TITLE **ENGINE MOUNT ANALYSIS REPORT**			REPORT NO. ER-01002	REV LTR. -	© 2003 All Rights Reserved
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REVISION LOG			
Rev.	Description	Approval	Date
-	Initial Release	M. F.	10/25/03

1.0 SCOPE

Purpose

The purpose of this report is to analyze the engine mount and attachment bolts to verify there is adequate strength for all operating conditions. The analysis will also optimize tubing sizes to minimize structural weight.

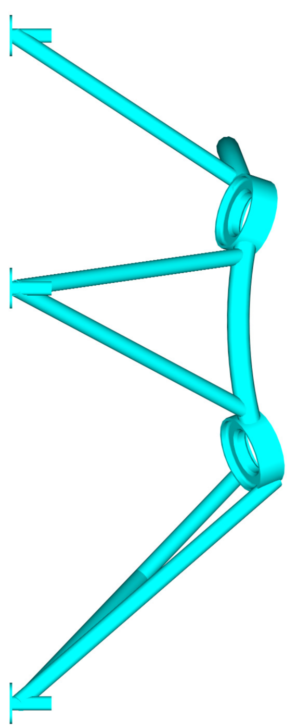
Discussion

Strength requirements for engine mounts are defined several places in FAR Part 23. Some requirements are not specific to engine mounts, but apply to all aircraft structure. Categories include: gyroscopic and aerodynamic loads, maneuvering limit loads, gust loads, torque loads, side loads and crash loads. All these conditions will be examined to identify critical load cases. Preliminary analysis shows that a five-point engine mount is the best configuration for mitigating crash loads. Supporting data is presented in *Appendix A - Crash Load Trade Study*.

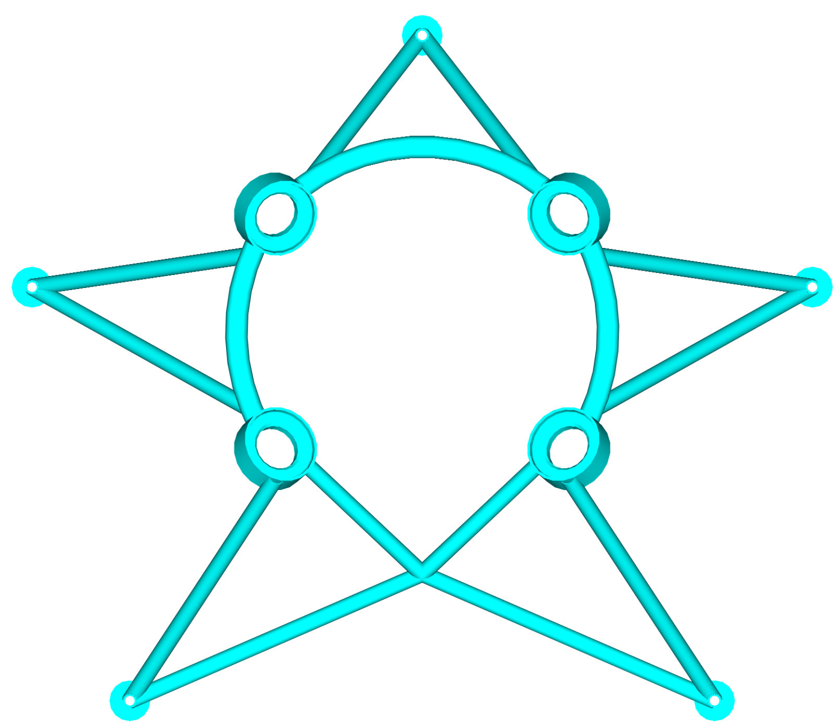
2.0 DESCRIPTION

Engine Mount

The engine mount is a conventional truss structure built up from .75” diameter 4130 steel tubing. On the engine side, truss members are welded to a Type 1 dynafocal ring with 3.5” diameter isolator cups. On the firewall side, truss members are welded to .625” diameter stub tubes at the five attach points. All attachment bolts are 7/16-20 UNF hardware per AN7 military specifications. Refer to the images below:



SIDE VIEW



FORWARD VIEW

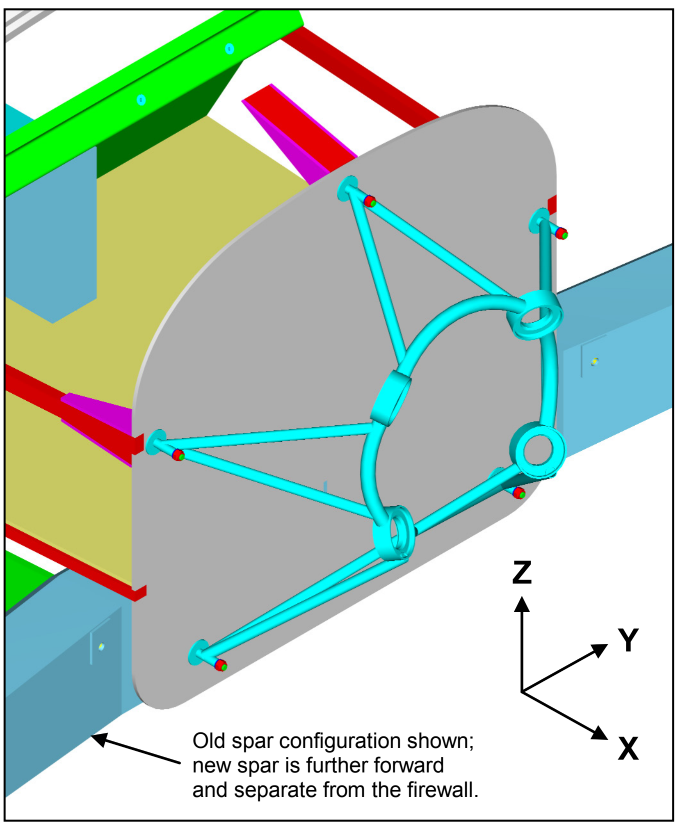
Support Structure

The engine mount is bolted to five aluminum hardpoints in the firewall bulkhead. Starting from the engine side, the firewall bulkhead is constructed from .016” thick stainless steel, one layer of .080” fiberfrax, and .25” thick aircraft grade plywood. The plywood is sealed with two plies of fiberglass/epoxy on each side prior to installing the fiberfrax/steel fire barrier. Aluminum hardpoints are embedded in the plywood core to distribute loads and prevent local crushing.

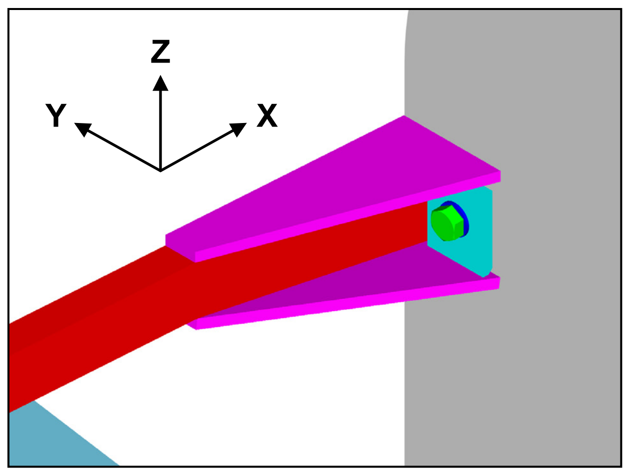
The top center engine mount bolt passes through the firewall bulkhead and attaches to a composite bathtub fitting. The bathtub fitting includes a .25” thick aluminum plate washer for load distribution. Ply reinforcements are added to distribute the loads into the turtleback skin.

The two outboard engine mount bolts pass through the firewall bulkhead and attach to composite bathtub fittings on the upper longerons. Each bathtub fitting includes ply reinforcements and a .25” thick aluminum plate washer for load distribution.

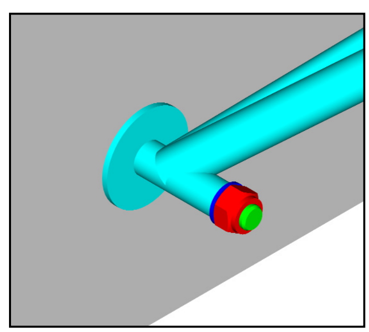
The two lower engine mount bolts pass through the firewall bulkhead and attach to composite bathtub fittings located at the corner of the firewall and the fuselage floor. There are ply reinforcements and .25” thick aluminum plate washers for load distribution. Refer to the images below:



Engine Mount Installation



Longeron Bathtub Fitting



Stub Tube Detail

3.0 STRUCTURE REQUIREMENTS

General

Strength requirements are specified in terms of limit loads and ultimate loads. FAR 23.301(a) states that ultimate loads are derived by multiplying the limit load by a factor of safety of 1.5 (from FAR 23.303).

FAR 23.305 requires all structure to support limit loads without detrimental or permanent deformation, and structures must withstand ultimate loads without failure for at least three seconds.

Load Factors

Maneuvering limit load factors per FAR 23.337 (see note 1)

	Limit	Ultimate (1.5 x limit)
Positive	**6.0**	**9.0**
Negative	**3.0**	**4.5**

Engine mount side load per FAR 23.363 (see note 2)

	Limit	Ultimate
Load factor	**1.33**	**2.0**

Emergency landing (crash) load factors per FAR 23.561 (see note 3)

	Limit	Ultimate
Upward	**-**	**3.0**
Forward	**-**	**18.0**
Sideward	**-**	**4.5**
Downward	**-**	**6.0**

Gust loads per FAR 23.333(c) and 23.341 (see note 4)

Engine torque per FAR 23.361 (see note 5)

Gyroscopic and aerodynamic loads per FAR 23.371 (see note 6)

Notes:

1) Acrobatic load factors are used for increased margin of safety; aircraft is not approved for aerobatics.
2) Crash loads dominate engine mount side load requirements.
3) FAR 23.561(b)(3) is specific to items of mass “within the cabin” that could injure occupants. The engine mount is not “in the cabin” so reduced load factors could be justified. Higher load factors are used to provide additional crash safety. See *Appendix A - Crash Load Trade Study.*
4) Preliminary calculations show that gust loads do not exceed limit load factors of +6 or –3 g’s.
5) Engine torque requirements are embedded in the analysis spreadsheet.
6) Gyroscopic and aerodynamic loads are embedded in the analysis spreadsheet.

