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HEAVY-LIFT TIP TURBOJET ROTOR SYSTEM
VOLUME V
STRUCTURAL ANALYSIS

October 1965

U. S. ARMY AVIATION MATERIEL LABORATORIES

FORT EUSTIS, VIRGINIA

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HILLER AIRCRAFT COMPANY, INC.

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VOLUME V

STRUCTURAL ANALYSIS

Hiller Engineering Report No. 64-45

Prepared by

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Subsidiary of Fairchild Hiller Corporation
Palo Alto, California

For

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

(U. S. Army Transportation Research Command when report prepared)

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SYMBOLS

A	Area, in^2
b	Panel length, in.
C.F.	Centrifugal force, lb.
d,D	Diameter, in.
E	Modulus of elasticity, lb/in^2
f	Stress, lb/in^2
f_s	Steady stress, lb/in^2
f_a	Alternating stress, lb/in^2
$f_{a_{eq}}$	Equivalent alternating stress, lb/in^2
$F_{e\ell}$	Endurance limit, lb/in^2
F_{su}	Allowable shear stress, lb/in^2
F_{tu}	Allowable tensile stress, lb/in^2
g	Gravitational units, 32.2 ft/sec^2
GJ	Torsional rigidity, $lb\text{-}in^2$
I	Area moment of inertia, in^4
K_t	Theoretical fatigue notch factor
$K_{f_{av}}$	Available fatigue notch factor
L_c	Calculated fatigue life, hr.
M	Moment, in-lb.
M.S.	Margin of safety
n	Load factor, multiples of g
N	Fatigue cycles
P	Load, lb.
R	Stress ratio

SYMBOLS (CONTINUED)

r	Radius, in.
S.L.	Service life, hr.
T	Torque, in-lb.
V_T	Rotor blade tip speed, f.p.s.
W	Weight, lb.
Z	Section modulus, in^3
Ω	Rotor angular velocity, rad/sec.

1.0 SUMMARY

1.1 General Description

The tip turbojet helicopter described herein is designed as a cargo helicopter having a payload of 12 tons and a gross weight of 72,000 pounds. The rotor system utilizes four blades with two engines mounted in an over-under configuration on each blade tip. Each blade has a radius of 56 feet to the centerline of the engines and a chord of 6.5 feet. Titanium alloys are used, when practical, throughout the rotor system construction.

1.2 Structural Design Philosophy

1.2.1 Purpose

It is the purpose of this part of the report to provide sufficient load and stress data to illustrate the feasibility of the rotor system from a structural viewpoint.

1.2.2 Scope

The information contained herein is to provide a brief preliminary survey of loads and stresses in the major structural components of the rotor system. The rotor system is analyzed progressing in order of force transmission from the engine nacelles at the tip to the gimbal at the rotor shaft.

Due to the lack of structural design details, certain refinements must be eliminated in this preliminary effort which will be considered in a production or prototype effort. Simplifying assumptions are clearly defined in the appropriate places within the report.

The special engine environmental vertical load factor of ±40g used in the rotor system static analysis was conservatively estimated during development design and prior to the dynamic load study completion. The rotor system stiffness has increased such that the engine environmental vertical load factor computed during the dynamic load study has a magnitude of ±3.4g. Therefore, the static rotor system stress analysis is conservatively presented herein using the ±40g loading factor in conjunction with the rotor-overspeed-operation, both-engines-operating condition.

1.2.3 Fatigue Considerations

The fatigue analysis primarily consists of an investigation of the rotor system to illustrate that the alternating stresses developed during normal flight conditions are below the component material endurance limit and nondamaging.

In cases where attachment bolt static margins of safety are low for centrifugal force loading, a start-stop fatigue investigation is conducted to determine the required bolt diameter to establish a 10,000-hour service life when a fatigue notch factor of 2.0 is applied to the S-N data.

The prediction of component fatigue strength, in some cases, is accomplished through the use of available fatigue notch factors. Only those components which are major structural members are considered from a fatigue viewpoint.

2.0 CONCLUSIONS

2.1 Critical Static Design Conditions

2.1.1 Engine Nacelle and Attachments

Although the structural significance of the engine nacelles is recognized at this time, it is felt that they do not contribute structurally as rotor system major load carry members, and as such are not analyzed during Phase I.

2.1.2 Engine Mount System and Attachment

The critical engine mount system and attachment loading occurs during the rotor limit speed condition, and during the rotor-overspeed-operation, both-engines-operating condition. The critical engine mount areas are the attachment bolts and lugs and the heat expansion fitting.

2.1.3 Main Rotor Blade Tip Engine Retention Structure

The critical main rotor blade tip and attachments loading occurs during the rotor limit speed condition and during the rotor-overspeed-operation, two-engines-operating condition. The critical areas are the attachment lugs.

2.1.4 Main Rotor Blade Typical Section

The critical main rotor blade typical section is at rotor station 170.00 during the static droop condition.

2.1.5 Main Rotor Blade Root Retention Structure

The critical main rotor blade root retention structure loading occurs during the rotor limit speed condition. The critical areas are the tension-torsion strap and its retention bolt.

2.1.6 Stub Blade and Retention

The critical stub blade loading occurs during the transient cyclic stick whirl condition. The adjustable link attachment lug is critical.

2.1.7 Main Rotor Hub Assembly

The critical main rotor hub assembly loading occurs during forward flight, 41 miles per hour, 2.5g, 562 feet per second tip velocity condition. The critical hub areas are the blade retention lugs and pins.

2.1.8 Gimbal and Attachments

The critical gimbal and attachments loading occurs during the 2.5g loading condition. The critical areas are the bearings and Section 17-17 as defined on page 87.

2.1.9 Restraint Spring Assembly

The outside spring fiber stress is critical

2.1.10 Static Margins of Safety

A summary of the critical static margins of safety is presented in tabular form on the following page.

2.2 Critical Fatigue Design Conditions

The start-stop condition is the critical main rotor system fatigue design condition considered in this analysis. The critical rotor system components during the start-stop condition are the engine to mount attachment heat-expansion fitting, the rotor system component attachment bolts, pins, and lugs, and the tension-torsion strap assembly.

The alternating stresses developed during a steady-state, in trim, normal flight condition are below the rotor system component material endurance limit and nondamaging.

Further fatigue consideration of the main rotor system will be conducted upon the accomplishment of strain-measured flight maneuvers.

TABLE 1
SUMMARY - CRITICAL STATIC MARGINS OF SAFETY

Section	Name	Margin of Safety		Page No.
		Due to	M.S.	
4.1.2	Engine Mount System and Attachments			
	Engine to mount attachment bolts	Bolt bending	.01	16
	Main mount, Section 2-2	Combined load	.20	20
	Heat expansion fitting, Sect. 4-4	Combined load	.15	28
	Inner lug to mount weldment	Combined load	.16	33
	Mount to tip attachment bolts	Bolt bending	.01	36
4.1.3	Main Rotor Blade Tip Engine Retention Structure			
	Mount pickup fittings	Lug shear-out	.16	43
	Pickup fittings, Section 8-8	Combined load	.17	47
4.1.4	Main Rotor Blade Typical Section	Buckling	.10	62
4.1.5	Main Rotor Blade Root Retention			
	Tension-torsion strap retention pin	Pin bending	.24	65
	Tension-torsion strap assembly	Tension	.00	69
4.1.6	Stub Blade and Retention			
	Adjustable link attachment lug	Lug shear-out	.60	74
4.1.7	Main Rotor Hub Assembly			
	Stub blade to hub attachment lugs	Lug shear-out	.03	77
	Stub blade to hub attachment pin	Pin bending	.02	78
	Adjustable link lug analysis	Lug shear-out	.23	82
4.1.8	Gimbal and Attachments			
	Gimbal ring, Section 17-17	Combined load	.27	88
	Pivot pin bearing	Radial bearing load	.14	89
	Rotor shaft bearing lug, Section 18-18	Combined load	.18	91
4.1.9	Restraint Spring Assembly			
	Outside spring	Spring load	.09	92

3.0 RECOMMENDATIONS

3.1 Static Testing

In those cases where design of components and/or assemblies results in the requirement of an ultraconservative static analysis, structural substantiation will be accomplished by static test.

3.2 Fatigue Testing

A minimum of four specimens will be fatigue tested to obtain the required fatigue data for component service life estimation.

4.0 STATIC STRUCTURAL ANALYSIS

4.1 Introduction

It is the purpose of this part of the report to provide sufficient load and stress data to insure the integrity of the rotor system design from the static structural viewpoint.

The main rotor system is substantiated to the criteria of Reference 1 utilizing the loads developed in Reference 2. The methods of analysis used in this report are those which are generally accepted throughout the airframe and missile industry. Whenever possible, the latest edition of Reference 4, "Metallic Materials and Elements for Flight Vehicle Structures," MIL-HDBK-5, August 1962, is referred to for the material mechanical properties.

A fitting factor of 1.15 is not used in the lug analyses margins of safety calculations for the rotor limit speed condition (tip velocity of 813 feet per second) as it is felt that the conservatism is too extreme. The rotor limit speed tip velocity is 1.25 times the maximum design tip speed velocity, and an ultimate safety factor of 1.50 is used in the margins of safety calculations.

The rotor system is analyzed in order of force transmission progressing from the engine nacelle at the tip to the hub and gimbal at the rotor shaft.

4.1.1 Engine Nacelle and Attachments

An engine nacelle is provided for aerodynamic streamlining purposes and does not carry primary structural loads. Since the intent of the Phase I rotor system stress analysis is to provide a brief preliminary survey of loads and stresses in the major structural components and since the function of the engine nacelles is aerodynamic in nature, this portion of the rotor system stress analysis is not presented for Phase I considerations.

4.1.2 Engine Mount System and Attachment

This portion of the main rotor system stress analysis is concerned with the static structural substantiation of the engine mount system, including the mount system to engines and blade-tip attachments. The engine mounts are analyzed in order of force transmission, progressing from the engine attachments to the blade-tip attachments.

4.1.2.1 Loading Analysis

Preliminary static stress analysis completed during the engine main mount system development design period established that the maximum engine main mount loading occurred during the rotor limit speed condition, and during the rotor-overspeed-operation, two-engines-operating condition. The aft engine mount experiences its highest loading during the rotor overspeed operation.

The engine mount system loads presented in Figures 1, 2, and 3 are developed below for the following critical loading conditions:

Condition 1: Rotor limit speed (813 f.p.s. tip velocity).
Condition 2: Design maximum rotor speed - one engine out.
Condition 3: Rotor overspeed operation - both engines operating.

The loads presented at a given location in Figures 1, 2, and 3 are the applied loads acting at that point.

Condition 1:

Engine weight: $W = 370$ lb/engine (limit)

Blade tip velocity: $V_T = 813$ f.p.s.

$$\Omega = \frac{813}{56} = 14.51 \text{ rad/sec.}$$

Centrifugal force load per engine:

$$\text{C.F.} = \frac{(14.51)^2}{32.2}(370)(56) = 136{,}000 \text{ lb. (limit)}$$

The additional load due to centrifugal force acting on the main mount and that portion of the nacelle supported by the main mount is assumed to be acting at rotor station 661.00.

Weight: $W = 100$ lb. (limit)

$$\text{C.F.} = \frac{(14.51)^2}{32.2}(100)(55.6) = 36{,}700 \text{ lb. (limit)}$$

Condition 2:

Engine weight: $W = 370$ lb. (limit)

Blade tip velocity: $V_T = 650$ f.p.s.

$$\Omega = \frac{650}{56} = 11.60 \text{ rad/sec.}$$

$$\text{C.F.} = \frac{(11.60)^2}{32.2}(370)(56) = 87{,}000 \text{ lb. (limit)}$$

The gyroscopic moment tends to force the rotor blade leading edge down and has a magnitude of:

$$M_x = (1.67)(2{,}304)\frac{650}{56} = 44{,}700 \text{ in-lb. (limit)}$$

The centrifugal force load due to the main mount and nacelle acts at rotor station 661.00 and has a magnitude of:

$$C.F. = \frac{(11.60)^2}{32.2}(100)(55.6) = 23,300 \text{ lb. (limit)}$$

The engine thrust is 1,500 pounds (limit) per engine. The moment about the z axis due to the centerline of the engine thrust being located 11.25 inches outboard of the engine to main mount attachment centerline is reacted as a couple by the main and aft engine mounts.

$$P_x = \frac{11.25}{15.125}(1,500) = 1,125 \text{ lb. (limit)}$$

The aft mount is assumed to weigh 11.2 pounds (limit). The centrifugal force load due to the aft mount weight is:

$$C.F. = \frac{(11.60)^2}{32.2}(11.2)(55.6) = 2,600 \text{ lb. (limit)}$$

Condition 3:

Engine weight: $W = 370$ lb. (limit)

Blade tip velocity: $V_T = 683$ f.p.s.

$$\Omega = \frac{683}{56} = 12.20 \text{ rad/sec.}$$

$$C.F. = \frac{(12.20)^2}{32.2}(370)(56) = 95,800 \text{ lb. (limit)}$$

The gyroscopic moment per engine tends to force the rotor blade leading edge down and has a magnitude of:

$$M_x = (1.67)(2,304)\frac{683}{56} = 47,000 \text{ in-lb. (limit)}$$

The centrifugal force load due to the main mount and nacelle weight acts at rotor station 661.00 and has a magnitude of:

$$C.F. = \frac{(12.20)^2}{32.2}(100)(55.6) = 25,700 \text{ lb. (limit)}$$

The static engine thrust is 1,700 pounds (limit) per engine. Per the discussion on page 9 of Volume IV, an additional in-plane load normal to the blade axis is applied.

$$P_y = 1,700 + (\pm 5)(370) = 3,550 \text{ lb. (limit) per engine}$$

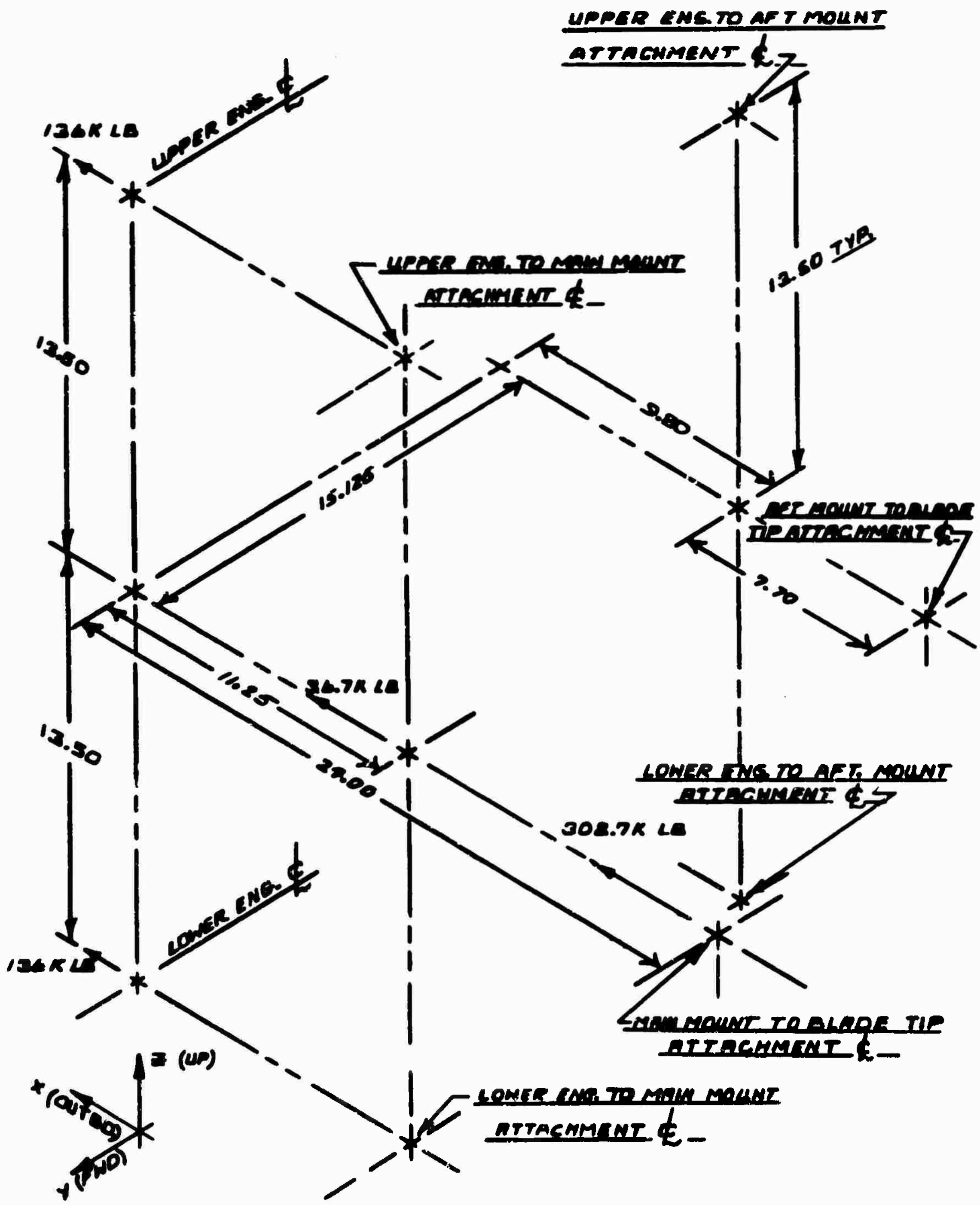


Figure 1. Engine-Mount-System Loading Diagram - Rotor Limit Speed of 813 f.p.s. Tip Velocity.

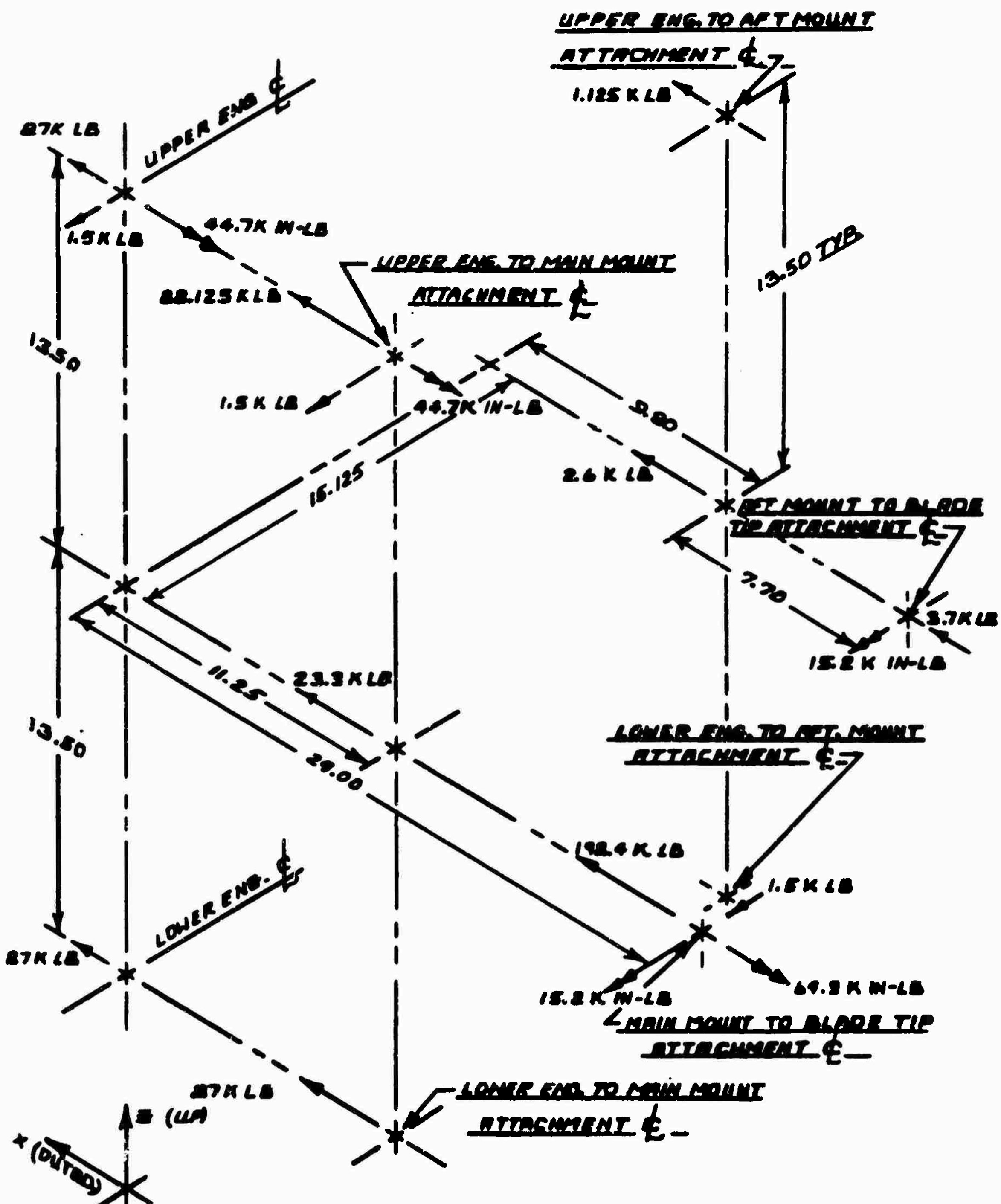


Figure 2. Engine-Mount-System Loading Diagram - Design Maximum Rotor Speed - One Engine Out.

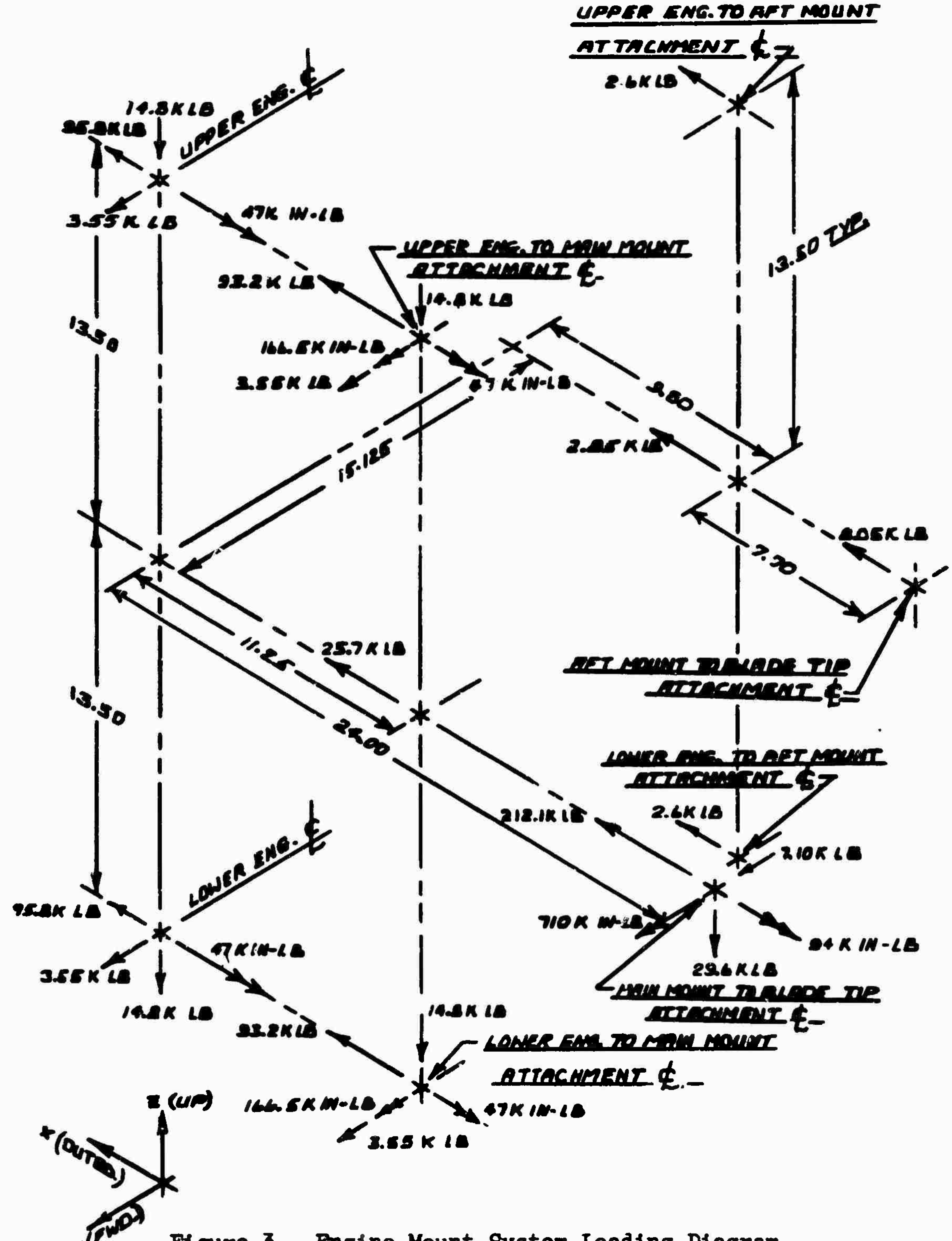


Figure 3. Engine-Mount-System Loading Diagram - Rotor Overspeed Operation - Both Engines Operating.

The vertical engine load normal to the blade axis is:

$$P_z = (\pm 40)(370) = \pm 14,800 \text{ lb. (limit) per engine}$$

The aft mount reacts the in-plane engine loading.

$$P_x = \frac{11.25}{15.125}(3,550) = 2,600 \text{ lb. (limit) per engine}$$

The centrifugal force load due to the aft mount weight of 11.2 pounds (limit) is:

$$C.F. = \frac{(12.20)^2}{32.2}(11.2)(55.6) = 2,850 \text{ lb. (limit)}$$

4.1.2.2 <u>Stress Analysis for Main Engine Mount</u> - (Ref. Volume III, Figure 20)

This portion of the main rotor system stress analysis is concerned with the static structural substantiation of the main engine mount, including the engine and blade-tip-to-mount attachments.

The material properties presented below are obtained from Reference 10.

Material type: Ti-6AL-4V titanium

At room temperature:

$$\left.\begin{aligned} F_{tu} &= 162,000 \text{ p.s.i.} \\ F_{su} &= 94,000 \text{ p.s.i.} \end{aligned}\right\} \text{(Ref. 10, page 39)}$$

At 400°F. temperature:

$$\left.\begin{aligned} F_{tu} &= (.785)(162,000) = 127,000 \text{ p.s.i.} \\ F_{su} &= (.785)(94,000) = 74,000 \text{ p.s.i.} \end{aligned}\right\} \text{(Ref. 10, pages 52,53)}$$

4.1.2.2.1 <u>Loading Analysis</u>

Per the discussion on page 7, the critical engine mount centrifugal force loads occur during the rotor limit speed condition.

The general geometry and loading sketch, presented in Figure 4 on the following page, establishes the upper lug attachments for both engines to be critical for the rotor limit speed condition centrifugal force.

C.F. = 308,700 lb. (limit)

W_T = 370 lb. (limit)

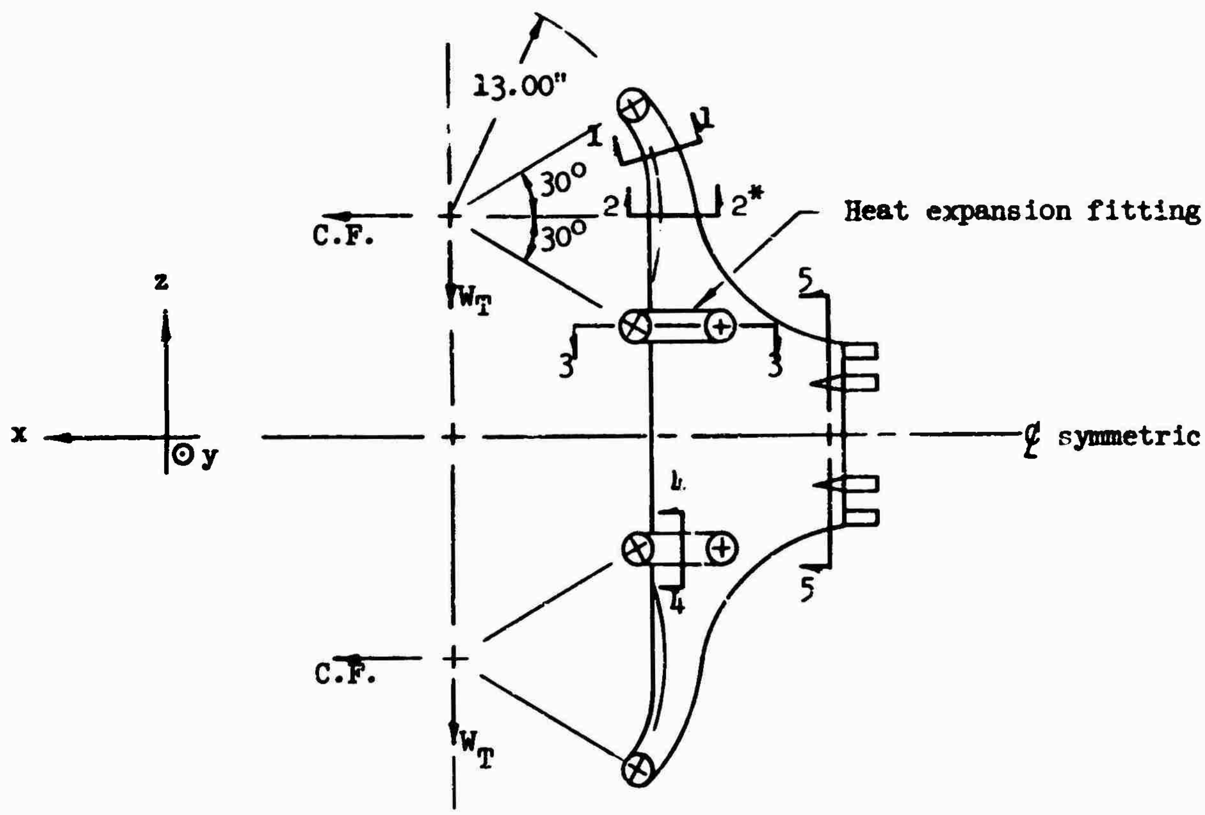


Figure 4. Main Engine Mount - General Geometry and Loading Sketch.

4.1.2.2.2 <u>Engine-to-Mount Attachment Bolts</u>

1.25 dia. bolt, NAS 464; $A = 1.23\ in^2$; $Z = .192\ in^3$

Material type: 1AL-8V-5F_e titanium alloy
HiTi 20 series bolts

The material properties are obtained from Reference 7, page 2.

At room temperature: F_{tu} = 200,000 p.s.i.

F_{su} = 120,000 p.s.i.

At 400°F. temperature: F_{tu} = (.8)(200,000) = 160,000 p.s.i.

F_{su} = (.8)(120,000) = 96,000 p.s.i.

The maximum bolt loads occur during the rotor limit speed condition:

$$P_x = \frac{C.F.}{2} + \frac{11.25W_T}{13.00} = \frac{136,000}{2} + \frac{(11.25)(370)}{13.00}$$

$$= 68,300 \text{ lb. (limit)}$$

The bolts are loaded in double shear.

$$A_s = \frac{(2)(\pi)(1.25)^2}{4} = 2.45 \text{ in}^2$$

$$f_s = \frac{68,300}{2.45} = 27,900 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{96,000}{(1.5)(27,900)} - 1 = \quad 1.30$$

Bolt Bending-Stress Investigation:

The method of analysis used in determining the bolt bending-stress level is obtained from pages 160 through 164 of Reference 9. In the following analysis, the applied load is assumed to tend to "peak-up" on the inner lug near the shear planes rather than be carried as a uniform load across the inner lug.

