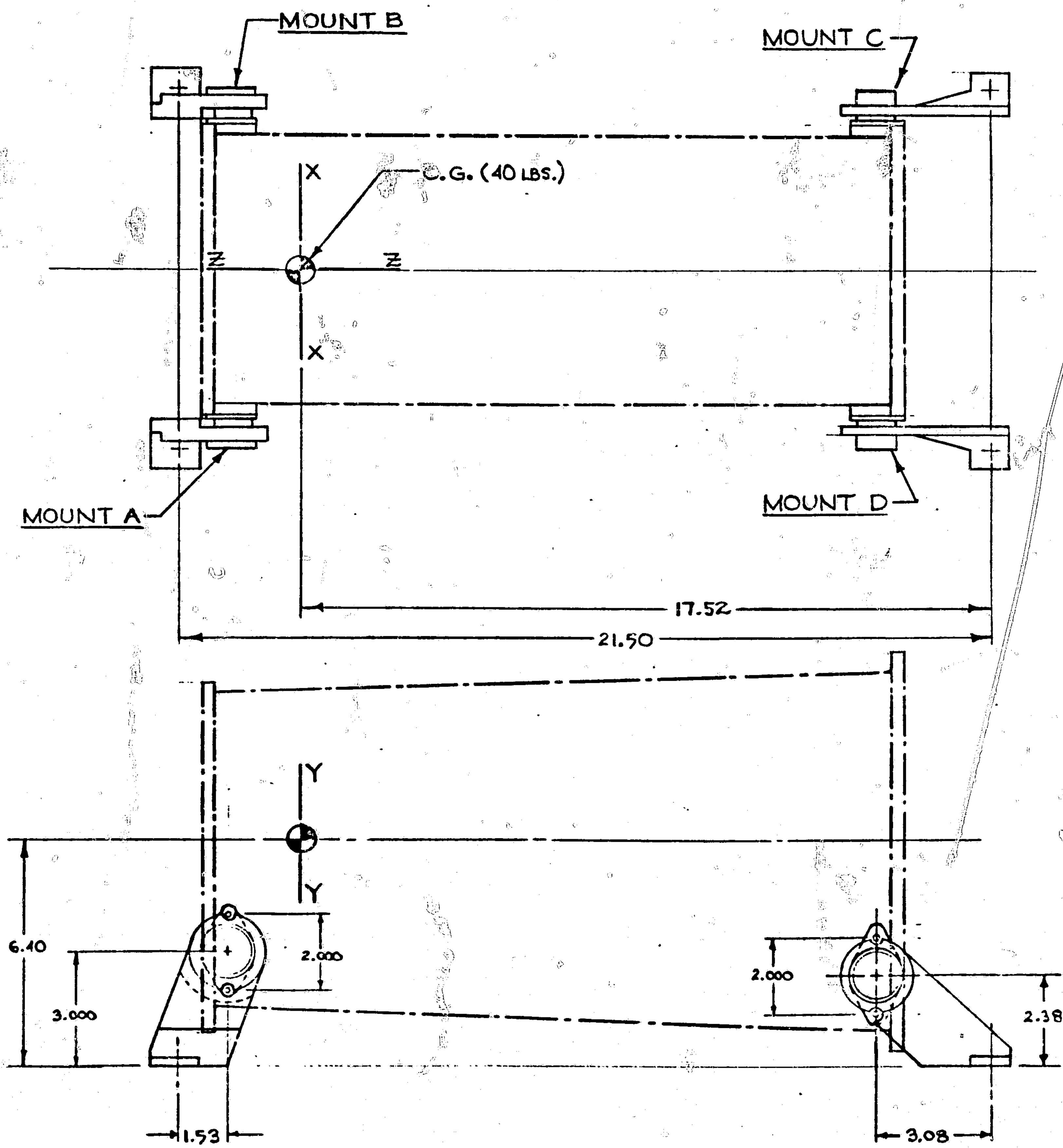
THERE ARE 14 FASTENERS. LOAD PER FASTENER IS

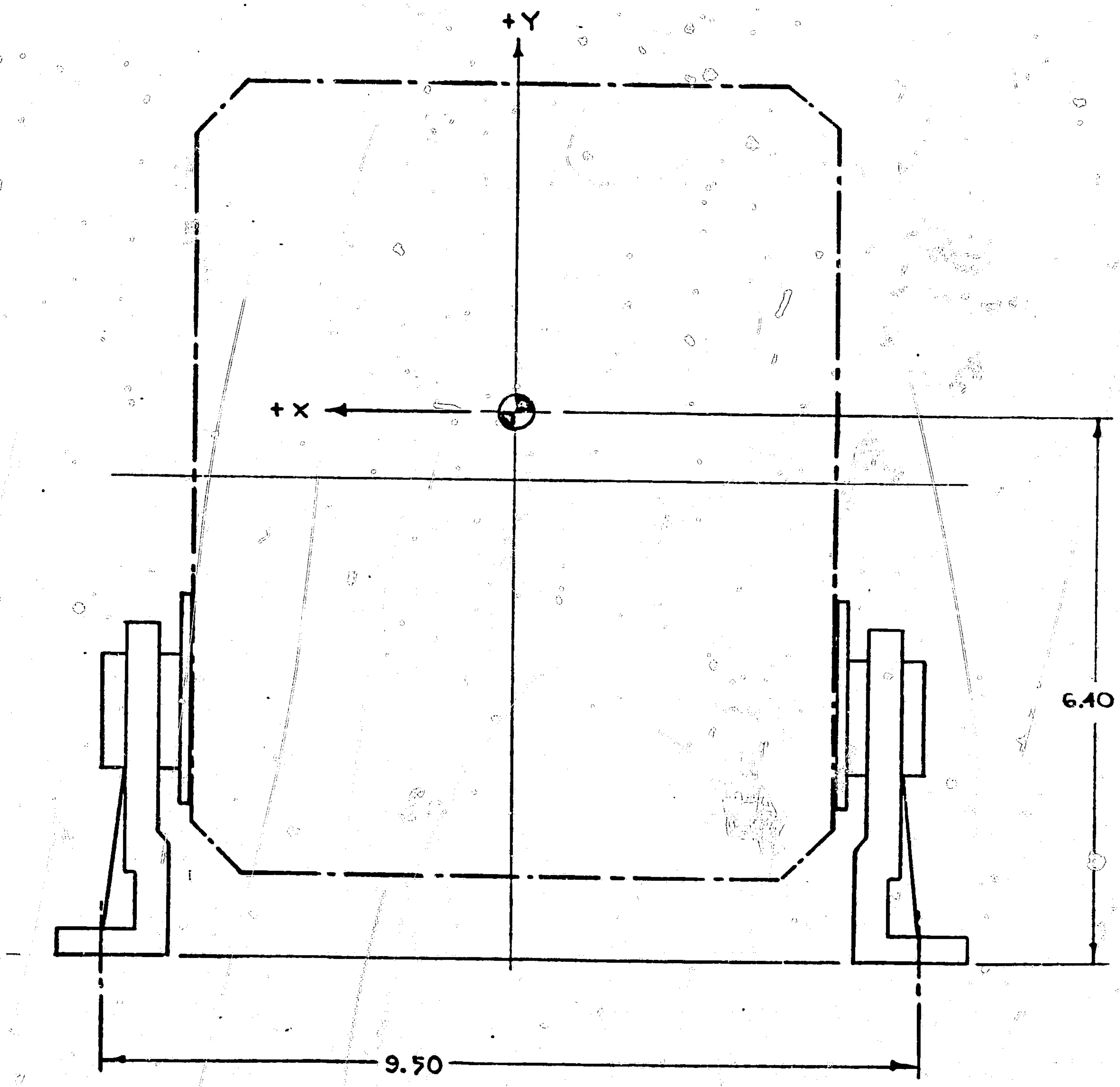
${}_{Z}P'_{1Y} = (652.9/14) = 46.6$ LBS. (YIELD)

ALLOWABLE P_Y = 336.0 LBS (REF. PG. 130)

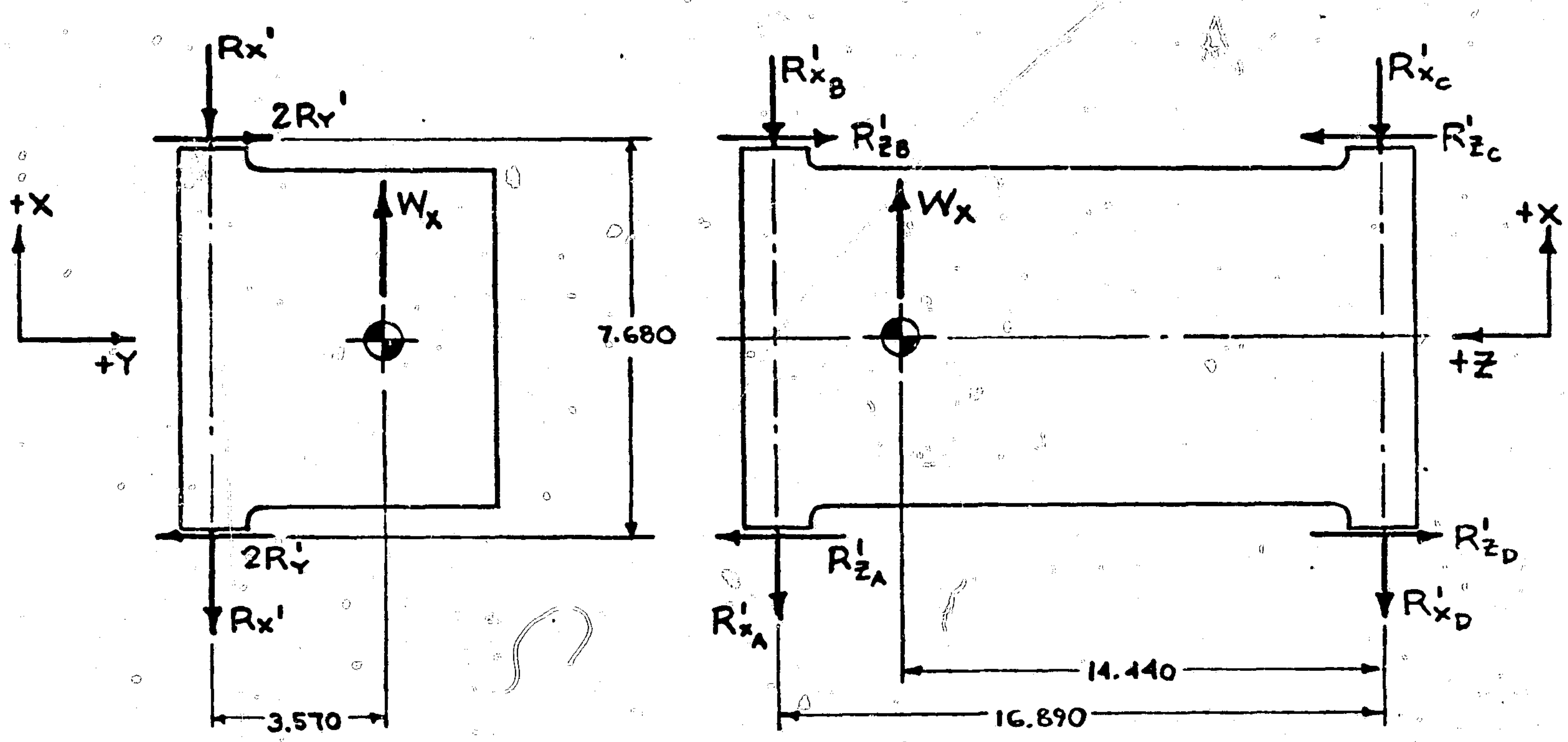
$$M.S. = \frac{P_Y}{{}_{Z}P'_{1Y}} - 1 = \frac{336.0}{46.6} - 1 = \text{LARGE}$$

11.3 ATTACHMENT OF ISOLATOR MOUNTS TO HOUSING.




+Y
+X
6.40
9.50

11.3.1 X-AXIS LOADING.



LET R_x', R_y', R_z' INDICATE REACTIONS ON HOUSING TO APPLIED LOAD W_x.

ALSO, LET $(R_x')_A = (R_x')_B = R_L$

$(R_x')_C = (R_x')_D = R_R$

$(R_z')_A = (R_z')_B = R_{Z_L}$

$(R_z')_C = (R_z')_D = R_{Z_R}$

$$\Sigma M_A = 0 = W_x(3.570) - 2R_Y'(7.680)$$

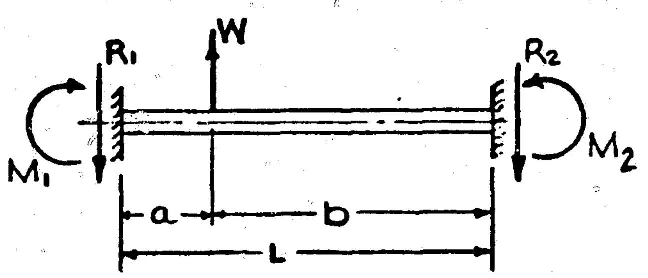
$$15.360 R_Y' = 3.570 W_x$$

$$R_Y' = \underline{0.2324 W_x}$$

FOR STATIC LOADING WITH W_x = 40 LBS.

$$R_Y' = \underline{9.30 \text{ LBS.}}$$

ASSUME THE X & Z REACTIONS MAY BE DETERMINED BY CONSIDERING THE STRUCTURE TO BE A BEAM WITH BUILT-IN ENDS UNDER AN APPLIED INTERMEDIATE CONCENTRATED LOAD.