Material Properties

All engine mount tubing is normalized 4130 alloy steel per MIL-T-6736.
Material properties for wall thickness under 0.187" from MIL-HDBK-5:

Ftu = 95 ksi, Fty = 75 ksi, Fcy = 75 ksi.

Knockdown for heat affected zones near welds, per MIL-HDBK-5:

Ftu = 80 ksi

The baseline configuration uses the following tubing sizes:

Dynafocal ring = 1.00" O.D. x .058 wall
Truss members = .75" O.D. x .049 wall
Stub tubes = .625" O.D. x .094 wall

4.0 LOAD CASE CALCULATIONS

Load Path Discussion

All loads exerted on the engine and propeller are distributed through the engine crankcase and reacted at the four crankcase mounting pads. Loads are transferred from the mounting pads to the dynafocal ring through elastomeric isolators. The dynafocal ring transfers loads to the engine mount truss structure. The truss structure reacts out the loads at five attach points on the firewall.

The engine is assumed to be a rigid body with inertial, thrust, torque and gyroscopic loads exerted at different locations on the crankcase. To simplify load input, only the focal point of the dynafocal cups is used for load application. All loads are resolved as forces and moments that can be applied at the focal point, now defined as the **load application point.** Spreadsheets are used to document the calculations. Measurement units are English (inches, pounds or inch-lbs) unless otherwise stated. The table below shows the engine parameters and component locations.

Engine Parameters										
Maximum Rotation Speed (rpm)	2800	(positive for pusher with conventional engine)								
Propeller Diameter (in)	66									
Stall Speed (kts)	55	(for calculating max thrust)								
Engine Max. Power (hp)	200									
Propeller Efficiency	0.85									
Max Thrust @ Stall (lb)	**1007**									
		X	**Y**	**Z**						
Load Application Point		53.21	0.00	6.81	Load Point is the Focal Point of the Dynafocal Ring Cups					
(From Aircraft CG)										
Part Location in Local Coordinates		(distance from load point)								
Part	**Weight**	**x**	**y**	**z**						
O-360 Engine	315.0	-5.06	0.00	-1.19						
Propeller	18.0	16.79	0.00	0.25						
Prop Ext / Crush Plate / Spinner	11.0	15.49	0.00	0.20						
Part in Aircraft CG Coordinates					Mass Moments of Inertia about the Part CG (slug-in^2)					
Part	**Weight**	**X**	**Y**	**Z**	**Ixx**	**Iyy**	**Izz**	**Ixy**	**Iyz**	**Izx**
O-360 Engine	315.0	48.15	0.00	5.62	551.00	490.00	786.00	-55.00	1.90	31.48
Propeller	18.0	70.00	0.00	7.06	203.11	0.00	0.00	0.00	0.00	0.00
Prop Ext / Crush Plate / Spinner	11.0	68.70	0.00	7.01	0.00	0.00	0.00	0.00	0.00	0.00
Total	**344.0**	**49.95**	**0.00**	**5.74**	**768.09**	**994.61**	**1276.62**	**-55.00**	**1.90**	**94.02**

Load Case Spreadsheet #1
Generating loads for application to the Finite Element Model is a critical step in the analysis process. Load cases must accommodate the FAR's plus any special requirements. The spreadsheet below documents the load case matrix used for this analysis.

	Roll	Pitch	Yaw	Forward	Sideways	Vertical	
	Angular Velocities (rad/s)			**Linear Accelerations (g)**			**% Thrust**
Load Case	**omega_x**	**omega_y**	**omega_z**	**a_X (g)**	**a_Y (g)**	**a_Z (g)**	**(0-1)**
1	0.00	**1.00**	**2.50**	0.00	0.00	**2.50**	**1**
2	0.00	0.00	**2.50**	0.00	0.00	**2.50**	**1**
3	0.00	**-1.00**	**2.50**	0.00	0.00	**2.50**	**1**
4	0.00	**1.00**	0.00	0.00	0.00	**2.50**	**1**
5	0.00	**-1.00**	0.00	0.00	0.00	**2.50**	**1**
6	0.00	**1.00**	**-2.50**	0.00	0.00	**2.50**	**1**
7	0.00	0.00	**-2.50**	0.00	0.00	**2.50**	**1**
8	0.00	**-1.00**	**-2.50**	0.00	0.00	**2.50**	**1**
9	0.00	**1.00**	**2.50**	0.00	0.00	**2.50**	0
10	0.00	0.00	**2.50**	0.00	0.00	**2.50**	0
11	0.00	**-1.00**	**2.50**	0.00	0.00	**2.50**	0
12	0.00	**1.00**	0.00	0.00	0.00	**2.50**	0
13	0.00	**-1.00**	0.00	0.00	0.00	**2.50**	0
14	0.00	**1.00**	**-2.50**	0.00	0.00	**2.50**	0
15	0.00	0.00	**-2.50**	0.00	0.00	**2.50**	0
16	0.00	**-1.00**	**-2.50**	0.00	0.00	**2.50**	0
17	0.00	0.00	0.00	0.00	0.00	**6.00**	**1**
18	0.00	0.00	0.00	0.00	0.00	**-3.00**	**1**
19	0.00	0.00	0.00	0.00	0.00	**6.00**	0
20	0.00	0.00	0.00	0.00	0.00	**-3.00**	0
21	0.00	0.00	0.00	0.00	0.00	**12.00**	**1**
22	0.00	0.00	0.00	0.00	0.00	**-6.00**	**1**
23	0.00	0.00	0.00	0.00	0.00	**12.00**	0
24	0.00	0.00	0.00	0.00	0.00	**-6.00**	0
25	0.00	0.00	0.00	0.00	**4.50**	0.00	**1**
26	0.00	0.00	0.00	0.00	**4.50**	0.00	0
27	0.00	0.00	0.00	**40.00**	0.00	0.00	**1**
28	0.00	0.00	0.00	**40.00**	0.00	0.00	0
29	0.00	0.00	0.00	**40.00**	0.00	**20.00**	**1**
30	0.00	0.00	0.00	**40.00**	0.00	**20.00**	0
31	0.00	0.00	0.00	0.00	**4.50**	**1.00**	**1**
32	0.00	0.00	0.00	0.00	**4.50**	**1.00**	0
33	0.00	0.00	0.00	**40.00**	0.00	**1.00**	**1**
34	0.00	0.00	0.00	**40.00**	0.00	**1.00**	0

Notes:

1) Load cases for thrust-on and thrust-off conditions are included. The thrust-on condition will produce torque loads required by FAR 23.361.
2) Load cases 1 thru 16 combine pitch and yaw velocities with 2.5 g's vertical load factor as required by FAR 23.371(b). Different combinations of pitch-up, pitch-down, yaw-left, yaw-right, thrust-on, thrust-off produce gyroscopic and aerodynamic loads per FAR 23.371.
4) Load cases 17 thru 20 are maneuvering limit load factors per FAR 23.337.
5) Load cases 21 thru 24 are ultimate load factors. A factor of safety of 2.0 was applied to the limit loads, which exceeds the requirements of FAR 23.303.
6) Load cases 25 and 26 are emergency landing (crash) sideward load factors. These are ultimate loads per FAR 23.561(b)(3).

7) Load cases 27 thru 30 are emergency landing load factors that greatly exceed FAR 23.561(b)(3). Refer to *Appendix A - Crash Load Trade Study.*
8) Load cases 31 thru 34 are the same as load cases 25 thru 28 except for 1 g vertical load factor.

Load Case Spreadsheet #2

The spreadsheet below calculates forces and moments for the conditions defined on spreadsheet #1. Inertial loads are exerted on the propulsion system center-of-gravity. Thrust loads are exerted along the crankshaft centerline. Gyroscopic and torque moments can be resolved anywhere on the crankcase body.