$P = 68,300$ lb. (limit)

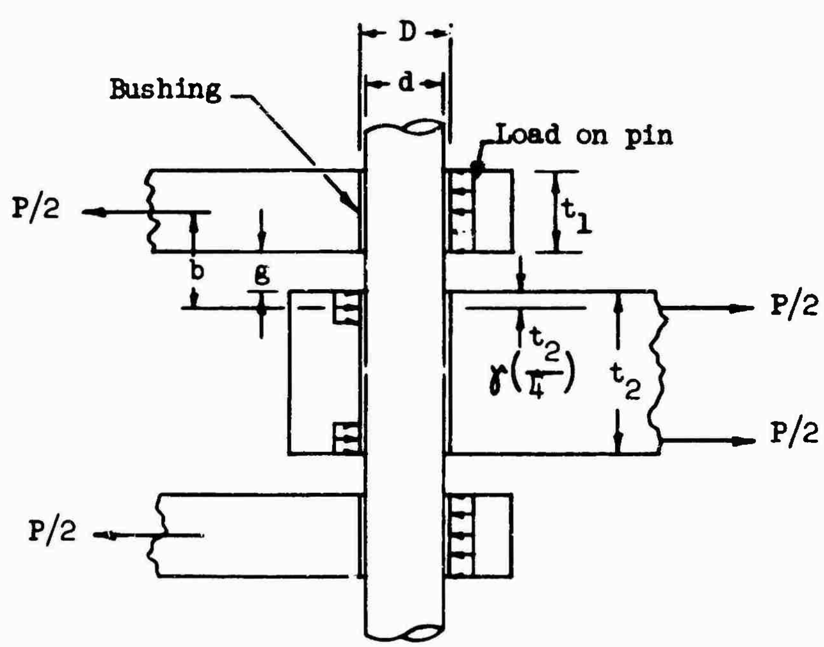


Figure 5. Bolt-and-Lug Loading Sketch - Engine-to-Main-Mount Attachment.

The inner lug is a portion of the turbojet engine ring and is loaded at an angle of 30 degrees. The outer lugs are a part of the main engine mount and are loaded at an angle of 60 degrees. The bolt bending stress analysis presented on the following page assumes the inner and outer lugs to be axially loaded and considers the load "peaking" effects on the inner lug only.

If the oblique loadings in the attachment system are resolved into axial and transverse components, the value of the minimum allowable ultimate load ($P'_{u_{min}}$) is reduced which results in a shorter moment arm and a smaller bolt bending stress. In addition, if the excess of strength in the outer lugs is considered, an additional moment arm reduction is realized.

Preliminary analysis has established that when the excess outer lug strength is considered, assuming axial lug loading, the vector quantity of the bolt horizontal shear and reduced bending stresses results in a slightly higher margin of safety than that obtained on the following page.

In view of the above discussion, the bolt bending stress analysis presented on the following page is considered to be satisfactory.

Lug material properties:

$F_{su} = 74{,}000$ p.s.i.

$F_{tu} = 127{,}000$ p.s.i.

$P'_{br_u} = (3.24)(74{,}000)$
$= 240{,}000$ lb.

$P'_{tu} = (1.936)(127{,}000)$
$= 246{,}000$ lb.

$P'_{u_{min}} = 240{,}000$ lb.

$$\frac{P'_{u_{min}}}{A_{br}F_{tu}} = .585$$

$\gamma = .68$ (Ref. 9, page 162, Figure 4)

$d = 1.25$ in.
$D = 1.44$ in.
$g = .065$ in.
$W = 2.30$ in.
$a = 1.15$ in.
$t_1 = 1.1$ in.
$t_2 = 2.25$ in.

$$r = \left[\frac{a}{D} - \frac{1}{2}\right]\frac{D}{t_2} = \left[\frac{1.15}{1.44} - \frac{1}{2}\right]\frac{1.44}{2.25}$$
$$= .192$$

$A_{br} = Dt_2 = (1.44)(2.25) = 3.24 \text{ in}^2$

$A_t = (W - D)t_2 = (2.3 - 1.44)(2.25)$
$= 1.936 \text{ in}^2$

$$b = \frac{t_1}{2} + g + \gamma\left(\frac{t_2}{4}\right)$$
$$= \frac{1.1}{2} + .065 + (.68)\left(\frac{2.25}{4}\right) = .997 \text{ in.}$$

$$M = (P/2)b = \frac{68{,}300}{2}(.997) = 34{,}400 \text{ in-lb. (limit)}$$

$$f_b = \frac{M}{Z} = \frac{34{,}000}{.192} = 177{,}000 \text{ p.s.i. (limit)}$$

The bending modulus of rupture factor (K = 1.68) is obtained from page 115 of Reference 3.

$$M.S. = \frac{KF_{tu}}{1.5\ f_b} - 1 = \frac{(1.68)(160{,}000)}{(1.5)(177{,}000)} - 1 = \qquad +.01$$

4.1.2.2.3 Upper Lug Stress Investigation - Upper Engine

$P_x = 68{,}300$ lb. (limit) (Ref. page 15)

$P_y = W_T = 370$ lb. (limit)

$$P = \sqrt{P_x^2 + P_y^2} \approx 68{,}300 \text{ lb. (limit)}$$

4.1.2.2.3.1 Link Tear-Out

$A = (1.1)(2.30 - 1.44)(2) = 1.89 \text{ in}^2$

$$f_t = \frac{68{,}300}{1.89} = 36{,}100 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{127{,}000}{(1.5)(36{,}100)} - 1 = \quad +1.51$$

4.1.2.2.3.2 Link Shear-Out

$$A = (1.1)(2.30 - 1.44)(2) = 1.89 \text{ in}^2$$

$$f_s = \frac{68{,}300}{1.89} = 36{,}100 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{74{,}000}{(1.5)(36{,}000)} - 1 = \quad +.46$$

4.1.2.2.4 Section 1-1 Stress Investigation

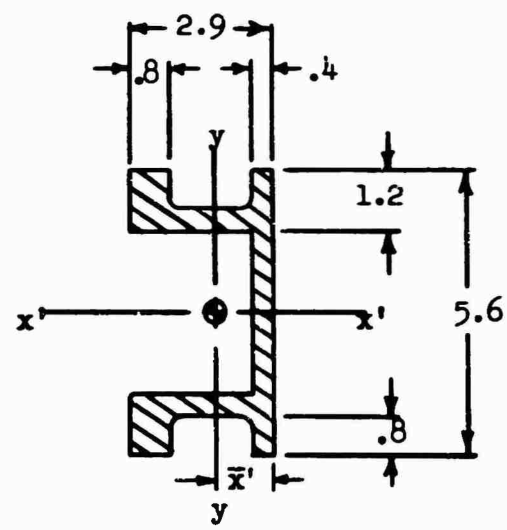


Figure 6. Main Engine Mount - Section 1-1 Geometry.

Section Properties:

$$A = (2.9)(5.6) - (2)(1.7)(.8) - (2.5)(3.2)$$
$$= 16.23 - 2.72 - 8.00 = 5.51 \text{ in}^2$$

$$I_{x'-x'} = \frac{(2.9)(5.6)^3}{12} - 2\left[\frac{(1.7)(.8)^3}{12} + (1.36)(2.4)^2\right] - \frac{(2.5)(3.2)^3}{12}$$
$$= 42.4 - 2(.0725 + 7.84) - 6.83$$
$$= 42.4 - 22.65$$
$$= 19.75 \text{ in}^4$$

$$\bar{x}' = \frac{(16.23)(1.45) - 2(1.36)(1.25) - (8.00)(1.65)}{5.51}$$

$$= \frac{23.58 - 3.40 - 13.20}{5.51} = \frac{6.98}{5.51} = 1.263 \text{ in.}$$

$$I_{y-y} = \frac{(5.6)(2.9)^3}{12} - 2\left[\frac{(.8)(1.7)^3}{12} + (1.36)(.013)^2\right] - \frac{(3.2)(2.5)^3}{12} - (8.00)(.39)^2 + (16.23)(.19)^2$$

$$= 11.38 + .59 - 2(.33 + 0) - 4.17 - 1.22 = 11.97 - 6.05$$

$$= 5.92 \text{ in}^4$$

4.1.2.2.4.1 **Loading Analysis**

$$\alpha = \tan^{-1}\frac{.85}{3.45} = 13.62^\circ$$

$$P_{x'} = P_x \cos\alpha + P_z \sin\alpha$$

$$= 68{,}300 \cos 13.62^\circ + 370 \sin 13.62^\circ$$

$$= 66{,}400 + 87 \approx 66{,}500 \text{ lb. (limit)}$$

$$P_{z'} = P_x \sin\alpha \mp P_z \cos\alpha$$

$$= 68{,}300 \sin 13.62^\circ - 370 \cos 13.62^\circ$$

$$= 15{,}700 \text{ lb. (limit)}$$

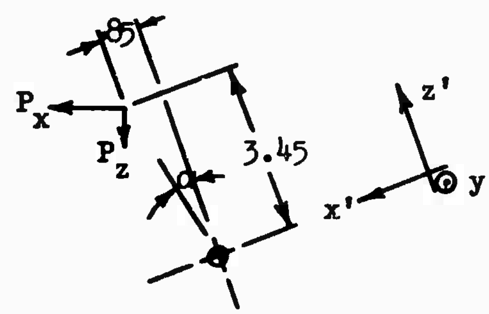


Figure 7. Main Engine Mount - Section 1-1 Loading Sketch.

4.1.2.2.4.2 **Stress Analysis**

$$M_{y-y} = 3.45P_{x'} - .85P_{z'}$$

$$= (3.45)(66{,}400) - (.85)(15{,}700)$$

$$= 215{,}700 \text{ in-lb. (limit)}$$

$$f_b = \frac{(215{,}700)(1.26)}{5.92} = 46{,}000 \text{ p.s.i. (limit)}$$

$$f_t = \frac{16{,}700}{5.51} = 2800 \text{ p.s.i. (limit)}$$

$$f_n = f_b + f_t = 48{,}800 \text{ p.s.i. (limit)}$$

$$f_s = \frac{66{,}500}{5.51} = 12{,}100 \text{ p.s.i. (limit)}$$

$$R_t = \frac{(1.5)(48{,}800)}{127{,}000} = .576 \,, \quad R_s = \frac{(1.5)(12{,}100)}{74{,}000} = .245$$

Combining the tensile and shear stresses vectorially, the margin of safety is:

$$M.S. = \frac{1}{\sqrt{R_t^2 + R_s^2}} - 1 = \quad .53$$

4.1.2.2.5 <u>Section 2-2 Stress Investigation</u>

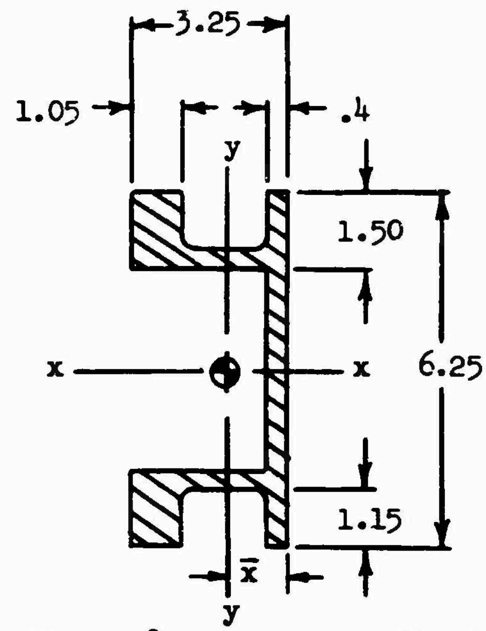


Figure 8. Main Engine Mount - Section 2-2 Geometry.

<u>Section Properties:</u>

$$A = (3.25)(6.25) - (2)(1.8)(1.15) - (2.85)(3.25)$$
$$= 20.35 - 4.14 - 9.26 = 7.05 \text{ in}^2$$

$$I_{x-x} = \frac{(3.25)(6.25)^3}{12} - 2\left[\frac{(1.8)(1.15)^3}{12} + (2.07)(2.75)^2\right] - \frac{(2.85)(3.25)^3}{12}$$
$$= 66.10 - 2(.23 + 15.12) - 8.16$$
$$= 66.10 - 38.86 = 37.24 \text{ in}^4$$

$$\bar{x} = \frac{(20.35)(1.625) - (2)(2.07)(1.3) - (9.26)(1.825)}{7.05}$$
$$= \frac{33.00 - 5.39 - 16.90}{7.05} = \frac{10.71}{7.05}$$
$$= 1.52 \text{ in.}$$

$$I_{y-y} = \frac{(6.25)(3.25)^3}{12} + (20.35)(.105)^2 - 2\left[\frac{(1.15)(1.8)^3}{12} + (2.07)(.22)^2\right] - \frac{(3.25)(2.85)^3}{12} - (9.26)(.305)^2$$

$$= 17.88 + .22 - 2(.459 + .10) - 6.27 - .86$$

$$= 18.10 - 8.25 = 9.85 \text{ in}^4$$

Loading Analysis:

$$M_{y-y} = 6.55\, P_x + 2.55\, P_z$$

$$= (6.55)(68300) + (2.55)(370)$$

$$= 447{,}000 + 940$$

$$= 447{,}900 \text{ in-lb. (limit)}$$

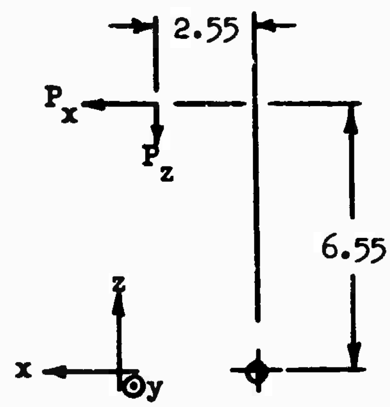


Figure 9. Main Engine Mount - Section 2-2 Loading Sketch.

$$f_b = \frac{(447{,}900)(1.52)}{9.85}$$

$$= 69{,}000 \text{ p.s.i. (limit)}$$

$$f_c = \frac{370}{7.05} = 52.5 \text{ p.s.i. (limit)}$$

negligible

$$f_s = \frac{68{,}300}{7.05} = 9700 \text{ p.s.i. (limit)}$$

$$R_t = \frac{(1.5)(69{,}000)}{127{,}000} = .815, \quad R_s = \frac{(1.5)(9700)}{74{,}000} = .197$$

Combining the bending and shear stresses vectorially, the margin of safety is:

$$\text{M.S.} = \frac{1}{\sqrt{R_t^2 + R_s^2}} - 1 = +.20$$

4.1.2.2.6 Section 3-3 Stress Investigation

4.1.2.2.6.1 Section Properties

$$A = 2\left[(5.95)(1.5) - (.5)(1.25)(.875) - (1.44)(1.5)\right] + (.58)(3.3)$$

$$= 2(8.925 - .547 - 2.16) + 1.914 = 14.35 \text{ in}^2$$

$$\bar{x} = \frac{2\left[(8.925)(2.975) - (.547)(5.53) - (2.16)(2.15)\right] + (1.914)(.29)}{14.35}$$

$$= \frac{2(26.55 - 3.03 - 4.65) + .56}{14.35} = \frac{38.30}{14.35} = 2.67 \text{ in.}$$

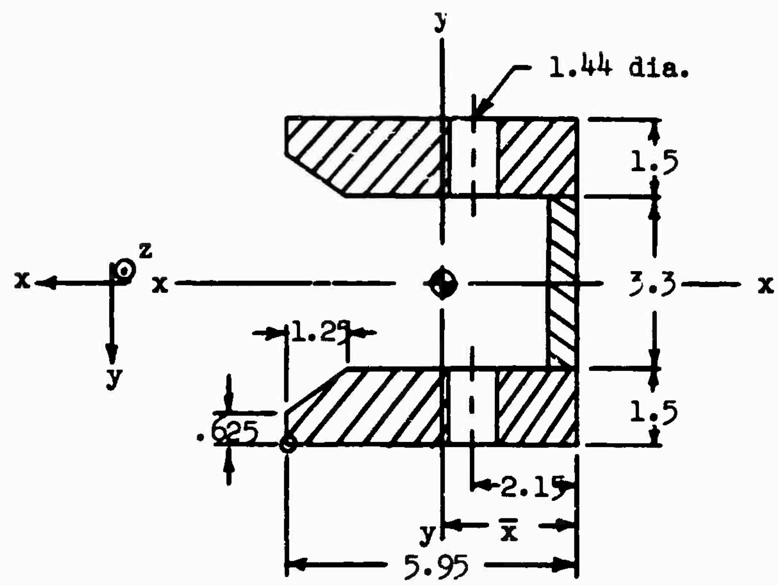


Figure 10. Main Engine Mount - Section 3-3 Geometry.

$$I_{y-y} = 2\left[\frac{(1.5)(5.95)^3}{12} + (8.925)(.305)^2 - \frac{(.875)(1.25)^3}{36} - (.547)(2.86)^2 - \frac{(1.5)(1.44)^3}{12} - (2.16)(.52)^2\right] + \frac{(3.3)(.58)^3}{12} + (1.914)(2.38)^2$$

$$= 2(26.35 + .83 - .05 - 4.48 - .58 - .49) + .05 + 10.85 = 54.06 \text{ in}^4$$

$$I_{x-x} = 2\left[\frac{(5.95)(1.5)^3}{12} + (8.925)(2.40)^2 - \frac{(1.25)(.875)^3}{36} - (.547)(1.36)^2 - \frac{(1.44)(1.5)^3}{12} - (2.16)(2.40)^2\right] + \frac{(.58)(3.3)^3}{12} + (1.914)(0)^2$$

$$= 2(1.67 + 51.40 - .02 - 1.01 - .41 - 12.42) + 1.74 + 0 = 80.16 \text{ in}^4$$

4.1.2.2.6.2 Loading Analysis

Rotor Limit Speed Condition: -

C.F. = 136,000 lb. (limit)

W_T = 370 lb. (limit)

M_y = (13.00)(68300)+(4.32)(370)
= 890,000 in-lb. (limit)

P_x = 136,000 lb. (limit)

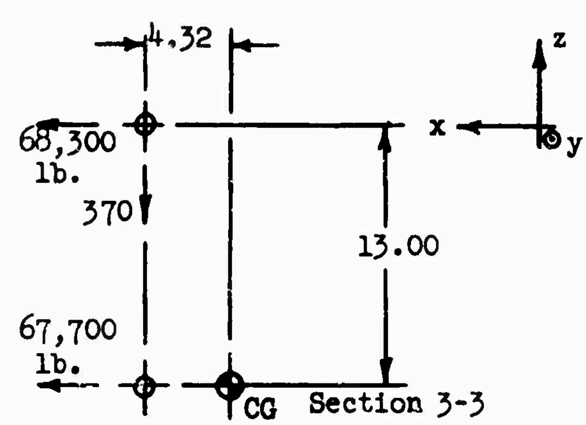


Figure 11. Main Engine Mount - Section 3-3 Loading Sketch (Cond. 1).

Rotor Overspeed Operation - Both Engines Operating: -

P_x = 93,200 lb. (limit)
P_y = 3,550 lb. (limit)
P_z = 14,800 lb. (limit)
M = 47,000 in-lb. (limit)
M_y = 166,500 in-lb. (limit)

(Ref. Figure 3, page 12)

$$\circlearrowright\!\!+\ M_{y-y} = M_y + 4.32\ P_z + 6.5\ P_x$$
$$= 166{,}500 + (4.32)(14800) + (6.5)(93{,}200)$$
$$= 836{,}500 \text{ in-lb. (limit)}$$

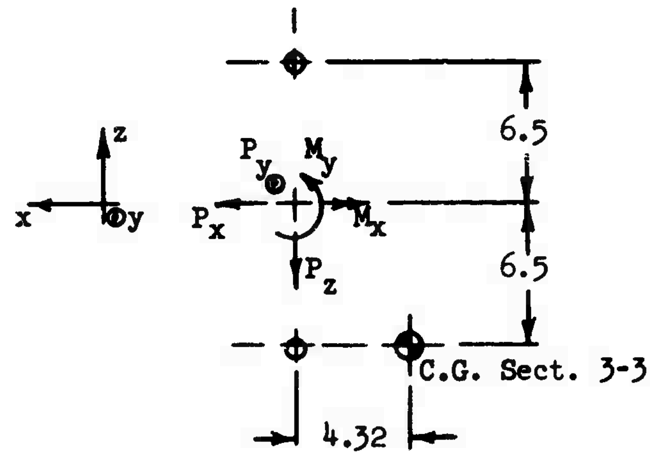


Figure 12. Main Engine Mount - Section 3-3 Loading Sketch (Cond. 3).

$$\circlearrowright\!\!+\ M_{x-x} = M_x + \frac{13P_y}{2}$$
$$= 47{,}000 + \frac{(13)(3550)}{2}$$
$$= 70{,}100 \text{ in-lb. (limit)}$$

$$\circlearrowright\!\!+\ T_z = 4.32P_y = (4.32)(3550)$$
$$= 15{,}350 \text{ lb-in. (limit)}$$

4.1.2.2.6.3 Stress Analysis (at point A)

Rotor Limit Speed Condition: -

$$M_y = 890{,}000 \text{ in-lb. (limit)}$$
$$f_b = \frac{(8.9)(10^5)(3.28)}{54.06} = 54{,}000 \text{ p.s.i. (limit)}$$

The shear load is assumed to be carried by the two members extending along the x-axis.

$$A_s = 2(8.925 - .547 - 2.16) = 12.44 \text{ in}^2$$
$$P_x = 136{,}000 \text{ lb. (limit)}$$
$$f_s = \frac{136{,}000}{12.44} = 10{,}900 \text{ p.s.i. (limit)}$$

The material properties are obtained from page 13, and are used in the stress ratios presented below.

$$R_b = \frac{(1.5)(54{,}000)}{127{,}000} = .638\ , \qquad R_s = \frac{(1.5)(10{,}900)}{74{,}000} = .221$$

Combining the stress ratios vectorially, the margin of safety is:

$$M.S. = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1 = +.48$$

Rotor Overspeed Operation - Both Engines Operating:

$$M_{yy} = 836,500 \text{ in-lb. (limit)}$$

$$F_{b_y} = \frac{(8.365)(10^5)(3.28)}{54.06} = 50,700 \text{ p.s.i. (limit)}$$

$$M_{xx} = 70,100 \text{ in-lb. (limit)}$$

$$F_{b_x} = \frac{(7.01)(10^4)(3.15)}{80.16} = 2,760 \text{ p.s.i. (limit)}$$

To determine the torsional stress distribution into the two members extending along the x-axis, the member thicknesses are reduced by the area of the removed material.

$$t_1 = \frac{8.925 - .547 - 2.16}{5.95} = 1.045 \text{ in.}$$

$$T = 15,350 \text{ lb-in. (limit)}$$

$$f_{s_T} = \frac{3T}{\Sigma bt^2} = \frac{(3)(15,350)}{(5.95)(1.045)^2(2) + (3.3)(.58)^2}$$

$$= 3,250 \text{ p.s.i. (limit)}$$

$$f_{s_x} = \frac{93,200}{12.44} = 7,500 \text{ p.s.i. (limit)}$$

$$f_{s_y} = \frac{3,550}{1.941} = 1,800 \text{ p.s.i. (limit)}$$

$$f_c = \frac{14,800}{14.35} = 1,030 \text{ p.s.i. (limit)}$$

The maximum compressive stress is:

$$f_{c_{max}} = f_{b_y} + f_{b_x} + f_c = 54,500 \text{ p.s.i. (limit)}$$

The maximum shear stress is:

$$f_{s_{max}} = f_{s_T} + f_{s_x} + f_{s_y} = 12,600 \text{ p.s.i. (limit)}$$

$$R_b = \frac{(1.5)(54,000)}{127,000} = .644 , \qquad R_s = \frac{(1.5)(12,600)}{74,000} = .256$$

Combining the compressive and shear stresses vectorially, the margin of safety is:

$$M.S. = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1 = +.44$$

4.1.2.2.7 Heat-Expansion-Fitting Stress Investigation

The heat expansion fitting is designed to rotate in the x-z plane due to engine thermal expansion. It can carry tension or compression loads and bending moments about the x or z axis.

4.1.2.2.7.1 Loading Analysis

Rotor Limit Speed Condition:

$$P_x = 68{,}300 \text{ lb. (limit)} \quad \text{(Ref. page 16)}$$

Rotor Overspeed Operation - Both Engines Operating:

The loading imposed during condition 3 is obtained from Figure 3 presented on page 12.

$$P_{CF} = 93{,}200 \text{ lb. (limit)}$$

$$P_y = 3{,}550 \text{ lb. (limit)}$$

$$P_z = 14{,}800 \text{ lb. (limit)}$$

$$M_x = 47{,}000 \text{ in-lb. (limit)}$$

$$M_y = 166{,}500 \text{ in-lb. (limit)}$$

The heat expansion lug on the lower engine attachment is critical.

$$P_x = \frac{P_{CF}}{2} + \frac{M_y}{13} = \frac{93{,}200}{2} + \frac{166{,}500}{13} = 59{,}400 \text{ lb. (limit)}$$

$$P_y = \frac{P_y}{2} + \frac{M_x}{13} = \frac{3{,}550}{2} + \frac{47{,}000}{13} = 5{,}400 \text{ lb. (limit)}$$

4.1.2.2.7.2 Engine-Fitting Stress Analysis

Rotor Limit Speed Condition:

$$P_x = 68{,}300 \text{ lb. (limit)}$$

For link tensile tear-out:

$$A_t = (2)(1.15 - .72)(1.3)(2) = 2.24 \text{ in}^2$$

$$f_t = \frac{68{,}300}{2.24} = 30{,}500 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{127{,}000}{(1.5)(30{,}500)} - 1 = \quad 1.78$$

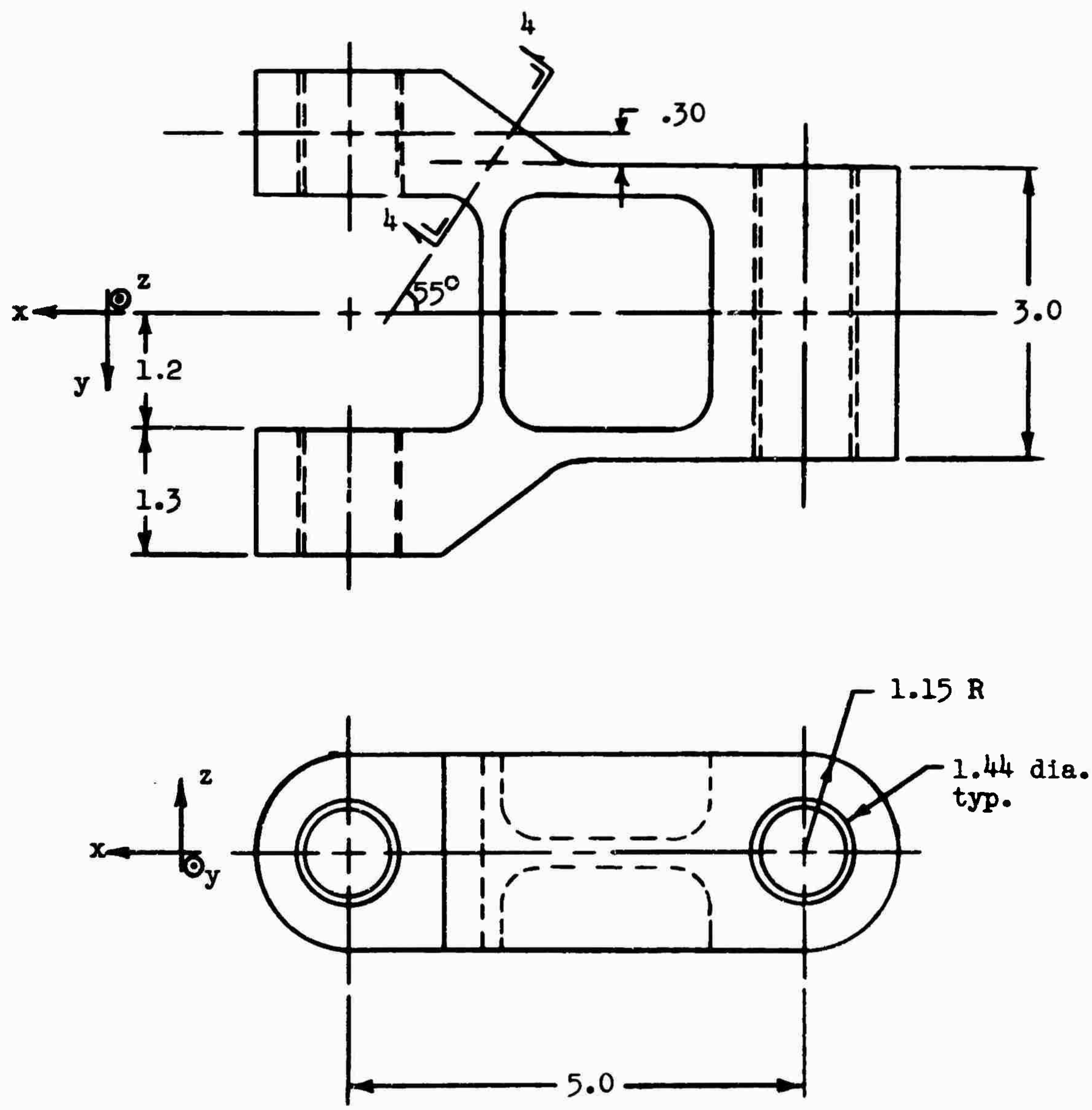


Figure 13. Heat-Expansion-Fitting General Geometry Sketch.

For link shear-out:

$$A_s = A_t = 2.24 \text{ in}^2$$

$$f_s = \frac{68,300}{2.24} = 30,500 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{74,000}{(1.5)(30,500)} - 1 = \quad .62$$

<u>Rotor Overspeed Operation - Both Engines Operating:</u>

$$P_x = 59,400 \text{ lb. (limit)}$$

The engine-fitting lugs are satisfactory for the condition 3 loads by comparison to the higher loading developed during the rotor limit speed condition.

4.1.2.2.7.3 Section 4-4 Stress Investigation

4.1.2.2.7.3.1 Section Properties

$$A = (2.0)(.78) = 1.56 \text{ in}^2$$

$$I_{y-y} = \frac{(2.0)(.78)^3}{12} = .079 \text{ in}^4$$

$$I_{z'-z'} = \frac{(.78)(2.0)^3}{12} = .52 \text{ in}^4$$

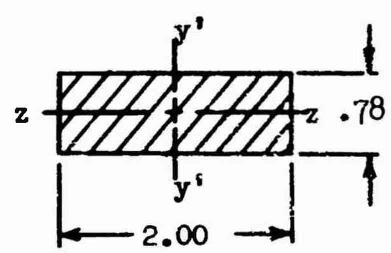


Figure 14. Expansion Fitting - Section 4-4 Geometry.

4.1.2.2.7.3.2 Loading Analysis

<u>Rotor Limit Speed Condition:</u>

$$P_x = \frac{68,300}{2} = 34,200 \text{ lb. (limit)}$$

$$M_z = (.30)(34,200) = 10,260 \text{ in-lb. (limit)}$$

$$P_{x'} = P_x \cos 35^\circ = 34,200 \cos 35^\circ = 28,000 \text{ lb. (limit)}$$

$$P_{y'} = P_x \sin 35^\circ = 34,200 \sin 35^\circ = 19,600 \text{ lb. (limit)}$$

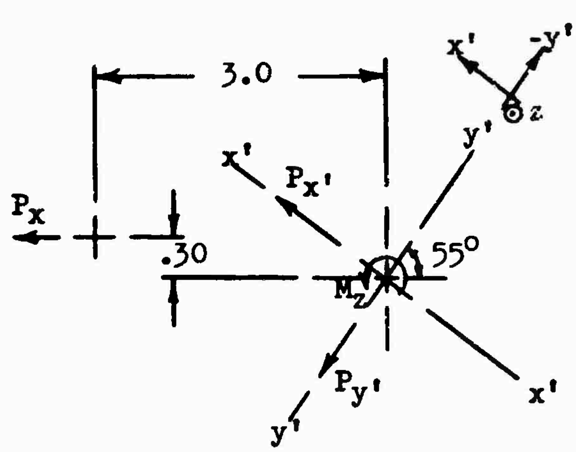


Figure 15. Expansion Fitting - Section 4-4 Loading Sketch (Cond. 1).