$a = 2.450$

$b = 14.440$

$L = 16.890$

$$R_1 = W\frac{b^2}{L^3}(3a+b) = 0.9430W$$

$$R_2 = W\frac{a^2}{L^3}(3b+a) = 0.0570W$$

$$R_L = (R_1/2) = 0.4715W = \underline{18.86 \text{ LBS.}}$$

$$R_R = (R_2/2) = 0.0285W = \underline{1.14 \text{ LBS.}}$$

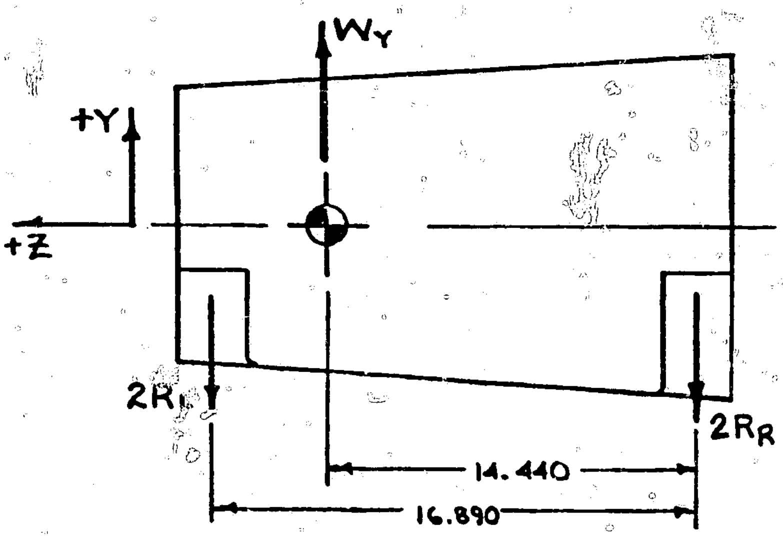
$$M_1 = W(ab^2/L^2) = 1.791W$$

$$M_2 = W(a^2b/L^2) = 0.304W$$

$$R_{Z_L} = (M_1/7.680) = 0.2332W = \underline{9.33 \text{ LBS.}}$$

$$R_{Z_R} = (M_2/7.680) = 0.0396W = \underline{1.58 \text{ LBS.}}$$

11.3.2 Y-AXIS LOADING.



$\Sigma F_Y = 0 = W_Y - 2(R_L + R_R)$

$R_L + R_R = (W_Y/2)$

$\Sigma M_A = 0 = 2R_R(16.890) - W_Y(2.450)$

$33.780\, R_R = 2.450\, W_Y$

$R_R = \underline{0.0725\, W_Y}$

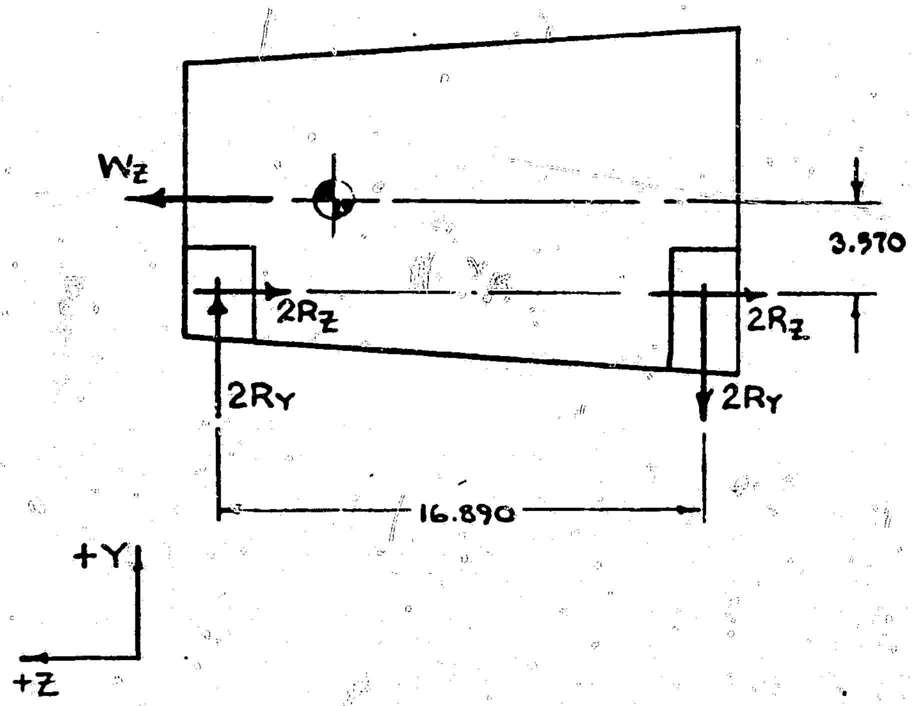
$R_L = \underline{0.4275\, W_Y}$

FOR STATIC LOADING, WITH W_Y = 40 LBS.

$R_R = \underline{2.90 \text{ LBS.}}$

$R_L = \underline{17.10 \text{ LBS.}}$

11.3.3 Z-AXIS LOADING.



$\Sigma F_Z = 0 = W_Z - 4R_Z$

$4R_Z = W_Z$

$R_Z = \underline{0.2500\, W_Z}$

$\Sigma M_A = 0 = 3.570\, W_Z - 16.890(2R_Y)$

$33.780\, R_Y = 3.570\, W_Y$

$R_Y = \underline{0.1057\, W_Y}$

FOR STATIC LOADING, WITH W_Z = 40 LBS.

$R_Z = \underline{10.00 \text{ LBS.}}$

$R_Y = \underline{4.23 \text{ LBS.}}$

11.3.4 TABULATION OF APPLIED LOADS AT INTERFACE MTG. POINTS., LBS.*

	DUE TO LOAD W_X			DUE TO LOAD W_Y			DUE TO LOAD W_Z		
	P_X	P_Y	P_Z	P_X	P_Y	P_Z	P_X	P_Y	P_Z
POINT A	18.86	9.30	-9.33	0	17.10	0	0	-4.23	10.00
POINT B	18.86	-9.30	9.33	0	17.10	0	0	-4.23	10.00
POINT C	1.14	-9.30	-1.58	0	2.90	0	0	4.23	10.00
POINT D	1.14	9.30	1.58	0	2.90	0	0	4.23	10.00

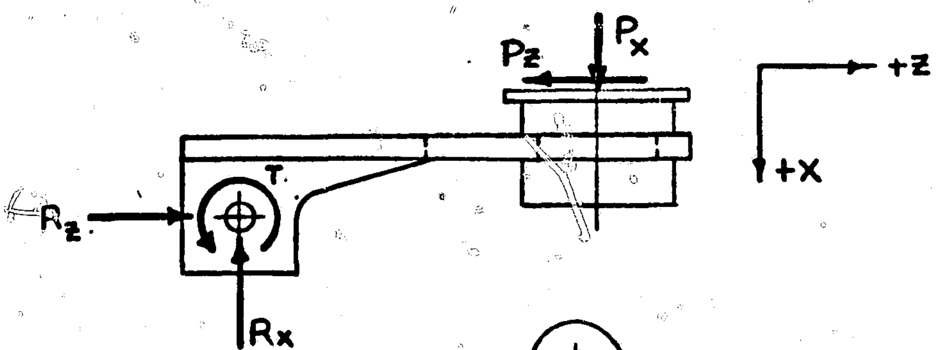
* LOADS ARE STATIC LOADS @ INTERFACE OF MOUNTING FEET AND HOUSING.

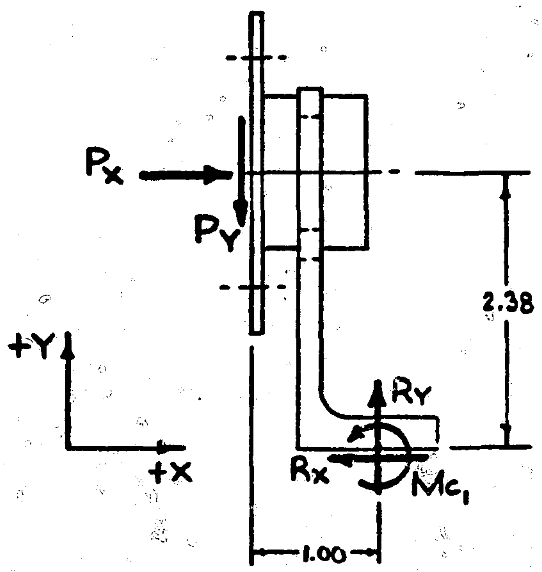
11.3.5 MOUNTING FEET LOADS ANALYSIS.

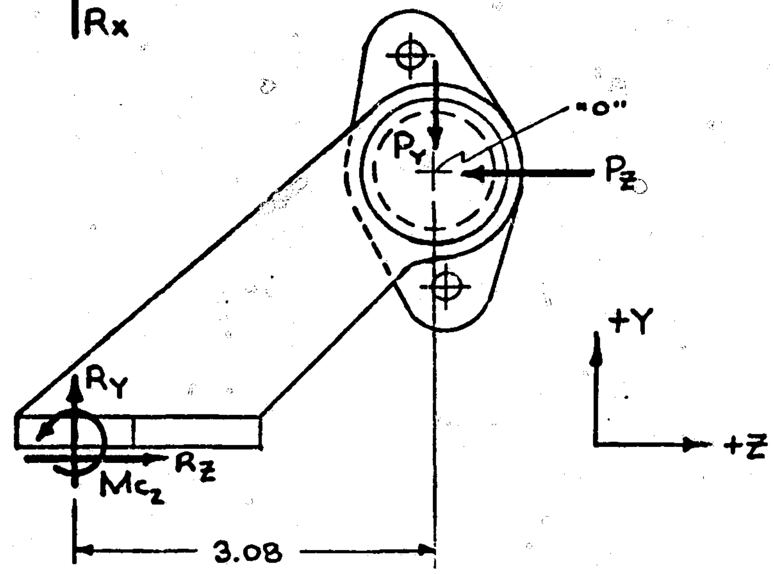
11.3.5.1 FOR X-AXIS LOADING.

11.3.5.1.1 MOUNTS C & D (MOUNT C SHOWN).

$P_X = 0.0285W = 1.14$ LBS.
$P_Y = 0.2324W = 9.30$ LBS.
$P_Z = 0.0396W = 1.58$ LBS.







FOR STATIC LOADING,

$\Sigma F_X = 0 = P_X - R_X$
$R_X = P_X = \underline{1.14 \text{ LBS.}}$

$\Sigma F_Z = 0 = R_Z - P_Z$
$R_Z = P_Z = \underline{1.58 \text{ LBS.}}$

$\Sigma F_Y = 0 = R_Y - P_Y$
$R_Y = P_Y = \underline{9.30 \text{ LBS.}}$

$\Sigma M_Z = 0 = 2.38P_X - 1.00P_Y - M_{C_1}$
$M_{C_1} = 2.38(1.14) - 1.00(9.30) = \underline{-6.59 \text{ IN-LBS.}}$

$\Sigma M_X = 0 = 3.08P_Y - 2.38P_Z - M_{C_2}$
$M_{C_2} = 3.08(9.30) - 2.38(1.58) = \underline{24.88 \text{ IN-LBS.}}$

$$M_{C_R} = \sqrt{M_{C_1}^2 + M_{C_2}^2} = \underline{25.74 \text{ IN-LBS.}}$$

$\Sigma M_Y = 0 = 3.08P_X - T - 1.00P_Z$
$T = 3.08(1.14) - 1.00(1.58) = \underline{1.93 \text{ IN-LBS.}}$

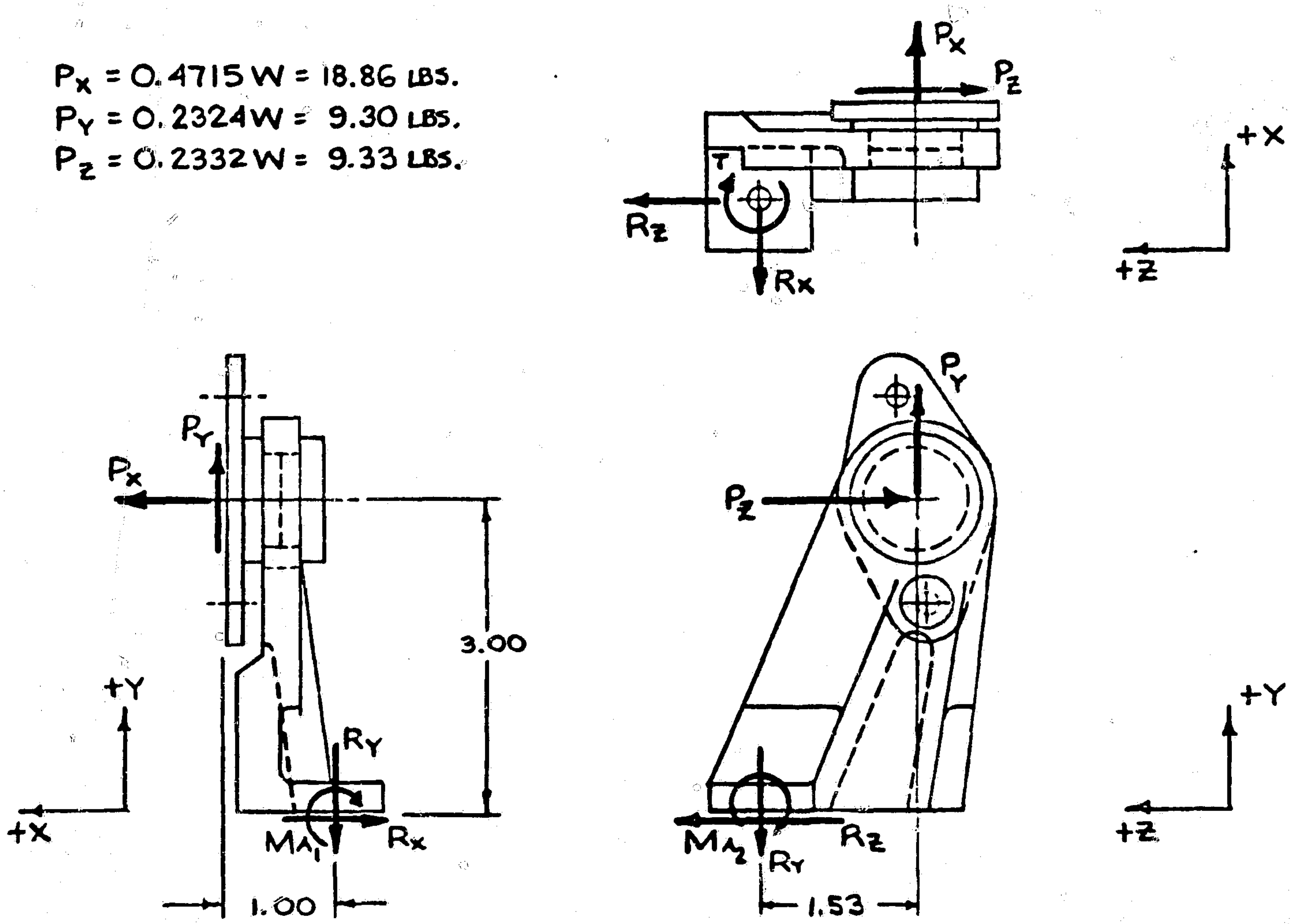
@ HOUSING, $P_{SHEAR} = \sqrt{R_Y^2 + R_Z^2} = \underline{9.43 \text{ LBS.}}$

@ NR BKT., $P_{SHEAR} = \sqrt{R_X^2 + R_Z^2} = \underline{1.95 \text{ LBS.}}$

FOR X- AXIS LOADING.