	Inertial Loads at Engine CG						Propeller Gyroscopic Moments			Thrust Load	Engine Torque
Load Case	**Fx**	**Fy**	**Fz**	**Mx**	**My**	**Mz**	**Mx**	**My**	**Mz**	**Fx**	**Mx**
1	**322.7**	**-12.8**	**-854.9**	**-59.6**	**143.5**	**-14.6**	0	**-12407**	**4963**	**-1007**	**-5364**
2	**278.2**	0.0	**-860.0**	**-1.0**	**49.0**	0.0	0	**-12407**	0	**-1007**	**-5364**
3	**322.7**	**12.8**	**-854.9**	**57.9**	**-45.5**	**23.8**	0	**-12407**	**-4963**	**-1007**	**-5364**
4	**44.5**	0.0	**-854.9**	**0.2**	0.0	**4.6**	0	0	**4963**	**-1007**	**-5364**
5	**44.5**	0.0	**-854.9**	**0.2**	0.0	**4.6**	0	0	**-4963**	**-1007**	**-5364**
6	**322.7**	**12.8**	**-854.9**	**57.9**	**-45.5**	**23.8**	0	**12407**	**4963**	**-1007**	**-5364**
7	**278.2**	0.0	**-860.0**	**-1.0**	**49.0**	0.0	0	**12407**	0	**-1007**	**-5364**
8	**322.7**	**-12.8**	**-854.9**	**-59.6**	**143.5**	**-14.6**	0	**12407**	**-4963**	**-1007**	**-5364**
9	**322.7**	**-12.8**	**-854.9**	**-59.6**	**143.5**	**-14.6**	0	**-12407**	**4963**	0	0
10	**278.2**	0.0	**-860.0**	**-1.0**	**49.0**	0.0	0	**-12407**	0	0	0
11	**322.7**	**12.8**	**-854.9**	**57.9**	**-45.5**	**23.8**	0	**-12407**	**-4963**	0	0
12	**44.5**	0.0	**-854.9**	**0.2**	0.0	**4.6**	0	0	**4963**	0	0
13	**44.5**	0.0	**-854.9**	**0.2**	0.0	**4.6**	0	0	**-4963**	0	0
14	**322.7**	**12.8**	**-854.9**	**57.9**	**-45.5**	**23.8**	0	**12407**	**4963**	0	0
15	**278.2**	0.0	**-860.0**	**-1.0**	**49.0**	0.0	0	**12407**	0	0	0
16	**322.7**	**-12.8**	**-854.9**	**-59.6**	**143.5**	**-14.6**	0	**12407**	**-4963**	0	0
17	0.0	0.0	**-2064.0**	0.0	0.0	0.0	0	0	0	**-1007**	**-5364**
18	0.0	0.0	**1032.0**	0.0	0.0	0.0	0	0	0	**-1007**	**-5364**
19	0.0	0.0	**-2064.0**	0.0	0.0	0.0	0	0	0	0	0
20	0.0	0.0	**1032.0**	0.0	0.0	0.0	0	0	0	0	0
21	0.0	0.0	**-4128.0**	0.0	0.0	0.0	0	0	0	**-1007**	**-5364**
22	0.0	0.0	**2064.0**	0.0	0.0	0.0	0	0	0	**-1007**	**-5364**
23	0.0	0.0	**-4128.0**	0.0	0.0	0.0	0	0	0	0	0
24	0.0	0.0	**2064.0**	0.0	0.0	0.0	0	0	0	0	0
25	0.0	**-1548.0**	0.0	0.0	0.0	0.0	0	0	0	**-1007**	**-5364**
26	0.0	**-1548.0**	0.0	0.0	0.0	0.0	0	0	0	0	0
27	**-13760.0**	0.0	0.0	0.0	0.0	0.0	0	0	0	**-1007**	**-5364**
28	**-13760.0**	0.0	0.0	0.0	0.0	0.0	0	0	0	0	0
29	**-13760.0**	0.0	**-6880.0**	0.0	0.0	0.0	0	0	0	**-1007**	**-5364**
30	**-13760.0**	0.0	**-6880.0**	0.0	0.0	0.0	0	0	0	0	0
31	0.0	**-1548.0**	**-344.0**	0.0	0.0	0.0	0	0	0	**-1007**	**-5364**
32	0.0	**-1548.0**	**-344.0**	0.0	0.0	0.0	0	0	0	0	0
33	**-13760.0**	0.0	**-344.0**	0.0	0.0	0.0	0	0	0	**-1007**	**-5364**
34	**-13760.0**	0.0	**-344.0**	0.0	0.0	0.0	0	0	0	0	0

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Load Case Spreadsheet #3

The spreadsheet below summarizes all forces and moments after loads are resolved at the load application point. This data is entered into the Finite Element Model to simulate loads exerted on the engine mount.

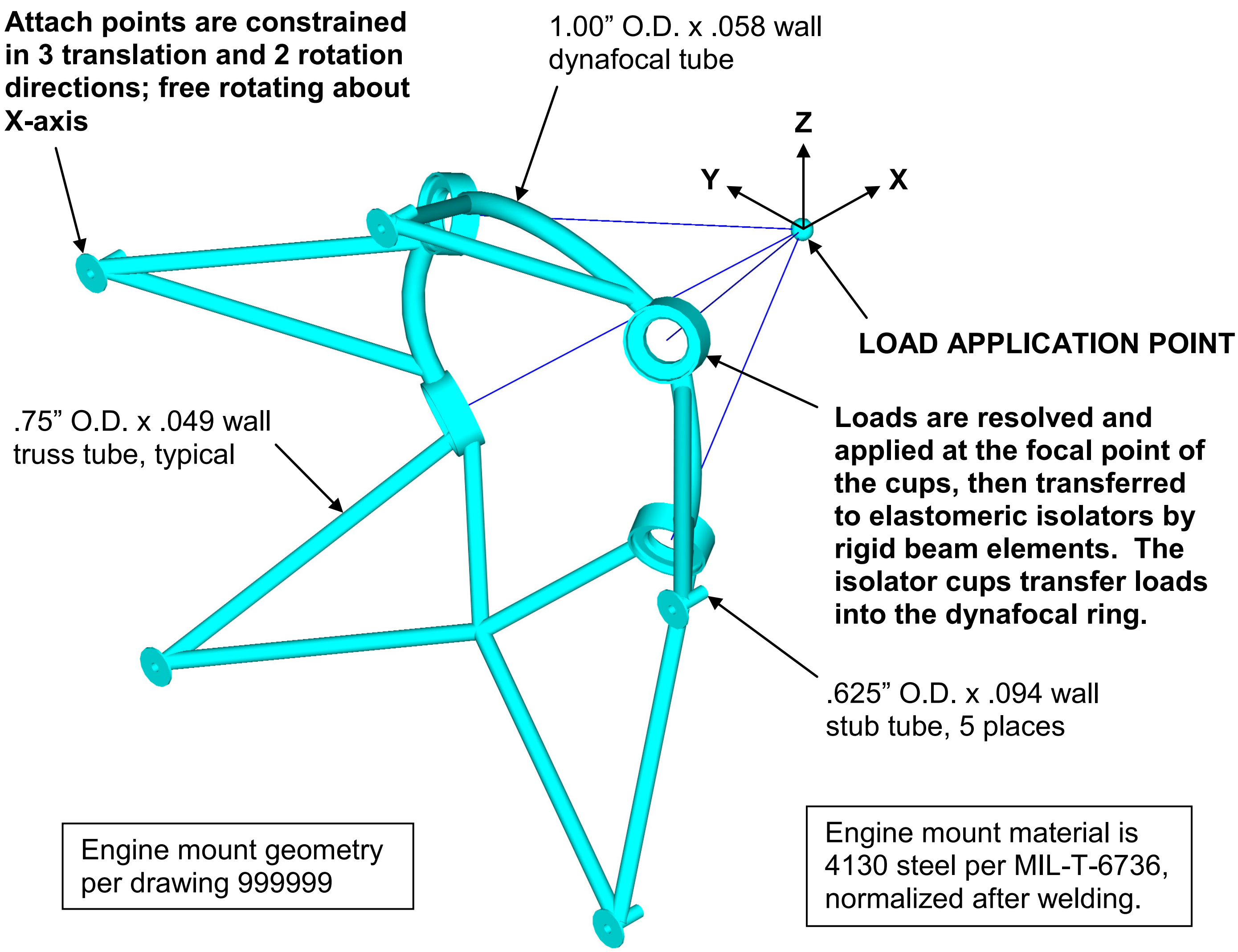
	Applied Loads at Load Point					
Load Case	Fx	Fy	Fz	Mx	My	Mz
1	-684.5	-12.8	-854.9	-5437	-14318	4990
2	-729.0	0.0	-860.0	-5365	-14381	0
3	-684.5	12.8	-854.9	-5292	-14506	-4981
4	-962.7	0.0	-854.9	-5364	-1755	4967
5	-962.7	0.0	-854.9	-5364	-1755	-4958
6	-684.5	12.8	-854.9	-5292	10308	4945
7	-729.0	0.0	-860.0	-5365	10433	0
8	-684.5	-12.8	-854.9	-5437	10497	-4936
9	322.7	-12.8	-854.9	-73	-15398	4990
10	278.2	0.0	-860.0	-1	-15462	0
11	322.7	12.8	-854.9	72	-15587	-4981
12	44.5	0.0	-854.9	0	-2836	4967
13	44.5	0.0	-854.9	0	-2836	-4958
14	322.7	12.8	-854.9	72	9227	4945
15	278.2	0.0	-860.0	-1	9353	0
16	322.7	-12.8	-854.9	-73	9416	-4936
17	-1007.2	0.0	-2064.0	-5364	-5651	0
18	-1007.2	0.0	1032.0	-5364	4447	0
19	0.0	0.0	-2064.0	0	-6732	0
20	0.0	0.0	1032.0	0	3366	0
21	-1007.2	0.0	-4128.0	-5364	-12383	0
22	-1007.2	0.0	2064.0	-5364	7813	0
23	0.0	0.0	-4128.0	0	-13464	0
24	0.0	0.0	2064.0	0	6732	0
25	-1007.2	-1548.0	0.0	-7025	1081	5049
26	0.0	-1548.0	0.0	-1661	0	5049
27	-14767.2	0.0	0.0	-5364	15845	0
28	-13760.0	0.0	0.0	0	14764	0
29	-14767.2	0.0	-6880.0	-5364	-6595	0
30	-13760.0	0.0	-6880.0	0	-7676	0
31	-1007.2	-1548.0	-344.0	-7025	-41	5049
32	0.0	-1548.0	-344.0	-1661	-1122	5049
33	-14767.2	0.0	-344.0	-5364	14723	0
34	-13760.0	0.0	-344.0	0	13642	0

5.0 FINITE ELEMENT ANALYSIS

General

The engine mount was modeled in Cosmos-M FEA software. The engine crank-case was constructed from rigid beam elements that transfer loads from the load application point to the elastomeric isolators. The isolators were made from beam elements that mimic the axial and rotational stiffness of the isolator assembly. The isolators were connected to cup elements on the dynafocal ring using short beam elements to evenly distribute loads. The isolators attenuate impulse loads and reduce engine mount vibration that causes fatigue.

Forces and moments were applied using values from load case spreadsheet #3. Stress and buckling analysis results were recorded for each load case. The model appeared to provide accurate results based on the analyst's past experience.



FEA Results

The spreadsheet below summarizes FEA results and calculates margins of safety. Margin values should be 0 or greater. Negative margins indicate the part does not meet the specified factor of safety or is not strong enough for that load condition.