Rotor Overspeed Operation - Both Engines Operating:

$$P_x = \frac{59,400}{2} = 29,700 \text{ lb. (limit)}$$

$$P_y = \frac{5,400}{2} = 2,700 \text{ lb. (limit)}$$

$$\begin{aligned} M_z &= (.30)P_x + (3.0)P_y \\ &= (.3)(29,700) + (3)(2,700) \\ &= 17,000 \text{ in-lb. (limit)} \end{aligned}$$

$$\begin{aligned} P_{y'} &= P_x \cos 55^\circ + P_y \sin 55^\circ \\ &= 29,700 \cos 55^\circ + 2,700 \sin 55^\circ \\ &= 17,000 + 2,200 \\ &= 19,200 \text{ lb. (limit)} \end{aligned}$$

$$\begin{aligned} P_{x'} &= P_x \sin 55^\circ - P_y \cos 55^\circ \\ &= 29,700 \sin 55^\circ - 2,700 \cos 55^\circ \\ &= 24,300 - 1,600 \\ &= 22,700 \text{ lb. (limit)} \end{aligned}$$

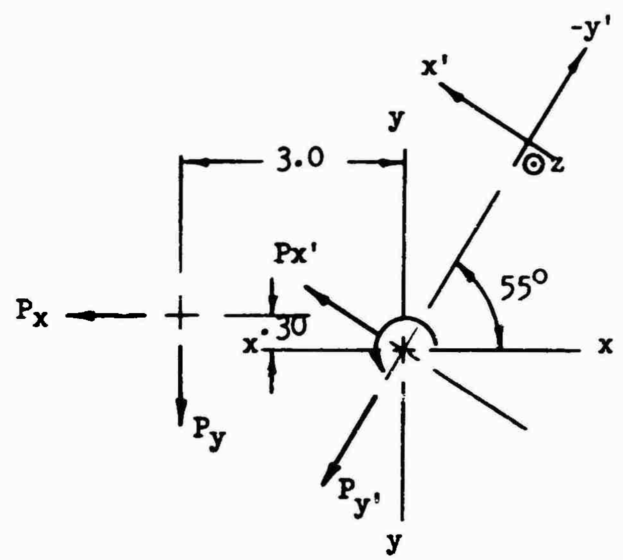


Figure 16. Expansion Fitting - Section 4-4 Loading Sketch (Cond. 5).

4.1.2.2.7.3.3 **Stress Analysis**

Rotor Limit Speed Condition:

$$f_b = \frac{(10,260)(.39)}{.079} = 50,600 \text{ p.s.i. (limit)}$$

$$f_t = \frac{28,000}{1.56} = 18,000 \text{ p.s.i. (limit)}$$

$$f_s = \frac{19,600}{1.56} = 12,600 \text{ p.s.i. (limit)}$$

The material properties are obtained from page 13 of this report and are used in the stress ratios presented below.

$$R_b = \frac{(1.5)(50,600 + 18,000)}{127,000} = .810$$

$$R_s = \frac{(1.5)(12,600)}{74,000} = .256$$

Combining the bending and shear stress ratios, the margin of safety is:

$$M.S. = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1 = +.18$$

Rotor Overspeed Operation - Both Engines Operating:

$$f_b = \frac{(17,000)(.39)}{.079} = 84,000 \text{ p.s.i. (limit)}$$

$$f_t = \frac{22,700}{1.56} = 14,600 \text{ p.s.i. (limit)}$$

$$f_s = \frac{19,200}{1.56} = 12,300 \text{ p.s.i. (limit)}$$

The bending modulus of rupture factor for rectangular sections is 1.5 (Ref. 6, page 320). Using the 1.5 bending modulus of rupture factor, the bending stress ratio is:

$$R_b = \frac{(1.5)(84,000)}{(1.5)(127,000)} = .661$$

$$R_t = \frac{(1.5)(14,600)}{127,000} = .1725$$

$$R_s = \frac{(1.5)(12,300)}{74,000} = .25$$

The margin of safety is obtained by combining the shear stress ratio with the bending and tensile stress ratios.

$$\text{M.S.} = \frac{1}{\sqrt{(R_b + R_t)^2 + R_s^2}} - 1 = \quad .15$$

4.1.2.2.7.4 Mount-Fitting-Lug Stress Analysis

Rotor Limit Speed Condition:

$$P_x = 68,300 \text{ lb. (limit)}$$

For link tensile tear-out:

$$A_t = (3.0)(1.15 - .72)(2) = 2.58 \text{ in}^2$$

$$f_t = \frac{68,300}{2.58} = 26,500 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{127,000}{(1.5)(26,500)} - 1 = \quad 2.20$$

For link shear-out:

$$A_s = A_t = 2.58 \text{ in}^2$$

$$f_s = \frac{68,300}{2.58} = 26,500 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{74,000}{(1.5)(26,500)} - 1 = \quad .87$$

Rotor Overspeed Operation - Both Engines Operating:

$$P_x = 59{,}400 \text{ lb. (limit)}$$

The mount-fitting lug is satisfactory for the condition 3 loads by comparison to the higher loading developed during the rotor limit speed condition.

4.1.2.2.8 Section 5-5 Stress Investigation

Note: The four .25-inch thick webs are neglected in the section properties calculations.

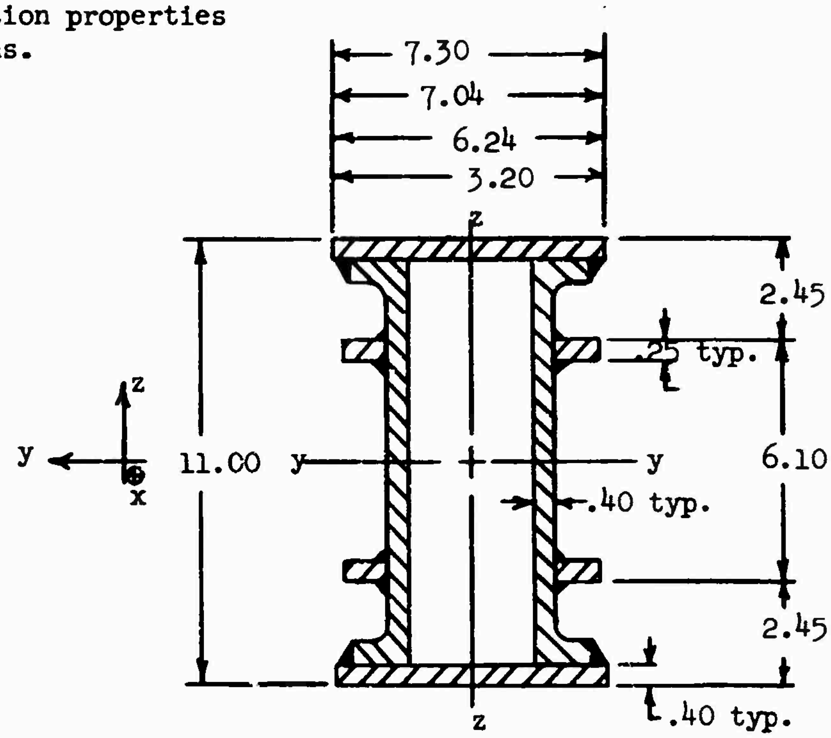


Figure 17. Main Engine Mount - Section 5-5 Geometry.

4.1.2.2.8.1 Section Properties

$$A = (7.3)(11.00) - (3.2)(10.2) - 2\left[(1.52)(9.4) + (.53)(10.2)\right]$$
$$= 80.30 - 32.6 - 2(14.3 + 5.4) = 80.30 - 72.00$$
$$= 8.30 \text{ in}^2$$

$$I_{y-y} = \frac{(7.3)(11.0)^3}{12} - \frac{(3.2)(10.2)^3}{12} - 2\left[\frac{(1.5)(9.4)^3}{12} + \frac{(.53)(10.2)^3}{12}\right]$$
$$= 810 - 283 - 2(105 + 47) = 810 - 587 = 223.0 \text{ in}^4$$

$$I_{z-z} = \frac{(11.0)(7.3)^3}{12} - \frac{(10.2)(3.2)^3}{12} - 2\left[\frac{(9.4)(1.52)^3}{12} + (14.3)(2.36)^2 + \frac{(10.2)(.53)^3}{12} + (5.4)(3.38)^2\right]$$

$$= 356 - 27.9 - 2(2.85 + 79.8 + .13 + 61.7) = 356 - 27.9 - 288.8$$

$$= 39.3 \text{ in}^4$$

4.1.2.2.8.2 Loading Analysis

Rotor Limit Speed Condition:

P_{CF} = 308,700 lb. (limit) (Ref. Fig. 1, page 10)

Rotor Overspeed Operation - Both Engines Operating:

From Fig. 3, page 12:

P_x = 212,100 lb. (limit)
P_y = 7,100 lb. (limit)
P_z = 29,600 lb. (limit)

M_x = 94,000 in-lb. (limit)
M_y = 710,000 in-lb. (limit)

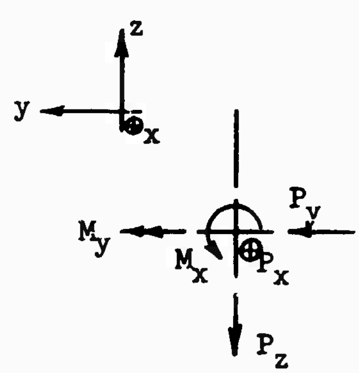


Figure 18. Main Engine Mount - Section 5-5 Loading Sketch (Cond.3).

4.1.2.2.8.3 Stress Analysis

Rotor Limit Speed Condition:

$$A = 8.30 \text{ in}^2$$

$$f_t = \frac{P_{CF}}{A} = \frac{308,700}{8.30} = 37,200 \text{ p.s.i. (limit)}$$

Using a 1.15 weld factor, the margin of safety is:

$$M.S. = \frac{127,000}{(1.15)(1.5)(37,200)} - 1 = \quad .98$$

Rotor Overspeed Operation - Both Engines Operating:

$A = 8.30 \text{ in}^2$, $I_{y-y} = 223 \text{ in}^4$, $I_{z-z} = 39.3 \text{ in}^4$
$C_y = 5.5$ in. $C_z = 3.65$ in.

$$f_{b_y} = \frac{(710{,}000)(5.5)}{223.0} = 17{,}500 \text{ p.s.i. (limit)}$$

$$f_{s_T} = \frac{M_x}{2K} = \frac{94{,}000}{(2)(15.25)} = 3{,}080 \text{ p.s.i. (limit)}$$

where: $K = (.4)(10.6)(3.6) = 15.25 \text{ in}^3$

$$f_{s_z} = \frac{29{,}600}{8.30} = 3{,}570 \text{ p.s.i. (limit)}$$

$$f_t = \frac{212{,}100}{8.30} = 25{,}600 \text{ p.s.i. (limit)}$$

Using a 1.15 weld factor, the ultimate stress ratios are:

$$R_b = \frac{(1.15)(1.5)(17{,}500 + 25{,}600)}{127{,}000} = .585$$

$$R_s = \frac{(1.15)(1.5)(3{,}080 + 3{,}570)}{74{,}000} = .155$$

Combining the bending and shear stress ratios, the margin of safety is:

$$\text{M.S.} = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1 = \quad .68$$

4.1.2.2.9 Engine-Mount-to-Blade-Tip Attachment Lugs

Material properties for Ti-6AL-4V titanium alloy are obtained from page 13.

At 400° F.: $F_{tu} = 127{,}000$ p.s.i.

$F_{su} = 74{,}000$ p.s.i.

4.1.2.2.9.1 Loading Analysis

Rotor Limit Speed Condition:

$P_{CF} = 308{,}700$ lb. (limit) (Ref. page 10, Figure 1)

Rotor Overspeed Operation - Both Engines Operating:

The condition 3 loading is obtained from page 12, Figure 3.

$P_x = 212{,}100$ lb. (limit)

$P_y = 7{,}100$ lb. (limit)

$P_z = 29{,}600$ lb. (limit)

$M_x = 94{,}000$ in-lb. (limit)

$M_y = 710{,}000$ in-lb. (limit)

4.1.2.2.9.2 Section Properties

Note: Nc scale.

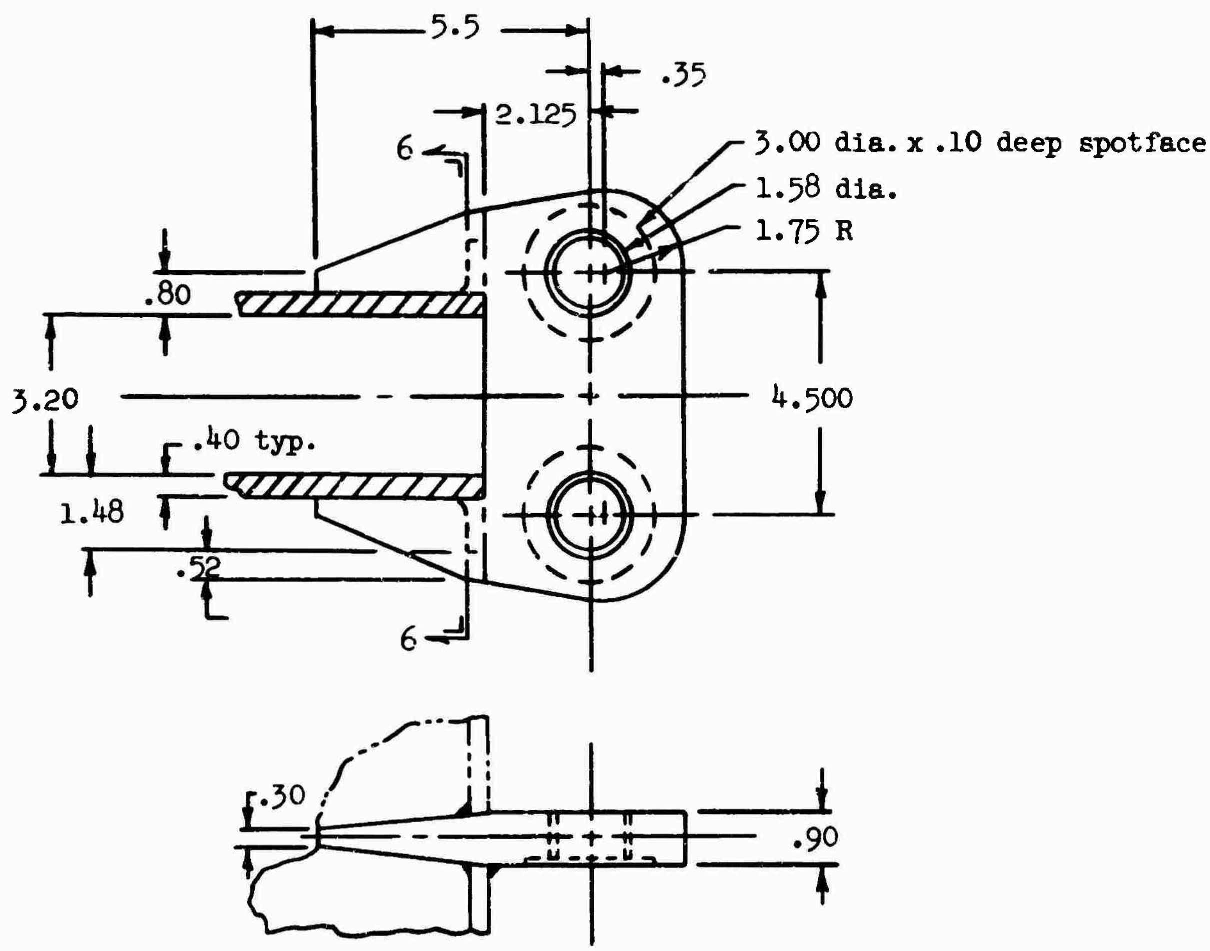


Figure 19. Inner Lug - General Geometry Sketch.

The section properties at the section under analysis are presented in that section's stress analysis.

4.1.2.2.9.3 Inner-Lug-to-Mount-Weldment Stress Analysis

Preliminary analysis has established condition 3 to be critical.

$$\text{Weld area} = (4)(.20)(5.50 - 2.125) = 2.70 \text{ in}^2$$

$$P_x = \frac{212,100}{4} + \frac{710,000}{(2)(7.6)} = 99,750 \text{ lb. (limit)}$$

$$f_s = \frac{99,750}{2.70} = 37,000 \text{ p.s.i. (limit)}$$

Using a 1.15 weld factor, the margin of safety is:

$$\text{M.S.} = \frac{74,000}{(1.15)(1.5)(37,000)} - 1 = \quad +.16$$

4.1.2.2.9.4 Section 6-6 Stress Analysis

Preliminary analysis has established condition 3 to be critical.

$$\text{Area} = (2)(1.60)(.85) = 2.72 \text{ in}^2$$

$$P_x = 99,750 \text{ lb. (limit)}$$

$$f_t = \frac{99,750}{2.72} = 36,700 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{127,000}{(1.5)(36,700)} - 1 = \quad +1.31$$

4.1.2.2.9.5 Bolt Hole Stress Analysis

The load per bolt is:

$$P = \frac{99,750}{2} = 50,000 \text{ lb. (limit)}$$

Lug Tension Tear-Out:

The lug tensile load is conservatively assumed to be carried as shown in the loading sketch.

$$\frac{P}{2} = \frac{50,000}{2} = 25,000 \text{ lb. (limit)}$$

$$A = (1.00)(.80) = .80 \text{ in}^2$$

$$f_t = \frac{25,000}{.80} = 31,200 \text{ p.s.i. (limit)}$$

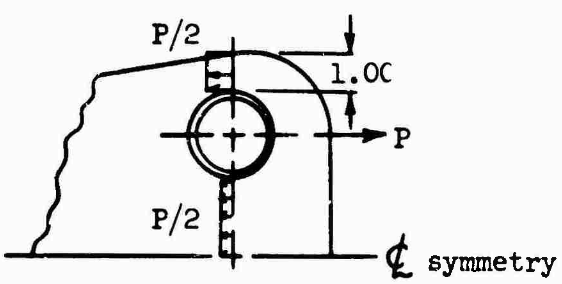


Figure 20. Inner Lug Loading Distribution.

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{127,000}{(1.15)(1.5)(31,200)} - 1 = \quad 1.36$$

Lug Shear-Out:

Shear area: $A = (2)(.80)(1.31) = 2.10 \text{ in}^2$

$$f_s = \frac{50,000}{2.10} = 23,800 \text{ p.s.i. (limit)}$$

Using a fitting factor of 1.15, the margin of safety is:

$$M.S. = \frac{74,000}{(1.15)(1.5)(23,800)} - 1 = \quad .80$$

4.1.2.2.9.6 Outer Lug-to-Mount Weldment Stress Analysis

This portion of the main engine mount stress analysis is satisfactory by comparison to the inner lug to mount weldment stress analysis.

4.1.2.2.10 Main Engine Mount to Blade Tip Attachment Bolt Analysis

Bolt dia.: = 1.375 in.
Mat'l type: H1 Ti 20 series bolts

The material properties are obtained from page 14 of this report.

At 400° F. temperature: $F_{tu} = 160,000$ p.s.i.
$F_{su} = 96,000$ p.s.i.

4.1.2.2.10.1 Bolt Shear Stress Analysis

$$A = \frac{\pi}{4}(1.375)^2 = 1.485 \text{ in}^2$$

Preliminary analysis has established condition 3 to be critical. The load per bolt is:

$P_x = 99,750$ lb. (limit) (Ref. page 33)

$$f_s = \frac{99,750}{1.485} = 33,600 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{96,000}{(1.15)(1.5)(33,600)} - 1 = \quad .66$$

4.1.2.2.10.2 Bolt Bending Stress Analysis

The method of analysis used in determining the bolt bending-stress level is obtained from pages 160 through 164 of Reference 9. In the following analysis, the applied load is assumed to "peak-up" on the inner lug near the shear planes rather than be carried as a uniform load across the inner lug.

P = 99,750 lb. (limit)

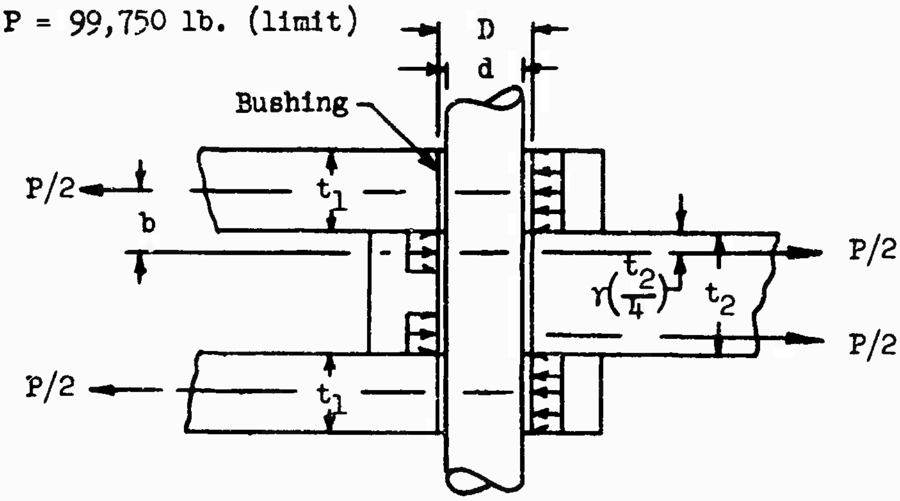


Figure 21. Bolt and Lug Loading Sketch Main-Mount-to-Blade-Tip Attachment

Lug material properties:

F_{tu} = 127,000 p.s.i.

F_{su} = 74,000 p.s.i.

d = 1.38 in.

D = 1.58 in. $\qquad t_1$ = .80 in.

W = 3.6 in. $\qquad t_2$ = 1.00 in.

a = 2.1 in.

$$r = \left[\frac{a}{D} - \frac{1}{2}\right]\frac{D}{t_2} = \left[\frac{2.1}{1.58} - \frac{1}{2}\right]\frac{1.58}{1.00} = 1.31$$

$$A_{br} = Dt_2 = (1.58)(1.00) = 1.58 \text{ in}^2$$

$$A_t = (W-D)t_2 = (3.6 - 1.58)(1.00) = 2.02 \text{ in}^2$$

$$P'_{br_u} = A_{br}F_{su} = (1.58)(74,000) = 117,000 \text{ lb.}$$

$$P'_{tu} = A_t\, F_{tu} = (2.02)(127000) = 263,000 \text{ lb.}$$

Since $P'_{br_u} < P'_{tu}$

$$P_{u_{min}} = P'_{br_u} = 117{,}000 \text{ lb.}$$

$$\frac{P_{u_{min}}}{A_{br}F_{tu}} = .583$$

$$\gamma = .28 \quad \text{(Ref. 9, page 162, Figure 4)}$$

$$b = \frac{t_1}{2} + \gamma\left(\frac{t_2}{4}\right) = \frac{.80}{2} + (.28)\left(\frac{1.00}{4}\right) = .47 \text{ in.}$$

$$M = \left(\frac{P}{2}\right)b = \frac{99{,}750}{2}(.47) = 23{,}400 \text{ in-lb. (limit)}$$

$$Z = .2552 \text{ in}^3$$

$$f_b = \frac{23{,}400}{.2552} = 91{,}800 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{160{,}000}{(1.15)(1.5)(91{,}800)} - 1 = +.01$$

4.1.2.3 Aft Engine Mount Stress Analysis - (Ref. Volume III, Figure 21)

This portion of the main rotor system stress analysis is concerned with the static structural substantiation of the aft engine mount, including the engine and the blade-tip-to-mount attachments.

The material properties presented below are obtained from Reference 10.

Material type: Ti-6AL-4V titanium alloy

At room temperature:

$$\left.\begin{array}{l} F_{tu} = 162{,}000 \text{ p.s.i.} \\ F_{su} = 94{,}000 \text{ p.s.i.} \end{array}\right\} \text{(Ref. 10, page 39)}$$

At 400°F. temperature:

$$\left.\begin{array}{l} F_{tu} = (.785)(162{,}000) = 127{,}000 \text{ p.s.i.} \\ F_{su} = (.785)(94{,}000) = 74{,}000 \text{ p.s.i.} \end{array}\right\} \text{(Ref. 10, pages 52, 53)}$$

4.1.2.3.1 Loading Analysis

From a review of References 1 and 2, the critical aft engine mount loads occur during the rotor overspeed operation with both engines operating. A loading analysis is presented in each section and/or components analysis.

4.1.2.3.2 Engine-to-Mount-Attachment Bolts

.625 dia. bolts, NAS 464, $A = .3068 \ in^2$, $Z = .024 \ in^3$

Mat'l type: 7AL-12ZR titanium alloy

The material properties presented below are obtained from Reference 12.

At 400°F. Temperature:

$$F_{tu} = (.97)(145,000) = 141,000 \text{ p.s.i.}$$
$$F_{su} = (.97)(90,000) = 87,000 \text{ p.s.i.}$$

The maximum bolt load is obtained from Figure 3 presented on page 12.

$$P_x = 2,600 \text{ lb. (limit)}$$

Bolt Shear Stress Analysis:

The bolt is loaded in double shear.

$$A_s = (2)(.3068) = .6136 \ in^2$$
$$f_s = \frac{2,600}{.6136} = 4,250 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{87,000}{(1.15)(1.5)(4250)} - 1 = \quad \text{High}$$

Bolt Bending Stress Analysis:

The method of analysis used in determining the bolt bending stress level is obtained from pages 160 through 164 of Reference 9. In the following analysis, the applied load is conservatively assumed to be carried as a uniform load across the lugs.

$$P = 2,600 \text{ lb. (limit)}$$
$$b = .125 + .225 + .1375 = .4875 \text{ in.}$$
$$M = \frac{P}{2}(b) = \frac{2,600}{2}(.4875) = 634 \text{ in-lb. (limit)}$$
$$f_b = \frac{634}{.024} = 26,400 \text{ p.s.i. (limit)}$$

Using a fitting factor of 1.15, the margin of safety is:

$$M.S. = \frac{141,000}{(1.15)(1.5)(26,400)} - 1 = \quad 2.10$$

4.1.2.3.3 Engine-to-Aft-Mount Attachment Links

The aft-mount-to-engine attachment lugs are designed to carry axial tension or compression loads. Freedom of movement is provided by a spherical or monoball type bearing.

Material type: Ti-6AL-4V titanium alloy
The material properties presented below are obtained from page 13.

At 400°F. temperature: F_{tu} = 127,000 p.s.i.

F_{su} = 74,000 p.s.i.

4.1.2.3.3.1 Loading Analysis

$$P_x = 2{,}600 \text{ lb. (limit)}$$

4.1.2.3.3.2 Lug Shear Stress Analysis

Shear area: $A_s = (2)(.15)(.55) = .165 \text{ in}^2$

$$f_s = \frac{2{,}600}{.165} = 15{,}800 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{74{,}000}{(1.15)(1.5)(15{,}800)} - 1 = \quad 1.72$$

4.1.2.3.3.3 Lug Tensile Stress Analysis

Tensile area: $A_t = .165 \text{ in}^2$

$$f_t = \frac{2{,}600}{.165} = 15{,}800 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{127{,}000}{(1.15)(1.5)(15{,}800)} - 1 = \quad 3.66$$

4.1.2.3.4 Engine to Aft Mount Attachment Lugs

Material type: Ti-6AL-4V titanium alloy
The material properties presented below are obtained from page 13.

At 400° F. temperature:

F_{tu} = 127,000 p.s.i.

F_{su} = 74,000 p.s.i.

4.1.2.3.4.1 Loading Analysis

$$P_x = 2{,}600 \text{ lb. (limit)}$$

4.1.2.3.4.2 Lug Shear Stress Analysis

Shear area: $A_s = (2)(.25)(.48)(2) = .48 \text{ in}^2$

$$f_s = \frac{2{,}600}{.48} = 5{,}420 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{74{,}000}{(1.15)(1.5)(5420)} - 1 =$$ High

4.1.2.3.4.3 Lug Tensile Stress Analysis

The lug is satisfactory for the applied tensile stress by comparison to the shear stress analysis presented above.

4.1.2.3.5 Section 7-7 Stress Analysis

Section Properties:

$$A = (2)(2.0)(.3)+(.2)(2.3)$$
$$= 1.2 + .46$$
$$= 1.66 \text{ in}^2$$

$$I_{xx} = \frac{(.20)(2.3)^3}{12} + 2\left[\frac{(2.0)(.3)^3}{12} + (2)(.3)(1.3)^2\right]$$
$$= .2025 + 2(.0045 + 1.015) = 2.24 \text{ in}^4$$

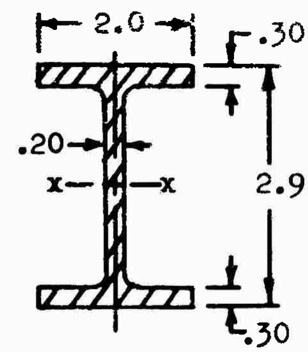


Fig. 22. Aft Engine Mount - Section 7-7 Geometry.

Loading Analysis:

$$M = 8.8P_x = (8.8)(2{,}600)$$
$$= 23{,}900 \text{ in-lb. (limit)}$$

Shear load:

$$P_s = P_x \cos 45^\circ = 2{,}600 \cos 45^\circ$$
$$= 1{,}840 \text{ lb. (limit)}$$

Compressive load:

$$P_c = P_x \sin 45^\circ$$
$$= 1{,}840 \text{ lb. (limit)}$$

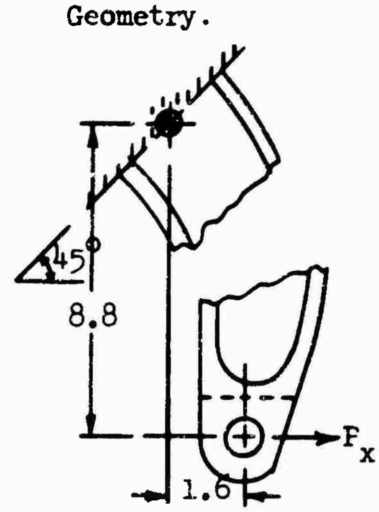


Fig. 23. Aft Engine Mount - Section 7-7 Loading Sketch.

Stress Analysis:

$$f_b = \frac{(23,900)(1.45)}{2.24} = 15,500 \text{ p.s.i. (limit)}$$

$$f_s = \frac{1,840}{1.66} = 1,110 \text{ p.s.i. (limit)}$$

$$f_c = \frac{1,840}{1.66} = 1,110 \text{ p.s.i. (limit)}$$

Combining the bending and tensile stress directly, and using the material ultimate tensile and shear strengths, the stress ratios are:

$$R_b = \frac{(1.5)(15,500 + 1,110)}{127,000} = .196$$

$$R_s = \frac{(1.5)(1,110)}{74,000} = .0225$$

$$\text{M.S.} = \frac{1}{\sqrt{R_b^2 + R_s^2}} - 1 = \quad \text{High}$$

4.1.2.3.6 Aft-Mount-to-Blade-Tip Attachment Lug

Loading Analysis:

P_x = 8,050 lb. (limit) (Ref. Figure 3, page 12)

The load per lug is:

$$P = \frac{8,050}{2} = 4,025 \text{ lb. (limit)}$$

Lug Shear-Out Stress Analysis:

The shear area is conservatively taken as the tensile area.

$$A_s = (2)(1.75 - .825)(.5)(2) = 1.85 \text{ in}^2$$

$$f_s = \frac{4,025}{1.85} = 2,180 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{74,000}{(1.15)(1.5)(2,180)} - 1 = \quad \text{High}$$

The remainder of the aft mount system is satisfactory by inspection.

4.1.2.3.7 Aft-Engine-Mount-to-Blade-Tip Attachment Bolts

.625 dia. bolts, type NAS 464

Mat'l type: 7AL-12ZR titanium alloy

$A = .3064 \text{ in}^2, \quad Z = .024 \text{ in}^3, \quad P = 4{,}050$ lb. (limit)

At 400°F. temperature: $\left.\begin{array}{l} F_{tu} = 141{,}000 \text{ p.s.i.} \\ F_{su} = 87{,}000 \text{ p.s.i.} \end{array}\right\}$ (Ref. page 37)

Bolt Shear Stress Analysis:

The bolt is loaded in double shear.

$$A_s = (2)(.3068) = .6136 \text{ in}^2$$

$$f_b = \frac{4{,}025}{.6136} = 6{,}570 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{87{,}000}{(1.15)(1.5)(6570)} - 1 =$$ High

Bolt Bending Stress Analysis:

The method of analysis presented below is obtained from page 16.

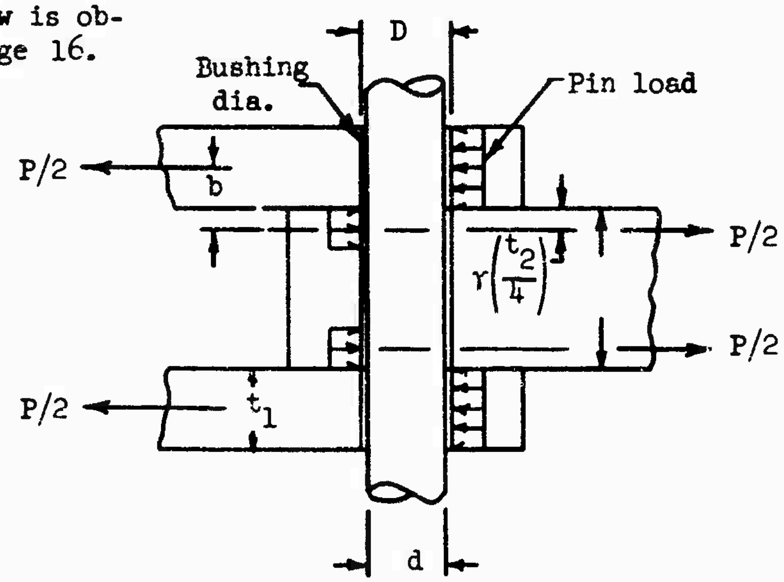


Figure 24. Aft-Mount-to-Blade-Tip Attachment - Lug and Bolt Loading Sketch.