11.3.5.1.2 MOUNTS A & B (MOUNT A SHOWN).

$P_X = 0.4715\,W = 18.86$ LBS.
$P_Y = 0.2324\,W = 9.30$ LBS.
$P_Z = 0.2332\,W = 9.33$ LBS.



FOR STATIC LOADING

$\Sigma F_x = 0 = P_x - R_x$

$R_x = P_x = \underline{18.86 \text{ LBS.}}$

$\Sigma F_Z = 0 = P_Z - R_Z$

$R_Z = P_Z = \underline{9.33 \text{ LBS.}}$

$\Sigma F_Y = 0 = P_Y - R_Y$

$R_Y = P_Y = \underline{9.30 \text{ LBS.}}$

$\Sigma M_Z = 0 = 1.00 P_Y - 3.00 P_x + M_{A_1}$

$M_{A_1} = 3.00(18.86) - 1.00(9.30) = \underline{47.28 \text{ IN-LBS.}}$

$\Sigma M_x = 0 = 1.53 P_Y - 3.00 P_Z - M_{A_2}$

$M_{A_2} = 1.53(9.30) - 3.00(9.33) = \underline{-13.76 \text{ IN-LBS.}}$

$$M_{A_R} = \sqrt{M_{A_1}^2 + M_{A_2}^2} = \underline{49.24 \text{ IN-LBS.}}$$

$\Sigma M_Y = 0 = T - 1.53 P_x + 1.00 P_Z$

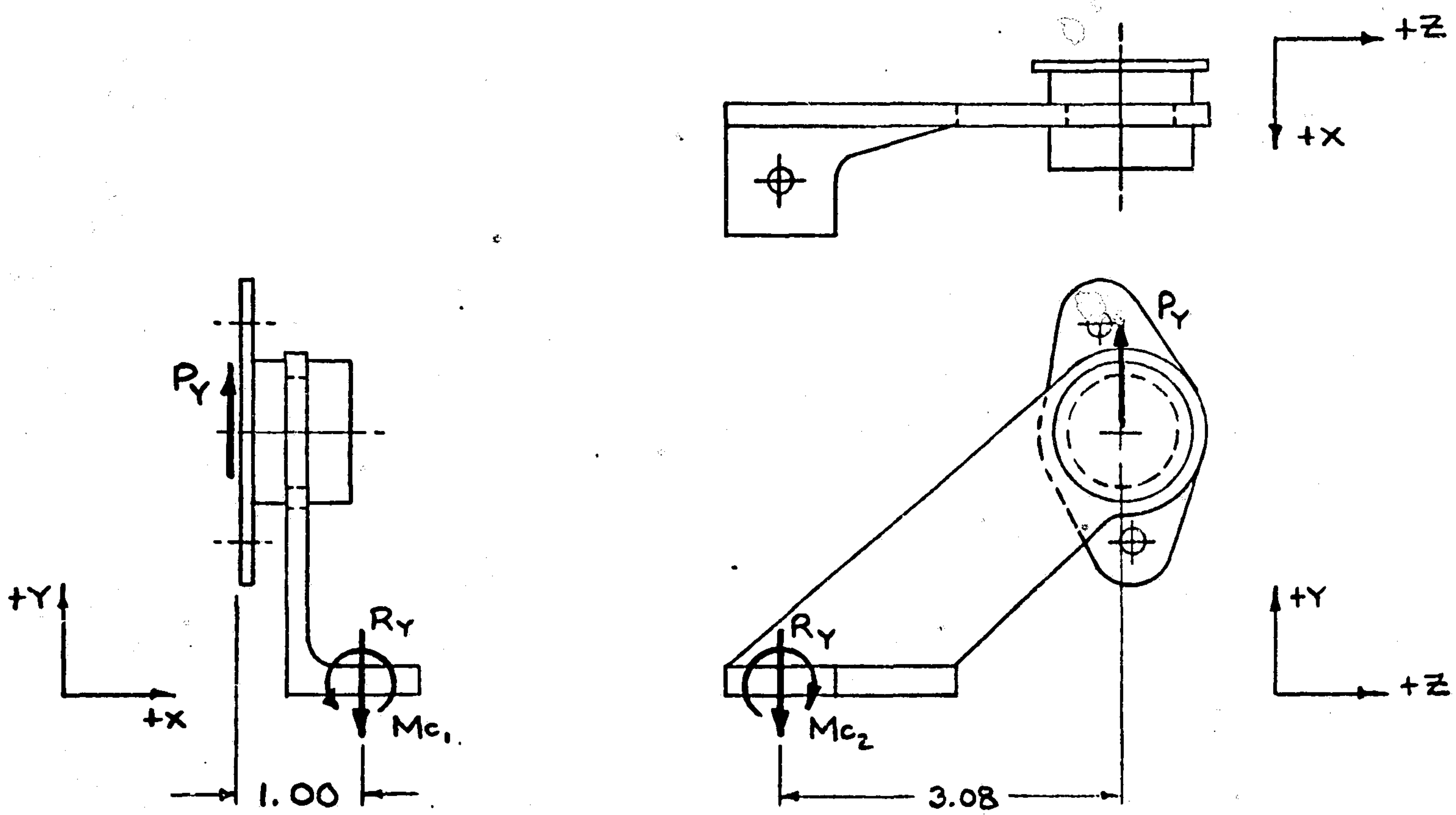
$T = 1.53(18.86) - 1.00(9.33) = \underline{19.53 \text{ IN-LBS.}}$

@ HOUSING, $P_{SHEAR} = \sqrt{R_Y^2 + R_Z^2} = \underline{13.17 \text{ LBS.}}$

@ NR BKT., $P_{SHEAR} = \sqrt{R_x^2 + R_Z^2} = \underline{21.04 \text{ LBS.}}$

11.3.5.2 FOR Y-AXIS LOADING.

11.3.5.2.1 MOUNTS C & D (MOUNT C SHOWN).



FOR STATIC LOADING

$$\Sigma F_Y = 0 = P_Y - R_Y$$

$$R_Y = P_Y = \underline{2.90 \text{ LBS.}}$$

$$\Sigma M_Z = 0 = Mc_1 - 1.00 P_Y$$

$$Mc_1 = 1.00(2.90) = \underline{2.90 \text{ IN-LBS.}}$$

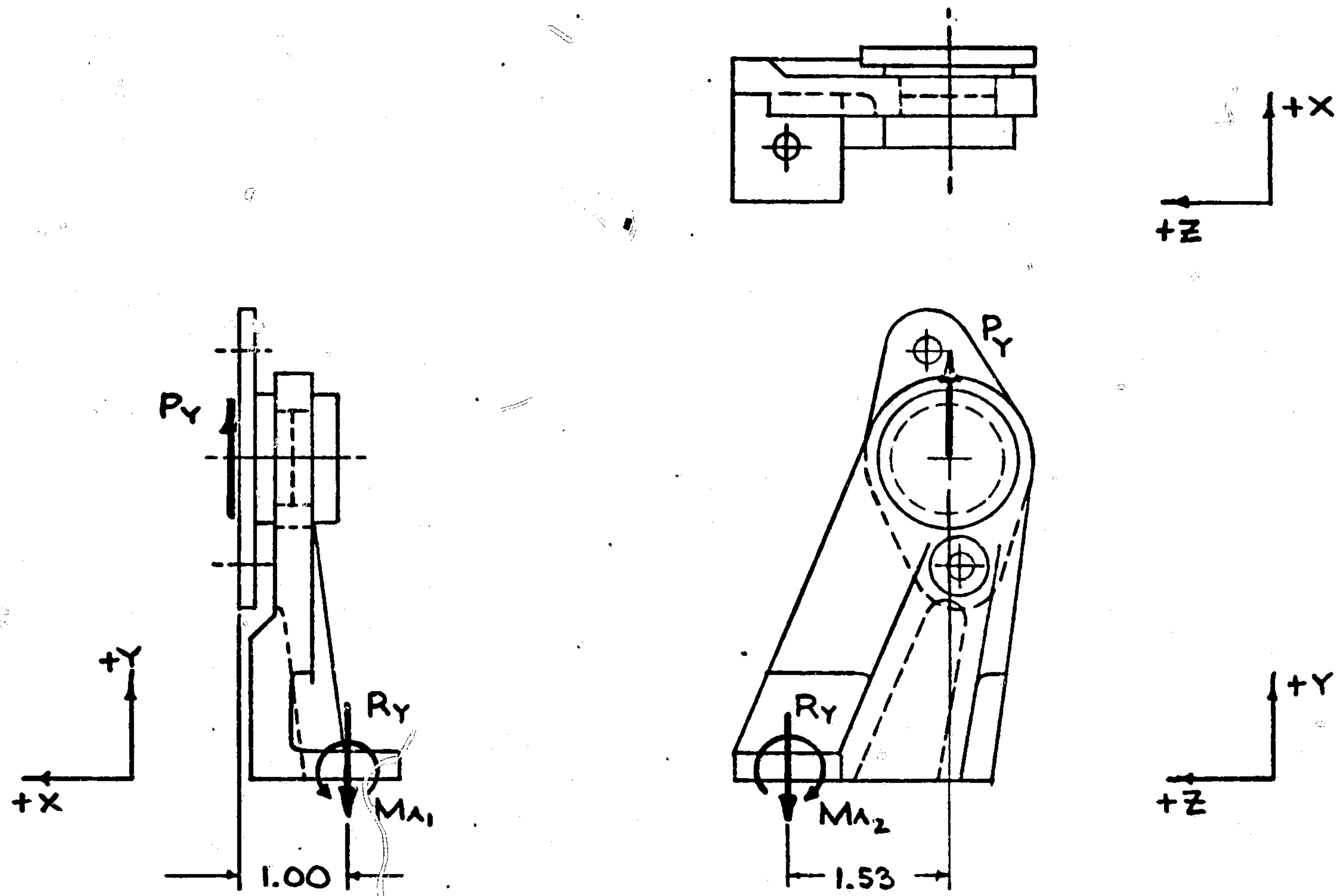
$$\Sigma M_X = 0 = Mc_2 - 3.08 P_Y$$

$$Mc_2 = 3.08(2.90) = \underline{8.93 \text{ IN-LBS.}}$$

$$Mc_R = \sqrt{Mc_1^2 + Mc_2^2} = \underline{9.39 \text{ IN-LBS.}}$$

FOR Y-AXIS LOADING.