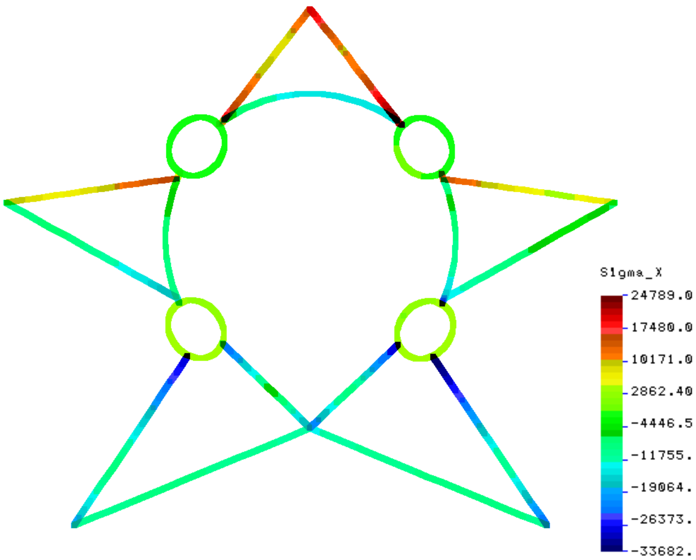
Load Case	Load Type	Max Tension Stress (psi)	Max Comp. Stress (psi)	FEM Buckling Factor	Stress M.S.	Buckling M.S.
1	Limit	23,476	28,731	6.024	0.74	2.01
2	Limit	24,081	27,004	32.171	0.85	15.09
3	Limit	24,817	25,126	5.684	0.99	1.84
4	Limit	8,929	17,918	7.757	1.79	2.88
5	Limit	14,519	15,948	7.539	2.14	2.77
6	Limit	25,082	23,458	6.340	0.99	2.17
7	Limit	23,060	25,101	5.774	0.99	1.89
8	Limit	21,202	26,318	6.260	0.90	2.13
9	Limit	20,867	20,851	6.005	1.40	2.00
10	Limit	20,424	19,124	30.732	1.45	14.37
11	Limit	21,161	20,997	5.868	1.36	1.93
12	Limit	13,418	12,627	22.077	2.73	10.04
13	Limit	13,417	12,604	22.071	2.73	10.04
14	Limit	20,550	19,599	10.248	1.43	4.12
15	Limit	18,528	18,626	9.490	1.68	3.75
16	Limit	20,704	19,843	10.147	1.41	4.07
17	Limit	24,789	33,682	4.613	0.48	1.31
18	Limit	14,104	20,212	4.982	1.47	1.49
19	Limit	25,144	28,390	7.167	0.76	2.58
20	Limit	14,195	12,572	11.986	2.52	4.99
21	Ultimate	46,290	59,460	2.876	0.26	1.88
22	Ultimate	28,275	32,780	4.016	1.29	3.02
23	Ultimate	50,289	56,780	3.583	0.32	2.58
24	Ultimate	28,390	25,144	5.993	1.64	4.99
25	Crash	18,930	24,433	5.868	2.07	4.87
26	Crash	18,889	18,905	9.641	2.97	8.64
27	Crash	30,570	62,717	1.525	0.20	0.53
28	Crash	28,482	56,242	1.637	0.33	0.64
29	Crash	33,362	136,540	0.993	**-0.45**	**-0.01**
30	Crash	29,979	131,250	1.042	**-0.43**	0.04
31	Crash	21,111	22,249	6.849	2.37	5.85
32	Crash	21,070	16,720	11.916	2.56	10.92
33	Crash	27,743	64,731	1.481	0.16	0.48
34	Crash	25,656	58,256	1.586	0.29	0.59

Note 2 applies to shaded area

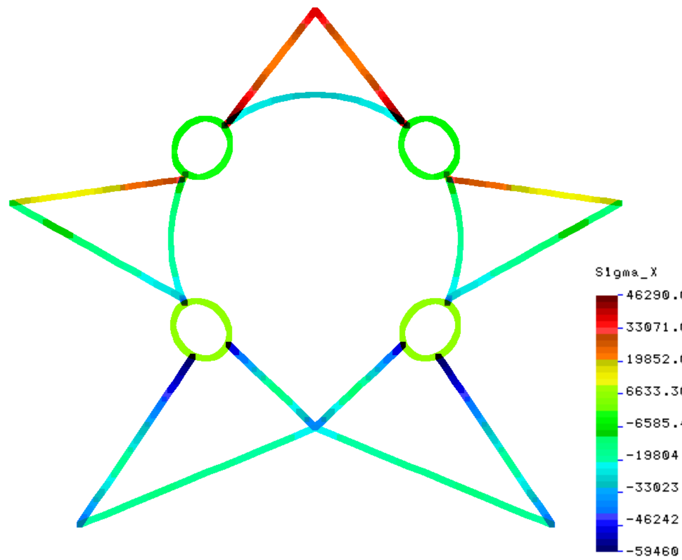
Notes:

1) Margins for limit loads (load cases 1-20) include factors of safety shown in the following equations:
   Stress M.S. = 75,000 / (max stress X 1.5 factor of safety) – 1
   Buckling M.S. = FEM buckling factor / 2.0 factor of safety – 1
2) Margins for ultimate loads (load cases 21-34) do not include factors of safety, as calculated below:
   Stress M.S. = 75,000 / max stress – 1
   Buckling M.S. = FEM buckling factor – 1

Flight Loads Stress Review
Results for load cases 1-16 (gyroscopic and aerodynamic loads) are relatively benign. Load cases 17-24 (maneuvering loads) show higher stress levels, but all margins are positive. Of most interest to pilots are load cases 17 (+6 g limit load) and 21 (+12 g ultimate load). Graphical results for these two are presented below.



Stress: Load Case 17
+6 G Limit Load



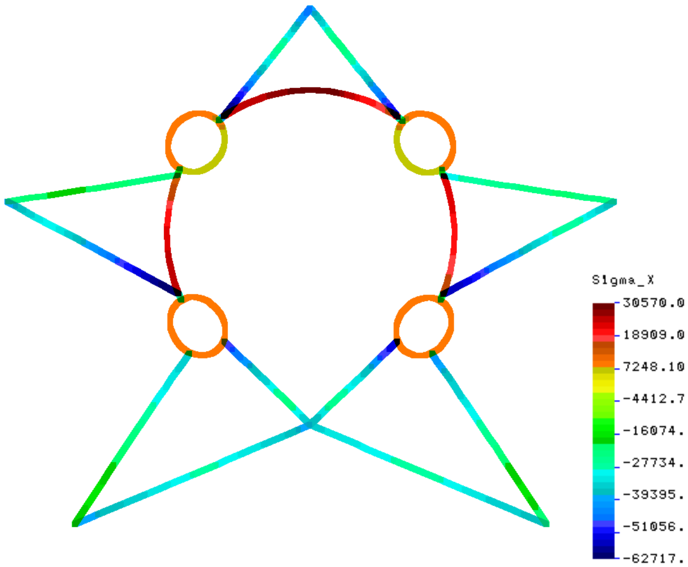
Stress: Load Case 21
+12 G Ultimate Load

Colors represent the following: The red spectrum depicts areas dominated by tension stress and the blue spectrum depicts areas dominated by compression stress. Yellow and green indicate areas with relatively low stress.

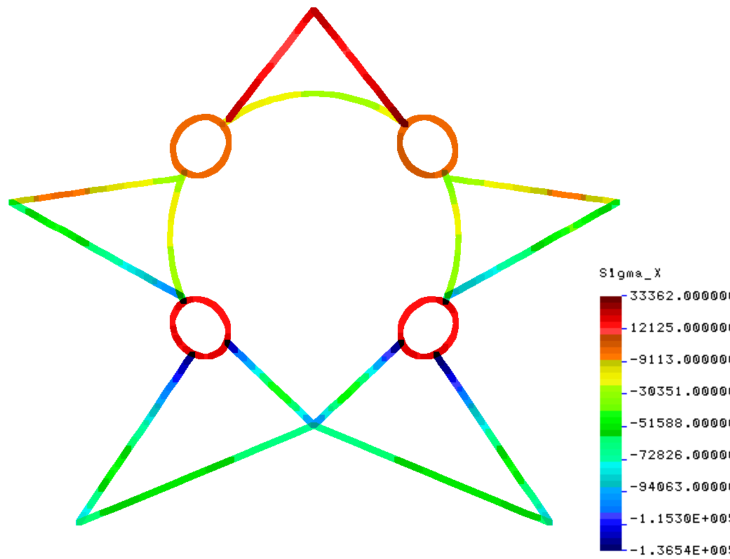
As expected for positive load factors, the upper truss members carry tension loads and the lower truss members are in compression. The areas of highest stress occur where the truss tubing attaches to the isolator cups. However, stress remains below the material yield point even for the 12 g condition. Because the engine mount was sized for crash loads, it appears to exceed FAR requirements for flight conditions.

Crash Loads Stress Review
Margins for loads cases 25-34 (crash loads) are positive except for load cases 29 and 30. Both cases represent crash loads of 40 g's forward and 20 g's downward, with stress for load case 29 slightly higher due to the thrust-on condition. Graphical results for load case 29 are shown on the next page along with load case 27.

Load case 27 is of interest because it depicts a 40 g forward deceleration. This crash condition drove the design of the engine mount and was the primary reason for performing the *Crash Load Trade Study.*



Stress: Load Case 27
40 G Forward Crash



Stress: Load Case 29
40 G Fwd, 20 G Downward

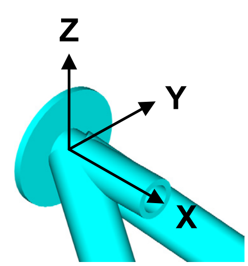
For load case 27, there are high compression loads in the truss tubes and stress levels are nearing the yield point of 75 ksi. The engine mount appears to survive 40 g forward deceleration loads, thus protecting occupants from the engine.