Lug material properties: -

$$F_{tu} = 127{,}000 \text{ p.s.i.}, \quad F_{su} = 74{,}000 \text{ p.s.i.}$$

$$P'_{br_u} = (.825)(74{,}000) = 61{,}000 \text{ lb.}$$

$$P'_{tu} = (.925)(127{,}000) = 118{,}000 \text{ lb.}$$

$$\therefore \quad P'_{u_{min}} = 61{,}000 \text{ lb.}$$

$$\frac{P'_{u_{min}}}{A_{br}F_{tu_x}} = \frac{61,000}{(.825)(126,500)} = .585$$

$D = .825$ in.
$d = .625$ in.
$W = 1.75$ in.
$a = .85$ in.
$t_1 = .50$ in.
$t_2 = 1.00$ in.

$P = 9,220$ lb. (limit)

$\gamma = .68$ (Ref. 9, page 162, Figure 4)

$$r = \left[\frac{a}{D} - \frac{1}{2}\right]\frac{D}{t_2} = \left[\frac{.85}{.825} - \frac{1}{2}\right]\frac{.825}{1.75} = .217$$

$$b = \frac{t_1}{2} + \gamma\left(\frac{t_2}{4}\right) = \frac{.50}{2} + (.68)\ \frac{1.00}{4} = .42 \text{ in.}$$

$$A_{br} = Dt_2 = (.825)(1.00) = .825 \text{ in}^2$$

$$A_t = (W-D)(t_2) = (.925)(1.00) = .925 \text{ in}^2$$

$$M = \frac{P}{2}(b) = \frac{4,025}{2}(.42) = 845 \text{ in-lb. (limit)}$$

$$f_b = \frac{845}{.024} = 35,200 \text{ p.s.i. (limit)}$$

Using a fitting factor of 1.15, the margin of safety is:

$$M.S. = \frac{141,000}{(1.15)(1.5)(35,200)} - 1 =$$ 1.32

4.1.3 Main Rotor Blade Tip Engine Retention Structure - (Ref. Volume III, Figure 8)

This portion of the main rotor system stress analysis is concerned with the static structural substantiation of the main rotor blade tip, including the engine mount system pickup fittings and their attachment to the blade tip. The blade tip is analyzed in order of force transmission progressing from the engine mount system attachments to the blade primary structure.

The blade primary structure, including the engine mount system pickup fittings, is constructed of Ti-8AL-1Mo-1V titanium alloy The material properties presented below are obtained from Reference 11.

Material type: Ti-8AL-1Mo-1V titanium alloy

$F_{tu} = 130,000$ p.s.i. $E = 18.5 \times 10^6$ p.s.i.

$F_{su} = .57F_{tu} = 74,000$ p.s.i.

4.1.3.1 Loading Analysis

From a review of Section 4.1.2.1, the maximum blade-tip loading occurs during the rotor limit speed condition and during the rotor overspeed condition with both engines operating.

The engine-mount-system loads presented in Figures 1, 2, and 3 are applicable to the blade tip and attachments, and are used in the blade tip analysis.

4.1.3.2 Engine-Main-Mount Pickup Fittings

4.1.3.2.1 Lug Analysis

The pickup-fittings lugs are analyzed for shear tear-out during the rotor overspeed operation-both engines operating.

$P = 99{,}750$ lb. (limit) (Ref. page 33.)

Shear area: $A_s = (2)(1.35)(1.00) = 2.70 \text{ in}^2$

$$f_s = \frac{99{,}750}{2.70} = 36{,}900 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{74{,}000}{(1.15)(1.5)(36{,}900)} - 1 = \quad +.16$$

The attachment lugs are satisfactory for the applied tension and bearing loads by comparison to the lug shear analysis presented above.

4.1.3.2.2 Section 8-8 Stress Investigation

The section under analysis is located on Drawing Number 1108-532, presented as Figure 26 on page 44.

4.1.3.2.2.1 Section Properties

Note: No scale

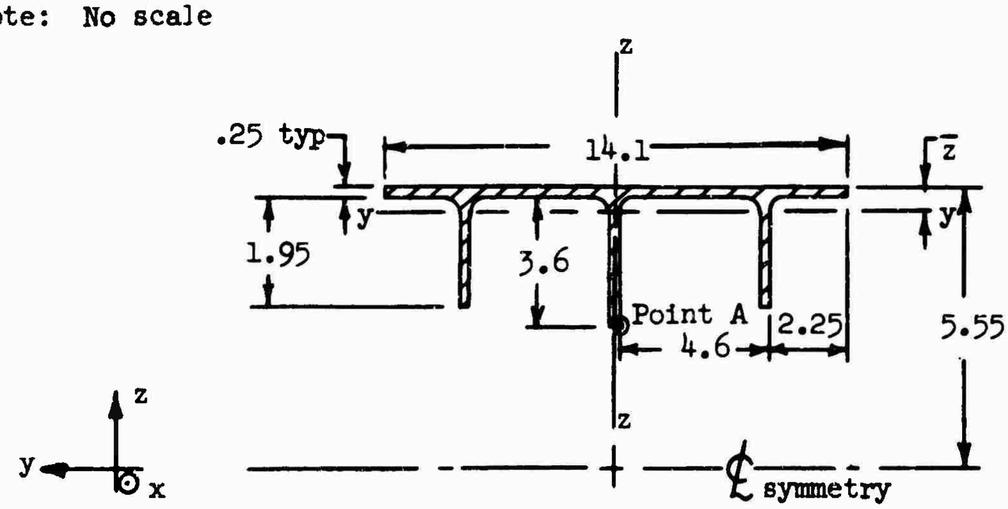


Figure 25. Upper-Main-Mount Pickup Fittings - Section 8-8 Geometry.

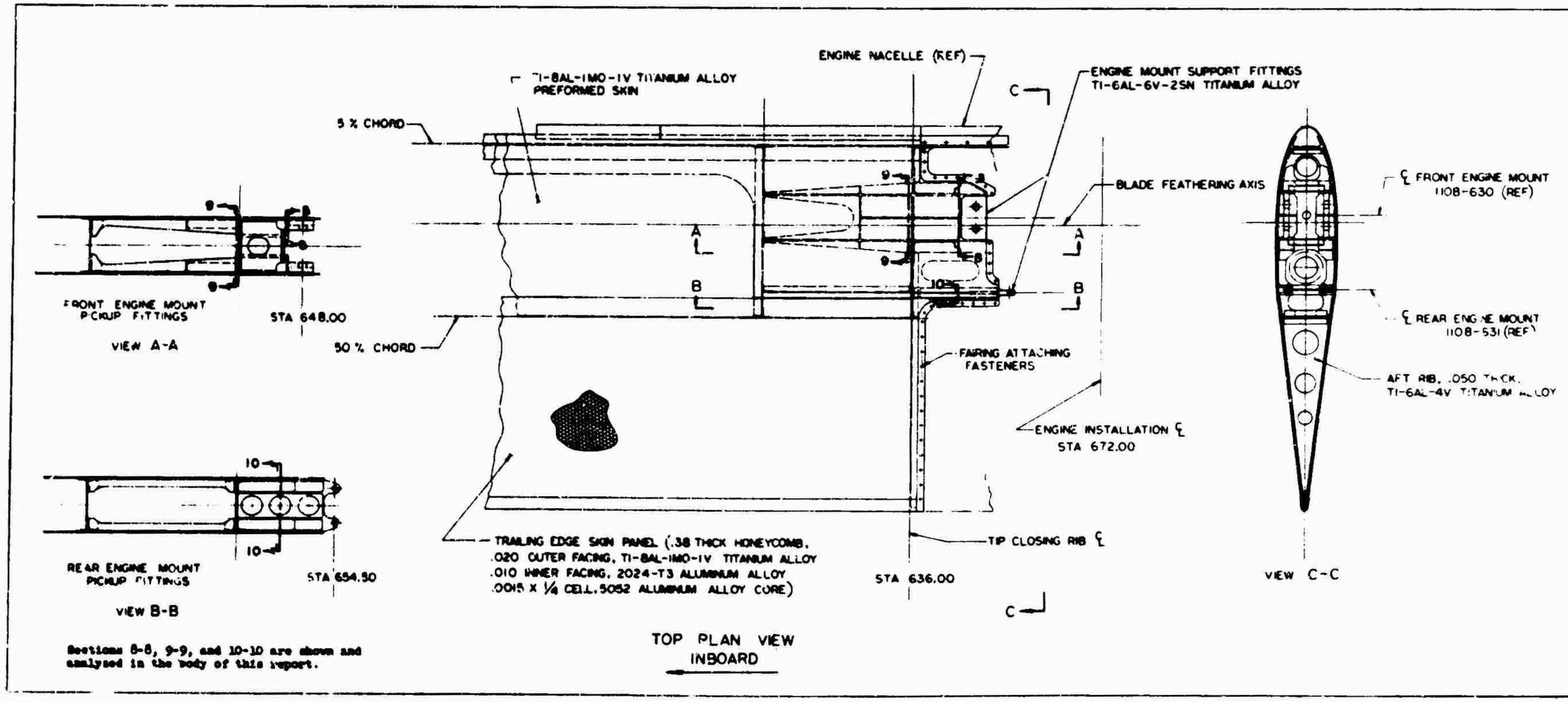


Figure 26. Engine Retention Structure - Main Rotor Blade Tip.

$$A = .25(14.1 + 1.95 + 3.6 + 1.95) = 5.40 \text{ in}^2$$

$$I_{z-z} = \frac{(.25)(14.1)^3}{12} + 2\left[\frac{(1.95)(.25)^3}{12} + (.25)(1.95)(4.6)^2\right] + \frac{(3.6)(.25)^3}{12}$$

$$= 54.4 + 2(.003 + 10.3) + .005 = 75.0 \text{ in}^4$$

$$\bar{z} = \frac{(.25)(14.1)(.125) + 2(.25)(1.95)(1.225) + (.25)(3.6)(2.05)}{5.40}$$

$$= \frac{.44 + .478 + 1.846}{5.40} = .401 \text{ in.}$$

$$I_{y-y} = \frac{(14.1)(.25)^3}{12} + (3.525)(.276)^2 + 2\left[\frac{(.25)(1.95)^3}{12} + (.4875)(.824)^2\right] + \frac{(.25)(3.6)^3}{12} + (.9)(1.649)^2$$

$$= .0184 + .269 + 2(.1545 + .331) + .971 + 2.45$$

$$= 4.70 \text{ in}^4$$

4.1.3.2.2.2 Loading Analysis

Rotor Limit Speed Condition:

P_{CF} = 308,700 lb. (limit) (Ref. page 10, Figure 1)

Load/fitting: $P_x = \frac{308,200}{2}$ = 154,400 lb. (limit)

Rotor Overspeed Operation - Both Engines Operating:

The engine-mount-pickup-fittings loads presented below are obtained from Figure 3, page 12.

P_x = 212,100 lb. (limit)

P_y = 7,100 lb. (limit)

P_z = 29,600 lb. (limit)

M_x = 94,000 in-lb. (limit)

M_y = 710,000 in-lb. (limit)

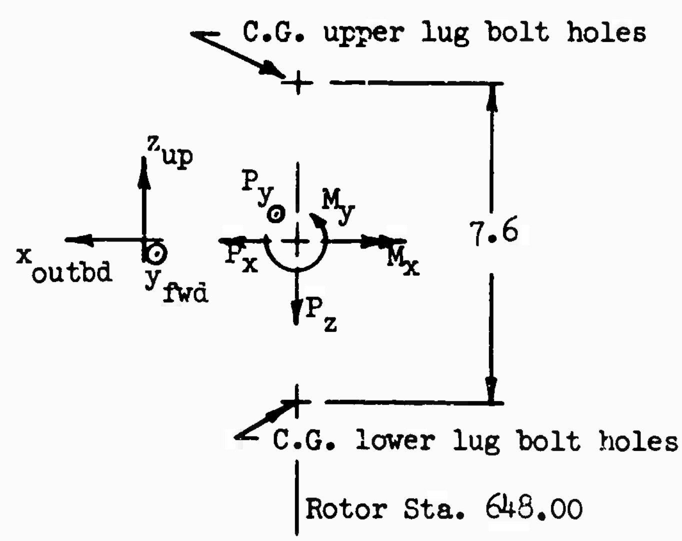


Figure 27. Engine-Mount-Pickup - Fittings Loading Sketch (Cond. 3)

The upper lug loading is established below.

$$P_{x_u} = \frac{P_x}{2} + \frac{M_y}{7.6} = \frac{212,100}{2} + \frac{710,000}{7.6} = 199,500 \text{ lb. (limit)}$$

$$P_{z_u} = \frac{P_z}{2} = \frac{29,600}{2} = 14,800 \text{ lb. (limit)}$$

$$P_{y_u} = \frac{P_y}{2} + \frac{M_x}{7.6} = \frac{7,100}{2} + \frac{94,000}{7.6} = 15930 \text{ lb. (limit)}$$

The applied loads at Section 8-8 are:

P_x = 199,500 lb. (limit)

P_y = 14,800 lb. (limit)

P_z = 15,930 lb. (limit)

M_y = (3.3)(14,800) = 48,800 in-lb. (limit)

M_z = (3.3)(15930) = 52,600 in-lb. (limit)

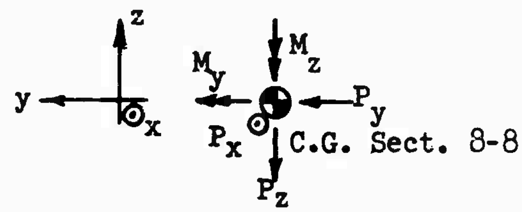


Fig. 28. Section 8-8 Loading Sketch - (Cond. 3)

4.1.3.2.2.3 Section 8-8 Stress Analysis

Rotor Limit Speed Condition:

P_x = 154,400 lb. (limit)

$$f_t = \frac{154,400}{5.40} = 28,600 \text{ psi (limit)}$$

$$\text{M.S.} = \frac{130,000}{(1.5)(28,600)} - 1 = \quad 2.03$$

Rotor Overspeed Operation - Both Engines Operating:

P_x = 199,500 lb. (limit)

P_y = 15,930 lb. (limit)

P_z = 14,800 lb. (limit)

M_y = 48,800 in-lb. (limit)

M_z = 52,600 in-lb. (limit)

$$f_t = \frac{199,500}{5.40} = 37,000 \text{ p.s.i. (limit)}$$

The shear stress due to P_z is assumed to be carried by the three vertical flanges.

$$A_s = (.25)(1.95 + 1.95 + 3.6) = 1.875 \text{ in}^2$$

$$f_{s_z} = \frac{14,800}{1.875} = 7,900 \text{ p.s.i. (limit)}$$

The shear stress due to P_y is assumed to be carried by the horizontal flange.

$$A_s = (.25)(14.10) = 3.525 \text{ in}^2$$

$$f_{s_y} = \frac{15,930}{3.525} = 4,520 \text{ p.s.i. (limit)}$$

$$f_{b_{y_{max}}} = \frac{(48,800)(3.45)}{4.7} = 35,800 \text{ p.s.i. (limit)}$$

$$f_{b_{z_{max}}} = \frac{(52,600)(7.05)}{75.0} = 4,950 \text{ p.s.i. (limit)}$$

The maximum combined stresses occur at point "A" as defined in Figure 25 on page 43.

$$f_{n_t} = f_t + f_{b_{y_{max}}} = 72,800 \text{ p.s.i. (limit)}$$

$$f_{s_{max}} = 7,900 \text{ p.s.i. (limit)}$$

The stress ratios are:

$$R_t = \frac{(1.5)(72,800)}{130,000} = .840$$

$$R_s = \frac{(1.5)(7,900)}{74,000} = .160$$

Combining the tensile and shear stress ratios, the margin of safety is:

$$M.S. = \frac{1}{\sqrt{R_t^2 + R_s^2}} - 1 = \quad .17$$

4.1.3.2.3 Section 9-9 Stress Investigation

From a review of Figure 26 on page 44, both the upper and lower engine mount fittings are considered to act as a unit at Section 9-9 due to doubler installations.

4.1.3.2.3.1 Section Properties

$$A = (2)(.25)(14.1 + 1.95 + 1.8 + 1.95) = 9.90 \text{ in}^2$$

$$\frac{I_{zz}}{2} = \frac{(.25)(14.1)^3}{12} + 2\left[\frac{(1.95)(.25)^3}{12} + (.25)(1.95)(4.6)^2\right] + \frac{(1.8)(.25)^3}{12}$$

Note: No Scale.

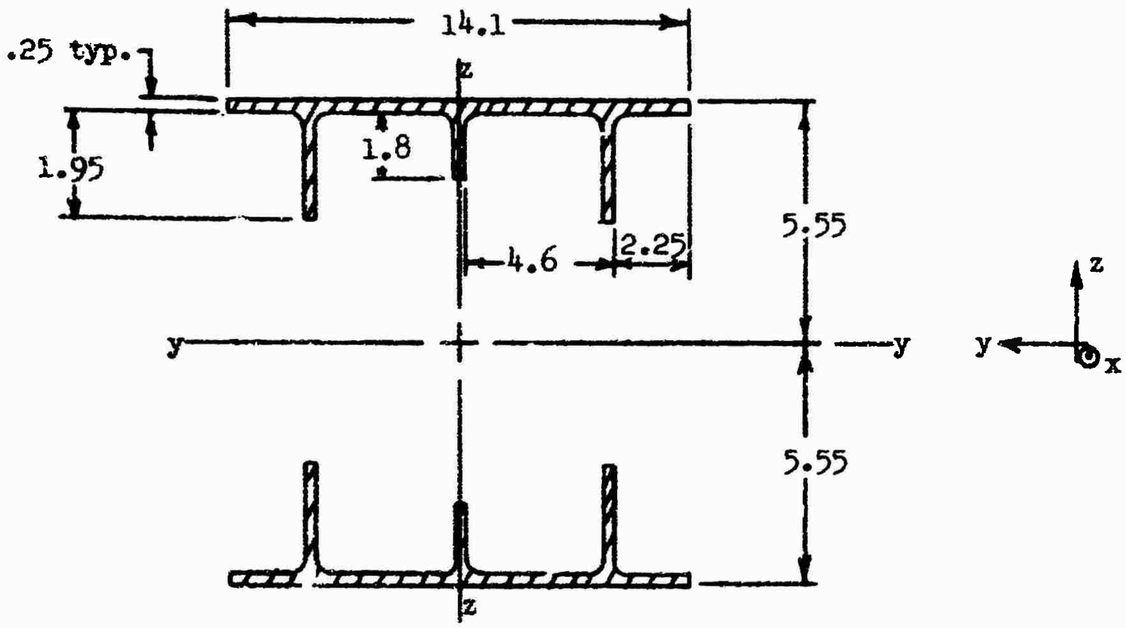


Figure 29. Section 9-9 Geometry.

$$I_{z-z} = 2\left[54.4 + 2(.003 + 10.3) + .0025\right] = 150 \text{ in}^4$$

$$\frac{I_{y-y}}{2} = \frac{(14.1)(.25)^3}{12} + (3.525)(5.425)^2 + 2\left[\frac{(.25)(1.95)^3}{12} + (.4875)(4.325)^2\right] + \frac{(.25)(1.8)^3}{12} + (.45)(4.40)^2$$

$$I_{y-y} = 2\left[.0184 + 104 + 2(.1545 + 9.13) + .122 + 8.7\right] = 263 \text{ in}^4$$

4.1.3.2.3.2 Loading Analysis

Rotor Limit Speed Condition:

P_{CF} = 308,700 lb. (limit) (Ref. Fig. 1, page 10)

Rotor Overspeed Operation - Both Engines Operating:

The engine-mount-pickup - fittings loads presented below are obtained from Figure 3, page 12.

P_x = 212,100 lb. (limit) M_x = 94,000 in-lb. (limit)
P_y = 7,100 lb. (limit) M_y = 710,000 in-lb. (limit)
P_z = 29,600 lb. (limit)

The engine mount fittings loading due to an applied load in the z-direction is shown in Figure 30 presented below.

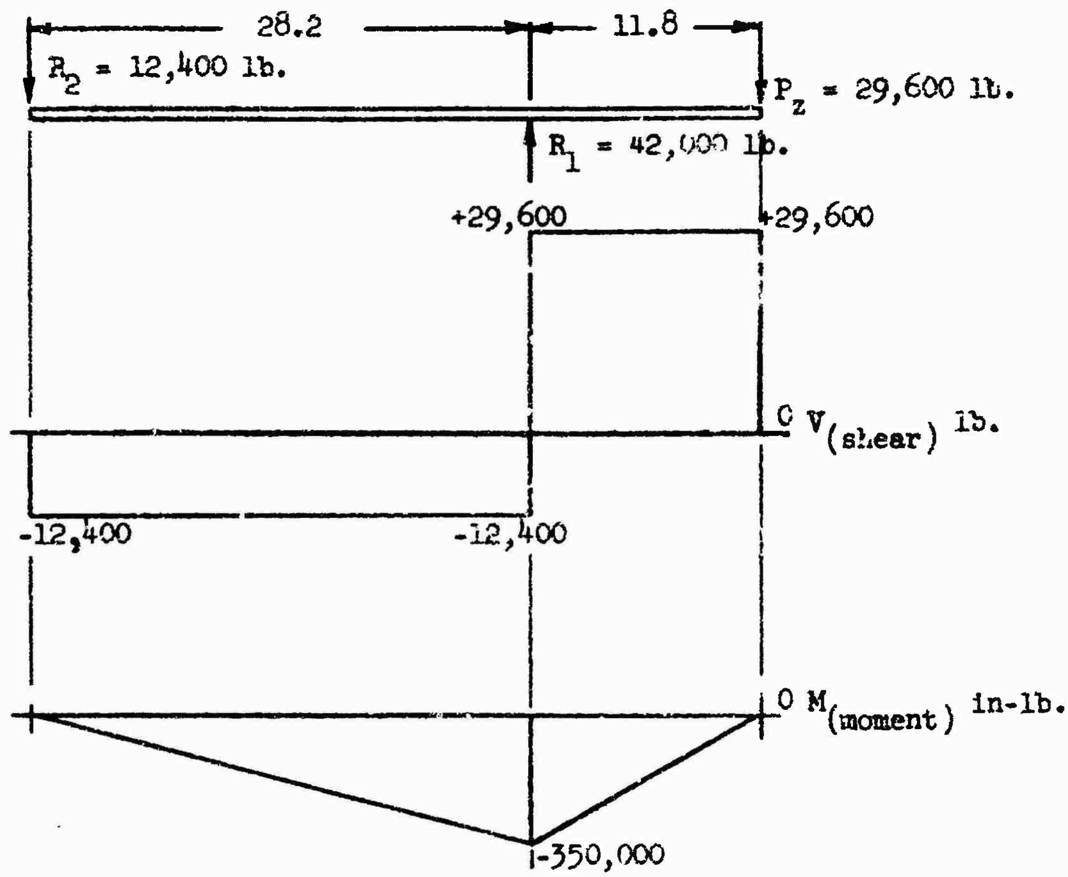


Figure 30. Engine-Mount-Fittings Loading Diagram.

The applied loading at Section 9-9 is:

P_x = 212,100 lb. (limit)

$M_{y-y} = M_y + \frac{27.8}{28.2}(350,000) = 710,000 + 345,000$

$= 1,055,000$ in-lb. (limit)

Torsion: $T = M_x = 94,000$ lb-in. (limit)

Shear: $P_z = 12,400$ lb. (limit)

4.1.3.2.3.3 Stress Analysis

Rotor Limit Speed Condition:

P_x = 308,700 lb. (limit) (Ref. page 48)

A = 9.90 in^2 (Ref. page 47)

$$f_t = \frac{308,700}{9.90} = 31,200 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{130,000}{(1.5)(31,200)} - 1 = \quad 1.78$$

Rotor Overspeed Operation - Both Engines Operating:

P_x = 212,100 lb. (limit) (Ref. page 49)

$$f_t = \frac{212,100}{9.9} = 21,400 \text{ p.s.i. (limit)}$$

M = 1,055,000 in-lb. (limit) (Ref. page 49)

$$f_b = \frac{(1.055)(10^6)(5.55)}{263} = 22,200 \text{ p.s.i. (limit)}$$

The shear load due to P_z is assumed to be carried by the six vertical flanges.

A_s = (.25)(1.95 + 1.8 + 1.95)(2) = 2.85 in^2

P_z = 12,400 lb. (limit) (Ref. page 49)

$$f_s = \frac{12,400}{2.85} = 4,350 \text{ p.s.i. (limit)}$$

The torsional shear stress is assumed to be carried by the two horizontal members.

A_s = (.25)(14.1)(2) = 7.05 in^2

T = 94,000 lb-in. (limit) (Ref. page 49)

$$f_{st} = \frac{T}{2AL} = \frac{94,000}{(2)(7.05)(10.85)} = 616 \text{ p.s.i. (limit)}$$

The maximum tensile stresses are:

f_t = 21,400 + 22,200 = 43,600 p.s.i. (limit)

and the tensile stress ratio is:

$$R_t = \frac{(1.5)(43,600)}{130,000} = .503$$

The maximum shear stress ratio is:

$$R_s = \frac{(1.5)(4,350)}{74,000} = .0882$$

Combining the tensile and shear stress ratios, the margin of safety is:

$$M.S. = \frac{1}{\sqrt{R_t^2 + R_s^2}} - 1 = \quad +.96$$

4.1.3.2.4 Engine-Mount-Fittings-to-Rib Attachment at Rotor Sta. 636.00

The fittings-to-rib attachment at rotor station 636.00 is provided by two doublers bonded to the outboard vertical flanges of the fittings and to the outboard face of the station 636.00 rib. An additional doubler is bonded to the inboard face of the rib and fittings at rotor station 636. To facilitate analysis, the inboard doubler is neglected in the stress analysis presented below.

4.1.3.2.4.1 Section Properties

Shear area: $A_s = (4)(3.25)(3.25) = 42.4 \text{ in}^2$

4.1.3.2.4.2 Loading Analysis

From a review of Figure 26, page 44, the mount fittings to station 636.00 rib attachment doublers are designed to react loads having a direction along the z-axis. The maximum loading occurs during the rotor overspeed condition with two engines operating.

$P_z = 42,000$ lb. (limit) (Ref. Figure 30, page 49)

4.1.3.2.4.3 Stress Analysis

$P_z = 42,000$ lb. (limit)

$A = 42.4 \text{ in}^2$

$$f_s = \frac{42,000}{42.4} = 993 \text{ p.s.i. (limit)}$$

The type of adhesive to be used in the bonded joints has not been selected at this time. However, the mechanical properties of the bonded joint are established per Reference 13.

$F_{su} = 3,600$ p.s.i. (Ref. 13, page 7)

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{3,600}{(1.15)(1.5)(993)} - 1 = \quad +1.10$$

The remainder of the engine mount fittings to rib attachment at rotor station 636.00 is satisfactory by inspection.

4.1.3.2.5 Engine-Mount-Fittings-to-Rib Attachment at Rotor Sta. 607.80

The fittings-to-rib attachment at rotor station 607.80 is provided by two tee-sections bonded to the outboard vertical flanges of the fittings and to the outboard face of the station 607.80 rib. From a review of Figure 26, page 44, the mount fittings to station 607.80 rib attachments are designed to react loads having a direction along the z-axis.

4.1.3.2.5.1 Section Properties

Shear area: $A_s = (4)\ (.7)(3.5)\ + (1.3)(1.1)\ + (.5)(1.3)(2.4)$

$= 21.76 \text{ in}^2$

4.1.3.2.5.2 Loading Analysis

$P_z = 12{,}400$ lb. (limit) (Ref. Figure 30, page 49)

4.1.3.2.5.3 Stress Analysis

$P_z = 12{,}400$ lb. (limit)

$A = 21.76 \text{ in}^2$

$f_s = \frac{12{,}400}{21.76} = 570$ p.s.i. (limit)

$f_{su} = 3{,}600$ p.s.i. (Ref. page 51)

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{3{,}600}{(1.15)(1.5)(570)} - 1 = \quad 2.66$$

4.1.3.2.6 Attachments for Engine-Mount Fittings to Blade Spar and Doublers

The engine-mount fittings are attached to the blade spar doubler with a structural adhesive. From a review of Figure 26, the horizontal flanges on the fittings are structurally bonded to the blade spar doubler. The attachment is designed to carry shear loading in the chordwise and spanwise directions.

4.1.3.2.6.1 Section Properties

Shear area per fitting:

$$A_s = (14)(15) + (10)(13.3) + (2)(16.5)(2.3) - (3.4)(2.1)(.5)$$
$$= 210 + 133 + 76 - 4$$
$$= 415 \text{ in}^2$$

4.1.3.2.6.2 Loading Analysis

Rotor Limit Speed Condition:

Load/fitting: P_x = 154,400 lb. (limit)

Rotor Overspeed Operation - Both Engines Operating:

$$\left.\begin{array}{l} P_x = 199{,}500 \text{ lb. (limit)} \\ P_y = 15{,}930 \text{ lb. (limit)} \end{array}\right\} \text{(Ref. page 46)}$$

$$P = \sqrt{P_x^2 + P_y^2} \approx 200{,}000 \text{ lb. (limit)}$$

4.1.3.2.6.3 Stress Analysis

Rotor Limit Speed Condition:

P_x = 154,400 lb. (limit)

A_s = 415 in^2

$$f_s = \frac{154{,}400}{415} = 370 \text{ p.s.i. (limit)}$$

F_{su} = 3,600 p.s.i. (Ref. page 51)

Using a fitting factor of 1.15, the margin of safety is:

$$\text{M.S.} = \frac{3{,}600}{(1.15)(1.5)(370)} - 1 =$$ High

Rotor Overspeed Operation - Both Engines Operating:

P = 200,000 lb. (limit)

A_s = 415 in^2

$$f_s = \frac{200{,}000}{415} = 482 \text{ p.s.i. (limit)}$$

F_{su} = 3,600 p.s.i. (Ref. page 51)

Using a fitting factor of 1.15, the margin of safety is:

$$\text{M.S.} = \frac{3{,}600}{(1.15)(1.5)(482)} - 1 =$$ 3.33

4.1.3.3 Engine-Aft-Mount Pickup Fittings

The material properties presented below are obtained from page 42 of this report.

F_{tu} = 130,000 p.s.i., F_{su} = 74,000 p.s.i.

4.1.3.3.1 Lug Analysis

The aft-mount-pickup-fittings lugs are analyzed for tensile and shear tear-out during rotor overspeed operation with two engines operating.

$$P_x = 4{,}050 \text{ lb. (limit)} \quad \text{(Ref. page 41)}$$
$$A = (1.00)(2.00 - .875) = 1.125 \text{ in}^2$$

Lug Tensile Tear-Out:

$$f_t = \frac{4{,}050}{1.125} = 3{,}600 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{130{,}000}{(1.15)(1.5)(3{,}600)} - 1 = \quad \text{High}$$

Lug Shear-Out:

$$f_s = f_t = 3{,}600 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{74{,}000}{(1.15)(1.5)(3{,}600)} - 1 = \quad \text{High}$$

4.1.3.3.2 Section 10-10 Stress Investigation

The section under analysis is located on Figure 26, page 44.

Section Properties:

Note: No scale.