11.3.5.2.2 MOUNTS A & B (MOUNT A SHOWN).



FOR STATIC LOADING

$$\Sigma F_Y = 0 = P_Y - R_Y$$

$$R_Y = P_Y = \underline{17.10 \text{ LBS.}}$$

$$\Sigma M_Z = 0 = 1.00 P_Y - M_{A_1}$$

$$M_{A_1} = 1.00(17.10) = \underline{17.10 \text{ IN-LBS.}}$$

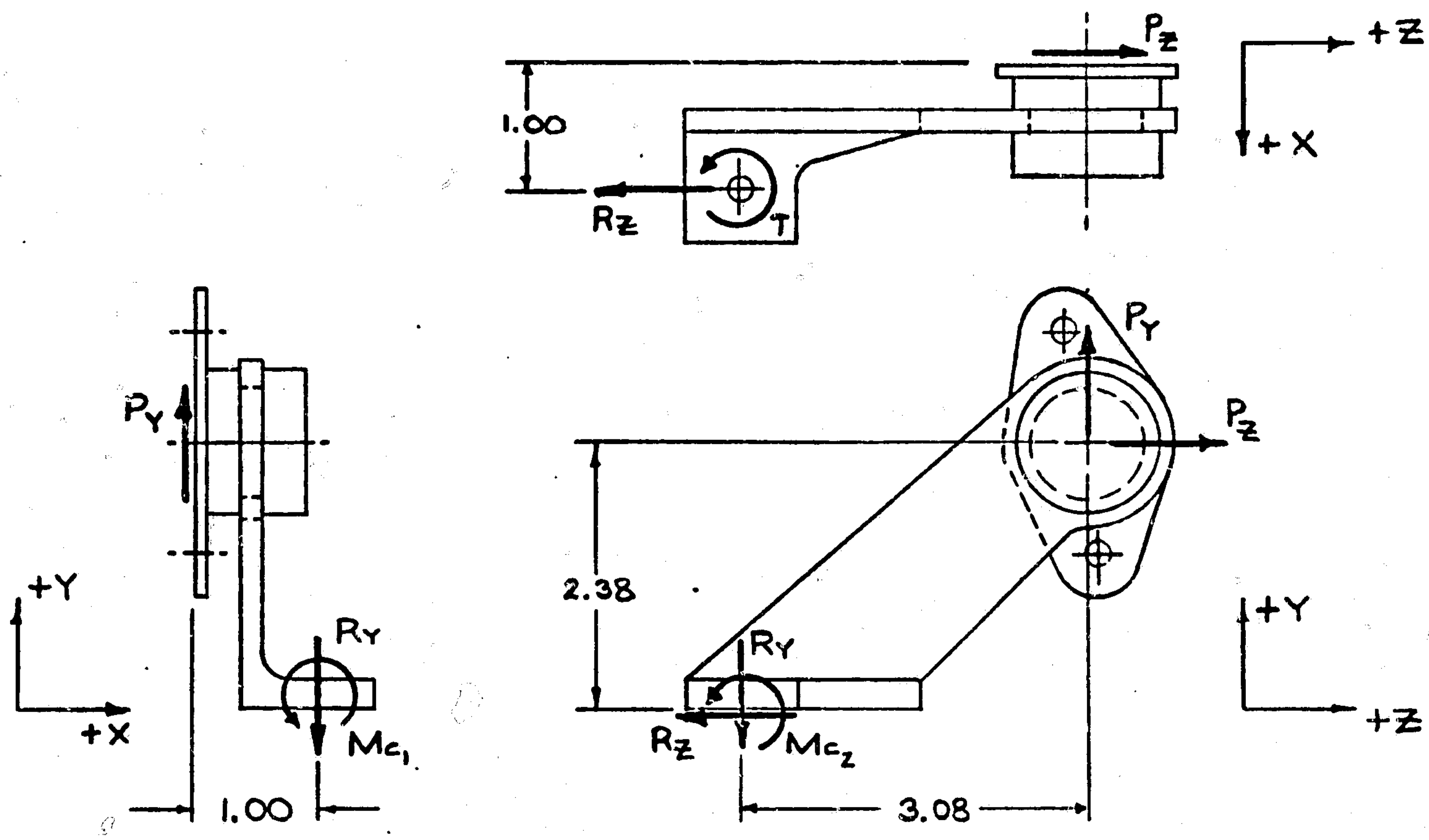
$$\Sigma M_X = 0 = 1.53 P_Y - M_{A_2}$$

$$M_{A_2} = 1.53(17.10) = \underline{26.16 \text{ IN-LBS.}}$$

$$M_{A_R} = \sqrt{M_{A_1}^2 + M_{A_2}^2} = \underline{31.26 \text{ IN-LBS.}}$$

11.3.5.3 FOR Z-AXIS LOADING.

11.3.5.3.1 MOUNTS C & D (MOUNT C SHOWN).



FOR STATIC LOADING

$\Sigma F_Y = 0 = P_Y - R_Y$

$R_Y = P_Y = \underline{4.23 \text{ LBS.}}$

$\Sigma F_Z = 0 = P_Z - R_Z$

$R_Z = P_Z = \underline{10.00 \text{ LBS.}}$

$\Sigma M_Z = 0 = M_{C_1} - 1.00 P_Y$

$M_{C_1} = 1.00(4.23) = \underline{4.23 \text{ IN-LBS.}}$

$\Sigma M_X = 0 = 2.38 P_Z - 3.08 P_Y - M_{C_2}$

$M_{C_2} = 2.38(10.00) - 3.08(4.23) = \underline{10.77 \text{ IN-LBS.}}$

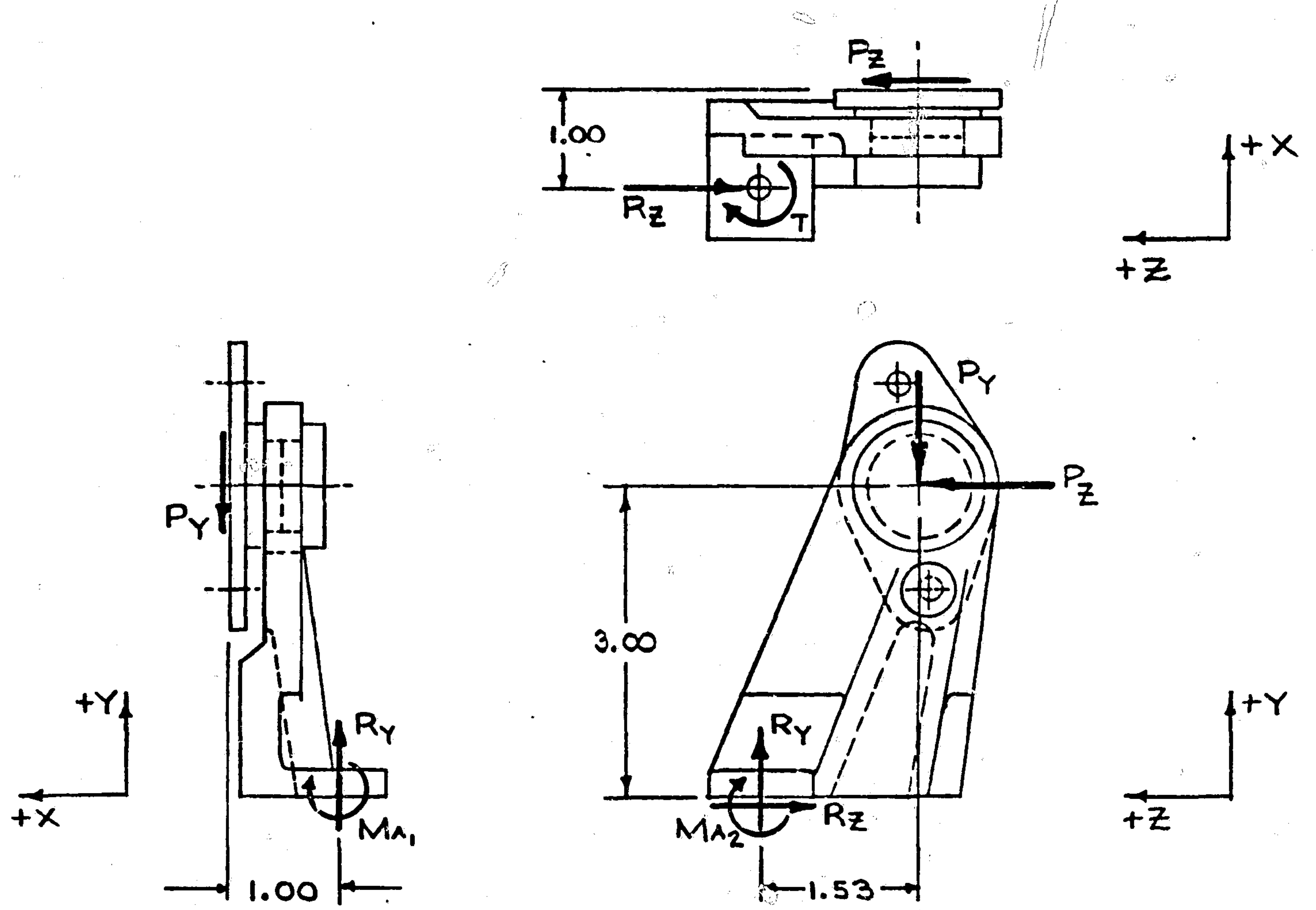
$$M_{C_R} = \sqrt{M_{C_1}^2 + M_{C_2}^2} = \underline{11.57 \text{ IN-LBS.}}$$

$\Sigma M_Y = 0 = 1.00 P_Z - T$

$T = 1.00(10.00) = \underline{10.00 \text{ IN-LBS.}}$

FOR Z-AXIS LOADING.

11.3.5.3.2 MOUNTS A & B (MOUNT A SHOWN).



FOR STATIC LOADING.

$\Sigma F_Y = 0 = R_Y - P_Y$

$R_Y = P_Y = \underline{4.23 \text{ LBS.}}$

$\Sigma F_Z = 0 = P_Z - R_Z$

$R_Z = P_Z = \underline{10.00 \text{ LBS.}}$

$\Sigma M_Z = 0 = M_{A_1} - 1.00 P_Y$

$M_{A_1} = 1.00(4.23) = \underline{4.23 \text{ IN-LBS.}}$

$\Sigma M_X = 0 = 1.53 P_Y - 3.00 P_Z + M_{A_2}$

$M_{A_2} = 3.00(10.00) - 1.53(4.23) = \underline{23.53 \text{ IN-LBS.}}$

$$M_{A_R} = \sqrt{M_{A_1}^2 + M_{A_2}^2} = \underline{23.91 \text{ IN-LBS.}}$$

$\Sigma M_Y = 0 = T - 1.00 P_Z$

$T = 1.00(10.00) = \underline{10.00 \text{ IN-LBS.}}$

11.3.6 SUMMARY OF CRITICAL STATIC LOADS.

AXIS	POINT*	SHEARING FORCE, LBS.	AXIAL FORCE, LBS.	RESULTANT MOMENT, M IN-LBS.	TWISTING MOMENT T IN-LBS.
X	A′	21.04	9.30	49.24	19.53
	B′	21.04	− 9.30	49.24	19.53
	C′	1.95	− 9.30	25.74	1.93
	D′	1.95	9.30	25.74	1.93
Y	A′	0	17.10	31.26	0
	B′	0	17.10	31.26	0
	C′	0	2.90	9.39	0
	D′	0	2.90	9.39	0
Z	A′	10.00	− 4.23	23.91	10.00
	B′	10.00	− 4.23	23.91	10.00
	C′	10.00	4.23	11.57	10.00
	D′	10.00	4.23	11.57	10.00

* POINTS ARE AT INTERFACE OF NORTHAMERICAN BRACKET & ISOLATOR MTG. FEET.

11.3.7 ATTACHMENT OF MOUNTING FEET TO HOUSING.

11.3.7.1 BOLT TENSION & JOINT FRICTION CAPACITY.

BOLT TENSION

EACH MOUNTING FOOT IS ATTACHED TO THE HOUSING BY TWO NAS1004 STAINLESS STEEL SCREWS. ($F_{TU_{MIN}}$ = 140000 psi). ASSUME A VALUE FOR F_{TY} = 100,000 psi. BASED ON A TENSILE STRESS AREA, A_T = 0.0364 in^2.

$$P_{YIELD} = F_{TY} \times A_T = 3640 \text{ LBS.}$$

ASSUME THE FASTENERS ARE TORQUED UP TO 90% OF YIELD

$$P' = 0.90(3640) = \underline{3276 \text{ LBS.}}$$

THE MAXIMUM APPLIED TENSILE LOAD RESULTS FROM X-AXIS LOADING AT POINTS A & B (REF. PAR. 11.3.4). THE STATIC LOAD PER FASTENER IS

$$P = (18.86/2) = 9.43 \text{ LBS.}$$

BASED ON A 1.15 YIELD LOAD FACTOR AND AN ACCELERATION OF 31.1g (REF. PG. 40)

$$P_{DYN} = 1.15(9.43)(31.1) = \underline{337 \text{ LBS. (YIELD)}}$$

$$M.S. = \frac{P'}{P_{DYN}} - 1 = \frac{3276}{337} - 1 = \underline{\text{LARGE}}$$

JOINT FRICTION CAPACITY

ASSUME A COEFFICIENT OF FRICTION μ = 0.30

FRICTION FORCE, $F' = \mu P' = 0.30(3276) = 983$ LBS.

JOINT FRICTION CAPACITY, $F_F = 2F' = \underline{1966 \text{ LBS.}}$

FROM PAR. 11.3.4 AND THE ACCELERATIONS TABULATED ON PG. 40, THE DYNAMIC SHEAR LOADS ARE:

$$P_{S_X} = 1.15\{(G_X = 31.1g)(P_S = 13.17 \text{ LBS})\} = \underline{471 \text{ LBS. (YIELD)}}$$

$$P_{S_Y} = 1.15\{(G_Y = 20.9g)(P_S = 17.10 \text{ LBS})\} = \underline{411 \text{ LBS. (YIELD)}}$$

$$P_{S_Z} = 1.15\{(G_Z = 31.4g)(P_S = \sqrt{(4.23)^2 + (10.00)^2} = 10.86 \text{ LBS.})\} = \underline{392 \text{ LBS (YIELD)}}$$

$$M.S. = \frac{F_F}{P_{S_Y}} - 1 = \frac{1966}{471} - 1 = \underline{\text{LARGE}}$$

BEARING STRESS IN HOUSING PAD.

ASSUME BEARING AGAINST WALL OF HOUSING IN PAD AREA

MAX. SHEAR FORCE, $(P_s)_{MAX.}$ = 471 LBS. (YIELD)

A_{br} = BEARING AREA = td

d = BOLT. DIAMETER = 0.25 IN

t = PAD DEPTH = 0.58 IN

$\therefore A_{br} = 0.25(0.58) = 0.145 \text{ IN.}^2$

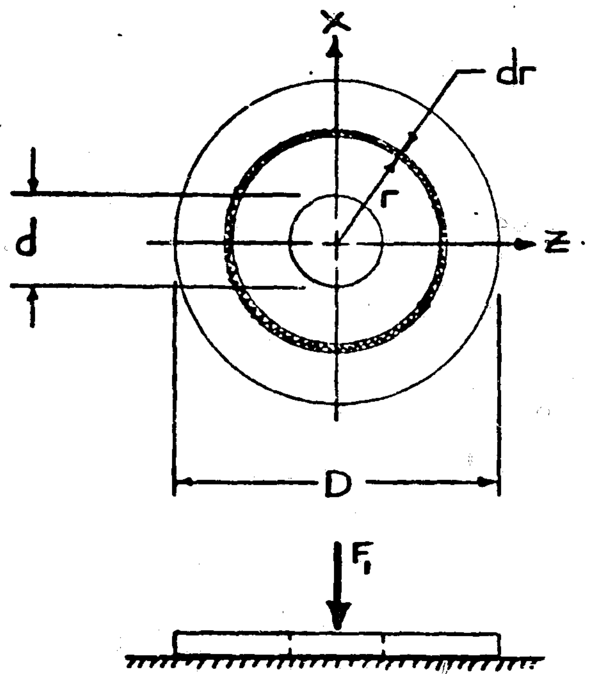
$$f_{bry} = \frac{(P_s)_{MAX}}{A_{br}} = \frac{471}{0.145} \approx \underline{3250 \text{ psi (YIELD)}}$$

FOR 356 AL. ALLOY CASTING, $(F_{bry})_{MIN\,(e/D=1.5)} = \underline{34000 \text{ psi}}$ (REF. C)

M.S. LARGE BY INSPECTION

11.3.8 MOUNTING FOOT FRICTION CAPACITY.

FORCE REQ'D. TO PREVENT SLIPPING DUE TO TORSION.