For load case 29, truss tubes supporting the lower dynafocal cups have four times the stress as the upper cup support tubes. This indicates the lower tubes will fail or the lower cups will tear out well before the upper tubes. Since the engine is cantilevered off the engine mount, this creates an aft-end-down moment for the engine. This is desirable because the aft end of the engine is directed towards the ground as the engine rotates about the upper attach points. Striking the ground is a good way to disperse the engine's kinetic energy.

Results so far indicate the dynafocal ring and truss structure meet or exceed all design requirements. To complete the analysis, the engine mount attachment bolts must be checked for adequate strength. Only the firewall bolts will be examined; the bolts attaching the engine to the dynafocal ring are specified by the engine manufacturer and are used throughout the aircraft industry.

Reaction Forces

The FEA output file includes reaction forces (Fx, Fy, Fz) and moments (Mx, My, Mz) for each mounting point. The local axis system used at each mounting point is shown at right. Forces Fy and Fz create shear loads on the bolt, while +Fx results in compression loads on the firewall and –Fx causes bolt tension loads (descriptions of +Fx and –Fx may seem reversed because reaction vectors are the opposite direction of load vectors). Moments My and Mz result in bolt bending loads, while Mx attempts rotation about the bolt centerline.



Local Axis for Top Center Stub Tube

The fixed stub tube configuration is designed to minimize bending stress created by shear loads. With the truss cluster acting upon any bolt, the true bending moment is difficult to quantify. The stress will not be significant if the cluster weldment is located as close as practical to the stub tube washer. In other words, the truss tube centerlines (line-of-action) should intersect as close as possible to the firewall.

After reviewing the FEA output data, four load cases were identified as having the highest tension, shear and moment loads, or combination thereof. The reaction forces for these load cases are presented below. Force units are pounds and moments are inch-lbs.

REACTION FORCE FOR LOAD CASE 23: ***ULTIMATE +12 G's***

								----- Applied Loads -----		
								Tension	**Shear**	**Moment**
Node Name	**Node No.**	**Fx**	**Fy**	**Fz**	**Mx**	**My**	**Mz**	**Fx**	**Fyz**	**Myz**
Top Center	1275	-2624	0	1790	0	-483	0	2,624	1,790	483
Right Longeron	21	-263	194	-398	0	58	-204	263	443	212
Left Longeron	84	-263	-193	-398	0	58	204	263	443	212
Bottom Left	1102	1575	1974	1567	0	-298	236	0	2,520	380
Bottom Right	1146	1575	-1974	1568	0	-297	-236	0	2,521	379

REACTION FORCE FOR LOAD CASE 24: ***ULTIMATE NEG 6 G's***

								Tension	**Shear**	**Moment**
Node Name	**Node No.**	**Fx**	**Fy**	**Fz**	**Mx**	**My**	**Mz**	**Fx**	**Fyz**	**Myz**
Top Center	1275	1312	0	-895	0	241	0	0	895	241
Right Longeron	21	132	-97	199	0	-29	102	0	221	106
Left Longeron	84	131	97	199	0	-29	-102	0	221	106
Bottom Left	1102	-787	-987	-784	0	149	-118	787	1,260	190
Bottom Right	1146	-788	987	-784	0	149	118	788	1,260	190

REACTION FORCE FOR LOAD CASE 29: ***CRASH 40 G's FWD, 20 G's DOWN***

								Tension	**Shear**	**Moment**
Node Name	**Node No.**	**Fx**	**Fy**	**Fz**	**Mx**	**My**	**Mz**	**Fx**	**Fyz**	**Myz**
Top Center	1275	-740	-96	524	0	-325	-40	740	532	327
Right Longeron	21	2949	-3050	-1460	0	270	-699	0	3,381	749
Left Longeron	84	2972	3071	-1607	0	307	710	0	3,466	774
Bottom Left	1102	4781	6593	4697	0	-426	252	0	8,095	495
Bottom Right	1146	4806	-6518	4726	0	-481	-281	0	8,051	557

REACTION FORCE FOR LOAD CASE 33: ***CRASH 40 G's FWD*** *(thrust on)*

								Tension	**Shear**	**Moment**
Node Name	**Node No.**	**Fx**	**Fy**	**Fz**	**Mx**	**My**	**Mz**	**Fx**	**Fyz**	**Myz**
Top Center	1275	3414	-96	-2311	0	439	-39	0	2,313	441
Right Longeron	21	3365	-3357	-830	0	179	-376	0	3,458	416
Left Longeron	84	3388	3377	-976	0	216	386	0	3,515	442
Bottom Left	1102	2288	3467	2216	0	45	-122	0	4,115	130
Bottom Right	1146	2313	-3392	2244	0	-11	93	0	4,067	94

Since reaction vectors are the opposite direction of load vectors, bolt tension loads are indicated by negative Fx values. Positive Fx values can be ignored because they represent compression loads that do not stress the bolts. Fy and Fz may be combined into a single shear value using the equation below. Likewise, My and Mz may be combined into one moment using the second equation.

$$\textbf{Shear, Fyz} = \sqrt{(Fy^2 + Fz^2)} \qquad \textbf{Moment, Myz} = \sqrt{(My^2 + Mz^2)}$$

Maximum tension, shear and moment loads are calculated and shown in the three right columns above. These loads occur simultaneously for the conditions shown.

Each load creates a simple stress that can be accurately determined. But the combined loads are more difficult to analyze because the ultimate allowable for tension, shear and bending are different. Another complication is that loads interact differently. For example, shear and bending stresses don't normally interact to significantly reduce a bolt's strength from that which would result when considering the stresses individually, whereas tension and bending stresses combine directly. Shear and tension loads also interact, but not as directly as tension and bending.

Combined Loads Analysis
The most practical method for determining stress conditions of combined loads is to use stress ratios and interaction equations. Stress ratios denote the ratio of applied stress (or load) to the corresponding allowable stress (or load) for each load type. Interaction equations are based on theoretical analysis and empirical tests that determine the stress state for different combinations of loads.

When performing calculations based on limit loads, stress ratios should include factors of safety applied to the design (limit) stress prior to dividing by the ultimate allowable. Since the four load cases being examined are for ultimate and crash loads, factors of safety are not required.

Interaction equations are generally expressed in the form:

$$R_a^x + R_b^y = R \leq 1$$

where R_a and R_b are stress ratios for corresponding loads, and x and y are exponents with values that depend upon the types of interacting stresses. As long as the resultant value is less than 1, there is a positive margin. Interaction equations are fully explained in the book ***Analysis and Design of Flight Vehicle Structures*** by E. F. Bruhn.

Margin of safety equations can be derived from interaction equations. They are generally expressed in the form: **M.S. = 1 / R – 1**, where R is the interaction equation. These equations are from the ***Northrop Grumman Structures Manual:***

For Tension and Shear combined: $$\mathbf{M.S.} = \frac{1}{\sqrt{Rt^2 + Rs^2}} - 1$$

For Tension and Bending combined: $$\mathbf{M.S.} = \frac{1}{Rt + Rb} - 1$$

For Tension, Bending and Shear combined: $$\mathbf{M.S.} = \frac{1}{\sqrt{(Rt + Rb)^2 + Rs^2}} - 1$$

The bolt analysis can now be completed using these equations. The spreadsheet on page 14 calculates simple stress for tension, shear and bending (ftu, fsu, fbu) using the applied loads from page 12. Stress ratios Rt, Rs and Rb are then calculated. Margins of safety are calculated and shown in the three right columns. Section properties, stress equations and variable names are shown above the spreadsheet border.

Bolt Shank Minimum Dia =	**0.433**				
Shank Area, A =	0.147254	Tension Stress, ftu =	Fx/A	Ult. Tensile, Ftu =	**125,000**
Moment 0f Inertia, I =	0.001726	Shear Stress, fsu =	Fyz/A	Ult. Shear, Fsu =	**75,000**
Section Modulus, Z =	0.00797	Bending Stress, fbu =	Myz/Z	Ult. Bending, Fbu =	**180,000**

Variable Names:	ftu	fsu	fbu	Rt	Rs	Rb	MS (t+s)	MS (t+b)	MS (t+b+s)
LOAD CASE 23: **Node Name**	**Tension Stress (psi)**	**Shear Stress (psi)**	**Bending Stress (psi)**	**ftu/Ftu stress ratio**	**fsu/Fsu stress ratio**	**fbu/Fbu stress ratio**	**Margin for ftu + fsu combined**	**Margin for ftu + fbu combined**	**Margin for ftu+fbu+fsu combined**
Top Center	17,820	12,156	60,602	0.14256	0.16208	0.33668	3.63	1.09	0.98
Right Longeron	1,786	3,008	26,610	0.01429	0.04010	0.14783	22.49	5.17	4.99
Left Longeron	1,785	3,007	26,610	0.01428	0.04010	0.14783	22.50	5.17	4.99
Bottom Left	0	17,116	47,695	0	0.22821	0.26497	3.38	2.77	1.86
Bottom Right	0	17,120	47,596	0	0.22827	0.26442	3.38	2.78	1.86
LOAD CASE 24: **Node Name**	**Tension Stress**	**Shear Stress**	**Bending Stress**	**ftu/Ftu ratio**	**fsu/Fsu ratio**	**fbu/Fbu ratio**	**Margin for ftu + fsu**	**Margin for ftu + fbu**	**Margin for ftu+fbu+fsu**
Top Center	0	6,078	30,238	0	0.08104	0.16799	11.34	4.95	4.36
Right Longeron	0	1,504	13,305	0	0.02005	0.07392	48.88	12.53	12.06
Left Longeron	0	1,504	13,305	0	0.02005	0.07392	48.88	12.53	12.06
Bottom Left	5,347	8,558	23,847	0.04277	0.11410	0.13249	7.21	4.71	3.78
Bottom Right	5,348	8,560	23,847	0.04278	0.11413	0.13249	7.20	4.71	3.78
LOAD CASE 29: **Node Name**	**Tension Stress**	**Shear Stress**	**Bending Stress**	**ftu/Ftu ratio**	**fsu/Fsu ratio**	**fbu/Fbu ratio**	**Margin for ftu + fsu**	**Margin for ftu + fbu**	**Margin for ftu+fbu+fsu**
Top Center	5,027	3,615	41,085	0.04021	0.04820	0.22825	14.93	2.72	2.67
Right Longeron	0	22,963	94,018	0	0.30618	0.52232	2.27	0.91	0.65
Left Longeron	0	23,538	97,054	0	0.31384	0.53919	2.19	0.85	0.60
Bottom Left	0	54,973	62,101	0	0.73298	0.34501	0.36	1.90	0.23
Bottom Right	0	54,675	69,894	0	0.72900	0.38830	0.37	1.58	0.21
LOAD CASE 33: **Node Name**	**Tension Stress**	**Shear Stress**	**Bending Stress**	**ftu/Ftu ratio**	**fsu/Fsu ratio**	**fbu/Fbu ratio**	**Margin for ftu + fsu**	**Margin for ftu + fbu**	**Margin for ftu+fbu+fsu**
Top Center	0	15,708	55,298	0	0.20943	0.30721	3.77	2.26	1.69
Right Longeron	0	23,484	52,250	0	0.31312	0.29028	2.19	2.45	1.34
Left Longeron	0	23,872	55,498	0	0.31829	0.30832	2.14	2.24	1.26
Bottom Left	0	27,943	16,315	0	0.37257	0.09064	1.68	10.03	1.61
Bottom Right	0	27,620	11,750	0	0.36826	0.06528	1.72	14.32	1.67