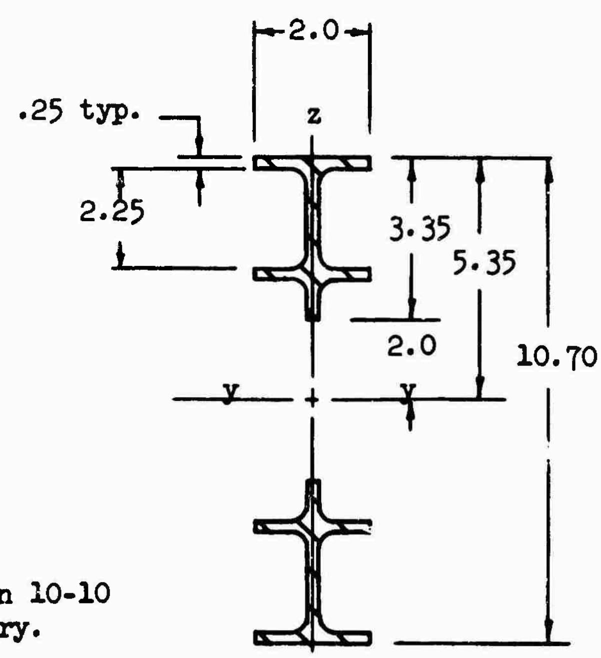


Figure 31. Section 10-10 Geometry.

$$A = (4)(2.0)(.25) + (2)(2.25)(.25) + (2)(.6)(.25) = 2.0 + 1.125 + .30$$
$$= 3.425 \text{ in}^2$$

$$I_{yy} = 2\left[\frac{(2)(.25)^3}{12} + (2)(.25)(5.225)^2 + \frac{(.25)(2.25)^3}{12} + (.25)(2.25)(3.975)^2 + \frac{(2)(.25)^3}{12} + (2)(.25)(2.725)^2 + \frac{(.25)(.6)^3}{12} + (.25)(.6)(2.3)^2\right]$$

$$= 2(.0026 + 13.65 + .2375 + 8.88 + .0026 + 3.72 + .0045 + .794)$$
$$= 2(27.29) = 54.58 \text{ in}^4$$

Loading Analysis:

The loading data presented below is obtained from Figure 3 on page 12.

$$P_x = 8{,}050 \text{ lb. (limit)}$$

Stress Analysis:

$$f_c = \frac{8{,}050}{3.425} = 2{,}350 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{130{,}000}{(1.5)(2{,}350)} - 1 =$$ High

4.1.3.3.3 Aft Fitting to Blade Spar and Doublers Attachments

The engine mount aft fitting is attached to the blade spar doubler with a structural adhesive. From a review of Figure 26, page 44, the upper and lower flanges of the fitting are structurally bonded to the blade spar doubler. The attachment is designed to carry shear loading in the spanwise direction.

Section Properties:

Shear area per flange: $A_s = (2.00)(15.0) = 30.0 \text{ in}^2$

Loading Analysis:

The maximum loading occurs during the rotor overspeed operation with both engines operating.

$$P = 4{,}025 \text{ lb. (limit) (Ref. page 40)}$$

Stress Analysis:

$$f_s = \frac{4{,}025}{30.0} = 134 \text{ p.s.i. (limit)}$$

$$F_{su} = 3{,}600 \text{ p.s.i. (Ref. page 51)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{3,600}{(1.15)(1.5)(134)} - 1 = \quad \text{High}$$

4.1.3.4 <u>Blade Spar Doubler to Spar Attachment</u>

This portion of the main rotor blade tip and attachments is satisfactory structurally by inspection.

4.1.3.5 <u>Blade Spar Stress Investigation</u>

The spar and doubler are assumed to carry all of the loads transferred from the engines to the blade primary structure. The spar section under analysis is located at rotor station 606.00.

The material properties presented below are obtained from page 42 of this report.

$E = 18.5 \times 10^6$ p.s.i. $\qquad F_{tu} = 130,000$ p.s.i.

$G = 6.2 \times 10^6$ p.s.i. $\qquad F_{su} = 74,000$ p.s.i.

4.1.3.5.1 <u>Section Properties</u>

Chordwise: $EI_C = 5.78 \times 10^{10}$ lb-in^2

Flapwise: $EI_F = 4.58 \times 10^9$ lb-in^2

Torsional: $GJ_T = 3.96 \times 10^9$ lb-in^2

Chordwise: $I_C = 3,125$ in^4

Flapwise: $I_F = 248$ in^4

Torsional: $J = 639$ in^4

Area: $A = 17.71$ in^2

4.1.3.5.2 <u>Loading Analysis</u>

<u>Rotor Limit Speed Condition:</u>

The centrifugal force loading due to the engines and mounts is obtained from Figure 1, page 10.

$P_{CF} = 308,700$ lb. (limit)

The centrifugal force loading due to the engine mount system tip attachment hardware is established below.

Weight = 68.7 lb. (limit)

$$P_{CF} = \frac{\Omega^2}{g}(W)(r) = \frac{(14.5)^2}{32.2}(68.7)(56) = 25,300 \text{ lb. (limit)}$$

where:

$\Omega = 14.5$ rad/sec.

$g = 32.2$ ft/sec^2

$W = 68.7$ lb. (limit)

$r = 56.0$ ft.

The total centrifugal force load is:

$$P = 308,700 + 25,300 = 334,000 \text{ lb. (limit)}$$

Rotor Overspeed Operation - Both Engines Operating

The fittings loads presented below are obtained from Figure 3 on page 12. Use right hand rule on all moments and observe coordinate axis definitions shown on Figure 3.

Main-mount-pickup-fittings loading:

$P_{x_1} = 212,100$ lb. (limit)

$P_{y_1} = 7,100$ lb. (limit)

$P_{z_1} = -29,600$ lb. (limit)

$M_{x_1} = -47,000$ in-lb. (limit)

$M_{y_1} = 710,000$ in-lb. (limit)

Aft-mount-pickup-fittings loading:

$P_{x_2} = 8,050$ lb. (limit)

The centrifugal force loading due to the engine mount system tip attachment hardware is established below.

$$P_{CF} = \frac{\Omega^2}{g}(W)(r) = \frac{(12.2)^2}{32.2}(68.7)(56) = 17,800 \text{ lb. (limit)}$$

The applied loading at rotor station 606.00 is established below.

$P_x = P_{x_1} + P_{x_2} + P_{CF} = 238,000$ lb. (limit)

$P_y = P_{y_1} = 7,100$ lb. (limit)

$P_z = P_{z_1} = 29,600$ lb. (limit)

$M_x = 2.55P_{z_1} + M_{x_1} = (2.55)(29,600) + 47,000$

$= 120,500$ in-lb. (limit)

$$M_y = M_{y_1} + 40.2\ P_{z_1} = 710{,}000 + (40.2)(29{,}600)$$
$$= 1{,}900{,}000 \text{ in-lb. (limit)}$$

$$M_z = -40.2\ P_{y_1} + 2.55\ P_{x_1} - 12.7\ P_{x_2}$$
$$= -(40.2)(7{,}100) + (2.55)(212{,}100) - (12.7)(8{,}050)$$
$$= -286{,}000 + 541{,}000 - 102{,}000$$
$$= 153{,}000 \text{ in-lb. (limit)}$$

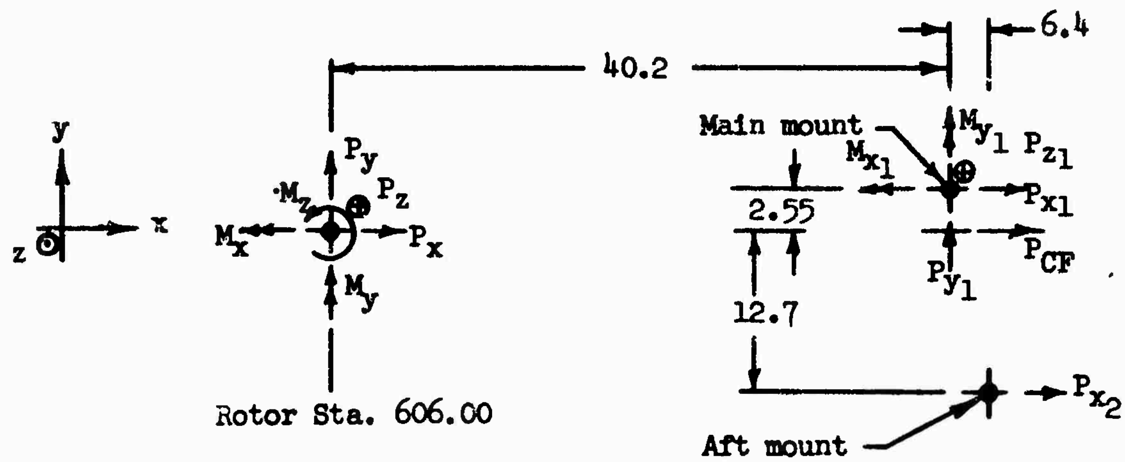


Figure 32. Rotor Station 606.00 Loading Geometry.

4.1.3.5.3 <u>Stress Analysis</u>

<u>Rotor Limit Speed Condition:</u>

$P = 334{,}000$ lb. (limit) (Ref. page 57)

$A = 17.71 \text{ in}^2$ (ref. page 56)

$f_t = \dfrac{334{,}000}{17.71} = 18{,}900$ p.s.i. (limit)

$F_{tu} = 130{,}000$ p.s.i. (ref. page 56)

The margin of safety is:

$$M.S. = \frac{130{,}000}{(1.5)(18{,}900)} - 1 = \quad 3.59$$

<u>Rotor Overspeed Operation - Both Engines Operating:</u>

$$f_t = \frac{P_x}{A} = \frac{238{,}000}{17.71} = 13{,}400 \text{ p.s.i. (limit)}$$

$$f_{by} = \frac{M_y C_z}{I_F} = \frac{(1.9)(10^6)(5.88)}{248} = 45{,}100 \text{ p.s.i. (limit)}$$

$$f_{bz} = \frac{M_z C_y}{I_C} = \frac{(153{,}000)(18.5)}{3{,}125} = 905 \text{ p.s.i. (limit)}$$

$$f_{sz} = \frac{P_z}{A} = \frac{29{,}600}{17.71} = 1{,}670 \text{ p.s.i. (limit)}$$

$$f_{st} = \frac{M_x C}{J} = \frac{(120{,}500)(18.5)}{639} = 3{,}490 \text{ p.s.i. (limit)}$$

The maximum tensile stress is:

$$f_{t_{max}} = f_t + f_{b_y} + f_{b_z} = 59{,}400 \text{ p.s.i. (limit)}$$

The maximum shear stress is:

$$f_{s_{max}} = f_{s_z} + f_{s_t} = 5{,}160 \text{ p.s.i. (limit)}$$

The stress ratios are:

$$R_t = \frac{(1.5)(59{,}400)}{130{,}000} = .685, \qquad R_s = \frac{(1.5)(5{,}160)}{74{,}000} = .105$$

Combining the stress ratios, the margin of safety is:

$$M.S. = \frac{1}{\sqrt{R_t^2 + R_s^2}} - 1 = +.44$$

The remainder of the main rotor blade tip and attachments is satisfactory structurally by inspection.

4.1.4 Main Rotor Blade Typical Section

This portion of the main rotor system stress analysis is concerned with the static structural substantiation of the blade basic section.

The blade basic section is constructed of Ti-8AL-1Mo-1V titanium alloy. The material properties presented below are obtained from Reference 11.

$$F_{tu} = 130{,}000 \text{ p.s.i.}$$
$$F_{su} = .57F_{tu} = 74{,}000 \text{ p.s.i.}$$
$$E = 18.5 \times 10^6 \text{ p.s.i.}$$

4.1.4.1 Loading Analysis

The blade basic section is analyzed for the following four loading conditions:

Cond. 1 - Rotor limit speed condition
Cond. 2 - Fwd. flight, 41 m.p.h., 2.5g, 562 f.p.s. tip speed
Cond. 3 - Static droop
Cond. 4 - Transient cyclic stick whirl

4.1.4.2 Section Properties

The section properties for the blade basic section presented below are obtained from Reference 11.

At rotor station 170.00:

Chordwise stiffness: $EI_C = 9.3 \times 10^{10}$ lb-in^2

Flapwise stiffness: $EI_F = 7.5 \times 10^9$ lb-in^2

Torsional stiffness: $GJ = 9.7 \times 10^9$ lb-in^2

Blade chord: = 78 in.

Blade depth: = (.15)(78) = 11.7 in.

$A = 16.92$ in^2

4.1.4.3 Stress Analysis

The blade basic section is analyzed for the four loading conditions presented in Section 4.1.4.1. A buckling analysis of the blade is conducted for the static droop condition and transient cyclic stick whirl.

Rotor Limit Speed Condition:

The centrifugal force load during the rotor limit speed condition is conservatively taken as the centrifugal force load at the tension-torsion bar retention pin.

C.F. = 890,000 lb. (limit) (Ref. page 64)

$$f_t = \frac{890{,}000}{16.92} = 52{,}500 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{130{,}000}{(1.5)(52{,}500)} - 1 = \quad .65$$

Fwd. Flight, 41 m.p.h., 2.5g, 562 f.p.s. Tip Speed:

$$\left.\begin{array}{l} M_F = 1.2 \times 10^6 \text{ in-lb. (limit)} \\ M_C = .1 \times 10^6 \text{ in-lb. (limit)} \\ C.F. = 4.64 \times 10^5 \text{ lb. (limit)} \end{array}\right\} \text{(Ref. 2, pages 87, 96, 53)}$$

$$f_t = \frac{4.64 \times 10^5}{16.92} = 27{,}400 \text{ p.s.i. (limit)}$$

$$f_{b_F} = \frac{(1.2)(10^6)(5.85)(18.5)(10^6)}{(7.5)(10^9)} = 17{,}300 \text{ p.s.i. (limit)}$$

$$f_{b_C} = \frac{(1.0)(10^5)(58.5)(18.5)(10^6)}{(9.3)(10^{10})} = 116 \text{ p.s.i. (limit)}$$

$$f_{max} = 27{,}400 + 17{,}300 = 44{,}700 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{130{,}000}{(1.5)(44{,}700)} - 1 = \quad .94$$

Static Droop:

The static droop flapwise moment at rotor station 170.00 is obtained from page 82, Figure 63, of Reference 2.

$$M_F = 1.08 \times 10^6 \text{ in-lb. (limit)}$$

The maximum compressive stress occurs on the lower skin where it forms a part of the spar. The equations used in determining the critical compressive buckling stress are obtained from pages 370 and 389 of Reference 6. The buckling stress is considered to be the sum of the buckling stress for a flat sheet simply suppoerted on four sides, and the buckling stress for a cylinder with a large radius.

$$F'_{C_{CR}} = KE\left(\frac{t}{b}\right)^2 = (3.62)(18.5)(10^6)\left(\frac{.136}{19}\right)^2 = 3{,}430 \text{ p.s.i.}$$

$K = 3.62$ $\qquad$ $t = .136$ in.

$E = 18.5 \times 10^6$ p.s.i. $\qquad$ $b = 19.0$ in.

$$F''_{C_{CR}} = E\left[9\left(\frac{t}{R}\right)^{1.6} + .16\left(\frac{t}{L}\right)^{1.3}\right]$$

$$= (18.5)(10^6)\left[9\left(\frac{.136}{120}\right)^{1.6} + .15\left(\frac{.136}{89}\right)^{1.3}\right]$$

$$= (18.5)(10^6)\left[(17.1)(10^{-5}) + (3.52)(10^{-5})\right]$$

$$= (18.5)(10^6)(2.062)(10^{-4}) = 3{,}820 \text{ p.s.i.}$$

$t = .136$ in.
$R = 120$ in.
$L = 89$ in. (rib spacing)

In order to preclude compressive buckling of the lower skin, a stringer is installed midway between the nose and rear spars and extends the length of the blade. The method of determining the stringer critical compressive buckling stress is obtained from pages 378 and 379 of Reference 6.

Stringer geometry and material type:

1.0 in. x 1.0 in. x .050 in. thick angle made from Ti-6AL-1Mo-1V titanium alloy.

Tangent modulus: $E_t = 19.35 \times 10^6$ p.s.i. (Ref. 11, Figure 15)

$$F_{C_{CR}}''' = \frac{KE_t}{(b/t)^2} = \frac{(.385)(19.35)(10^6)}{(1.0/.050)^2} = 18,600 \text{ p.s.i.}$$

K = .385
b = 1.0 in. (leg length)
t = .050 in.

$$F_{C_{CR}} = 3,430 + 3,820 + 18,600 = 25,850 \text{ p.s.i.}$$

The maximum applied compressive stress on the lower skin is established below.

$$f_{b_c} = \frac{(1.08)(10^6)(5.85)(18.5)(10^6)}{(7.9)(10^9)} = 15,600 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{25,850}{(1.5)(15,600)} - 1 = \quad .10$$

Transient Cyclic Stick Whirl:

The transient cyclic stick whirl maneuver is conducted at a tip speed of 650 feet per second. The low frequency in-plane bending moment is obtained from Reference 2, page 101, Figure 95, and has a magnitude of:

$M_c = \pm 3.25 \times 10^6$ in-lb. (limit)

C.F. = 5.38×10^5 lb. (limit) (Ref. 2, page 53, Figure 8)

$$f_t = \frac{(5.38)(10^5)}{16.92} = 31,800 \text{ p.s.i. (limit)}$$

$$f_b = \frac{(3.25)(10^6)(58.5)(18.5)(10^6)}{(9.3)(10^{10})} = \pm 37,800 \text{ p.s.i. (limit)}$$

The maximum tensile stress on the trailing edge is:

$$f_{t_{max}} = 37,800 + 31,800 = 69,600 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{130,000}{(1.5)(69,600)} - 1 = \quad .25$$

The maximum compressive stress on the trailing edge is:

$$f_{c_{max}} = 37,800 - 31,800 = 6,000 \text{ p.s.i. (limit)}$$

To simplify the buckling analysis, the trailing edge cap is assumed to be a hollow cylinder having an outside diameter of 1.187 inches and a wall thickness of .125 inches.

O.D. = 1.187 in., R = .5935 in.

t = .125 in., E = 18.5 x 10^6 p.s.i.

$$F_{C_{CR}} = .3E\left(\frac{t}{R}\right) \quad \text{(Ref. 6, page 389)}$$

$$= (.3)(18.5)(10^6)\,\frac{.125}{.5935} = 11{,}700 \text{ p.s.i.}$$

$$\text{M.S.} = \frac{11{,}700}{(1.5)(6{,}000)} - 1 = \quad +.30$$

The trailing edge cap will not buckle during the transient cyclic stick whirl condition.

4.1.5 Main Rotor Blade Root Retention Structure - (Ref. Vol. III, Figure 5)

This portion of the main rotor system stress analysis is concerned with the static structural substantiation of the blade root retention structure, including the tension-torsion bar assembly and its installation.

The blade skin and doublers are constructed of Ti-8AL-1Mo-1V annealed titanium alloy, and the primary structure is made of solution heat-treated Ti-6AL-4V titanium alloy.

For Ti-8AL-1Mo-1V annealed titanium alloy sheet:

$$\left.\begin{aligned} F_{tu} &= 130{,}000 \text{ p.s.i.} \\ F_{su} &= .57F_{tu} = 74{,}000 \text{ p.s.i.} \\ E &= 18.5 \times 10^6 \text{ p.s.i.} \end{aligned}\right\} \text{(Ref. 11)}$$

For Ti-6AL-4V solution heat-treated titanium alloy bar and sheet:

$$\left.\begin{aligned} F_{tu} &= 162{,}000 \text{ p.s.i.} \\ F_{su} &= 94{,}000 \text{ p.s.i.} \\ F_{br_u} &= 174{,}000 \text{ p.s.i.} \end{aligned}\right\} \text{(Ref. 10, page 39)}$$

4.1.5.1 Loading Analysis

The main rotor blade root retention system is analyzed for the following loading conditions:

Cond. 1 - Rotor limit speed condition
Cond. 2 - Fwd. flight, 2.5g, 41 m.p.h., 562 f.p.s. tip speed
Cond. 3 - Two engines inoperative in hover
Cond. 4 - Transient cyclic stick whirl

4.1.5.1.1 Condition 1 Loading

The rotor limit speed condition centrifugal force load presented below is obtained by multiplying the ratio of the squares of the tip velocities at 813 feet per second and at 650 feet per second times the centrifugal force load developed at a tip velocity of 650 feet per second.

$$C.F. = \left(\frac{813}{650}\right)^2 (570{,}000) = 890{,}000 \text{ lb. (limit)}$$

4.1.5.1.2 Condition 2 Loading

The chordwise moment at rotor station 71.00 during condition 2 is negligible, as the main rotor system is designed for the chordwise moment due to engine thrust to be reduced by the blade and engine nacelle drag such that the chordwise moment at the rotor system centerline is zero.

$M_F = 3.6 \times 10^6$ in-lb. (limit) (Ref. 2, Fig. 71, p. 87)
C.F. $= 4.0 \times 10^5$ lb. (limit) (Ref. 2, Fig. 6, p. 53)
Torque: $T = 1.78 \times 10^5$ lb-in. (limit) (Ref. 2, Fig. 9, p. 54)

4.1.5.1.3 Condition 3 Loading

$M_F = 6.3 \times 10^5$ in-lb. (limit) (Ref. 2, Fig. 64, p. 83)
$M_C = 1.18 \times 10^6$ in-lb. (limit) (Ref. 2, Fig. 85, p. 95)
C.F. $= 4.0 \times 10^5$ lb. (limit) (Ref. 2, Fig. 8, p. 53)
$T = 1.78 \times 10^5$ lb-in. (limit) (Ref. 2, Fig. 9, p. 54)

4.1.5.1.4 Condition 4 Loading

$M_F = 8.0 \times 10^5$ in-lb. (limit) (Ref. 2, Fig. 78, p. 91)
$M_C = \pm 3.4 \times 10^6$ in-lb. (limit) (Ref. 2, Fig. 95, p.101)
C.F. $= 4.0 \times 10^5$ lb. (limit) (Ref. 2, Fig. 8, p. 53)
$T = 1.78 \times 10^5$ lb-in. (limit) (Ref. 2, Fig. 9, p. 54)

4.1.5.2 Tension-Torsion Bar Retention Pin Analysis

The T-T bar retention pin provides the tension-torsion bar attachment to the rotor blade. The blade and tip attachments centrifugal forces and torsional moments are carried through the tension-torsion bar to the hub.

4.1.5.2.1 Section Properties

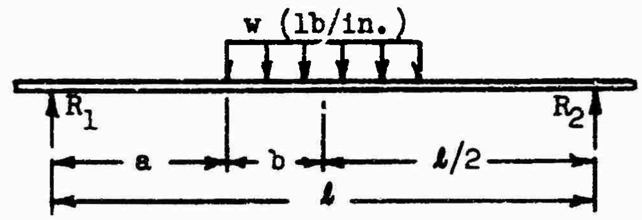
The attachment pin has an outside diameter of 6.50 inches, with a wall thickness of 1.25 inches at mid-span and .50 inches at the ends. The section properties are developed below.

At the ends: $A = \frac{\pi}{4}(6.50^2 - 5.50^2) = 9.425 \text{ in}^2$

At the mid-span: $Z = \frac{\pi}{32}\left(\frac{6.5^4 - 4.0^4}{6.5}\right) = 23.095 \text{ in}^3$

4.1.5.2.2 Loading Analysis

Preliminary analysis has established that the maximum pin loading occurs during the rotor limit speed condition.



ℓ = 11.70 in.

a = 3.225 in.

b = 2.625 in.

2b = 5.25 in.

Figure 33. T-T-Bar-to-Blade-Attachment-Pin Loading Geometry.

$R_1 = R_2 = \frac{890{,}000}{2} = 445{,}000$ lb. (limit)

$w = \frac{890{,}000}{5.25} = 173{,}000$ lb/in. (limit)

The maximum moment occurs at mid-span:

$$M = R_1(a + .5b) = 445{,}000\left[3.225 + (.5)(2.625)\right] = 2{,}020{,}000 \text{ in-lb. (limit)}$$

The maximum pin shear load is on the ends and has a magnitude of:

$$R_1 = R_2 = 445{,}000 \text{ lb. (limit)}$$

4.1.5.2.3 Stress Analysis

$$f_b = \frac{2{,}020{,}000}{23.1} = 87{,}500 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{162{,}000}{(1.5)(87{,}500)} - 1 = +.24$$

$$f_s = \frac{445{,}000}{9.425} = 47{,}200 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{94{,}000}{(1.5)(47{,}200)} - 1 = +.33$$

4.1.5.3 Blade T-T Bar Retention Pin Hole Analysis

4.1.5.3.1 Section Properties

Approximate composite thickness: t = 1.64 in. per side
Hole diameter: D = 6.75 in.

Bearing area: $A_{br} = (1.64)(6.75) = 11.08\ \text{in}^2$ per side

4.1.5.3.2 Stress Analysis

The centrifugal force load per side is:

$$P = \frac{890,000}{2} = 445,000 \text{ lb. (limit)}$$

$$f_{br} = \frac{445,000}{11.08} = 40,200 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{174,000}{(1.5)(40,200)} - 1 = \quad 1.88$$

4.1.5.4 Rotor Station 102.00 Analysis

Rotor station 102.00 carries flapwise and chordwise moments only, as it is located inboard of the tension-torsion bar retention pin hole.

4.1.5.4.1 Section Properties

The section properties presented below are obtained from Reference 2, page 51.

Flapwise $EI = 14.7 \times 10^9$ lb-in^2, $C_z = 5.15$ in.

Chordwise $EI = 9.85 \times 10^{10}$ lb-in^2, $C_y = 41.5$ in.

$E = 18.5 \times 10^6$ psi

4.1.5.4.2 Stress Analysis

Condition 2 Analysis:

$M_F = 3.6 \times 10^6$ in-lb. (limit) (ref. page 64)

$$f_b = \frac{(3.6)(10^6)(5.15)(18.5)(10^6)}{(14.7)(10^9)} = 23,300 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{130,000}{(1.5)(23,300)} - 1 = \quad 2.72$$

Condition 3 Analysis:

$$\left.\begin{array}{l} M_F = 6.3 \times 10^5 \text{ in-lb. (limit)} \\ M_C = 1.18 \times 10^6 \text{ in-lb. (limit)} \end{array}\right\} \text{(Ref. page 64)}$$

$$f_{b_F} = \frac{(6.3)(10^5)(5.15)(18.5)(10^6)}{(14.7)(10^9)} = 4100 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{130,000}{(1.5)(4100)} - 1 = \quad \text{High}$$

$$f_{b_C} = \frac{(1.18)(10^6)(41.5)(18.5)(10^6)}{(9.85)(10^{10})} = 9{,}200 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{130{,}000}{(1.5)(9{,}200)} - 1 = \quad \text{High}$$

Condition 4 Analysis:

$$M_C = \pm 3.4 \times 10^6 \text{ in-lb. (limit)} \quad \text{(ref. page 64)}$$

$$f_b = \frac{(3.4)(10^6)(41.5)(18.5)(10^6)}{(9.85)(10^{10})} = 26{,}500 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{130{,}000}{(1.5)(26{,}500)} - 1 = \quad 2.27$$

Rotor station 102.00 is satisfactory for flapwise bending by comparison to the condition 3 analysis presented on the previous page.

4.1.5.5 Main-Rotor-Blade-to-Stub-Blade Attachment

The main-rotor-blade-to-stub-blade attachment is provided by bearings located at rotor stations 42.80 and 99.20. The bearings react the flapwise and chordwise moments into the stub blade as couples. The critical flapwise and chordwise moments occur during condition 4.

4.1.5.5.1 Loading Analysis

$$\left.\begin{aligned} M_F &= 8.0 \times 10^5 \text{ in-lb. (limit)} \\ M_C &= \pm 3.4 \times 10^6 \text{ in-lb. (limit)} \end{aligned}\right\} \text{(Ref. page 64)}$$

$$\ell = 99.20 - 42.80 = 56.4 \text{ in.}$$

$$P = \frac{\sqrt{M_C^2 + M_F^2}}{\ell} = \frac{\sqrt{[(3.4)(10^6)]^2 + [(8.0)(10^5)]^2}}{56.4} = 62{,}000 \text{ lb. (limit)}$$

4.1.5.5.2 Section Properties

The section properties are presented in each individual item stress analysis.

4.1.5.5.3 Pitch Bearing Support Stress Analysis

Outboard Pitch Bearing Support:

Outside diameter: OD = 11.50 in.
Inside diameter: ID = 10.00 in.
Moment arm: = 3.0 in.

$$A = \frac{\pi}{4}(11.50^2 - 10.00^2)$$
$$= 25.2 \text{ in}^2$$

$$I = \frac{\pi}{64}(11.50^4 - 10.00^4)$$
$$= 368 \text{ in}^4$$

$$c = 5.75 \text{ in.}$$

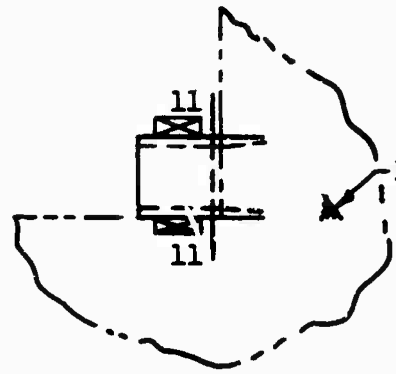


Figure 34. Outboard-Bearing Support Geometry Sketch.

Shear stress:

$$f_s = \frac{62{,}000}{25.2} = 2{,}480 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{94{,}000}{(1.5)(2{,}480)} - 1 = \text{High}$$

Bending stress:

$$f_b = \frac{(62{,}000)(3)(5.75)}{368} = 2{,}910 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{162{,}000}{(1.5)(2{,}910)} - 1 = \text{High}$$

<u>Inboard Pitch Bearing to Rotor Blade Attachment:</u>

Lug width: $w = 5.00$ in.
Lug thickness: $t = .50$ in.

Shear area: $A_s = (2)(5.00)(.50) = 5.00 \text{ in}^2$

$P = 62{,}000$ lb. (limit) (ref. page 67)

$$f_s = \frac{62{,}000}{5} = 12{,}400 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{94{,}000}{(1.15)(1.5)(12{,}400)} - 1 = 3.40$$

The remainder of the inboard-bearing-to-rotor-blade attachment is satisfactory by inspection.

4.1.5.5.4 Pitch Bearings Stress Analysis

The pitch bearings selection has not been finalized at this time. During Phase II a comprehensive study will be made to select structurally satisfactory bearings to adequately support the applied loading.

4.1.5.6 Tension-Torsion Strap Assembly

The tension-torsion strap assembly consists of high-strength wire wrapped around a flanged bushing at each end, in multiple layers to provide the proper width and thickness, and bonded together with urethane rubber. The material geometry and properties presented below are obtained from Reference 14.

.006 in. dia. AM 355 CRES steel wire.
Min. breaking strength per wire is 12.5 pounds.
Requires 107,000-107,300 wires per side, bonded with urethane rubber.

The ultimate strength of the strap is:

$$P_{T_u} = (12.5)(107,000) = 1,338,000 \text{ lb.}$$

The maximum tension-torsion strap load occurs during the rotor limit speed condition.

$$P_{CF} = 890,000 \text{ lb. (limit)} \quad \text{(ref. page 64)}$$

The margin of safety is: $M.S. = \dfrac{1,338,000}{(1.5)(890,000)} - 1 =$ 0.00

4.1.6 Stub Blade and Retention

This portion of the main rotor system stress analysis is concerned with the static structural substantiation of the stub blade and retention.

The stub-blade skin and doublers are constructed of Ti-8AL-1Mo-1V annealed titanium alloy, and the primary structure is made of Ti-6AL-4V titanium alloy.

For Ti-8AL-1Mo-1V annealed titanium alloy sheet:

$$\left.\begin{aligned} F_{tu} &= 130,000 \text{ p.s.i.} \\ F_{su} &= .57\,F_{tu} = 74,000 \text{ p.s.i.} \\ E &= 18.5 \times 10^6 \text{ p.s.i.} \end{aligned}\right\} \quad \text{(Ref. 11)}$$

For Ti-6AL-4V solution heat-treated titanium alloy bar and sheet:

$$\left.\begin{aligned} F_{tu} &= 162,000 \text{ p.s.i.} \\ F_{su} &= 94,000 \text{ p.s.i.} \\ F_{br_u} &= 174,000 \text{ p.s.i.} \end{aligned}\right\} \quad \text{(Ref. 10, page 39)}$$

4.1.6.1 Loading Analysis

From a review of Section 4.1.5.5, the critical stub-blade loading occurs during the transient cyclic stick whirl condition.

$$\left.\begin{array}{l} M_F = 8.0 \times 10^5 \text{ in-lb. (limit)} \\ M_C = \pm 3.4 \times 10^6 \text{ in-lb. (limit)} \end{array}\right\} \text{(Ref. page 64)}$$

4.1.6.2 Stub-Blade-to-Main-Rotor-Blade Attachment

This portion of the stub-blade analysis is adequate by comparison to the rotor blade to stub-blade attachment analysis presented in Section 4.1.5.5.