ASSUME THAT THE CONTACT AREA BETWEEN THE ISOLATOR MOUNTING FOOT AND THE NR MOUNTING BRACKET CAN BE APPROXIMATED BY A CIRCULAR AREA OF DIAMETER D = 1.000 IN. THE INSIDE DIAMETER, d, WILL BE TAKEN AS THE SINGLE HOLE CLEARANCE DIAMETER FOR THE SCREW USED IN THE ATTACHMENT. THE FORCE F_1 WILL BE DEFINED AS THAT PART OF THE FASTENER PRELOAD REQUIRED TO PREVENT ROTATION IN THE X-Z PLANE DUE TO A TORQUE ABOUT THE Y-AXIS. IT IS ASSUMED THAT THE PROBLEM IS ANALOGOUS TO THAT OF A FRICTION DISK SUBJECTED TO UNIFORM WEAR. THE MAXIMUM PRESSURE FOR SUCH A DISK OCCURS AT $r = d/2$.

LET p_a = MAXIMUM PRESSURE @ $r = d/2$

$$\therefore p = p_a \frac{(d/2)}{r} \qquad \text{(REF. O)}$$

WHERE p IS PRESSURE AT ANY RADIUS r.

AREA OF CIRCUMFERENTIAL ELEMENT, $dA = 2\pi r\,dr$

$\therefore$ NORMAL FORCE ON THIS ELEMENT, $dF_1 = p\,dA$

$$dF_1 = \left\{p_a \frac{(d/2)}{r}\right\}(2\pi r\,dr) = (\pi p_a d)\,dr$$

$$F_1 = \int_{d/2}^{D/2} (\pi p_a d)\,dr = (\pi p_a d)\int_{d/2}^{D/2} dr = \frac{\pi p_a d}{2}(D-d)$$

FRICTION FORCE = μdF_1

TORQUE, $dT = (\mu dF_1)r = (\mu \pi p_a d)\,r\,dr$

$$T = \mu \pi p_a d \int_{d/2}^{D/2} r\,dr = \frac{\mu \pi p_a d}{8}(D^2 - d^2) = \frac{\mu \pi p_a d}{8}(D+d)(D-d)$$

BUT $p_a = \dfrac{2F_1}{\pi d(D-d)}$

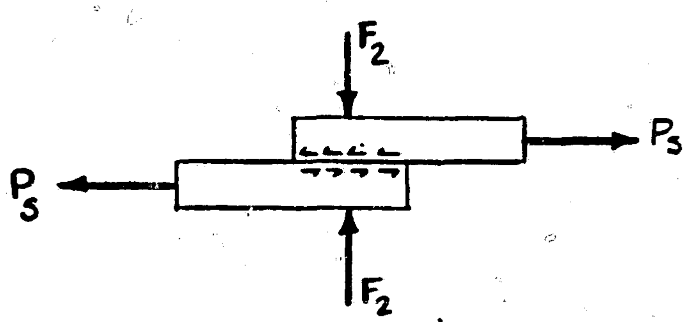
$$\therefore T = \frac{\mu d \pi}{8}(D+d)(D-d)\frac{2F_1}{\pi d(D-d)} = \frac{\mu F_1}{4}(D+d)$$

$$\boxed{F_1 = \frac{4T}{\mu(D+d)}}$$

ASSUMING A 9/32 DIAM. (0.281") CLEARANCE HOLE FOR A 1/4" DIA. SCREW AND A COEFFICIENT OF FRICTION $\mu = 0.35$ BETWEEN MATING SURFACES

$$F_1 = \frac{4T}{0.35(1.000 + 0.281)} = \underline{8.92T}$$

FORCE REQ'D. TO PREVENT SLIPPING DUE TO SHEAR.



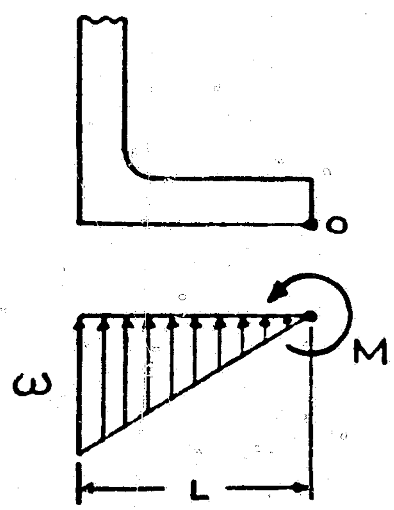
LET F_2 BE THAT PART OF THE TOTAL PRELOAD REQUIRED TO DEVELOP FRICTION FORCES LARGE ENOUGH TO PREVENT SLIPPAGE RELATIVE TO THE TWO SURFACES IN CONTACT AS A RESULT OF SHEARING FORCES P_S.

$$\mu F_2 = P_s$$

AGAIN ASSUMING $\mu = 0.35$

$$F_2 = (P_s/\mu) = \underline{2.86P_s}$$

FORCE REQ'D. TO PREVENT SLIPPING DUE TO AN OVERTURNING MOMENT.



ASSUME THE MOUNTING FOOT IS SUBJECTED TO AN OVERTURNING MOMENT AT POINT "O" OF MAGNITUDE M. LET F_3 BE THAT PART OF THE TOTAL PRELOAD REQUIRED TO PREVENT SEPARATION OF THE TWO SURFACES DUE TO M.

$$\Sigma M_o = 0 = M - \frac{1}{2}\omega L\left(\frac{2}{3}L\right)$$

BUT $\frac{1}{2}\omega L = F_3$

$$\therefore\ 0 = M - \frac{2}{3}F_3 L$$

$$F_3 = 1.50(M/L)$$

IF L = D = 1.250 IN.

$$F_3 = \underline{1.20M}$$

THUS, THE TOTAL PRELOAD REQUIRED TO MAINTAIN THE INTEGRITY OF THE JOINT IS

$$\Sigma F = F_1 + F_2 + F_3 = \underline{8.92T + 2.86P_s + 1.20M}$$

11.3.9 ATTACHMENT OF MOUNTS TO NR INTERFACE BRACKET.

MOUNTS A & B ARE CRITICAL FOR X-AXIS LOADING. THE APPLIED LOADS ARE

AXIAL FORCE, P_A	= 9.30 LBS.	REF. SECT. 11.3.6
SHEARING FORCE, P_S	= 21.04 LBS.	
OVERTURNING MOMENT, M	= 49.24 IN-LBS.	
TWISTING MOMENT, T	= 19.53 IN-LBS.	

FROM SECTION 11.3.8 THE TOTAL PRELOAD REQUIRED TO MAINTAIN THE INTEGRITY OF THE JOINT IS

$$\Sigma F = 8.92T + 2.86P_S + 1.20M$$

NOTE: BECAUSE OF THE CRITICALITY OF THIS ATTACHMENT AND THE USE OF FRICTION TO TRANSMIT LOAD SOME OF THE CONSERVATISM OF THE PREVIOUS ANALYSES (AND THOSE TO FOLLOW) MUST NOW BE ABANDONED. THE RESONANT FREQUENCY OF THE SINGLE-DEGREE OF FREEDOM SYSTEM ON THE ISOLATOR "SPRING" IS APPROXIMATELY 75 HZ. (INSTEAD OF THE 90 HZ.) AND THE ACTUAL TRANSMISSIBILITY, Q, IS 3.5 INSTEAD OF 10.0. THE 3σ-RANDOM VIBRATION RESPONSE IS MORE CRITICAL THAN THE SHOCK RESPONSE UNDER THESE CONDITIONS. THE REVISED RESPONSE IS

$$G_x = 3\sqrt{\frac{\pi}{2} f_n Q S_o} = 3\sqrt{\frac{\pi}{2}(75)(3.5)(0.075)}$$

$$G_x = \underline{16.7\,g}$$

BASED ON A 1.15 YIELD LOAD FACTOR AND AN ACCELERATION OF 16.7g, THE REQUIRED PRELOAD ΣF FOR DYNAMIC LOADING IS

$$\Sigma F = 1.15(16.7)\{8.92(19.53) + 2.86(21.04) + 1.20(49.24)\}$$
$$\Sigma F = \underline{5635 \text{ LBS.}}$$

THE FASTENER USED IS A 1/4" - 28 UNF BOLT FOR WHICH F_{ty} = 185000 PSI. THE BOLT HAS A TENSILE STRESS AREA, A_T = 0.0364 $IN.^2$ AND WILL BE TORQUED TO 90% OF YIELD. THE PRELOAD AXIAL FORCE IS

$$P_p = 0.9[185000 \times 0.0364] = \underline{6060 \text{ LBS.}}$$

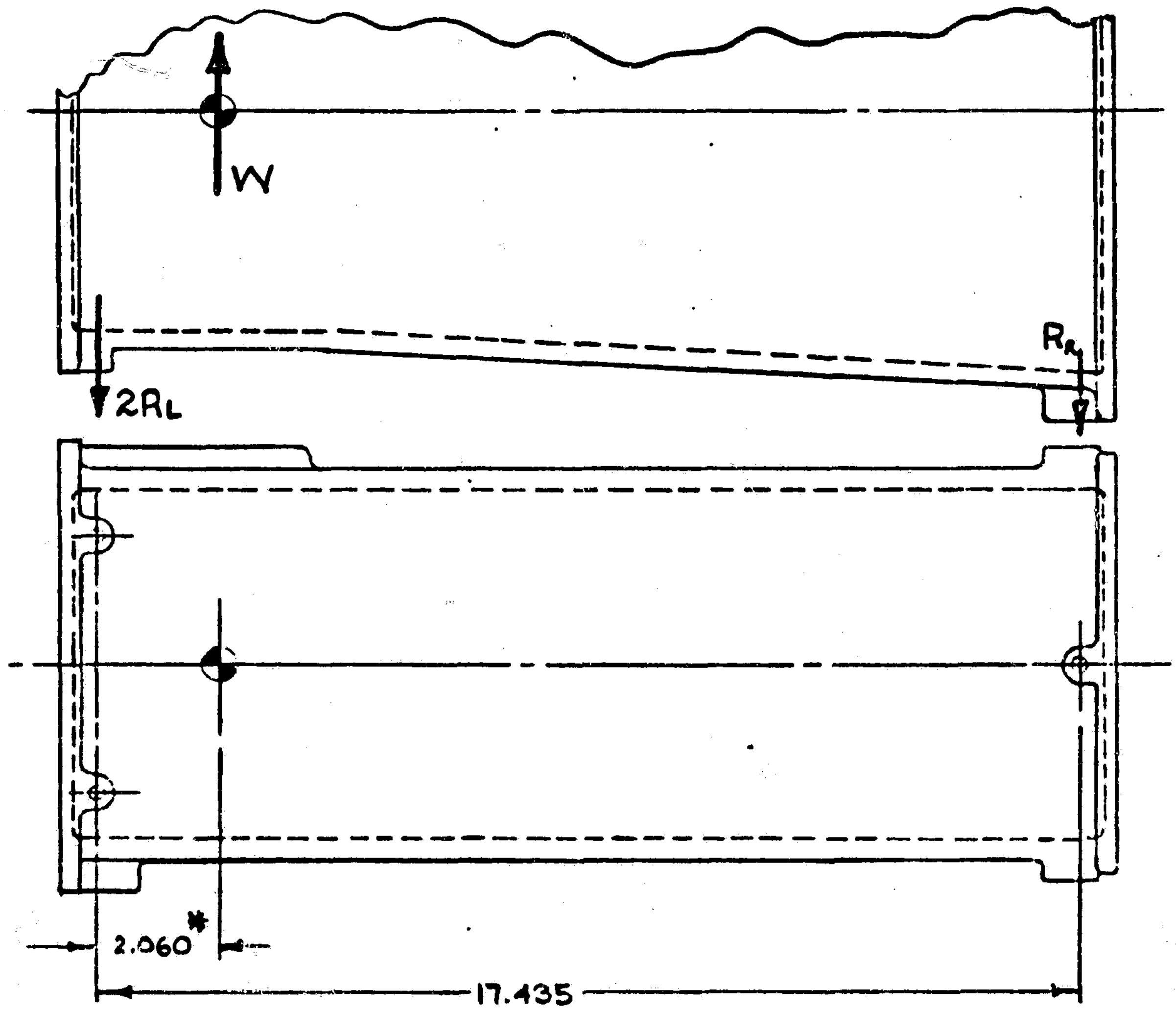
$$M.S. = \frac{P_p}{\Sigma F} - 1 = \frac{6060}{5635} - 1 = \underline{+0.075}$$