Margins are positive even for crash loads. The last column (ftu + fbu + fsu) has margins for the worst-case combination of loads. This approach is conservative because shear and bending stresses don't usually peak at the same location.

Conclusion

The attachment bolts have positive margins of safety for all flight and crash loads. Earlier analysis showed the dynafocal ring and the truss structure also meet or exceed design requirements. Because the engine mount was sized for 40 g crash loads, it exceeds FAR requirements under all operating conditions.

This document is not complete without the "Disclaimers and Disclosures" attachment. Such statement will inform the reader of methods, limitations, exclusions and waivers that apply to this report.

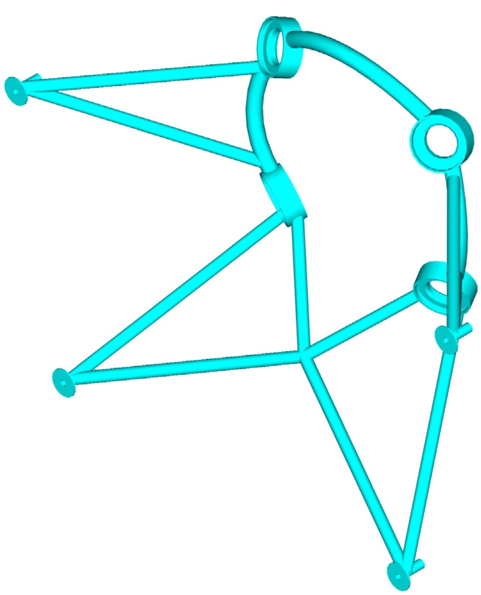
>> END OF REPORT <<

APPENDIX "A"

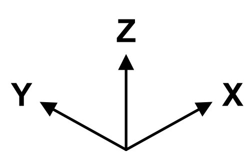
Crash Load Trade Study
With the engine mounted behind the occupants, some means must be taken to ensure the engine does not penetrate the seatback bulkhead during otherwise survivable accidents. Four possible solutions were identified:

1) The Long-EZ and other mid-wing canards place the center spar directly forward of the engine. The center spar is a major structural component that should prevent intrusion of the engine into the passenger area.
2) Design the engine mount and support structure to withstand 40 g crash loads along the longitudinal axis. This requirement exceeds the 18 g forward deceleration specified by FAR 23.561(b)(3), but there's no point in having 40 g seatbelt restraints if the occupants will be crushed at 18 g's.
3) Control engine deceleration by incorporating energy absorbing features into the engine mount and/or cabin structure. This approach would limit forward displacement of the engine by using deformable structure to dissipate the kinetic energy.
4) Design the engine mount to fail asymmetrically, thereby controlling the engine's trajectory. By allowing the engine to pitch forward and downward, it can be directed towards the baggage floor instead of the seatback.

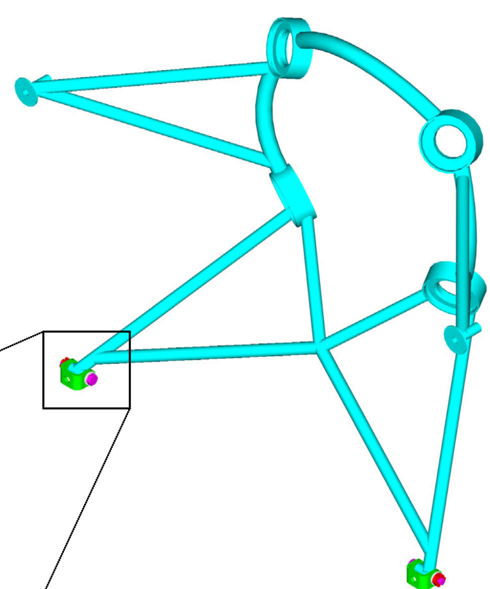
Option 1 can be eliminated because this option does not apply to low wing configurations like the Apollo. Option 3 requires extensive analysis and crash testing to ascertain actual performance, which would exceed the financial resources of this program. Options 2 and 4 appear to be viable, so they will be analyzed and compared. Their basic configurations are shown below.



Option 2
Fixed Engine Mount



Axis Orientation
for Both Views



Option 4
Hinged Engine Mount

Option 2 is a conventional fixed engine mount with four attach points. Option 4 is identical except the two lower stub tubes are rotated 90˚ and bolted to hinge brackets that allow the engine mount to pivot. During severe forward decelerations (crash loads), the upper truss members will fail and the hinged truss members will cause the engine to pitch downward as it moves forward. The kinetic energy will be dissipated as the engine tears out the cowling, firewall and turtleback structure.

Material Properties

Tubing for both configurations is normalized 4130 alloy steel per MIL-T-6736.
Material properties for wall thickness under 0.187" from MIL-HDBK-5:

Ftu = 95 ksi, Fty = 75 ksi, Fcy = 75 ksi.

Knockdown for heat affected zones near welds, per MIL-HDBK-5:

Ftu = 80 ksi

Both options use the following tubing sizes:

Dynafocal ring = .875" O.D. x .065 wall
Truss members = .75" O.D. x .049 wall
Stub tubes = .625" O.D. x .094 wall

Load Conditions

Four critical load cases were selected from the analysis spreadsheet to be used for this trade study:

Load case 17, maneuvering limit load, 6 g's positive
Load case 18, maneuvering limit load, 3 g's negative
Load case 27, crash load, 40 g's forward
Load case 28, crash load, 40 g's forward, 20 g's downward

After completing the trade study and selecting a baseline configuration, the engine mount will analyzed for all 34 load cases. Refer to the full analysis for more details.

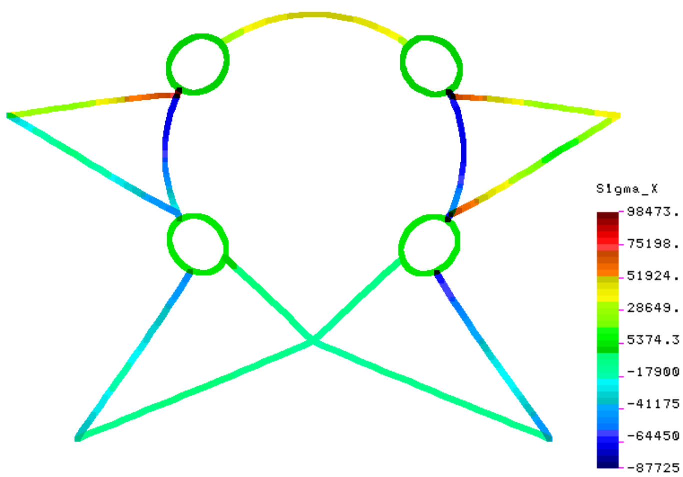
FEA Results for FIXED Engine Mount:

Load Case	Max Tension Stress (psi)	Max Compression Stress (psi)
17	98,473	87,725
18	77,579	89,026
27	280,210	363,460
29	52,858	163,320

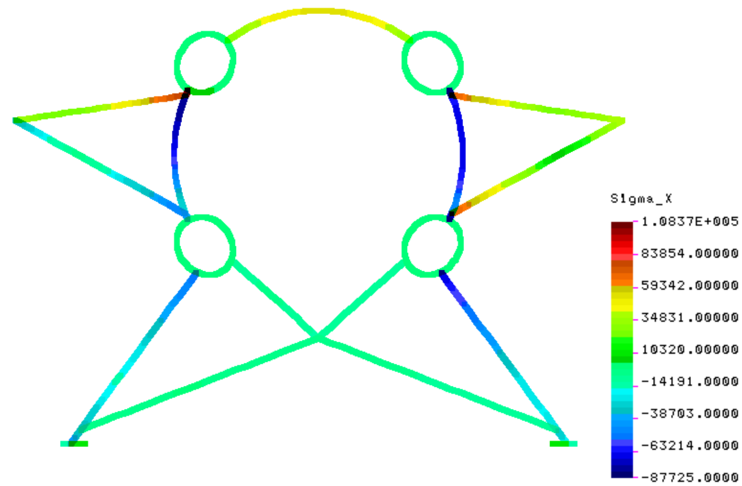
FEA Results for HINGED Engine Mount:

Load Case	Max Tension Stress (psi)	Max Compression Stress (psi)
17	108,370	87,725
18	80,432	93,310
27	285,600	373,000
29	60,266	268,090

Based on FEA results for load case 17 and the material property limits, both engine mounts will fail before reaching +6 g limit loads. Neither configuration survives the 40 g crash loads imposed by load case 27. To understand the stresses better, graphical results are presented below.