4.1.6.3 Stub-Blade Typical Section

The section properties for the stub blade at rotor station 71.25 are presented below.

$$EI_C = 13.8 \times 10^{10} \text{ lb-in}^2, \qquad C_y = 18.5 \text{ in.}$$

$$EI_F = 3.01 \times 10^{10} \text{ lb-in}^2, \qquad C_z = 8.25 \text{ in.}$$

$$E = 18.5 \times 10^6 \text{ p.s.i.}$$

Maximum chordwise bending stress:

$$f_{b_c} = \frac{(3.4)(10^6)(18.5)(18.5)(10^6)}{(13.8)(10^{10})} = 8,700 \text{ p.s.i.(limit)}$$

$$\text{M.S.} = \frac{130,000}{(1.5)(8700)} - 1 = \quad \text{High}$$

Maximum flapwise bending stress:

$$f_{b_f} = \frac{(8.0)(10^5)(8.25)(18.5)(10^6)}{(3.01)(10^{10})} = 4,100 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{130,000}{(1.5)(4100)} - 1 = \quad \text{High}$$

4.1.6.4 Stub-Blade-to-Hub-Attachment Lugs Analysis

4.1.6.4.1 Loading Analysis

The stub-blade-to-hub-attachment lugs are analyzed for the following loading conditions:

Cond. 2 - Fwd. flight, 2.5g, 41 m.p.h., 562 f.p.s. tip speed
Cond. 3 - Two engines inoperative in hover
Cond. 4 - Transient cyclic stick whirl

Condition 2 Loading:

The chordwise moment at rotor station 30.00 during condition 2 is negligible, as the main rotor system is designed for the chordwise moment due to engine thrust at the blade tip to be reduced by the blade and engine nacelle drag such that the chordwise moment at the rotor system centerline is zero.

$M_F = 4.4 \times 10^6$ in-lb. (limit) (Ref. 2, Fig. 71, p. 87)

Condition 3 Loading:

$M_F = 7.6 \times 10^5$ in-lb. (limit) (Ref. 2, Fig. 64, p. 83)

$M_C = 1.36 \times 10^6$ in-lb. (limit) (Ref. 2, Fig. 85, p. 95)

Condition 4 Loading:

$M_F = 9.8 \times 10^5$ in-lb. (limit) (Ref. 2, Fig. 78, p. 91)

$M_C = 3.5 \times 10^6$ in-lb. (limit) (Ref. 2, Fig. 95, p.101)

4.1.6.4.2 Section Properties

Tine thickness: $t = 1.30$ in.
Tine diameter: $D = 12.00$ in.
Hole diameter: $d = 6.90$ in.

Tensile area: $A_t = 1.3(12.00 - 6.90) = 6.63 \text{ in}^2$
Shear area: $A_s = (1.3)(2.9)(2) = 7.55 \text{ in}^2$

4.1.6.4.3 Stress Analysis

The flapwise moments and torsional moments are assumed carried as couples by the upper and lower lugs. The chordwise moments are assumed carried as a couple by the attachment lugs and the adjustable link.

Condition 2 Stress Analysis:

The load per lug due to flapwise bending is:

$$P_x = \frac{(4.4)(10^6)}{17.4} = 253{,}000 \text{ lb. (limit)}$$

Lug tensile tear-out:

$$f_t = \frac{253,000}{6.63} = 38,200 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{162,000}{(1.15)(1.5)(38,200)} - 1 = \quad 1.46$$

Lug shear-out:

$$f_s = \frac{253,000}{7.55} = 33,500 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{94,000}{(1.15)(1.5)(33,500)} - 1 = \quad .63$$

Condition 3 Stress Analysis:

The load per lug due to flapwise bending is:

$$P_x = \frac{(7.6)(10^5)}{17.4} = 40,000 \text{ lb. (limit)}$$

The load per lug due to chordwise bending is:

$$P_x = \frac{(1.36)(10^6)}{(2)(30)} = 23,700 \text{ lb. (limit)}$$

The maximum lug load is:

$$P = (40,000 + 23,700) = 63,700 \text{ lb. (limit)}$$

The lugs are satisfactory for the condition 3 loading by comparison to the higher loading developed during condition 2.

Condition 4 Stress Analysis:

The load per lug due to flapwise bending is:

$$P_x = \frac{(9.8)(10^5)}{17.4} = 56,400 \text{ lb. (limit)}$$

The load per lug due to chordwise bending is:

$$P_x = \frac{(3.5)(10^6)}{(2)(30)} = 58{,}400 \text{ lb. (limit)}$$

The maximum lug load is:

$$P = 56{,}400 + 58{,}400 = 115{,}000 \text{ lb. (limit)}$$

The lugs are satisfactory for the condition 4 loading by comparison to the higher loading developed during condition 2.

4.1.6.5 Stub-Blade-to-Adjustable-Link Attachment Analysis

4.1.6.5.1 Loading Analysis

From a review of Section 4.1.6.4, presented above, the maximum adjustable link load occurs during condition 4.

$$P_x = \frac{(3.5)(10^6)}{30} = 117{,}000 \text{ lb. (limit)}$$

4.1.6.5.2 Section Properties

Tine thickness:	t	$= .75$ in.
Tine diameter:	D	$= 4.00$ in.
Hole diameter:	d	$= 2.00$ in.
Tensile area:	A_t	$= (.75)(4.00 - 2.00)(2) = 3.00 \text{ in}^2$
Shear area:	A_s	$= (.75)(1.15)(4) = 3.45 \text{ in}^2$

4.1.6.5.3 Stress Analysis

Lug tensile tear-out:

$$f_t = \frac{117{,}000}{3.00} = 39{,}000 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{162{,}000}{(1.15)(1.5)(39{,}000)} - 1 = \quad 1.41$$

Lug shear-out:

$$f_s = \frac{117{,}000}{3.45} = 34{,}000 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{94{,}000}{(1.15)(1.5)(34{,}000)} - 1 = \quad +.60$$

The remainder of the attachment lug is satisfactory by inspection.

4.1.7 <u>Main Rotor Hub Assembly</u> - (Ref. Volume III, Figure 2)

This portion of the rotor system stress analysis is concerned with the static structural substantiation of the rotor hub and attachments.

The material properties presented below are obtained from page 39 of Reference 10 for Ti-6AL-4V titanium alloy.

$$F_{tu} = 162{,}000 \text{ p.s.i.}$$
$$F_{su} = 94{,}000 \text{ p.s.i.}$$

4.1.7.1 <u>General Loading Analysis</u>

From a review of Reference 2, pages 53 through 101, the critical hub loading occurs during the following conditions:

Cond. 1 - Rotor limit speed condition
Cond. 2 - Fwd. flight, 41 m.p.h., 2.5g, 562 f.p.s. tip speed
Cond. 3 - Two engines inoperative in hover

During the condition 1 maneuver a centrifugal force load per blade of 890,000 lb. (limit) is developed at each blade retention pin.

During the condition 2 maneuver the following loading is developed on each blade at rotor station 30.0:

$M_F = 4.40 \times 10^6$ in-lb. (limit) (Ref. 2, Fig. 71, p. 87)

$T = 1.81 \times 10^5$ lb-in. (limit) (Ref. 2, Fig. 8, p. 53)

$C.F. = 4.33 \times 10^5$ lb. (limit) (Ref. 2, Fig. 9, p. 54)

Loading assumptions:

The flapwise moments are carried as a couple by the upper and lower hub tines.

The torsion is carried as a couple by the upper and lower hub tines.

The centrifugal force loads are carried by the hub tines.

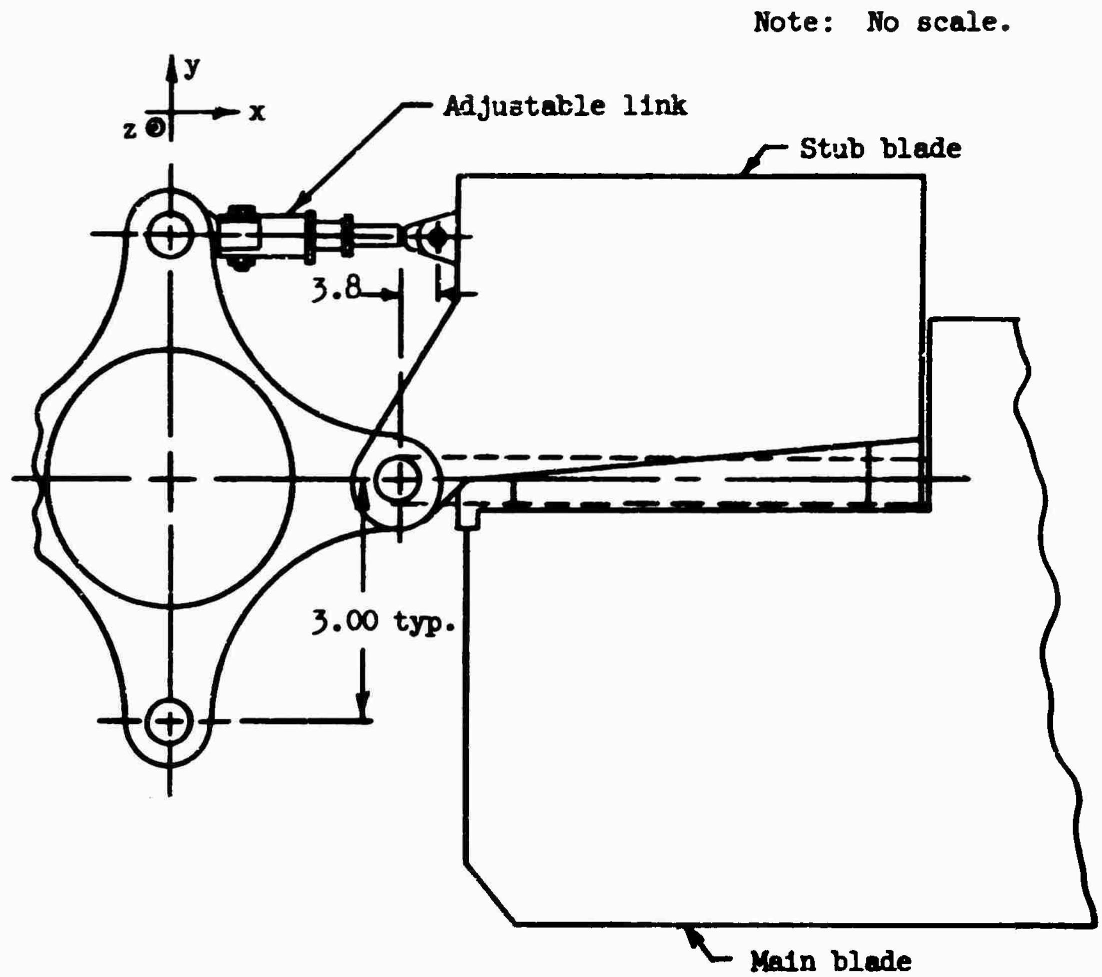


Figure 35. Hub General Geometry and Loading Sketch.

4.1.7.2 Lug Analysis

4.1.7.2.1 Loading Analysis

The highest lug loads are developed during condition 2.

$$M_F = 4.40 \times 10^6 \text{ in-lb. (limit) at rotor sta. } 30.0$$
$$T = 1.81 \times 10^5 \text{ in-lb. (limit) at rotor sta. } 30.0$$
$$C.F. = 4.33 \times 10^5 \text{ lb. (limit) at rotor sta. } 30.0$$

The load per lug tine due to flapwise bending is:

$$P_F = \frac{4.40 \times 10^6}{13.95} = 315{,}000 \text{ lb. (limit)}$$

The load per lug due to torsion is:

$$P_y = \frac{1.81 \times 10^5}{13.95} = 13,000 \text{ lb. (limit)}$$

The load per lug tine due to centrifugal force is:

$$P_x = \frac{4.33 \times 10^5}{2} = 217,000 \text{ lb. (limit)}$$

There also exists loading in the tines due to the restraint spring. During forward flight at 41 miles per hour the rotor system is tilted at an approximate angle of ± 4 degrees with respect to the rotor main mast. This results in the spring restraint load being applied at an angle of 31 degrees with respect to the retention pin centerline.

Spring rate = 364,000 lb. per radian of tilt.

$\alpha = 4^{\circ} = .0698$ rad.

$$P = (.0698)(364,000) = 25,400 \text{ lb. (limit)}$$

$$P_x = 25,400 \sin 31^{\circ} = 13,100 \text{ lb. (limit)}$$

$$P_z = 25,400 \cos 31^{\circ} = 21,800 \text{ lb. (limit)}$$

The maximum tensile load occurs on the lower tine and has a magnitude of:

$$P_x = (315,000 + 217,000 + 13,100 - \frac{13,000}{2}) + \frac{11,700}{2}$$
$$= 538,000 \text{ lb. (limit)}$$

4.1.7.2.2 Section Properties

Tine thickness: $t = 1.75$ in.
Tine diameter: $D = 12.00$ in.
Hole diameter: $d = 6.90$ in.

Tensile area: $A_t = (1.75)(12.00 - 6.90) = 8.92 \text{ in}^2$

Per the discussion on pages 165 and 166 of Reference 9, the lug shear failure is predicted to occur as shown in Figure 36 below.

Note: No scale.

$$A_s = (1.75)(2.9)(2)$$
$$= 10.15 \text{ in}^2$$

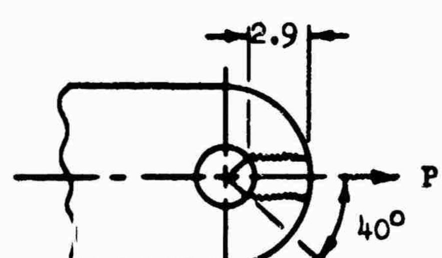


Figure 36. Predicted Lug Shear Failure Location.

4.1.7.2.3 Stress Analysis

Lug tensile tear-out:

$$f_t = \frac{538,000}{8.92} = 60,200 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{162,000}{(1.15)(1.5)(60,200)} - 1 =$$ .56

Lug shear-out:

$$f_s = \frac{538,000}{10.15} = 53,000 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{94,000}{(1.15)(1.5)(53,000)} - 1 =$$ .03

4.1.7.3 Rotor Blade Attachment Pin Assembly

This portion of the hub assembly stress analysis is concerned with the static structural evaluation of the rotor blade attachment pin.

The material properties presented below are obtained from page 39 of Reference 10 for Ti-6AL-4V titanium alloy.

$$F_{tu} = 162,000 \text{ p.s.i.}$$
$$F_{su} = 94,000 \text{ p.s.i.}$$

4.1.7.3.1 Loading Analysis

The maximum pin bending moment occurs during the rotor limit speed condition.

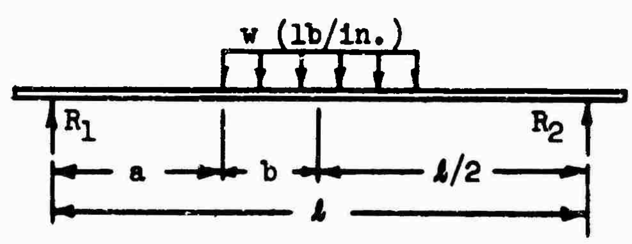


Figure 37. Attachment-Pin-Loading Geometry.

$$R_1 = R_2 = \frac{890,000}{2} = 445,000 \text{ lb. (limit)}$$

$$w = \frac{890,000}{5.25} = 173,000 \text{ lb/in. (limit)}$$

$$a = 4.375 \text{ in.}, \quad b = 2.625 \text{ in.}, \quad \ell = 14.0 \text{ in.}$$

The maximum moment occurs at mid-span:

$$M = R_1(a + .5b) = 445{,}000\left[4.375 + (.5)(2.625)\right]$$
$$= 2{,}530{,}000 \text{ in-lb. (limit)}$$

The maximum pin shear load occurs during condition 2 (see page 74 for definition of cond. 2). The maximum shear load is obtained from page 76.

$$P_s = 538{,}000 \text{ lb. (limit)}$$

4.1.7.3.2 Section Properties

The attachment pin has an outside diameter of 6.5 inches, with a wall thickness of 1.375 inches at mid-span and .625 inches at the ends.

At the mid-span: $Z = \frac{\pi}{(64)(3.25)}(6.5^4 - 3.75^4) = 24.000 \text{ in}^3$

At the ends: $A = \frac{\pi}{4}(6.5^2 - 3.75^2) = 11.536 \text{ in}^2$

4.1.7.3.3 Stress Analysis

$$f_b = \frac{2{,}530{,}000}{24.000} = 105{,}300 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{162{,}000}{(1.5)(105{,}300)} - 1 = +.02$$

$$f_s = \frac{538{,}000}{11.536} = 46{,}600 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{94{,}000}{(1.15)(1.5)(46{,}600)} - 1 = +.17$$

4.1.7.4 Section A-A Stress Investigation

Section A-A is located at rotor station 25.00 on the lower tine, and it carries the maximum loading.

4.1.7.4.1 Section Properties

$$A = (13.8)(1.0) = 13.8 \text{ in}^2$$

$$Z_y = \frac{bh^2}{6} = \frac{(13.8)(1.0)^2}{6} = 2.3 \text{ in}^3$$

$$Z_z = \frac{hb^2}{6} = \frac{(1.0)(13.8)^2}{6} = 31.8 \text{ in}^3$$

4.1.7.4.2 <u>Loading Analysis</u>

$$\left.\begin{array}{l} P_x = 538{,}000 \text{ lb. (limit)} \\ P_y = 13{,}000 \text{ lb. (limit)} \\ P_z = 10{,}900 \text{ lb. (limit)} \end{array}\right\} \text{(Ref. page 76)}$$

$$M_y = (5.0)(10{,}900) = 63{,}500 \text{ in-lb. (limit)}$$

$$M_z = (5.0)(13{,}000) = 65{,}000 \text{ in-lb. (limit)}$$

4.1.7.4.3 <u>Stress Analysis</u>

$$f_t = \frac{538{,}000}{13.8} = 39{,}000 \text{ p.s.i. (limit)}$$

$$f_{s_y} = \frac{13{,}000}{13.8} = 940 \text{ p.s.i. (limit)}$$

$$f_{s_z} = \frac{10{,}900}{13.8} = 790 \text{ p.s.i. (limit)}$$

$$f_{b_y} = \frac{63{,}500}{2.3} = 27{,}600 \text{ p.s.i. (limit)}$$

$$f_{b_z} = \frac{65{,}000}{31.8} = 2{,}040 \text{ p.s.i. (limit)}$$

The maximum tensile stress is:

$$f_t = 39{,}000 + 27{,}600 + 2{,}040 = 68{,}600 \text{ p.s.i. (limit)}$$

The maximum shear stress is:

$$f_s = \sqrt{940^2 + 790^2} = 1{,}230 \text{ p.s.i. (limit) - negligible}$$

$$\text{M.S.} = \frac{162\ 000}{(1.5)(63{,}600)} - 1 = +.57$$

4.1.7.5 <u>Section 12-12 Stress Investigation</u>

Section 12-12 is located at rotor station 16.35 on the lower tine and carries the maximum loading.

4.1.7.5.1 <u>Section Properties</u>

$$A = (22.0)(1.0) = 22.0 \text{ in}^2$$

$$Z_y = \frac{bh^2}{6} = \frac{(22.0)(1.0)^2}{6} = 3.67 \text{ in}^3$$

$$Z_z = \frac{hb^2}{6} = \frac{(1.0)(22.0)^2}{6} = 80.6 \text{ in}^3$$

4.1.7.5.2 Loading Analysis

$$\left.\begin{array}{l} P_x = 538{,}000 \text{ lb. (limit)} \\ P_y = 13{,}000 \text{ lb. (limit)} \\ P_z = 10{,}900 \text{ lb. (limit)} \end{array}\right\} \text{(Ref. page 76)}$$

$$M_y = (13.65)(10{,}900) = 149{,}000 \text{ in-lb. (limit)}$$

$$M_z = (13.65)(13{,}000) = 177{,}500 \text{ in-lb. (limit)}$$

4.1.7.5.3 Stress Analysis

$$f_t = \frac{538{,}000}{22.0} = 24{,}400 \text{ p.s.i. (limit)}$$

$$f_{b_y} = \frac{149{,}000}{3.67} = 40{,}600 \text{ p.s.i. (limit)}$$

$$f_{b_z} = \frac{177{,}500}{80.6} = 2{,}200 \text{ p.s.i. (limit)}$$

The maximum tensile stress is:

$$f_t = 24{,}400 + 40{,}600 + 2200 = 67{,}200 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{162{,}000}{(1.5)(67{,}200)} - 1 = \quad .61$$

4.1.7.6 Adjustable-Link Assembly

This portion of the hub-assembly stress analysis is concerned with the static structural substantiation of the adjustable link assembly.

The material properties presented below are obtained from page 39 of Reference 10 for Ti-6AL-4V titanium alloy.

$$F_{tu} = 162{,}000 \text{ p.s.i.}$$

$$F_{su} = 94{,}000 \text{ p.s.i.}$$

4.1.7.6.1 Loading Analysis

The maximum adjustable-link-assembly loading occurs during the cyclic stick whirl condition.

$$M_C = 3.5 \times 10^6 \text{ in-lb. (limit)} \quad \text{(Ref. page 71)}$$

The axial tension load on the adjustable link is:

$$P = \frac{3.5 \times 10^6}{30} = 117{,}000 \text{ lb. (limit)}$$

4.1.7.6.2 Adjustable-Link-to-Stub-Blade Attachment Bolt Analysis

Section Properties:

The minimum shear area for bolts loaded in double shear is:

Bolt dia. = 2.00 in.

$$A_s = \frac{2\pi D^2}{4} = \frac{\pi(2.00)^2}{2} = 6.28 \text{ in}^2$$

Stress Analysis:

$$f_s = \frac{117{,}000}{6.28} = 18{,}600 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{94{,}000}{(1.15)(1.5)(18{,}600)} - 1 = \qquad 1.93$$

4.1.7.6.3 Stub End Adjustable Link Lug

Section Properties:

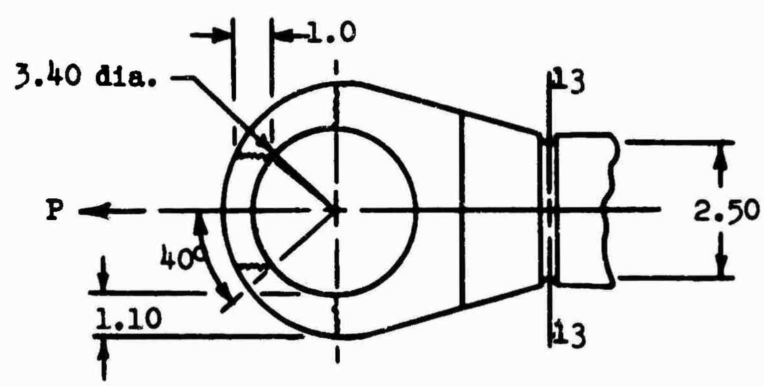


Figure 38. Lug General Geometry Sketch.

Lug thickness: $t = 1.20$ in.

Tension area: $A_t = (2)(1.2)(1.0) = 2.40 \text{ in}^2$

Shear area: $A_s = (2)(1.2)(1.1) = 2.64 \text{ in}^2$

Section 13-13: $A = \dfrac{\pi(2.5)^2}{4} = 4.9 \text{ in}^2$

Stress Analysis:

Lug tension tear-out:

$$f_t = \frac{117{,}000}{2.40} = 49{,}800 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{162,000}{(1.15)(1.5)(49,800)} - 1 = \qquad .89$$

Lug shear-out:

$$f_s = \frac{117,000}{2.64} = 44,400 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{94,000}{(1.15)(1.5)(44,400)} - 1 = \qquad .23$$

Section 13-13 tensile stress:

$$f_t = \frac{117,000}{4.9} = 24,000 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{162,000}{(1.5)(24,000)} - 1 = \qquad 3.50$$

The remainder of the adjustable-link assembly is satisfactory for the applied loading by inspection.

4.1.8 Gimbal and Attachments

This portion of the rotor system stress analysis is concerned with the static structural substantiation of the gimbal ring, the pivot pins, the gimbal bearings, the rotor shaft lugs, and the rotor hub bearing lugs.

The material properties presented below are obtained from page 39 of Reference 10 for Ti-6AL-4V titanium alloy.

$$F_{tu} = 162,000 \text{ p.s.i.} \qquad E = 18.5 \times 10^6 \text{ p.s.i.}$$
$$F_{su} = 94,000 \text{ p.s.i.}$$

4.1.8.1 Loading Analysis

The critical gimbal loading occurs during the 2.5g loading condition.

Total ship weight: $P_T = 72,000$ lb. (limit)

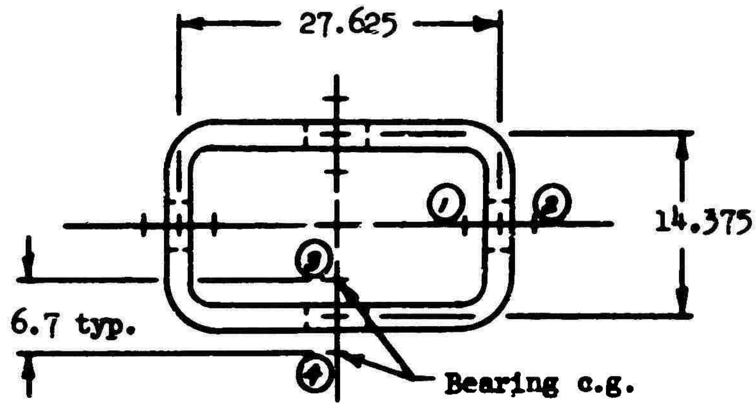
Rotor system weight: $P_R = 20,000$ lb. (limit)

Weight supported by gimbal:

$$P_W = 52,000 \text{ lb. (limit)}$$

For the 2.5g vertical loading condition, the load supported by the gimbal is:

$$P_z = (2.5)(52,000) = 130,000 \text{ lb. (limit)}$$



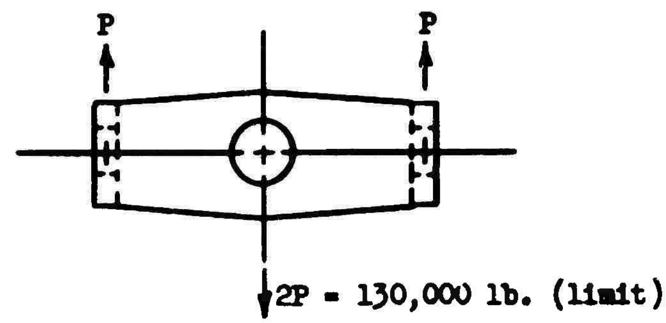


Figure 39. Gimbal-Ring General Loading Sketch.

Outside Bearing Loads:

Assuming the cantilevered load P rotates through an angle θ, and 1/2 of this rotation is taken out by a moment in the bearing and torsion in the gimbal.

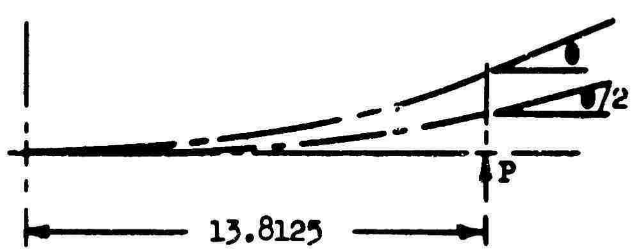


Figure 40. Bearing Loading Sketch.

$$\theta = \frac{P\ell^2}{2EI} = \frac{(65{,}000)(13.8125)^2}{(2)(18.5)(10^6)(100)} = .00336 \text{ rad.}$$

$\theta/2 = .00168$ rad., $I \approx 100$ in^4

The end rotation is imparted to the bearings by torsion in the gimbal.

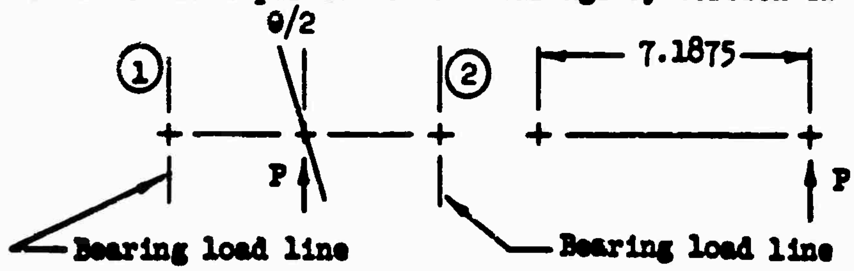


Figure 41. Bearing Loading Geometry.

From Ref. 6, page 331:

$$T = \frac{(\theta/2)\beta b t^3 G}{\ell} = \frac{(.00168)(.285)(9)(2)^3(6.2)(10^6)}{7.1875} = 29{,}800 \text{ lb-in. (limit)}$$

where:

$\theta/2$ = .00168 rad.
β = .285 rad.
b = 9.0 in.
t = 2.0 in.
G = 6.2×10^6 p.s.i.
ℓ = 7.1875 in.

Load on bearing 1:

$$P_1 = \frac{65{,}000}{2} + \frac{29{,}800}{6.7} = 37{,}950 \text{ lb. (limit)}$$

Load on bearing 2:

$$P_2 = \frac{65{,}000}{2} - \frac{29{,}800}{6.7} = 28{,}050 \text{ lb. (limit)}$$

The bearing loads for the 1g-loading condition are presented below.

$$P = 26{,}000 \text{ lb. (limit)}$$

$$\theta = (.00336)\left(\frac{26{,}000}{65{,}000}\right) = .001345 \text{ rad.}$$

$$T = (29{,}800)\left(\frac{.001345}{.00336}\right) = 11{,}900 \text{ lb-in. (limit)}$$

Load on bearing 1:

$$P_1 = \frac{26{,}000}{2} + \frac{11{,}900}{6.7} = 14{,}780 \text{ lb. (limit)}$$

Load on bearing 2:

$$P_2 = \frac{26{,}000}{2} - \frac{11{,}900}{6.7} = 11{,}220 \text{ lb. (limit)}$$

Inside Bearing Loads:

The same method of analysis is used on the inside bearings as on the outside bearings.

$$\theta = \frac{(65{,}000)(7.1875)^2}{(2)(18.5)(10^6)(100)} = .00091 \text{ rad.}$$

$$\theta/2 = .000455 \text{ rad.}$$

$$T = \frac{(.000455)(.285)(9)(2)^3(6.2)(10^6)}{13.820} = 4{,}190 \text{ lb-in. (limit)}$$

Load on bearing (3):

$$P_3 = \frac{65{,}000}{2} + \frac{4{,}190}{6.7} = 33{,}100 \text{ lb. (limit)}$$

Load on bearing (4):

$$P_4 = \frac{65{,}000}{2} - \frac{4{,}190}{6.7} = 31{,}900 \text{ lb. (limit)}$$

The bearing loads for the 1g-loading condition are presented below.

$$P = 26{,}000 \text{ lb. (limit)}$$

$$\theta = (.00091)\left(\frac{26{,}000}{65{,}000}\right) = .000364 \text{ rad.}$$

$$T = (4{,}190)\left(\frac{.000364}{.00091}\right) = 1{,}680 \text{ lb-in. (limit)}$$

Bearing (3): $P_3 = \frac{26{,}000}{2} + \frac{1{,}680}{6.7} = 13{,}250$ lb. (limit)

Bearing (4): $P_4 = \frac{26{,}000}{2} - \frac{1{,}680}{6.7} = 12{,}750$ lb. (limit)

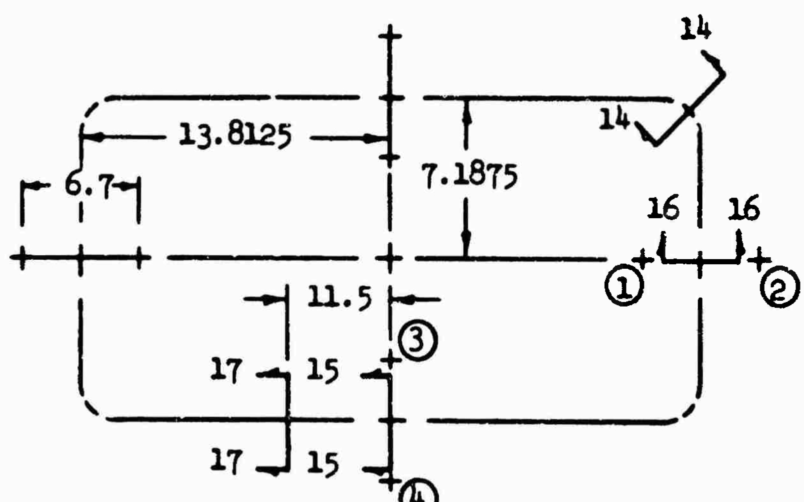


Figure 42. Gimbal-Bearings General Loading Sketch.