11.4 ATTACHMENT OF AFT PLATE TO HOUSING.

FROM PAGE 127 IT IS SEEN THAT THE WEIGHT OF THE AFT PLATE AND ITS MOUNTED COMPONENTS, P_2, IS MUCH LESS THAN THAT OF THE FORWARD PLATE AND ITS MOUNTED COMPONENTS, P_1. THE NUMBER OF PICK-UP POINTS AND THE FASTENER SIZE IS IDENTICAL FOR BOTH ATTACHMENTS. THUS, THE AFT PLATE ATTACHMENT LOADS ARE LOWER THAN THE FORWARD PLATE ATTACHMENT LOADS FOR WHICH AMPLE MARGINS OF SAFETY WERE EXHIBITED. THEREFORE,

M.S. LARGE BY INSPECTION

11.5 STRESSES DUE TO HANDLING LOADS.



* ESTIMATED

FROM STATICS

$$\Sigma F = 0 = W - 2R_L - R_R$$

$$2R_L + R_R = W$$

$$\Sigma M = 0 = 2.060W - 17.435R_R$$

$$17.435R_R = 2.060W$$

$$R_R = 0.118W$$

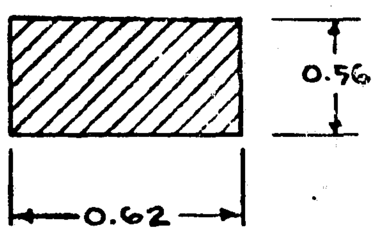
$$R_L = 0.441W$$

ASSUMING A 40 LB. SPECTROMETER AND A 2g HANDLING LOAD, THE MAX. LUG LOAD IS.

$$R_L = 0.441(40)(2) = \underline{35.3 \text{ LBS.}}$$

CONSERVATIVELY ASSUME THE LUG TO BE A CANTILEVER WITH A MOMENT ARM $e = 0.28$ IN.

$$\therefore M = R_L \times e$$
$$M = 35.3(0.28)$$
$$M = 9.88 \text{ IN-LBS.}$$



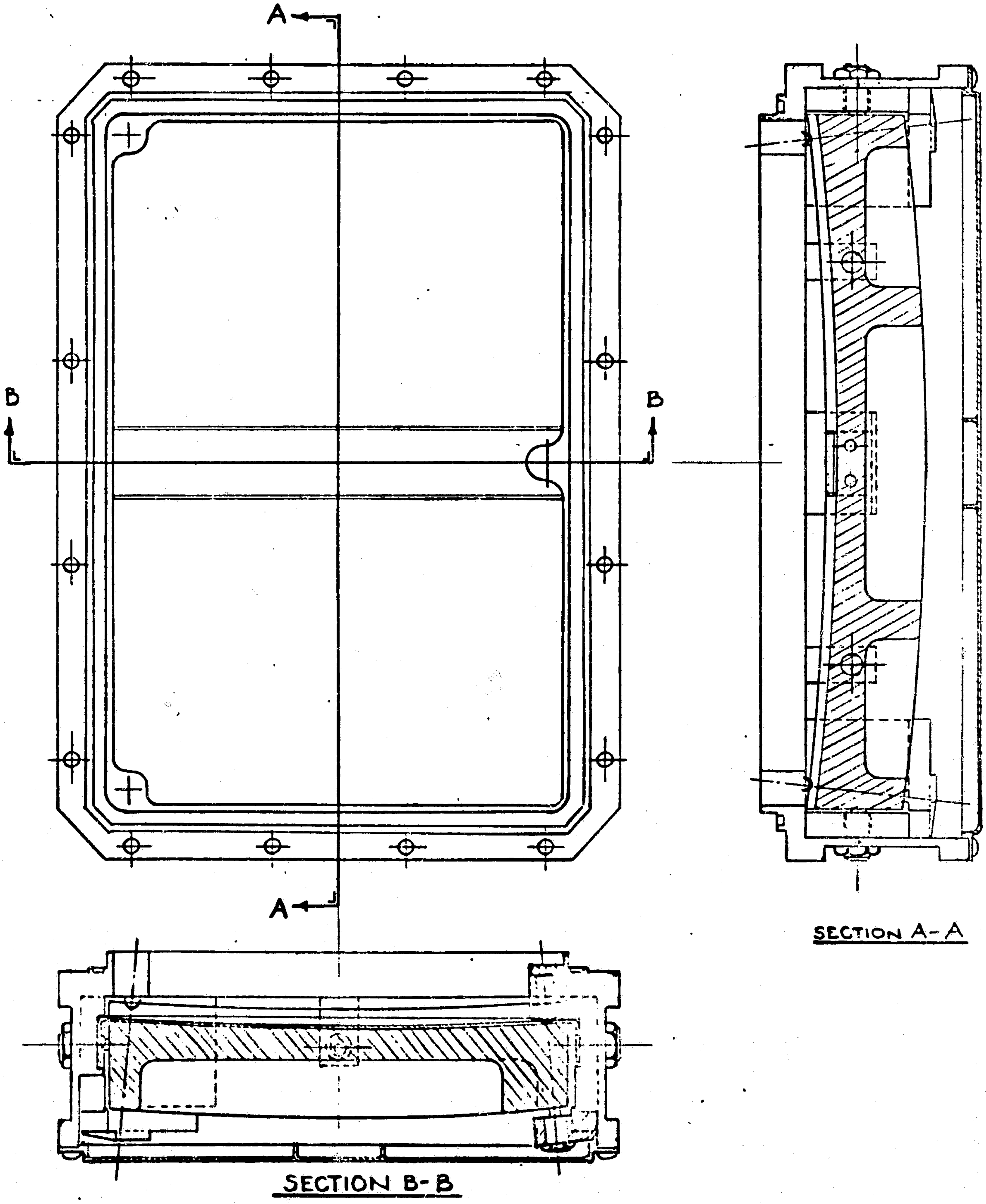
BENDING STRESS

$$f_b = \frac{6M}{bt^2} = \frac{6(9.88)}{0.62(0.56)^2}$$
$$f_b \approx 300 \text{ PSI}$$

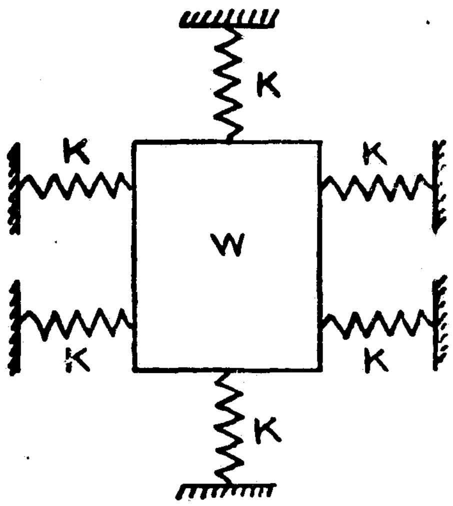
M.S. LARGE BY INSPECTION

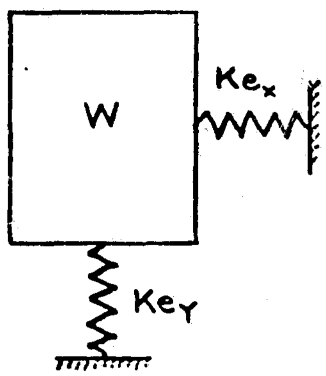
12.0 ANALYSIS OF MIRROR

(REF. RAY LEE DWG. 7232-0150 "MIRROR CELL ASS'Y.")



12.1 ANALYSIS OF EBERT MIRROR



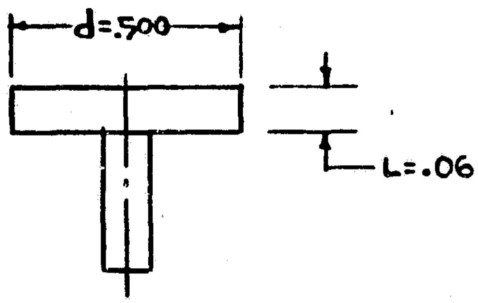


THE SCHEMATIC REPRESENTATION OF THE MIRROR SUPPORT FOR X AND Y AXES LOAD-ING IS SHOWN AT THE LEFT. IMMEDIATELY BELOW IS THE EQUIVALENT SPRING-MAS SYSTEM IN WHICH FOR SPRINGS IN PARALLEL $Ke_x = 4K$ AND $Ke_Y = 2K$ (K BEING THE "SPRING CONSTANT" FOR AN INDIVIDUAL PAD

ASSUMPTIONS:

1. FROM PG. 41 THE MIRROR ACCELERATI ARE AS FOLLOWS:
   $G_X = 34.2\ g$
   $G_Y = 21.0\ g$
   $G_Z = 34.1\ g$
2. MIRROR WEIGHT, W = 3.1 LBS.
3. FOR EXCITATION IN THE PLANE OF THE MIRROR, FRICTION FORCES NEGLIGI
4. KEL-F-81 PADS OR "SPRINGS" PRELOA

12.1.1 DETERMINATION OF SPRING CONSTANT.



PAD (7232-0160-3)

$$A = \frac{\pi}{4} d^2 = 0.7854 (0.500)^2$$

$A = 0.196$ IN.^{2}

$L = 0.06$ IN.

$E_{(KEL-F-81)} = 180{,}000$ PSI (R...

$$K = \frac{AE}{L} = \frac{0.196(180000)}{0.06}$$

$K = 5.88 \times 10^5$ LBS./IN.

$Ke_x = 4(5.88 \times 10^5) = \underline{2.352 \times 10^6 \text{ LBS./IN.}}$

$Ke_Y = 2(5.88 \times 10^5) = \underline{1.176 \times 10^6 \text{ LBS./IN.}}$

12.1.2 DYNAMIC LOADS

$$(W_{DYN.})_X = 1.5(3.1\text{ LBS.})(34.2g) = \underline{159.0\text{ LBS. (ULT.)}}$$
$$(W_{DYN.})_Y = 1.5(3.1\text{ LBS.})(21.0g) = \underline{97.7\text{ LBS. (ULT.)}}$$

DYNAMIC DISPLACEMENT

$$F = ma = (W_{DYN}) = KX$$

OR $X = (W_{DYN})/K$

$$X_X = \frac{(W_{DYN})}{Ke_X} = \frac{159.0}{2.352\times10^6} = 67.60\times10^{-6}\text{ IN.}$$

$$X_Y = \frac{(W_{DYN})}{Ke_Y} = \frac{97.7}{1.176\times10^6} = 83.08\times10^{-6}\text{ IN.}$$

ASSUME PRELOADING 25% ABOVE DYNAMIC DEFLECTION

$$(X_X)_P = 1.25(67.60\times10^{-6}) = 84.50\times10^{-6}\text{ IN.}$$
$$(X_Y)_P = 1.25(83.08\times10^{-6}) = 103.85\times10^{-6}\text{ IN.}$$

CORRESPONDING PRELOAD PER "SPRING"

$$(P_X)_P = (5.88\times10^5)(8.450\times10^{-5}) = \underline{49.7\text{ LBS.}}$$
$$(P_Y)_P = (5.88\times10^5)(10.385\times10^{-5}) = \underline{61.1\text{ LBS.}}$$

12.1.2.1 FOR X-AXIS LOADING

THE LOAD PER SPRING IS

$$(P_X)_D = [(W_{DYN})_X/4] = \underline{39.8\text{ LBS.}}$$

THUS, TOTAL LOAD ON TWO OF THE SPRINGS IS

$$\Sigma P_{X_I} = (P_X)_P + (P_X)_D = 49.7 + 39.8 = \underline{89.5\text{ LBS.}}$$

THE TOTAL LOAD ON THE OTHER TWO SPRINGS IS

$$\Sigma P_{X_{II}} = (P_X)_P - (P_X)_D = 49.7 - 39.8 = \underline{9.9\text{ LBS.}}$$

12.1.2.2 FOR Y-AXIS LOADING

THE LOAD PER SPRING IS

$$(P_Y)_D = [(W_{DYN})_Y / 2] = \underline{48.8 \text{ LBS.}}$$

THUS, TOTAL LOAD ON TWO OF THE SPRINGS IS

$$\Sigma P_{Y_I} = (P_Y)_P + (P_Y)_D = 61.1 + 48.8 = \underline{109.9 \text{ LBS.}}$$

THE TOTAL LOAD ON THE OTHER TWO SPRINGS IS

$$\Sigma P_{Y_{II}} = (P_Y)_P - (P_Y)_D = 61.1 - 48.8 = \underline{12.3 \text{ LBS.}}$$

12.1.2.2.1 COMPRESSIVE STRESS IN PAD.

THE MAXIMUM LOAD ON ANY ONE PAD IS 109.9 LBS.

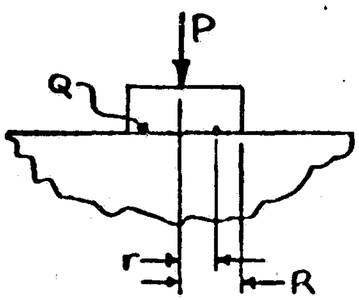
$$\therefore f_c = \frac{109.9}{0.196} = \underline{560 \text{ PSI (ULT.)}}$$

FOR KEL-F-81, F_{CY} = 5440 PSI (REF. 2)

$$\therefore M.S. = \frac{F_{CY}}{f_c} - 1 = \frac{5440}{560} - 1 = \underline{\text{LARGE}}$$

12.1.2.2.2 CONTACT STRESS IN MIRROR.

CONSERVATIVELY ASSUME THE PAD TO BE A RIGID CYLINDRICAL DIE OF RADIUS R UNDER TOTAL LOAD P = 109.9 LBS. THIS CORRESPONDS TO CASE 12, TABLE XIV OF REF. K.