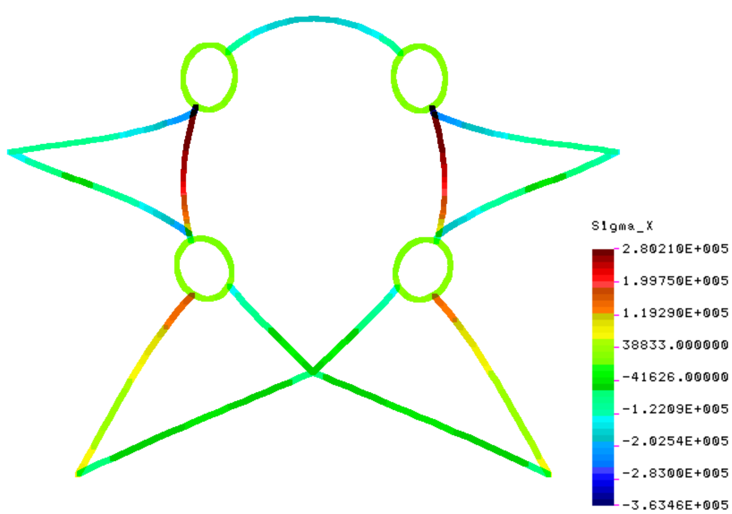
Stress: Load Case 17
Fixed Mount



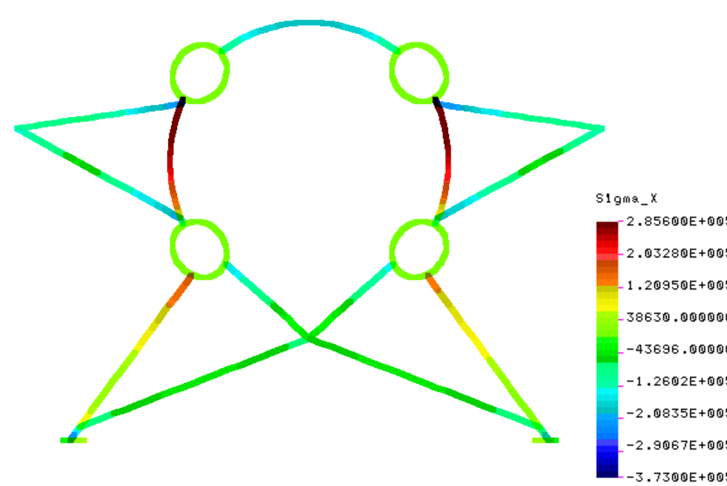
Stress: Load Case 17
Hinged Mount

As expected for a +6 g load factor, the upper truss members carry tension loads and the lower truss members are in compression. The left and right dynafocal tubes are highly stressed, especially where the tubing attaches to the isolator cups.

Results for load case 27 (40 g forward deceleration) are shown below. Note that stress values for color bars do not correlate with other results. Any distortions are highly exaggerated deformations that do not correlate with other views.



Stress: Load Case 27
Fixed Mount



Stress: Load Case 27
Hinged Mount

For load case 27, stress results for options 2 and 4 are within 3% of each other. The 40 g deceleration creates extremely high stress in the left and right dynafocal tubes as well as the upper truss members. The lower truss members exhibit relatively low stress.

Conclusions So Far…

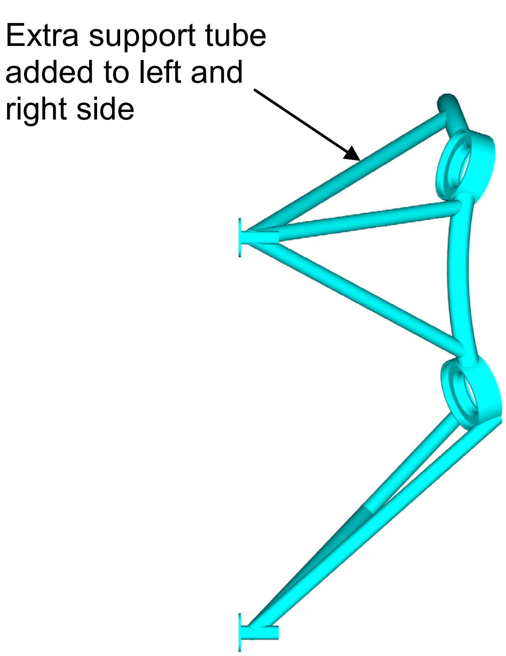
1) Neither configuration meets design requirements. Peak stress for crash loads are 500% higher than desired. Local reinforcements will be ineffective because high stress occurs over large sections of tubing. To fix this problem, the truss configuration must be modified or tube sizes must be increased.

2) For a 40 g forward crash, the hinged configuration may not offer much advantage over the fixed engine mount. High stresses in the upper truss elements cause them to fail long before the lower truss elements. Stress in the lower truss elements is nearly the same for both configurations. There is no indication that the hinged mount will be better at forcing the engine to pitch downward after the upper tubes have buckled.

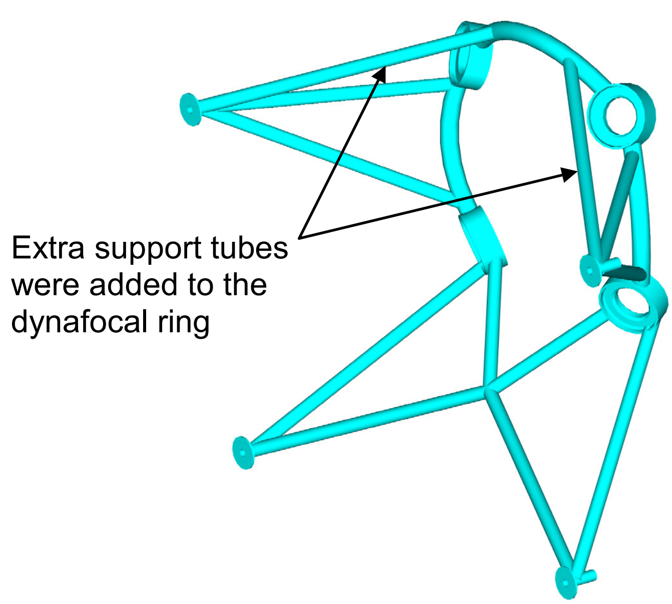
Truss tube diameters of .75” and wall thickness of .049” are fairly common for engine mounts, and that is what was used. It becomes evident that the simple truss used for this design is not adequate for these loads. High stresses in the dynafocal ring indicate that the .875” tube diameter should be increased.

Truss Modifications

To increase strength and reduce stress, two additional support tubes were added to the upper dynafocal ring. Similar truss configurations have been used on certified aircraft. The dynafocal tube diameter was left the same size to better compare it with earlier configurations. The modified engine mount is depicted below. The hinged configuration was not pursued due to its added complexity.



SIDE VIEW

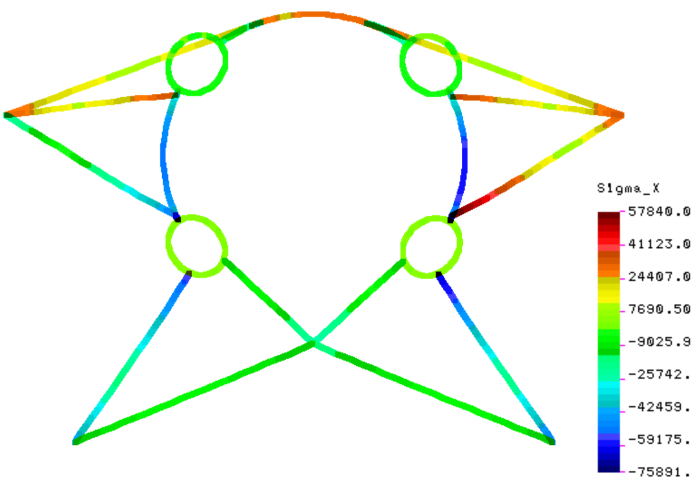


3D VIEW

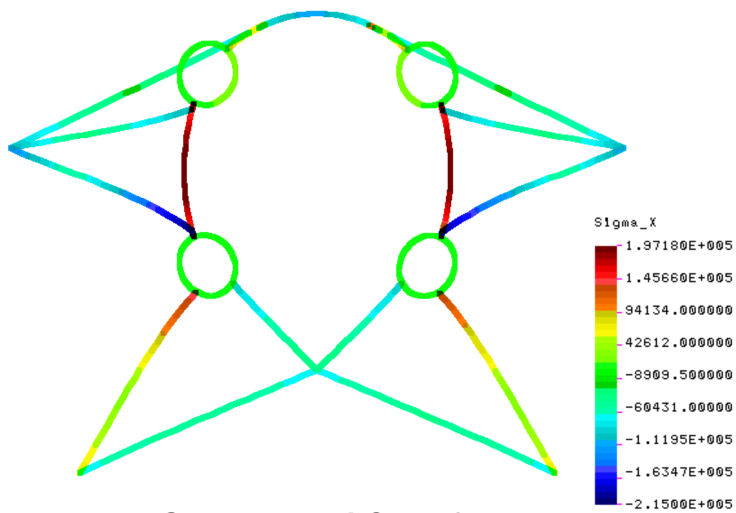
FEA Results for MODIFIED Engine Mount:

Load Case	Max Tension Stress (psi)	Max Compression Stress (psi)
17	57,840	75,891
18	60,442	55,076
27	197,180	215,000
29	21,931	162,140

Compared to the previous fixed mount, there is significant reduction in stress. For load case 17, max tension stress was reduced 41% and max compression stress was reduced 13%. For load case 27, max tension stress was reduced 29% and max compression stress was reduced 40%. To visualize where these stresses occur, graphical results are shown below.