4.1.8.2 <u>Gimbal Ring Analysis</u>

<u>Section 14-14:</u>

Section geometry:

b = 8.0 in.
t = 2.0 in.

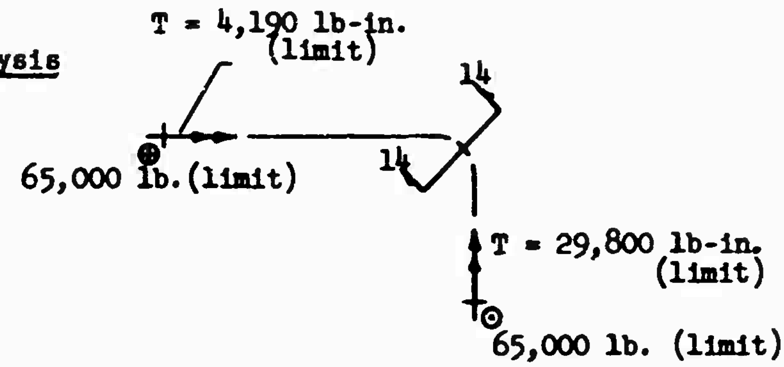


Figure 43. Gimbal Ring - Section 14-14 Loading and Geometry Sketch.

Section 14-14 is under normal and torsional shear.

$$f_{s_n} = \frac{65,000}{(2)(8)} = 4,060 \text{ p.s.i. limit)}$$

From Ref. 6, page 330:

$$f_{s_t} = \frac{3T}{bt^2} = \frac{(3)\sqrt{(29,800^2 + 4,190^2)}}{(8)(2)^2} = 2,820 \text{ p.s.i. (limit)}$$

The maximum shear stress is:

$$f_{s_{max}} = 4,060 + 2,820 = 6,880 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{94,000}{(1.5)(6,880)} - 1 = \quad \text{High}$$

<u>Section 15-15:</u>

Section 15-15 is under bending and torsion.

Section properties:

$$I = \frac{b(h_1^3 - h_2^3)}{12}$$

$$= \frac{(4)(9^3 - 4.125^3)}{12}$$

$$= 220 \text{ in}^4$$

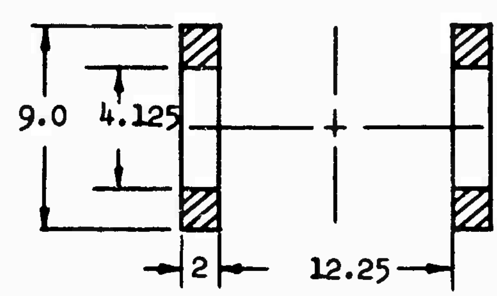


Figure 44. Gimbal Ring - Section 15-15 Geometry.

The bending stress is:

$$f_b = \frac{(13.8125)(65,000)(4.5)}{220} = 18,400 \text{ p.s.i. (limit)}$$

The torsional load is obtained from page 85, and is assumed to be carried as a couple by the lugs.

$$T = 4{,}190 \text{ lb-in. (limit)}$$

$$P = \frac{4{,}190}{14.25} = 294 \text{ lb. (limit)}$$

$$f_{s_t} = \frac{294}{(2)(9 - 4.125)} = 30 \text{ p.s.i. (limit) negligible}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{162{,}000}{(1.15)(1.5)(18{,}400)} - 1 = \quad \text{High}$$

Section 16-16:

Section 16-16 is under bending and torsion. The section properties are identical to Section 15-15 section properties.

$$I = 220 \text{ in}^4$$

$$f_b = \frac{(7.1875)(65{,}000)(4.5)}{220} = 9{,}560 \text{ p.s.i. (limit)}$$

The torsional load on Section 16-16 is obtained from page 84 and is assumed to be carried as a couple by the lugs.

$$T = 29{,}800 \text{ lb-in. (limit)}$$

$$P = \frac{29{,}800}{14.25} = 2{,}100 \text{ lb. (limit)}$$

$$f_{s_t} = \frac{2{,}100}{(2)(9 - 4.125)} = 216 \text{ p.s.i. (limit) negligible}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{162{,}000}{(1.15)(1.5)(9{,}560)} - 1 = \quad \text{High}$$

Section 17-17:

$$I_{x-x} = \frac{(2)(8)^3}{12} - \frac{(1.6)(7.2)^3}{12}$$

$$= 85 - 49.8$$

$$= 35.2 \text{ in}^4$$

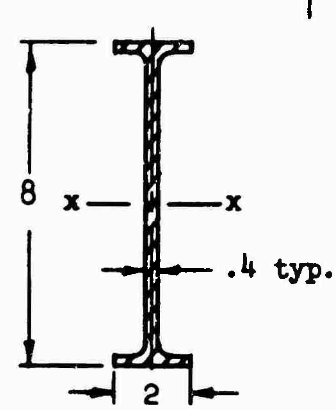


Figure 45. Gimbal Ring - Section 17-17 Geometry.

Bending stress:

$$f_b = \frac{(65,000)(11.5)(4)}{35.2} = 85,000 \text{ p.s.i. (limit)}$$

$$\text{M.S.} = \frac{162,000}{(1.5)(85,000)} - 1 = \quad .27$$

Longitudinal shear stress:

$$f_s = \frac{VQ}{Ib} = \frac{(65,000)(4.78)}{(35.2)(.4)} = 22,000 \text{ p.s.i. (limit)}$$

$$Q = \bar{Y}A = (2.13)(2.24) = 4.78 \text{ in}^3$$

$$\text{M.S.} = \frac{94,000}{(1.5)(22,000)} - 1 = \quad 1.84$$

4.1.8.3 <u>Gimbal Hardware Analysis</u>

4.1.8.3.1 <u>Pivot Pins and Bearings</u>

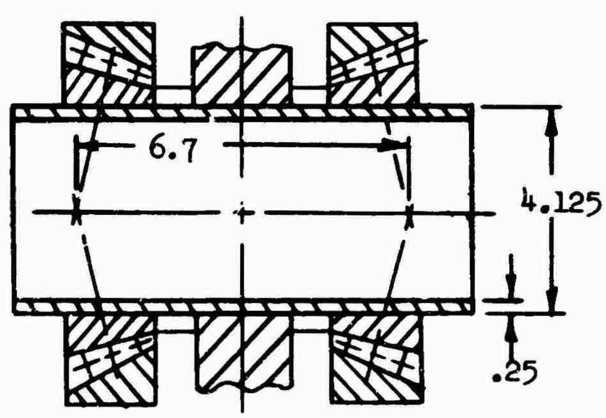


Figure 46. Pivot Pin and Bearings Geometry.

<u>Pin and Bearings Loading Analysis:</u>

Since the outer pin carries the highest loading, it is used to substantiate the inner gimbal support.

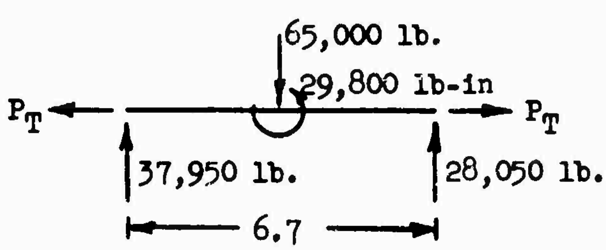


Figure 47. Pivot Pin Loading Sketch.

From Reference 11, page 174, and from the revision supplement dated 13 September 1962, pages 3 to 5:

$$K = 1.51$$

$$P_T = \frac{37,950}{1.51} = 25,200 \text{ lb. (limit) tension}$$

Bearing Stress Analysis:

From Reference 11, page 174, and from the revision supplement dated 13 September 1962, pages 3 to 5:

Radial basic rating = 13,400 lb.

Since the rotor speed is 100 revolutions per minute and the maximum rotor tilt is 10°, the true gimbal bearing speed is:

$$RPM = \frac{(\text{rotor r.p.m.})(2\alpha)}{360} = \frac{(100)(2)(10)}{360} = 5.6 \text{ r.p.m.}$$

The speed factor from Reference 11 is 3.233, and:

Radial bearing rating = (13,400)(3.233) = 43,400 lb.

$$M.S. = \frac{\text{radial bearing rating}}{\text{bearing load}} - 1$$

$$= \frac{43,400}{37,950} - 1 = \qquad + .14$$

Pin Stress Analysis:

The section modulus for a tube with a 4.00-inch outside diameter and a wall thickness of .25 inch is:

$$Z = 2.78 \text{ in}^3, \quad A = 3.04 \text{ in}^2$$

$$f_b = \frac{(37,950)(3.35)}{2.78} = 45,800 \text{ p.s.i. (limit)}$$

$$f_t = \frac{25,200}{3.04} = 8,300 \text{ p.s.i. (limit)}$$

$$f_{max} = 45,800 + 8,300 = 54,100 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{162,000}{(1.5)(54,100)} - 1 = \qquad + 1.00$$

The shear stress is:

$$f_s = \frac{37,950}{3.04} = 12,500 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{93,000}{(1.5)(12,500)} - 1 = \qquad + \text{High}$$

4.1.8.3.2 <u>Rotor-Shaft-Bearing Lugs</u>

The weakest lug with the highest applied load is analyzed below.

P_3 = 33,100 lb. (limit)

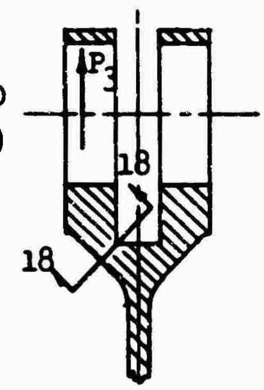


Figure 48. Rotor-Shaft-Bearing-Lug Geometry.

The material properties for Ti-6AL-4V titanium alloy are obtained from page 39 of Reference 10.

F_{tu} = 162,000 p.s.i.

F_{su} = 94,000 p.s.i.

<u>Lug Analysis:</u>

$P = 33{,}100$ lb. (limit)

$OD = 8.2$ in.

$ID = 7.125$ in.

$t = 2.5$ in.

Shear area: $A_s = 2.5(8.2 - 7.125) = 2.69 \text{ in}^2$

$$f_s = \frac{P}{A} = \frac{33{,}100}{2.69} = 12{,}300 \text{ p.s.i.}$$

Using a 1.15 fitting factor, the margin of safety is:

$$M.S. = \frac{95{,}000}{(1.15)(1.5)(12{,}300)} - 1 = \quad 3.48$$

<u>Section 18-18 Stress Analysis:</u>

$$Z_{xx} = \frac{(5)(.9)^2}{6} = .684 \text{ in}^3$$

$P_t = (33{,}100) \cos 30^\circ$
$= 28{,}700$ lb. (limit)

$P_b = (33{,}100) \sin 30^\circ$
$= 16{,}600$ lb. (limit)

$M = (3.5)(16{,}600)$
$= 58{,}100$ in-lb. (limit)

$\theta = 30^\circ$

$P_3 = 33{,}100$ lb. (limit)

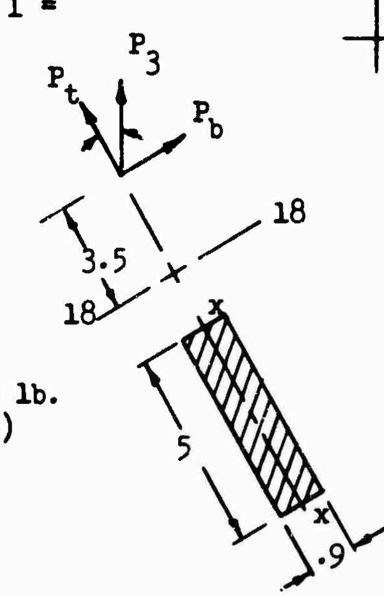


Figure 49. Section 18-18 Loading and Geometry Sketch.

$$f_t = \frac{28,700}{(5)(.9)} = 6,480 \text{ p.s.i. (limit)}$$

$$f_b = \frac{58,100}{.684} = 85,000 \text{ p.s.i. (limit)}$$

$$f_{max} = 85,000 + 6,480 = 91,500 \text{ p.s.i. (limit)}$$

$$M.S. = \frac{162,000}{(1.5)(91,500)} - 1 = \quad .18$$

4.1.8.3.3 Hub Bearing Lugs

This portion of the rotor system stress analysis is satisfactory by comparison to the analysis presented in Section 4.1.8.3.2.

4.1.9 Restraint Spring Assembly

4.1.9.1 Spring Analysis

From Reference 8, pages 49 to 52, nomograms for determining spring stress levels are presented. Assuming a load of 5,000 pounds on each spring, and using Chart 5, page 52 of Reference 8:

For the outside spring:

Mean diameter: 9.0 in.
Wire diameter: 1.6 in.
Number of coils: 5

Spring rate:

$$\delta_o = .39 \text{ in/coil}$$

$$\delta_{t_o} = (.39)(5) = 1.95 \text{ in.}$$

$$K_o = \frac{P}{\delta_{t_o}} = \frac{5,000}{1.95} = 2,570 \text{ lb/in.}$$

For the inside spring:

Mean diameter: 6.0 in.
Wire diameter: 1.1 in.
Number of coils: 9

Spring rate:

$$\delta_i = .525 \text{ in/coil}$$

$$\delta_{t_i} = (9)(.525) = 4.7 \text{ in.}$$

$$K_i = \frac{P}{\delta_{t_i}} = \frac{5,000}{4.7} = 1,070 \text{ lb/in.}$$

Total spring constant:

$$K_t = K_o + K_i = 2,570 + 1,070 = 3,640 \text{ lb/in.}$$

With a maximum load of 22,000 pounds (limit) on the restraint spring assembly, the deflection is:

$$\delta = \frac{P}{K_t} = \frac{22,000}{3,640} = 6.1 \text{ in.}$$

The outside spring loading is:

$$P_o = K_o\delta = (2,570)(6.1) = 15,700 \text{ lb. (limit)}$$

and from Chart 2, page 49 of Reference 8, the outside spring fiber stress is:

$$f = 88,000 \text{ p.s.i. (limit)}$$

The stress correction for curvature is found in Chart 3 of Reference 8:

$$f_c = 110,000 \text{ p.s.i. (limit)}$$

The inside spring loading is:

$$P_i = K_i\delta = (1,070)(6.1) = 6,500 \text{ lb. (limit)}$$

and from Chart 2, page 49 of Reference 8, the inside spring fiber stress is:

$$f = 76,000 \text{ p.s.i. (limit)}$$

The stress correction for curvature is found in Chart 3 of Reference 8:

$$f_c = 99,000 \text{ p.s.i. (limit)}$$

Using a spring steel with an ultimate shear stress of 180,000 p.s.i., the minimum margin of safety is:

$$M.S. = \frac{180,000}{(1.5)(110,000)} - 1 = \quad .09$$

4.1.9.1.2 <u>Lug Analysis</u>

The lug materials presented below are obtained from page 39 of Reference 10 for Ti-6AL-4V titanium alloy.

$$F_{tu} = 162,000 \text{ p.s.i.,} \qquad F_{su} = 94,000 \text{ p.s.i.}$$

<u>Section Properties:</u>

Lug diameter:	D = 3.00 in.
Hole diameter:	d = 1.00 in.
Lug thickness:	t = .50 in.

Tensile area: $A_t = (.50)(3.00 - 1.00) = 1.00 \text{ in}^2$

Shear area at 40°: $A_s = (.50)(2.0)(1.08) = 1.08 \text{ in}^2$

<u>Stress Analysis:</u>

Lug tensile tear-out:

$$f_t = \frac{22{,}000}{1.00} = 22{,}000 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{162{,}000}{(1.15)(1.5)(22{,}000)} - 1 = \qquad +3.26$$

Lug shear-out:

$$f_s = \frac{22{,}000}{1.08} = 20{,}400 \text{ p.s.i. (limit)}$$

Using a 1.15 fitting factor, the margin of safety is:

$$\text{M.S.} = \frac{94{,}000}{(1.15)(1.5)(20{,}400)} - 1 = \qquad +1.67$$

The remainder of the restraint spring assembly is satisfactory by inspection.

5.0 ROTOR SYSTEM FATIGUE ANALYSIS

5.1 Introduction

It is the purpose of this part of the report to provide fatigue data to predict the available fatigue strength of the rotor system components. Only those components which are major structural members are considered from a structural viewpoint.

The fatigue analysis primarily consists of an investigation of the rotor system to illustrate that the alternating stresses developed during normal flight conditions are below the component material endurance limit and nondamaging. In some cases a start-stop analysis of the centrifugal force is conducted. The normal flight condition is defined as: Forward flight, 1.0g, 650 feet per second tip speed, 144 miles per hour forward velocity.

During the stop-start analysis the assumed occurrence is four starts and stops per hour with the design maximum rotor speed centrifugal force occurring at 91 percent of the FAA loading spectrum, and the rotor overspeed centrifugal force occurring at 9 percent of the FAA loading spectrum.

A modified Goodman Diagram is used to define the operating boundary which indicates the maximum alternating stress or load that may be applied with a steady stress or load and still obtain unlimited life for a part. A modified Goodman Diagram is presented in Figure 50. If actual test data were plotted on the modified Goodman, a parabolic curve would be obtained. Usually, the data available is insufficient to draw an accurate boundary, and therefore the boundary is assumed as a straight line. The perfect specimen endurance boundary is established as a line drawn from the ultimate allowable to the unnotched endurance allowable and a line from the yield allowable drawn at 45 degrees. The latter portion of the boundary is imposed by the Federal Aviation Agency's rule which does not allow yielding.

The failure boundary on the modified Goodman Diagram is determined by dividing the slope of the perfect specimen endurance boundary by the fatigue component's theoretical fatigue notch factor. The operating boundary is determined by dividing the failure boundary slope by a safety factor (the Federal Aviation Agency requires a safety factor of 3).

Because the straight line is assumed, it is possible to express vibratory data in terms of equivalent alternating stress or load. The expression is, from similar triangles on the Goodman:

$$f_{a_{eq}} = \frac{F_{tu}}{F_{tu} - f_s} (f_a)$$

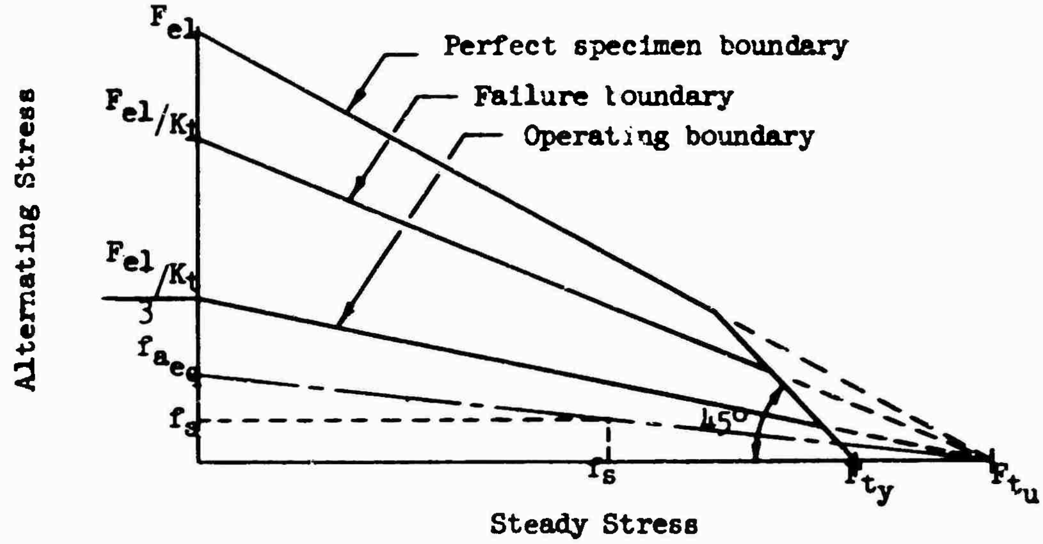


Figure 50. Modified Goodman Diagram.

Generally, an available fatigue notch factor is estimated for the normal flight condition stresses, and in some cases for the stop-start analysis. In cases where fatigue data permits, analysis of critical attachment bolts and pins is conducted to determine the required bolt diameter to approximately establish a 10,000-hour service life based on S-N data reduced by a factor of 2.0.

The rotor system is analyzed progressing in order of force transmission from the engine nacelles at the tip to the hub and gimbal at the rotor shaft.

From a review of the rotor system static stress analysis, presented in Section 4.0 of this report, the rotor system primary structure is constructed of solution heat-treated Ti-6AL-4V titanium alloy, and the secondary structure is constructed of Ti-8AL-1Mo-1V annealed titanium alloy.

The unnotched endurance limit for solution heat-treated Ti-6AL-4V titanium alloy is obtained from Reference 15.

$$F_{tu} = 162{,}000 \text{ p.s.i.,} \qquad F_{e\ell} = \pm 130{,}000 \text{ p.s.i.}$$

The unnotched endurance limit for Ti-8AL-1Mo-1V annealed titanium alloy is obtained from page 3.25.2-4 of Reference 16.

$$F_{tu} = 130{,}000 \text{ p.s.i.,} \qquad F_{e\ell} = \pm 70{,}000 \text{ p.s.i.}$$

5.1.1 Engine Nacelle and Attachment

Per the discussion on page 7 of this report, the engine nacelles are not considered from a structural viewpoint at this time.

5.1.2 Engine Mount System and Attachment

From a review of the engine-mount-system static analysis presented in Section 4.1.2 of this report, and based on previous helicopter component fatigue experience, the critical engine mount system fatigue areas are the attachment lugs and bolts during the start-stop condition. The attachment bolts and lugs are also analyzed for the normal level flight condition to illustrate that the applied fatigue stress is below the material endurance limit during normal flight conditions.

5.1.2.1 Engine-to-Mount Attachment Bolts

Preliminary fatigue analysis has established the engine-to-mount attachment bolts to be critical during the start-stop condition. An analysis is conducted below to determine the required bolt diameter to approximately establish a 10,000-hour service life when a fatigue notch factor of 2.0 is applied to the S-N data.

Material type: H1 T1 20 series bolts

$F_{tu} = 200{,}000$ p.s.i.

$F_{e\ell} = \pm 31{,}000$ p.s.i. (using a fatigue notch factor of 2.0)

Assumed bolt diameter:

$$D = 1.65 \text{ in.}$$

$$Z = \frac{\pi}{32}(1.65)^3 = .441 \text{ in}^3$$

The equations used to determine the moments and stress levels are obtained from page 16 of this report. The design maximum rotor speed centrifugal force is obtained from page 8.

$$C.F. = \pm 87{,}000 \text{ lb. per engine}$$

$$P = \frac{\pm 87{,}000}{2} = \pm 43{,}500 \text{ lb. per bolt}$$

$$M = \frac{Pb}{2} = \frac{(\pm 43{,}500)(.997)}{2} = \pm 21{,}700 \text{ in-lb.}$$

$$f_a = f_{a_{eq}} = \frac{\pm 21{,}700}{.441} = \pm 49{,}000 \text{ p.s.i.}$$

The rotor overspeed operation centrifugal force is:

$$C.F. = \left(\frac{683}{650}\right)^2 (\pm 87{,}000) = \pm 96{,}000 \text{ lb.}$$

$$P = \frac{\pm 96{,}000}{2} = \pm 48{,}000 \text{ lb. per bolt}$$

$$M = \frac{Pb}{2} = \frac{(\pm 48{,}000)(.997)}{2} = \pm 24{,}000 \text{ in-lb.}$$

$$f_a = f_{a_{eq}} = \frac{\pm 24{,}000}{.441} = \pm 54{,}000 \text{ p.s.i.}$$

The engine environmental alternating vertical load factor of ±40g used in the static analysis was conservatively estimated during development design and prior to the dynamic load study completion. The rotor system stiffness has increased such that the vertical load factor computed during the dynamic load study has a magnitude of ±3.4g (see Volume IV, page 44) for the 2.5g loading condition. The ±3.4g vertical load factor is conservatively used to compute the alternating vertical engine loads acting on the engine mount system during normal flight conditions.

Engine weight: $W_T = 370$ lb.

$$P_z = 370 + (\pm 3.4)(370) = 370 \pm 1{,}260 \text{ lb. per engine}$$

The in-plane load factor of ±5.0g is conservatively used in computing the alternating in-plane engine mount loads.

Thrust per engine = 1,500 lb.
Weight per engine = 370 lb.

$$P_y = 1{,}500 + (\pm 5)(370) = 1{,}500 \pm 1{,}850 \text{ lb. per engine}$$

The steady centrifugal force load per engine is obtained from page 8 of this report.

C.F. = 87,000 lb. per engine

The maximum load per bolt is the vector quantity of the x and z loading.

$$P_x = \frac{87{,}000}{2} + \frac{11.25}{13.00}(370 \pm 1{,}260) + \frac{11.25}{15.125}(1{,}500 \pm 1{,}850)$$

$$= 43{,}500 + (370 \pm 1{,}090) + (1{,}210 \pm 1{,}500)$$

$$= 45{,}000 \pm 2{,}590 \text{ lb.}$$

$$P_z = 370 \pm 1{,}260 \text{ lb.}$$

$$P = \sqrt{P_x^2 + P_z^2} = 45{,}000 \pm 2{,}900 \text{ lb.}$$

$$M = \frac{Pb}{2} = \frac{(45{,}000 \pm 2{,}900)(.997)}{2} = 22{,}400 \pm 1{,}435 \text{ in-lb.}$$

$$f_t = \frac{22,400 \pm 435}{.441} = 50,700 \pm 3,250 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{220}{200 - 53.0} (\pm 3250) = \pm 4,350 \text{ p.s.i.}$$

The S-N curve shape presented in Figure 51 is obtained from data in References 7 and 16. The S-N curve shape has been reduced by a factor of 2.0.

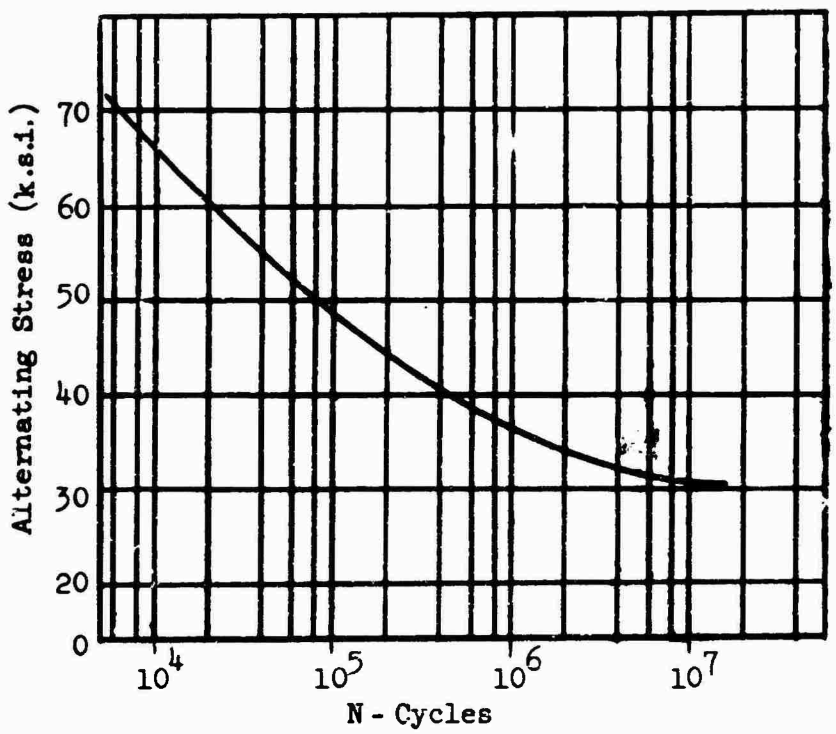


Figure 51. H1 T1 Series Bolts S-N Data.

TABLE 2 ENGINE TO MOUNT ATTACHMENT BOLT FINITE LIFE ESTIMATION					
(1)	(2)	(3)	(4)	(5)	(6)
$f_{a_{eq}}$	N-Cycles	% Occur.	Cycles/Hr.	Hour	% Occur/Hr.
	Figure 51	Page 94	Page 94	(2)/(4)	(3)/(5)
±49,000	1.01×10^5	91.0	4.0	2.53×10^4	.00360
±54,000	4.9×10^4	9.0	4.0	1.225×10	.00074
±4,350	∞	-	-	-	-
				$\Sigma =$	.00434

Calculated life: $L_c = \frac{100}{.00434} = 23,000$ hours

Service life: S.L. $= 1,250 + (.375)(23,000) = 9,900$ hours

5.1.2.2 Engine-to-Mount Attachment Lugs

The engine-to-mount attachment lugs are investigated for lug tensile tear-out during rotor overspeed operation for the start-stop condition and during normal flight condition.

Material type: Solution heat-treated Ti-6AL-4V titanium alloy

$$F_{tu} = 162,000 \text{ p.s.i.}$$

$$F_{e\ell} = \pm 130,000 \text{ p.s.i. (unnotched)}$$

The rotor overspeed condition centrifugal force load is obtained from page 97 of this report.

$$P = \pm 48,000 \text{ lb.}$$

$$f_a = f_{a_{eq}} = \frac{\pm 48,000}{1.89} = \pm 25,400 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 130,000}{\pm 25,400} = 5.12$$

The applied loading during the normal flight condition is obtained from page 97 of this report.

$$P = 45,000 \pm 2,900 \text{ lb.}$$

$$A = 1.89 \text{ in}^2 \text{ (Ref. page 17)}$$

$$f = \frac{45,000 \pm 2,900}{1.89} = 24,800 \pm 1,530 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{162}{162 - 24.8} (\pm 1,530) = \pm 1,810 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 130,000}{\pm 1,810} = 71.8$$

5.1.2.3 Heat Expansion Fitting - Section 4-4

From a review of pages 27 and 28 of this report, Section 4-4 is critical in bending. An investigation is conducted to estimate the available fatigue notch factor during the rotor overspeed condition and during the normal flight condition.

Material type: Solution heat-treated Ti-6AL-4V titanium alloy

$$F_{tu} = 162,000 \text{ p.s.i.}$$

$$F_{e\ell} = \pm 130,000 \text{ p.s.i. (unnotched)}$$

The rotor overspeed operation centrifugal force load is obtained from page 97 of this report.

$$2P_x = \pm 48{,}000 \text{ lb.}$$

$$M_z = \frac{.30}{2}(\pm 48{,}000) = \pm 7{,}200 \text{ in-lb.}$$

$$f_b = \frac{(\pm 7{,}200)(.39)}{2} = \pm 35{,}600 \text{ p.s.i.}$$

$$P_{x'} = \frac{\pm 48{,}000}{2} \cos 55^\circ = \pm 13{,}800 \text{ lb.}$$

$$f_b = \frac{\pm 13{,}800}{1.56} = \pm 8{,}850 \text{ p.s.i.}$$

$$f_a = f_{a_{eq}} = (\pm 35{,}600) + (\pm 8{,}850) = \pm 44{,}500 \text{ p.s.i.}$$

The available fatigue notch factor for the start-stop condition is:

$$K_{f_{av}} = \frac{\pm 130{,}000}{\pm 44{,}500} = 2.92$$

The normal flight condition applied stress is developed below.

$$2P_x = 45{,}000 \pm 2{,}590 \text{ lb.} \quad \text{(Ref. page 97)}$$

$$4P_y = 1{,}500 \pm 1{,}850 \text{ lb.} \quad \text{(Ref. page 97)}$$

$$\begin{aligned} M_z &= (.30)\frac{(45{,}000 \pm 2{,}590)}{2} + (3.0)\frac{(1{,}500 \pm 1{,}850)}{4} \\ &= (6{,}750 \pm 390) + (1{,}130 \pm 1{,}390) \\ &= 7{,}880 \pm 1{,}780 \text{ in-lb.} \end{aligned}$$

$$f_b = \frac{(7{,}880 \pm 1{,}780)(.39)}{.079} = 39{,}000 \pm 8{,}800 \text{ p.s.i.}$$

$$\begin{aligned} P_{x'} &= \frac{45{,}000 \pm 2{,}590}{2} \cos 55^\circ + \frac{1{,}500 \pm 1{,}850}{4} \sin 55^\circ \\ &= (12{,}900 \pm 740) + (300 \pm 380) \\ &= 13{,}200 \pm 1{,}120 \text{ lb.} \end{aligned}$$

$$f_t = \frac{13{,}200 \pm 1{,}120}{1.56} = 8{,}460 \pm 720 \text{ p.s.i.}$$

The maximum stress is:

$$f_{max} = f_b + f_t = 47{,}500 \pm 9{,}520 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{162}{162 - 47.5}(\pm 9{,}520) = \pm 13{,}500 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 130{,}000}{\pm 13{,}500} = 9.64$$

5.1.2.4 Mount-to-Blade-Tip Attachment Bolts

Preliminary fatigue analysis has established the mount-to-blade-tip attachment bolts to be critical during the start-stop condition. An analysis is conducted below to determine the required bolt diameter to establish approximately a 10,000-hour service life when a fatigue notch factor of 2.0 is applied to the S-N data. An investigation of the normal flight condition is included to demonstrate that the fatigue stress is below the material endurance limit and nondamaging.