AT ANY POINT Q ON THE SURFACE OF CONTACT, THE COMPRESSIVE STRESS IS GIVEN BY

$$f_c = \frac{P}{2\pi R\sqrt{R^2 - r^2}}$$

AT $r = 0$, $f_c = (f_c)_{MIN.} = P/2\pi R^2$

AT $r = R$, $f_c = (f_c)_{MAX.} = \infty$

CONSERVATIVELY ASSUME STRESS AT $r = 0.8R$ IS AVERAGE OVER ENTIRE CONTACT SURFACE

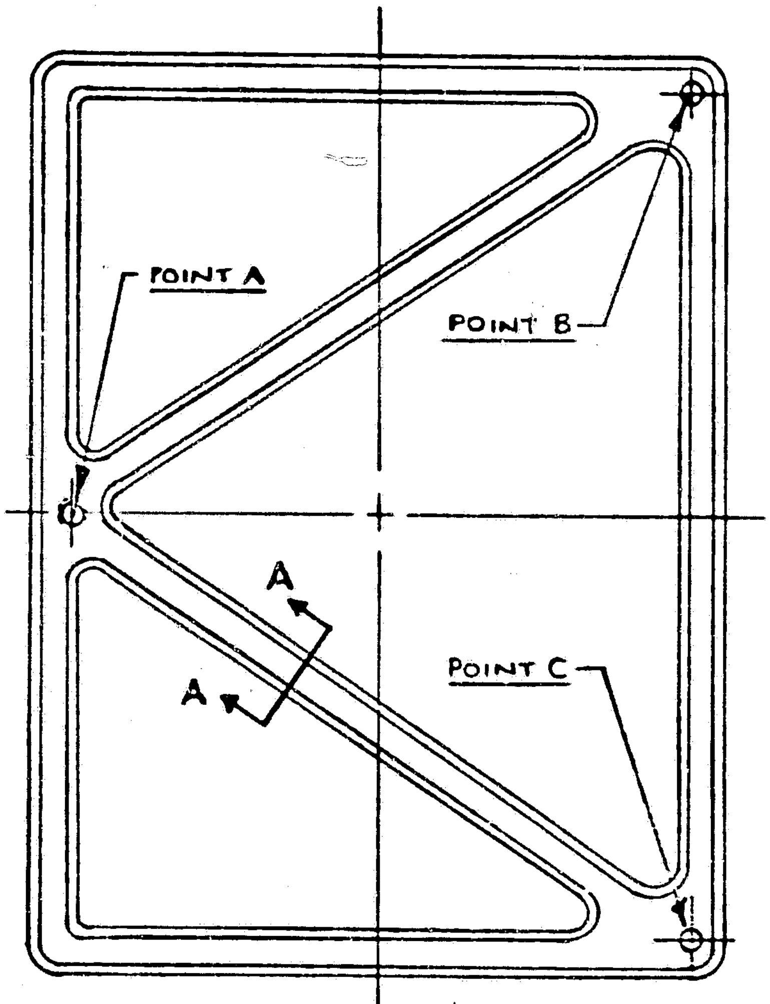
$$f_c = \frac{P}{2\pi R\sqrt{R^2-(0.8R)^2}} = \frac{P}{1.2\pi R^2} = \frac{109.9}{1.2\pi(0.25)^2} = \underline{466 \text{ PSI (ULT.)}}$$

CONSERVATIVELY ASSUME A COMPRESSIVE ALLOWABLE, F_c = 2500 PSI

$$M.S. = \frac{F_c}{f_c} - 1 = \frac{2500}{466} - 1 = \underline{\text{LARGE}}$$

12.1.2.3 Z-AXIS LOADING.

12.1.2.3.1 BENDING STRESS IN MIRROR.



MAT'L. CER-VIT (OWENS-ILLINOIS)

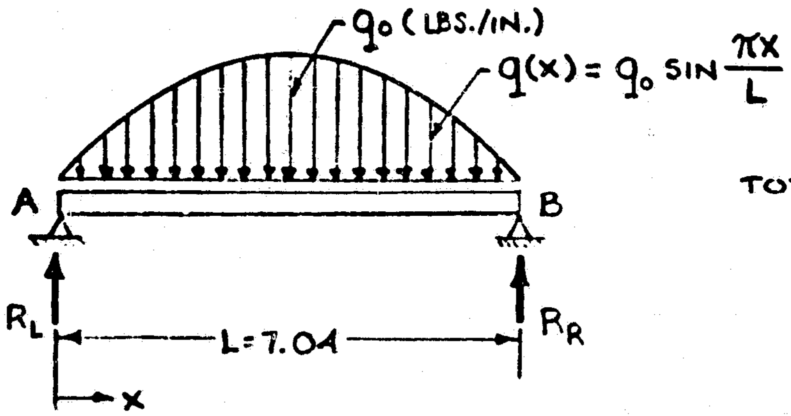
FOR Z-AXIS ACCELERATION, THE MIRROR IS SUPPORTED BY THREE PAIRS OF PARALLEL PRELOADED PADS OR "SPRINGS" AT POINTS A, B, AND C. IT IS ASSUMED THAT THE MIRROR CAN BE REPRESENTED BY THREE SIMPLY SUPPORTED BEAMS BETWEEN REACTION POINTS A, B AND C. IN THIS MODEL, ASSUME THAT BOTH MEMBERS AB AND AC CARRY 25% OF THE TOTAL MIRROR WEIGHT DUE TO THE OVERHUNG TRIANGULAR SECTIONS. THE REMAINING 50% OF THE MIRROR WEIGHT IS CONSIDERED TO BE DIVIDED EQUALLY BETWEEN ALL THREE LOAD CARRYING MEMBERS. THUS,

$$W_{AB} = (1/3)(0.50)(3.1)+(0.25)(3.1) = \underline{1.29 \text{ LBS.}}$$

$$W_{BC} = (1/3)(0.50)(3.1) = \underline{0.52 \text{ LBS.}}$$

$$W_{CA} = (1/3)(0.50)(3.1)+(0.25)(3.1) = \underline{1.29 \text{ LBS.}}$$

ASSUME THE LOAD TO BE DISTRIBUTED SINUSOIDALLY



TOTAL SHEAR, $V = \int q(x)\,dx$

$$V = \int_0^L q_o \sin(\pi x/L)\,dx$$

$$V = -\frac{q_o L}{\pi}\left[\cos\frac{\pi x}{L}\right]_0^L = \frac{2q_o L}{\pi}$$

$$V = (W_{AB})_{DYN} = 1.5(1.29)(34.1)$$

$$V = \underline{65.98 \text{ LBS. (ULT.)}}$$

$(2q_oL/\pi) = 65.98$

$$q_o = \frac{65.98\pi}{2L} = \frac{65.98(3.14)}{2(7.04)} = \underline{14.72 \text{ LBS/IN. (ULT.)}}$$

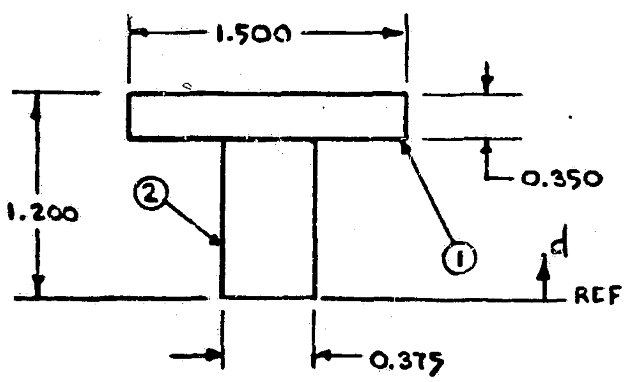
BENDING MOMENT, $M_x = \int V dx = -\int_0^L \frac{q_oL}{\pi}\cos\frac{\pi x}{L}dx = -\frac{q_oL^2}{\pi^2}\sin\frac{\pi x}{L}$

THE MAX. MOMENT OCCURS AT $x = (L/2)$

$$M = |M_{MAX}| = (q_oL^2/\pi^2) = \frac{14.72(7.04)^2}{(3.14)^2}$$

$$M = \underline{73.9 \text{ IN-LBS. (ULT.)}}$$

SECTION A-A



ITEM	DESCRIPTION	AREA, A	d	Ad	Ad²	Io
1	1.500 × 0.350	0.525	1.025	0.538	0.551	0.005
2	0.375 × 0.850	0.319	0.425	0.136	0.058	0.019
	Σ	0.844		0.674	0.609	0.024

$\bar{d} = C = (\Sigma Ad/\Sigma A) = (0.674/0.844) = \underline{0.799 \text{ IN.}}$

$I = \Sigma I_o + \Sigma Ad^2 - \bar{d}\cdot\Sigma Ad = 0.024 + 0.609 - 0.539 = \underline{0.094 \text{ IN}^4}$

$Z = (I/C) = (0.094/0.799) = \underline{0.118 \text{ IN}^3}$

BENDING STRESS, $f_b = \frac{M}{Z} = \frac{73.9}{0.118} = \underline{626 \text{ psi (ULT.)}}$

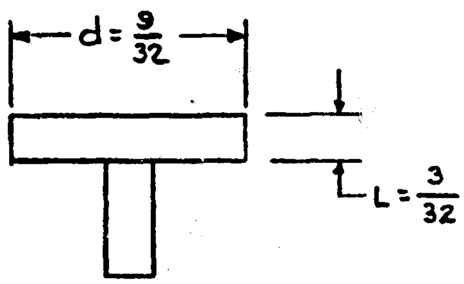
FOR A MINUMUM TENSILE ALLOWABLE STRESS OF 1,000 psi (AS PER MANUFACTURER'S SUGGESTION)

$$M.S. = \frac{F_T}{f_b} - 1 = \frac{1000}{626} - 1 = \underline{+0.60}$$

12.1.2.3.2 COMPRESSIVE STRESS IN PAD

THE TOTAL SHEAR WAS SHOWN TO BE 65.98 LBS. HALF OF THIS LOAD IS APPLIED AT POINT A ON BEAM AB. A SIMILAR LOAD IS APPLIED AT POINT A DUE TO THE LOADING OF BEAM AC. THUS THE TOTAL DYNAMIC LOAD ON THE PAD AT POINT A IS

$$(P_A)_D = 2\left\{\frac{1}{2}(65.98)\right\} = \underline{65.98 \text{ LBS.}}$$



PAD 7232-0156-3

$$A = \frac{\pi}{4} d^2 = 0.7854\,(9/32)^2 = 0.0621 \text{ IN}^2$$

$$E_{(KEL\text{-}F)} = 180000 \text{ PSI}$$

$$L = (3/32)''$$

$$K = \frac{AE}{L} = \frac{0.0621\,(180000)}{0.09375}$$

$$K = 1.19 \times 10^5 \text{ LBS./IN.}$$

$$Ke_z = 2\left\{3(1.19 \times 10^5)\right\} = 7.14 \times 10^5 \text{ LBS./IN.}$$

DYNAMIC DISPLACEMENT

$$x_z = \left[(P_A)_D / K\right]$$

$$(P_A)_D = 65.98 \text{ LBS. (ULT.)}$$

$$\therefore\ x_z = (65.98)/(1.19 \times 10^5) = 55.45 \times 10^{-5} \text{ IN.}$$

ASSUMING 25% ABOVE X_z FOR PRELOADING

$$(X_z)_P = 1.25\,(55.45 \times 10^{-5}) = 69.31 \times 10^{-5} \text{ IN.}$$

CORRESPONDING PRELOAD, $(P_A)_P = (1.19 \times 10^5)(69.31 \times 10^{-5})$

$$(P_A)_P = \underline{76.05 \text{ LBS.}}$$

TOTAL COMPRESSIVE LOAD

$$(P_A) = (P_A)_D + (P_A)_P$$

$$(P_A) = 65.98 + 76.05 = \underline{142.03 \text{ LBS}}$$

$$f_c = \frac{(P_A)}{A} = \frac{142.03}{0.0621} = \underline{2290 \text{ PSI (ULT.)}}$$

$$F_{CY} = 5440 \text{ PSI (REF. 2)}$$

$$M.S. = \frac{F_{CY}}{f_c} - 1 = \frac{5440}{2290} - 1 = \underline{\text{LARGE}}$$

12.1.2.3.3 CONTACT STRESS IN MIRROR

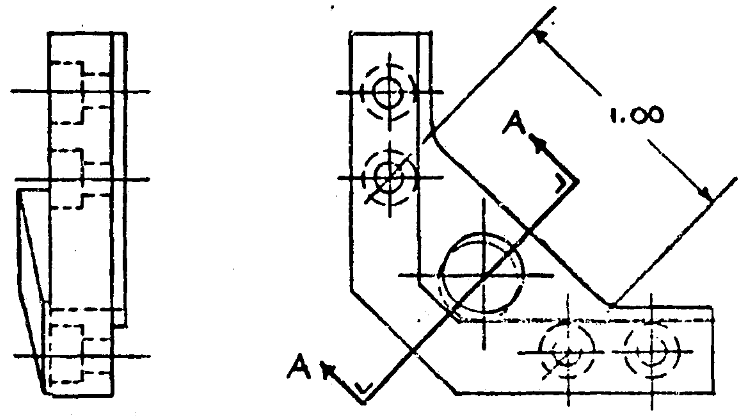
CONSERVATIVELY ASSUMING

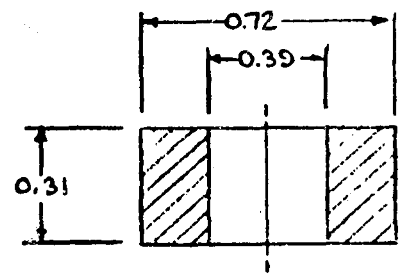
$$f_c = \frac{(P_A)}{1.2\pi R^2} \qquad \text{(REF. PG. 156)}$$

$$f_c = \frac{142.03}{1.2\pi(0.1406)^2} = \underline{1900 \text{ psi (ULT.)}}$$

$$\text{M.S.} = \frac{F_c}{f_c} - 1 = \frac{2500}{1900} - 1 = \underline{+0.32}$$

12.2 STRESS IN MIRROR CLAMP (SIDE).





SECTION A-A (CONSERVATIVELY MODIFIED FOR ANALYSIS)

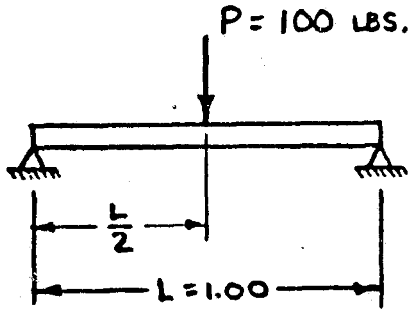
THE MAXIMUM LOAD APPLIED TO THE SIDE MIRROR CLAMP IS DETERMINED AS FOLLOWS:

FROM PG. 157 P WOULD BE EQUAL TO $\frac{1}{2}W_{AB} + \frac{1}{2}W_{BC}$

$$P = \frac{1}{2}\left\{1.50(34.1g)(1.29+0.52 \text{ LBS})\right\} = 46.3 \text{ LBS. (ULT.)}$$

ASSUMING AN ADDITIONAL LOAD DUE TO PRELOADING OF AT LEAST 46.3 LBS, USE P_{TOTAL} = 100 LBS.