Material type: H1 T1 20 series bolts.

$$F_{tu} = 200{,}000 \text{ p.s.i.}$$

$$F_{e\ell} = \pm 31{,}000 \text{ p.s.i.}$$ (using a fatigue notch factor of 2.0)

Assumed bolt diameter:

$$D = 1.365 \text{ in.}$$

$$Z = \frac{\pi}{32}(1.365)^3 = .25 \text{ in}^3$$

The equations used to determine the moments and stress levels are obtained from page 37 of this report. The normal rotor operating speed centrifugal force is obtained from page 8.

$$C.F. = \pm 198{,}400 \text{ lb.}$$

The load per bolt is:

$$P = \frac{\pm 198{,}400}{4} = \pm 49{,}600 \text{ lb. per bolt}$$

$$M = \frac{Pb}{2} = \frac{(\pm 49{,}600)(.4875)}{2} = \pm 12{,}100 \text{ in-lb.}$$

$$f_a = f_{a_{eq}} = \frac{\pm 12{,}100}{.25} = \pm 48{,}500 \text{ p.s.i.}$$

The rotor overspeed operation centrifugal force load is:

$$C.F. = (1.05)^2(\pm 198{,}400) = \pm 219{,}000 \text{ lb.}$$

The load per bolt is:

$$P = \frac{\pm 219{,}000}{4} = \pm 54{,}800 \text{ lb.}$$

$$M = \frac{Pb}{2} = \frac{(\pm 54{,}800)(.4875)}{2} = \pm 13{,}400 \text{ in-lb.}$$

$$f_a = f_{a_{eq}} = \frac{\pm 13{,}400}{.25} = \pm 53{,}600 \text{ p.s.i.}$$

The stress level developed during the normal flight condition is presented on the following page.

$C.F. = 198{,}400$ lb. (Ref. page 11)

The gyroscopic moment per engine tends to force the rotor blade leading edge down, and has a magnitude of:

$M_x = 94{,}000$ in-lb. (Ref. page 11)

The vertical engine load normal to the blade axis is:

$$M_y = (\pm 3.4)(24.00)(370) = \pm 29{,}400 \text{ in-lb.}$$
$$P_z = (\pm 3.4)(370) = \pm 1{,}260 \text{ lb.}$$
$$P_y = 2\ 1{,}500 + (\pm 5)(370) = 3{,}000 \pm 3{,}700 \text{ lb.}$$

The load per bolt is:

$$P = \sqrt{\left[\frac{CF}{4} + \frac{M_y}{2\ell}\right]^2 + \left[\frac{P_y}{4} + \frac{M_x}{2\ell}\right]^2}$$

$$P = \sqrt{\left[\frac{198{,}400}{4} + \frac{\pm 29{,}400}{(2)(7.6)}\right]^2 + \left[\frac{3{,}000 \pm 3{,}700}{4} + \frac{94{,}000}{(2)(7.6)}\right]^2}$$

$$= \sqrt{(49{,}600 \pm 1{,}945)^2 + (6{,}950 \pm 925)^2}$$

$$= 50{,}000 \pm 2{,}150 \text{ lb.}$$

$$M = \frac{Pb}{2} = \frac{(50{,}000 \pm 2{,}150)(.47)}{2} = 11{,}700 \pm 505 \text{ in-lb.}$$

$$f_b = \frac{11{,}700 \pm 505}{.25} = 46{,}900 \pm 2{,}020 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{200}{200 - 46.9}(\pm 2{,}020) = \pm 2{,}640 \text{ p.s.i.}$$

TABLE 3

MOUNT TO BLADE TIP ATTACHMENT BOLTS FINITE LIFE ESTIMATION

(1)	(2)	(3)	(4)	(5)	(6)
$f_{a_{eq}}$	N-Cycles	% Occur.	Cycles/Hr.	Hour	% Occur/Hr.
	Figure 51	Page 94	Page 94	(2)/(4)	(3)/(5)
±48,500	1.012×10^5	91.0	4.0	2.53×10^4	.00359
±53,600	5.2×10^4	9.0	4.0	1.30×10^4	.00069
±2,640	∞	-	-	-	-
					Σ = .00428

Calculated life: $L_c = \frac{100}{.00428} = 23{,}400$ hours

Service life: S.L. = 1,250 + (.375)(23,400) = 10,000 hours

5.1.2.5 Engine-to-Aft-Mount Attachment Bolts

The aft mount is designed to react only the engine thrust loads. Therefore, the engine-to-aft-mount attachment bolts are investigated during the normal level flight condition.

The engine thrust is:

$$P_y = 1{,}500 \pm 1{,}850 \text{ lb. per engine}$$

$$P = \frac{11.25}{15.125}(1{,}500 \pm 1{,}850) = 1{,}115 \pm 1{,}375 \text{ lb.}$$

The equations used to determine the moment and stress level are obtained from page 37 of this report.

$$M = \frac{Pb}{2} = \frac{(1{,}115 \pm 1{,}375)(.4875)}{2} = 272 \pm 336 \text{ in-lb.}$$

$$f_b = \frac{272 \pm 336}{.024} = 11{,}300 \pm 14{,}000 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{145}{145 - 11.3}(\pm 14{,}000) = \pm 15{,}200 \text{ p.s.i.}$$

The endurance limit for 7AL-12ZR titanium alloy is conservatively assumed to be the same as that for Ti-8AL-1Mo-1V annealed titanium alloy.

$$F_{e\ell} = \pm 70{,}000 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 70{,}000}{\pm 15{,}200} = 4.67$$

5.1.2.6 Aft-Mount-to-Tip Attachment Bolts

The load per bolt due to engine thrust is:

$$P_x = 1{,}115 \pm 1{,}375 \text{ lb.}$$

The centrifugal force load due to the aft mount weight is obtained from page 11 of this report.

$$P_{CF} = 2{,}600 \text{ lb.}$$

The maximum load is:

$$P = \frac{2{,}600}{2} + (1{,}115 \pm 1{,}375) = 2{,}415 \pm 1{,}375 \text{ lb.}$$

The equations used to determine the moment and stress level are obtained from page 42 of this report.

$$M = \frac{Pb}{2} = \frac{(2415 \pm 1375)(.42)}{2} = 508 \pm 289 \text{ in-lb.}$$

$$f_b = \frac{508 \pm 289}{.024} = 21{,}200 \pm 12{,}100 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{145}{145 - 21.2} (\pm 12{,}100) = \pm 14{,}200 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 70{,}000}{\pm 14{,}200} = 4.9$$

5.1.3 Main Rotor Blade Tip and Attachments

The blade tip fatigue analysis consists of an investigation of the main mount to tip attachment lugs for shear tear-out, using the alternating centrifugal force loading developed for the start-stop condition during rotor overspeed operation. An investigation of the normal flight condition is provided to demonstrate that the fatigue stress developed is below the material endurance limit and nondamaging

$$\left.\begin{aligned} F_{tu} &= 130{,}000 \text{ p.s.i.} \\ F_{e\ell} &= \pm 70{,}000 \text{ p.s.i.} \end{aligned}\right\} \quad \text{(Ref. page 95)}$$

$$\text{Shear area: } A_s = 2.70 \text{ in}^2$$

The centrifugal force load during the rotor overspeed condition is obtained from page 101 of this report

$$C.F. = \pm 219{,}000 \text{ lb.}$$

$$P = \frac{\pm 219{,}000}{4} = \pm 54{,}800 \text{ lb.}$$

$$f_s = f_{a_{eq}} = \frac{\pm 54{,}800}{2.70} = \pm 20{,}300 \text{ p.s.i.}$$

The material shear endurance limit is conservatively taken as 57 percent of the tensile endurance limit.

$$F_{e\ell_s} = (.57)(\pm 70{,}000) = \pm 40{,}000 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 40{,}000}{\pm 20{,}300} = 1.97$$

The lug load during the normal flight condition is obtained from page 102 of this report.

$$P = 50{,}000 \pm 2150 \text{ lb.}$$

$$f_s = \frac{50{,}000 \pm 2150}{2.70} = 18{,}600 \pm 797 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{74}{74 - 18.6} (\pm 797) = \pm 1{,}070 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 40{,}000}{\pm 1{,}070} = 37.2$$

5.1.4 Main Rotor Blade Typical Section

The blade typical section fatigue analysis consists of an investigation of the following conditions:

Condition 1. Rotor overspeed condition
Condition 2. Forward flight, 1g, 144 m.p.h. fwd. velocity
Condition 3. Transient cyclic stick whirl

The condition 1 investigation consists of a start-stop analysis using the alternating centrifugal force loads developed during the rotor overspeed condition.

The condition 2 analysis investigates the alternating flapwise stress developed during normal operating conditions.

The condition 3 analysis investigates the alternating stress developed on the blade trailing edge during the cyclic stick whirl condition.

The section properties for the blade basic section (rotor station 170.00) are obtained from page 60 of this report.

Chordwise stiffness: $EI_C = 9.3 \times 10^{10}$ lb-in^2
Flapwise stiffness: $EI_F = 7.5 \times 10^{9}$ lb-in^2
Torsional stiffness: $GJ = 9.7 \times 10^{9}$ lb-in^2

Blade chord = 78 in.
Blade depth = (.15)(78) = 11.7 in.
Area = 16.92 in^2

5.1.4.1 Rotor Overspeed Condition

A start-stop investigation of the blade basic section at rotor station 170.00 is conducted for the alternating centrifugal force load developed during the rotor overspeed condition.

$$P = (1.05)^2(\pm 570{,}000) = \pm 629{,}000 \text{ lb.}$$

$$f_a = \frac{629{,}000}{16.92} = \pm 37{,}100 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 70{,}000}{\pm 37{,}100} = 1.89$$

5.1.4.2 Forward Flight, 1g, 144 m.p.h. Forward Velocity

During condition 2, the maximum alternating stress is due to flapwise bending and centrifugal force, the torsional moment being negligible.

The maximum alternating flapwise moment occurs at rotor station 437.00.

$$M_{F_a} = \pm 3.66 \times 10^5 \text{ in-lb.}$$
$$M_{F_s} = 2.0 \times 10^5 \text{ in-lb.}$$

The centrifugal force load is obtained from page 53 of Reference 2.

$$P = 4.3 \times 10^5 \text{ lb.}$$
$$M = 2.0 \times 10^5 \pm 3.66 \times 10^5 \text{ in-lb.}$$

The blade section properties at rotor station 437.00 are obtained from Reference 2.

Flapwise stiffness: $EI_F = 5.2 \times 10^9 \text{ in-lb}^2$

Area $\approx 12.00 \text{ in}^2$

$$f_t = \frac{(4.3)(10^5)}{12.00} = 35{,}800 \text{ p.s.i.}$$

$$f_b = \frac{[(2.0)(10^5) \pm (3.66)(10^5)](5.85)(18.5)(10^6)}{(5.2)(10^9)} = 4{,}200 \pm 7{,}600 \text{ p.s.i.}$$

$$f_{max} = 35{,}800 + (4{,}200 \pm 7{,}600) = 40{,}000 \pm 7.600 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{130}{130 - 40}(\pm 7{,}600) = \pm 11{,}000 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 70{,}000}{\pm 11{,}000} = 6.36$$

5.1.4.3 Transient Cyclic Stick Whirl

During condition 3, the maximum alternating stress is due to chordwise bending and centrifugal force.

The centrifugal force and alternating chordwise moment are obtained from page 62 of this report.

$$M_{C_a} = \pm 3.25 \times 10^6 \text{ in-lb.}$$
$$P = 5.38 \times 10^5 \text{ lb.}$$
$$f_t = \frac{(5.38)(10^5)}{16.92} = 31{,}800 \text{ p.s.i.}$$

$$f_b = \frac{(3.25)(10^6)(58.5)(18.5)(10^6)}{(9.3)(10^{10})} = \pm 37,800 \text{ p.s.i.}$$

$$f = 31,800 \pm 37,800 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{130}{130 - 31.8} (\pm 37,800) = \pm 50,000 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 70,000}{\pm 50,000} = 1.40$$

5.1.5 Main Rotor Blade Root-Retention Structure

The root-retention-structure fatigue analysis consists of an investigation of the tension-torsion strap and its retention pin and an analysis of the blade at rotor station 102.00.

5.1.5.1 Tension-Torsion Strap Assembly

An analysis is conducted below to determine the required number of wires per side to establish approximately a 10,000-hour service life based on alternating centrifugal force loads developed during the start-stop condition.

The wire material is AM 355 CRES stainless steel.

$$F_{tu} = 475,000 \text{ p.s.i.}$$

The S-N curve shape presented in Figure 52 on page 108 was obtained from the Bendix Corporation.

Wire diameter: $d = .006$ in.

$$A = \frac{\pi}{4} (.006)^2 = 2.83 \times 10^{-5} \text{ in}^2$$

Preliminary analysis has established the required number of wires to be 117,000 to approximate a 10,000-hour service life.

The centrifugal force load developed during normal rotor operation is obtained from page 105 of this report.

$$P = \pm 570,000 \text{ lb.}$$

$$\text{Load/wire} = \frac{\pm 570,000}{117,000} = \pm 4.87 \text{ lb.}$$

$$f_a = f_{a_{eq}} = \frac{\pm 4.87}{(2.83)(10^{-5})} = \pm 172,000 \text{ p.s.i.}$$

The rotor overspeed condition centrifugal force is obtained from page 105 of this report.

$$P = \pm 629{,}000 \text{ lb.}, \quad \text{load/wire} = \frac{\pm 629{,}000}{117{,}000} = \pm 5.37 \text{ lb.}$$

$$f_a = f_{a_{eq}} = \frac{\pm 5.37}{(2.83)(10^{-5})} = \pm 190{,}000 \text{ p.s.i.}$$

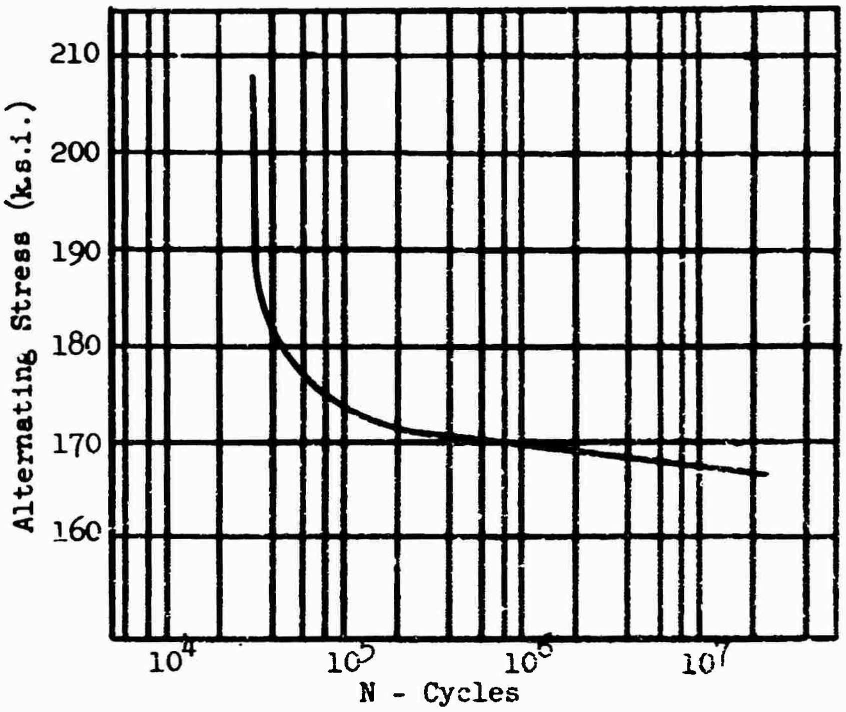


Figure 52. Tension-Torsion Strap S-N Data.

TABLE 4 TENSION-TORSION STRAP FINITE LIFE ESTIMATION					
(1)	(2)	(3)	(4)	(5)	(6)
$f_{a_{eq}}$	N-Cycles	% Occur.	Cycles/Hr.	Hour	% Occur/Hr.
	Figure 52	Page 94	Page 94	(2)/(4)	(3)/(5)
±172,000	1.35×10^5	91.0	4	3.38×10^4	.0027
±190,000	3.2×10^4	9.0	4	8.0×10^3	.00113
				Σ =	.00383

Calculated life: $L_c = \frac{100}{.00383} = 26{,}100$ hours

Service life: S.L. = 1250 + (.375)(26,100) = 11,000 hours.

5.1.5.2 Tension-Torsion Strap Retention Pin

A start-stop investigation of the retention pin is conducted to determine the required pin section properties to approximate a 10,000 hours service life when a fatigue notch factor of 2.0 is applied to the S-N data.

Material type: Ti-6AL-4V solution heat-treated titanium alloy.

$$F_{tu} = 162{,}000 \text{ p.s.i.}$$

$$F_{e} = \pm 130{,}000 \text{ p.s.i. (unnotched)}$$

The assumed bolt dimensions are presented below.

Outside diameter: $D = 6.50$ in.

Inside diameter at mid-span: $d_1 = 4.86$ in.

$$Z = \frac{\pi}{(32)(6.5)} = (6.5^4 - 4.86^4) = 18.52 \text{ in}^3 \text{ at mid-span}$$

The equations used to determine the moments and stress levels are obtained from pages 64 and 65 of this report. The normal rotor operating speed centrifugal force load is obtained from page 107 of this report.

$$P_{CF} = \pm 570{,}000 \text{ lb.}$$

$$M = \frac{\pm 570{,}000}{2}\left[(3.225) + (.5)(2.625)\right] = \pm 1{,}290{,}000 \text{ in-lb.}$$

$$f_a = f_{a_{eq}} = \frac{\pm 1{,}290{,}000}{18.52} = \pm 69{,}600 \text{ p.s.i.}$$

The rotor overspeed condition centrifugal force load is obtained from page 108 of this report.

$$P = \pm 629{,}000 \text{ lb.}$$

$$M = \frac{\pm 629{,}000}{2}\left[(3.225) + (.5)(2.625)\right] = \pm 1{,}420{,}000 \text{ in.-lb.}$$

$$f_a = f_{a_{eq}} = \frac{\pm 1{,}420{,}000}{18.52} = \pm 76{,}600 \text{ p.s.i.}$$

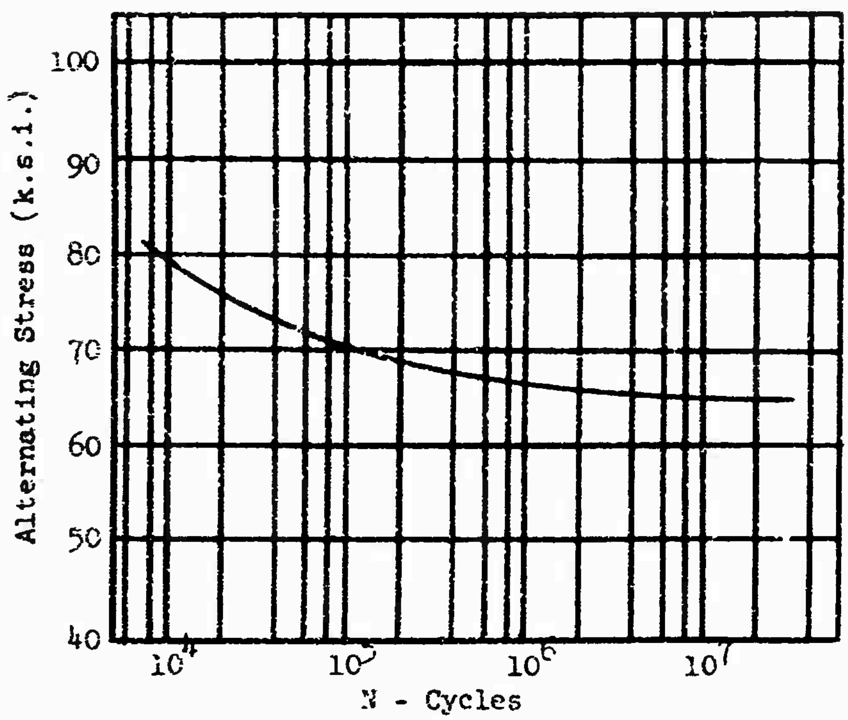


Figure 53. Ti-6AL-4V Titanium S-N Data.

TABLE 5
T-T STRAP RETENTION PIN FINITE LIFE ESTIMATION

(1)	(2)	(3)	(4)	(5)	(6)
$f_{a_{eq}}$	N-Cycles	% Occur.	Cycles/Hr.	Hour	% Occur/Hr.
	Figure 53	Page 94	Page 94	(2)/(4)	(3)/(5)
±69,600	1.82×10^5	91.0	4.0	4.55×10^4	.002
±76,600	1.95×10^4	9.0	4.0	4.88×10^3	.00184
				$\Sigma =$	.00384

Calculated life: $L_c = \frac{100}{.00384} = 26{,}000$ hours

Service life: S.L. = 1250 + (.375)(26,000) = 11,000 hours

5.1.5.3 Rotor Station 102.00 Analysis

Rotor station 102.00 carries flapwise and chordwise moments only as it is located inboard of the tension-torsion bar retention pin. The station 102.00 fatigue analysis consists of an investigation of the following two conditions:

Condition 1. Fwd. flight, 1g, 144 m.p.h. fwd. velocity
Condition 2. Transient cyclic stick whirl

Condition 1:

$$M_F = 7.0 \times 10^5 \pm 2.33 \times 10^5 \text{ in-lb. (Ref. Vol. IV)}$$

Flapwise $EI = 14.7 \times 10^9$ lb-in^2, $C_z = 5.15$ in. (Ref. page 66)

$$f_b = \frac{[(7.0)(10^5) \pm (2.33)(10^5)]\,(5.15)(18.5)(10^6)}{(14.7)(10^9)}$$

$$= 4{,}540 \pm 1{,}510 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{130}{130 - 4.5}(\pm 1{,}510) = \pm 1{,}565 \text{ p.s.i.}$$

The material endurance limit for Ti-8AL-1Mo-1V titanium alloy is obtained from page 94, and the available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 70{,}000}{\pm 1{,}565} = 44.8$$

Condition 2:

$$M_C = \pm 3.4 \times 10^6 \text{ in-lb. (Ref. page 67)}$$

Chordwise $EI = 9.85 \times 10^{10}$ lb-in^2, $C_y = 41.5$ in. (Ref. page 66)

$$f_a = f_{a_{eq}} = \frac{(\pm \text{[illegible]})(10^6)(41.5)(18.5)(10^6)}{(9.85)(10^{10})} = \pm 26{,}500 \text{ p.s.i.}$$

The available notch factor is:

$$K_{f_{av}} = \frac{\pm 70{,}000}{\pm 26{,}500} = 2.64$$

5.1.6 Stub Blade and Retention

The stub-blade fatigue analysis consists of an investigation of the stub-blade-to-hub attachment lugs and the adjustable-link-to-stub-blade attachment lugs.

5.1.6.1 Analysis of Stub-Blade-to-Hub Attachment Lugs

The stub-blade-to-hub attachment lugs are analyzed for the forward flight, 1g, 144 miles per hour forward velocity condition.

$$M_F = 9.5 \times 10^5 \pm 2.33 \times 10^5 \text{ lb-in. (Ref. Volume IV)}$$

$$P = \frac{(9.5)(10^5) \pm (2.33)(10^5)}{17.4} = 54{,}600 \pm 13{,}400 \text{ lb.}$$

$$f_t = \frac{54{,}600 \pm 13{,}400}{6.62} = 8{,}250 \pm 2{,}020 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{162}{162 - 8.3} (\pm 2{,}020) = \pm 2{,}130 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 130{,}000}{\pm 2{,}130} = 61.0$$

5.1.6.2 Adjustable-Link-to-Stub-Blade Attachment Lug

The adjustable link and the stub-blade-to-hub attachment pin react chordwise moments out as couples. The chordwise moment at rotor station 30.00 during the normal flight conditions is negligible, as the main rotor system is designed for the chordwise moment due to engine thrust at the blade tip to be reduced by the blade and engine nacelle drag such that the chordwise moment at the rotor system centerline is zero. However, a high alternating chordwise moment is experienced during the transient cyclic stick whirl condition.

$$P = \pm 117{,}000 \text{ lb.} \quad \text{(Ref. page 73)}$$

The available fatigue notch factor is established below for lug shear-out. The material shear endurance limit is conservatively taken as 57 percent of the tensile endurance limit.

$$F_{e\ell} = (.57)(\pm 130{,}000) = \pm 74{,}000 \text{ p.s.i.}$$

The shear area is obtained from page 73 of this report.

$$A_s = 3.45 \text{ in}^2$$

$$f_a = f_{a_{eq}} = \frac{\pm 117{,}000}{3.45} = \pm 34{,}000 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 74{,}000}{\pm 34{,}000} = 2.18$$

5.1.7 Main Rotor Hub Assembly

The hub-assembly fatigue analysis consists of an investigation of the blade retention pin, the retention pin lugs, and the adjustable link.

5.1.7.1 Rotor-Blade-Attachment-Pin Assembly

Preliminary fatigue analysis has established the rotor-blade-to-hub attachment pin to be critical during the start-stop condition. An analysis is conducted below to determine the required pin section properties to establish approximately a 10,000-hour service life when a fatigue notch factor of 2.0 is applied to the S-N data. An investigation of the normal flight condition is included to demonstrate that the fatigue stress is below the material endurance limit and nondamaging.

Material type: Ti-6AL-4V solution heat-treated titanium alloy.

$$F_{tu} = 162{,}000 \text{ p.s.i.}$$
$$F_{e\ell} = \pm 130{,}000 \text{ p.s.i. (unnotched)}$$

Preliminary fatigue analysis has established the following pin dimensions:

Outside diameter: D = 6.50 in.
Inside dia. at mid-span: d = 4.00 in.

$$Z = \frac{\pi}{(32)(6.5)}(6.5^4 - 4.00^4) = 23.1 \text{ in}^3$$

The equations used to determine the moments and stress levels are obtained from page 78 of this report. The normal rotor operating speed centrifugal force load is obtained from page 109 of this report.

$$P_{CF} = \pm 570{,}000 \text{ lb.}$$
$$M = \frac{\pm 570{,}000}{2}\left[(4.375) + (.5)(2.625)\right]$$
$$= \pm 1{,}620{,}000 \text{ in-lb.}$$
$$f_a = f_{a_{eq}} = \frac{\pm 1{,}620{,}000}{23.1} = \pm 70{,}000 \text{ p.s.i.}$$

The rotor overspeed condition centrifugal force loading is obtained from page 108 of this report.

$$P = \pm 629{,}000 \text{ lb.}$$
$$M = \frac{\pm 629{,}000}{2}\left[(4.375) + (.5)(2.625)\right] = \pm 1{,}790{,}000 \text{ in-lb.}$$
$$f_a = f_{a_{eq}} = \frac{\pm 1{,}790{,}000}{23.1} = \pm 77{,}500 \text{ p.s.i.}$$

The pin loading during the normal flight condition is obtained from Volume IV.

$$P_{CF} = 570{,}000 \text{ lb.}$$
$$M_F = 9.5 \times 10^5 \pm 2.32 \times 10^5 \text{ in-lb.}$$
$$T = -1.81 \times 10^5 \text{ lb-in.}$$

From a review of Figure 2, Volume III, the flapwise moment induces shear stress into the attachment pin, and the steady centrifugal force and torsion induce a steady bending moment into the pin. Therefore, the pin is analyzed for the alternating shear load due to flapwise bending developed during the normal flight condition.

$$P = \sqrt{\left[\frac{P_{CF}}{2} + \frac{M_F}{\ell}\right]^2 + \left[\frac{T}{\ell}\right]^2} \qquad \text{where: } \ell = 14.00 \text{ in.}$$

$$P = \sqrt{\left[\frac{(5.7)(10^5)}{2} + \frac{(9.5)(10^5) \pm (2.32)(10^5)}{14.00}\right]^2 + \left[\frac{(1.81)(10^5)}{14.00}\right]^2}$$

$$= \sqrt{(352{,}900 \pm 16{,}600)^2 + (12{,}900)^2} = 353{,}000 \pm 16{,}600 \text{ lb.}$$

$$A_s = \frac{\pi}{4}(6.9)^2 = 37.2 \text{ in}^2$$

$$f_s = \frac{353{,}000 \pm 16{,}600}{37.2} = 9{,}500 \pm 446 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{94}{94 - 9.5}(\pm 446) = \pm 495 \text{ p.s.i.}$$

The material shear endurance limit is conservatively taken as 57 percent of the tensile endurance limit. The material tensile endurance limit is obtained from Figure 53 of this report.

$$F_{e\ell_s} = (.57)(\pm 65{,}000) = \pm 37{,}000 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 37{,}000}{\pm 495} = 74.8$$

TABLE 6
BLADE ATTACHMENT PIN FINITE LIFE ESTIMATION

(1)	(2)	(3)	(4)	(5)	(6)
$f_{a_{eq}}$	N-Cycles	% Occur.	Cycles/Hr.	Hour	% Occur/Hr.
	Figure 53	Page 94	Page 94	(2)/(4)	(3)/(5)
±70,000	1.6×10^5	91.0	4.0	4.0×10^4	.00228
±77,500	1.5×10^4	9.0	4.0	3.75×10^3	.00240
±495	∞	-	-	-	-
					Σ = .00468

Calculated life: $L_c = \frac{100}{.00468} = 21,400$ hours

Service life: S.L. = 1,250 + (.375)(21,400) = 9,300 hours

5.1.7.2 Analysis of Blade-to-Hub Attachment Lug

The blade-to-hub attachment-lug analysis consists of an investigation of the alternating centrifugal force loading developed during the rotor overspeed operation for the start-stop condition and the normal flight condition to demonstrate that the equivalent alternating stress developed is below the material endurance limit and nondamaging.

The lug section properties are obtained from page 76 of this report.

Shear area: $A_s = 10.15 \text{ in}^2$

The centrifugal force loading during the rotor overspeed operation is obtained from page 108 of this report.

$$P_{CF} = \pm 629{,}000 \text{ lb.}$$

$$f_a = f_{a_{eq}} = \frac{\pm 629{,}000}{(2)(10.15)} = \pm 31{,}000 \text{ p.s.i.}$$

The material shear endurance limit is conservatively taken as 57 percent of tensile endurance limit.

$$F_{e\ell_s} = (.57)(\pm 130{,}000) = \pm 74{,}000 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 74\ 000}{\pm 31{,}000} = 2.38$$

The lug loading during the normal flight condition is obtained from page 114 of this report.

$$P = 353{,}000 \pm 16{,}600 \text{ lb.}$$

$$f_s = \frac{353{,}000 \pm 16{,}600}{10.15} = 34{,}800 \pm 1{,}640 \text{ p.s.i.}$$

$$f_{a_{eq}} = \frac{94}{94 - 34.8}(\pm 1{,}640) = \pm 2{,}600 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 74{,}000}{\pm 2{,}600} = 28.4$$

5.1.7.3 Adjustable-Link-Assembly Analysis

Per the discussion in Section 5.1.6.2, the maximum adjustable-link loading occurs during the transient cyclic stick whirl condition.

$$P = \pm 117{,}000 \text{ lb. (Ref. page 111)}$$

The lug shear area is obtained from page 81 of this report.

$$A_s = 6.28 \text{ in}^2$$

$$f_a = f_{a_{eq}} = \frac{\pm 117{,}000}{6.28} = \pm 18{,}600 \text{ p.s.i.}$$

The lug material is Ti-6AL-4V titanium alloy, and the shear endurance limit is conservatively taken as 57 percent of the tensile endurance limit.

$$F_{e\ell_s} = (.57)(\pm 130{,}000) = \pm 74{,}000 \text{ p.s.i.}$$

The available fatigue notch factor is:

$$K_{f_{av}} = \frac{\pm 74{,}000}{\pm 18{,}600} = 3.98$$

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13. ABSTRACT

Volume V of Heavy-Lift Tip Turbojet Rotor System discusses stress calculations for major structural components of this system. Tip engine attachment hardware is also analyzed.

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