ASSUME THAT THIS TOTAL LOAD IS APPLIED AT MID-SPAN OF A 1.00 IN. LONG BEAM SIMPLY SUPPORTED AT ITS ENDS WITH A CROSS-SECTION SHOWN IN A-A ABOVE.



$$M_{MAX} = \frac{PL}{4} = \frac{100(1.00)}{4}$$

M_{MAX} = 25 IN-LBS (ULT.)

FROM SECT. A-A

$C = (0.31/2) = 0.155$ IN.

$I = (0.72-0.39)(0.31)^3/12$

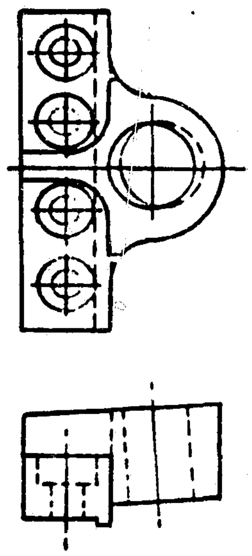
$I = 0.000828 \text{ IN}^4$

$$f_b = \frac{M_{MAX}C}{I} = \frac{25(0.155)}{0.000828} \approx \text{46[illegible] PSI (ULT)}$$

MAT'L. IS 6061-T6 AL. ALLOY PLATE

∴ M.S. LARGE BY INSPECTION.

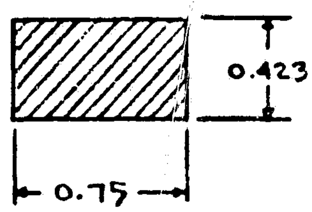
12.3 STRESS IN MIRROR CLAMP (CENTER).

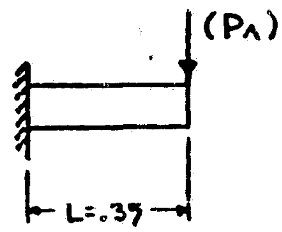


THE MAXIMUM LOAD APPLIED TO THE CENTER CLAMP IS TAKEN AS

$$(P_A) = 142.03 \text{ LBS.} \quad \text{(REF. PG. 159)}$$

CONSERVATIVELY ASSUME THAT THIS LOADS ACTS AT THE END OF A CANTILEVER OF LENGTH L = 0.35 IN. WHOSE CROSS-SECTION IS SHOWN BELOW:





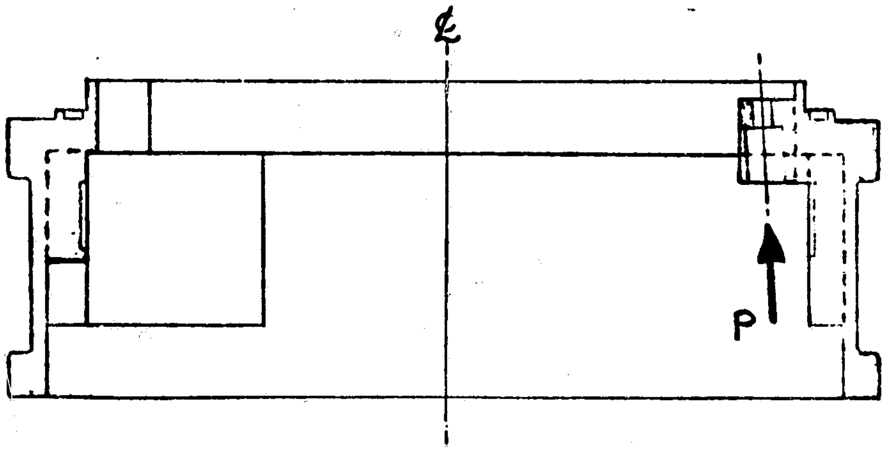
$$M_{MAX} = PL = 142.03(0.35) = 49.7 \text{ IN-LBS (ULT.)}$$

$$f_b = \frac{6M_{MAX}}{bt^2} = \frac{6(49.7)}{0.75(0.423)^2} = \underline{2225 \text{ psi (ULT.)}}$$

MAT'L. IS 6061-T6 AL. ALLOY PLATE

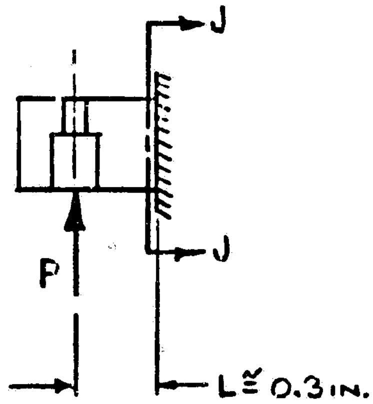
∴ M.S. LARGE BY INSPECTION.

12.4 ANALYSIS OF MIRROR CELL (AFT PLATE)
REF. DWG. NO. 7232-0151



SECTION B-B

FROM PG. 159 THE MAXIMUM VALUE OF P IS 142 LBS. (ULT.). ASSUME THE LOAD IS APPLIED IN SUCH A MANNER AS TO PRODUCE BENDING STRESSES IN THE CANTILEVERED "TAB" PORTION OF THE MIRROR CELL.



M = MAX. BENDING MOMENT = PL

M = 142 (0.3) = 42.6 IN-LBS. (ULT.)

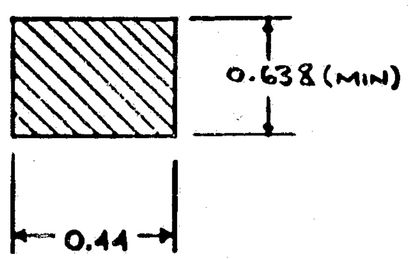
BENDING STRESS $f_b = \dfrac{6M}{bt^2}$

$$f_b = \frac{6(42.6)}{0.44(0.638)^2}$$

f_b = 1430 PSI (ULT.)

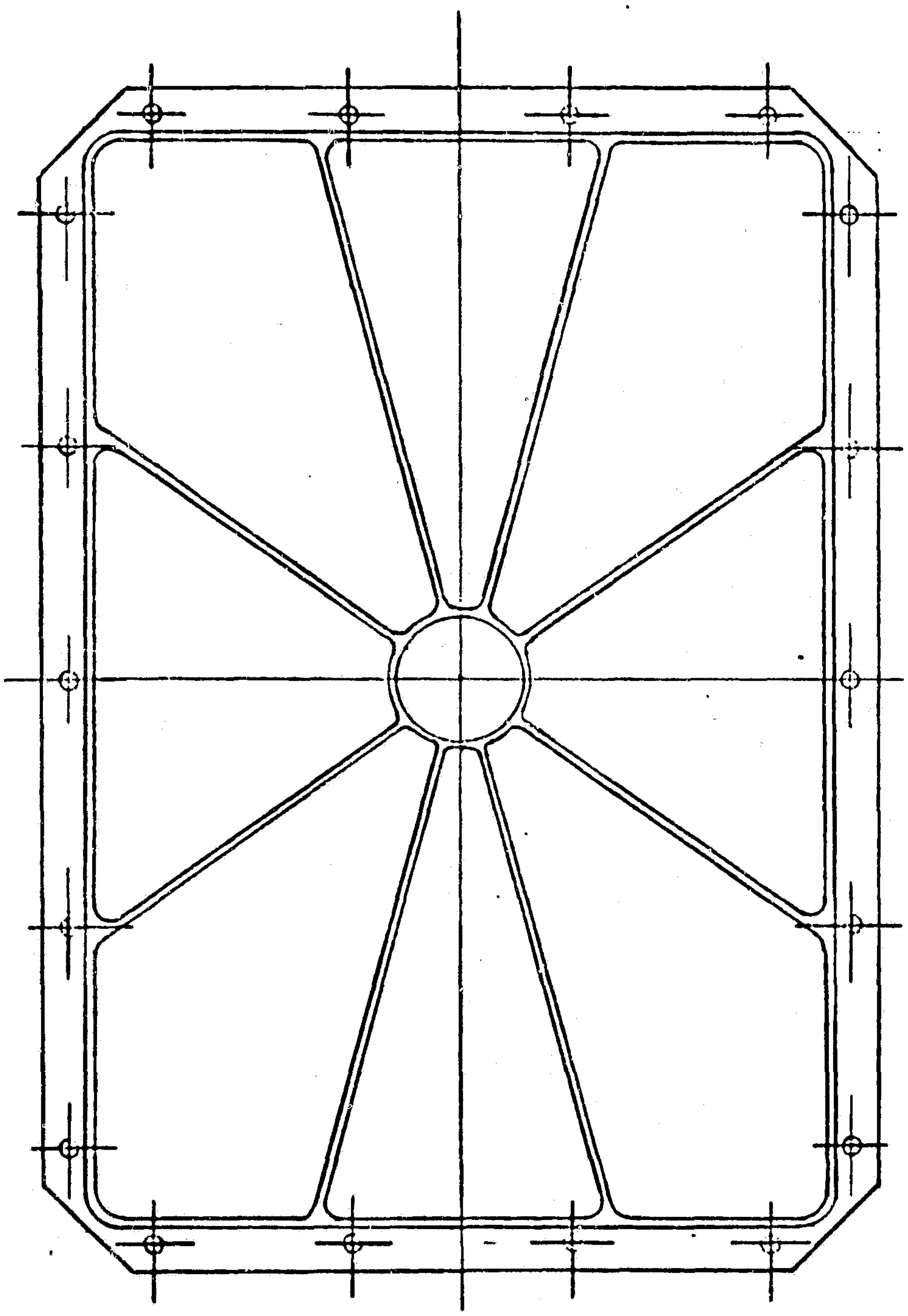
MAT'L. IS 6061-T6 ALUMINUM ALLOY

∴ M.S. LARGE BY INSPECTION



SECTION J-J

12.5 ANALYSIS OF MIRROR CELL REAR COVER (AFT END CAP).



12.5.1 STRESS IN REAR COVER.

THE MIRROR CELL REAR COVER WILL CONSERVATIVELY BE ANALYZED AS A RECTANGULAR FLAT PLATE, SIMPLY SUPPORTED ON ALL FOUR EDGES UNDER A UNIFORMLY DISTRIBUTED PRESSURE. THE STIFFENING EFFECT OF THE REINFORCING RIBS IS NEGLECTED.

THE MAXIMUM STRESS IS GIVEN IN REF. K, TABLE X - CASE 36.

$$f_b = \frac{\beta \omega b^2}{t^2}$$

b = SHORT SIDE OF PLATE = 6.87 IN.
a = LONG SIDE OF PLATE = 10.00 IN.
$(a/b) = 1.455$
t = PLATE THICKNESS = 0.03 IN.
β = COEFFICIENT = 0.471 FOR $(a/b) = 1.455$

$$\therefore f_b = \frac{0.471\,(6.87)^2\omega}{(0.03)^2} = 24717\omega$$

FOR DYNAMIC LOADING OF AFT ENDCAP

$G_Z = 36.1g$ (REF. PG. 41)

$\therefore W_{DYN} = 1.50(0.35\text{ LBS})(36.1g)$

$W_{DYN} = 18.95$ LBS (ULT.)

PLATE AREA, $A = ab = 68.70\text{ in}^2$

$\omega_{DYN} = \frac{W_{DYN}}{A} = 0.276$ psi (ULT)

CONSERVATIVELY ASSUME THAT AT THE TIME OF SIM DOOR JETTISON AN INTERNAL PRESSURE OF 1.0 psi EXISTS ON THE COVER DUE TO DIFFERENTIAL RATES OF PRESSURE LOSS WITHIN AND WITHOUT THE SPECTROMETER.

$$\therefore f_b = 24717(1.0) = \underline{24717\text{ psi}}$$

F_{TY} = 36000 psi FOR 6061-T6 AL. ALLOY SHEET (REF. C)

$$M.S. = \frac{F_{TY}}{f_b} - 1 = \frac{36000}{24717} - 1 = \underline{+0.45}$$

12.5.2 ATTACHMENT OF REAR COVER TO AFT PLATE.

TOTAL AREA UNDER PRESSURE = $68.70\ in^2$
PRESSURE LOAD ω = 1.0 PSI

TOTAL W = 68.70 LBS.

USING A 1.50 LOAD FACTOR

$$W' = 1.50(68.70) \cong \underline{103\ \text{LBS.}}$$

THERE ARE 18 SCREWS. THE LOAD PER SCREW IS

$$P = (W'/18) = \underline{5.7\ \text{LBS.}}$$

SCREWS ARE #6-32 UNC CRES. FOR WHICH $(P_Y)_{MIN}$ = 270 LBS. (REF. e).
ASSUMING SCREWS TORQUED TO 80% YIELD,

$$(P_Y) = 0.80(270) = \underline{216.0\ \text{LBS.}}$$

$$M.S. = \frac{(P_Y)}{P} - 1 = \frac{216.0}{5.7} - 1 = \underline{\text{LARGE}}$$