Stress: Load Case 17
Modified Mount



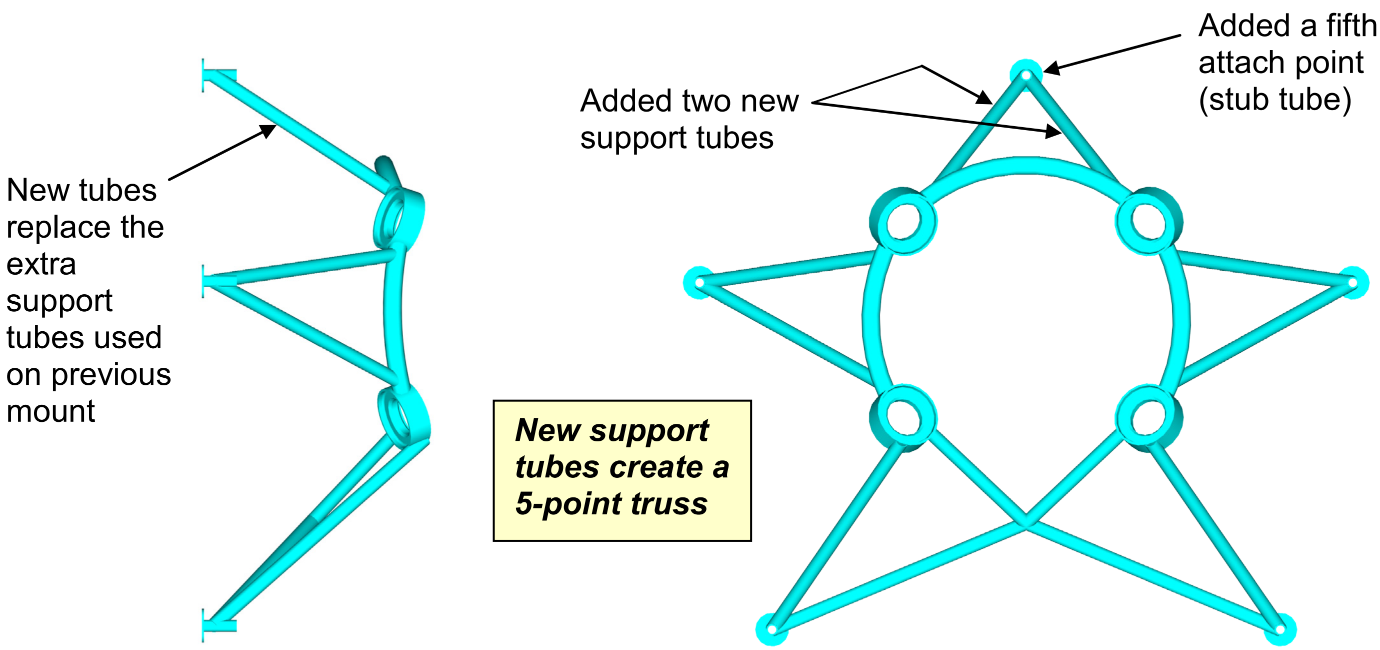
Stress: Load Case 27
Modified Mount

Stress distribution is similar to the previous engine mount and failure modes are apparent. In load case 17, the lower truss tubes have exceeded their compressive yield strength and will probably buckle before reaching +6 g limit loads. In load case 27, the middle truss tubes and dynafocal ring are failing much too early due to high tension and compression loads.

Conclusions on Modified Truss

To meet the crash load specifications, this configuration would require larger diameter tubing, thicker walls and/or local stiffeners. While the two extra support tubes reduce overall stress, they are structurally inefficient due to their shallow angle to the firewall. In order to clear the isolators, the new support tubes are welded to the dynafocal ring two inches away from the isolator cup. This induces bending loads in the dynafocal tube. Instead of adding weight to strengthen an inefficient truss, a better approach would be to use more efficient truss geometry.

Five Point Engine Mount
A fifth attach point was created by adding a stub tube near the top of the firewall and centered on the truss. New support tubes connect the stub tube to the dynafocal ring near the isolator cups. The new tubes are shorter than the extra support tubes used on the previous truss, so the 5-point mount is lighter by 0.20 pound. The new truss is shown below.



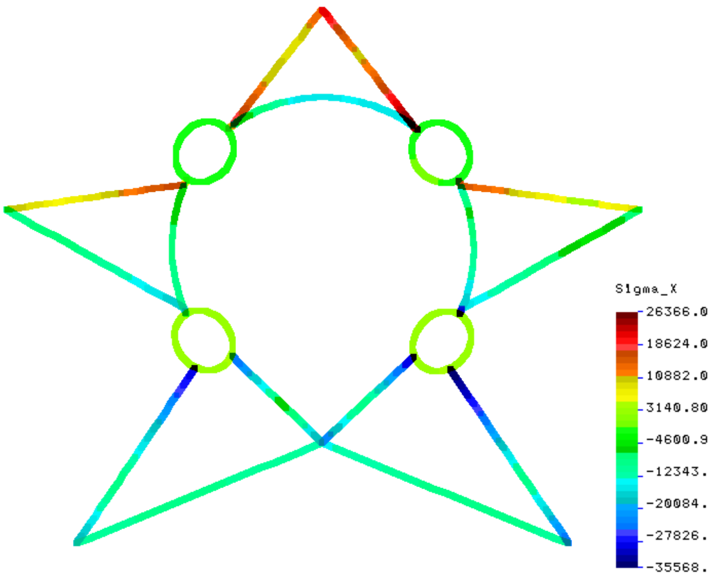
SIDE VIEW **FORWARD VIEW**

FEA Results for FIVE POINT Engine Mount:

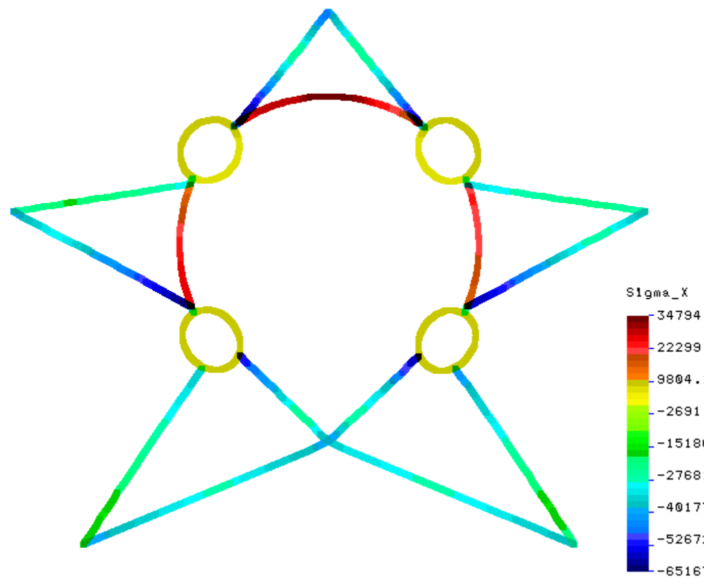
Load Case	Max Tension Stress (psi)	Max Compression Stress (psi)
17	26,366	35,568
18	17,441	21,478
27	34,794	65,167
29	35,738	142,470

There is a large reduction in stress when compared to the previous engine mount. For load case 17, max tension stress was reduced 54% and max compression stress was reduced 53%. For load case 27, max tension stress was reduced 82% and max compression stress was reduced 70%. Except for load case 29, all stress results are below the yield strength of 4130 steel.

The buckling failure resulting from high compression stress in load case 29 (crash deceleration of 40 g's forward and 20 g's downward) is a preferred failure mode. The explanation for this is on the next page with load case 29 graphical results.



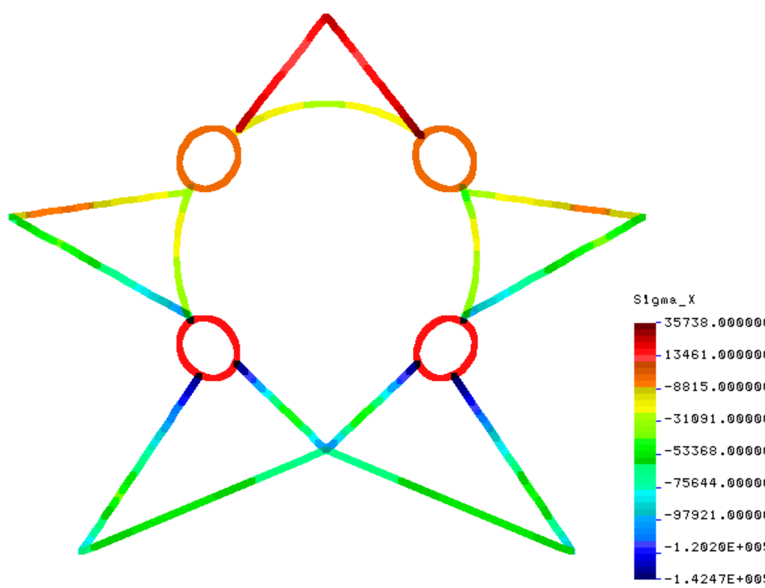
Stress: Load Case 17
Five Point Mount



Stress: Load Case 27
Five Point Mount

For load case 17 (+6 g limit load), the lower support tubes are in compression but stress is less than half the material yield point. In load case 27 (40 g forward crash) four of the upper and middle support tubes have high compression loads and stress levels are nearing the yield point. This should be acceptable for crash conditions; the full analysis will examine buckling factors to determine if further modifications are necessary.

Load case 29 results are shown at right. In this extreme crash (40 g forward, 20 g downward) the support tubes for the lower dynafocal cups have four times the stress as the upper cup support tubes. This indicates the lower tubes will fail or the lower cups will tear out well before the upper tubes. Since the engine is cantilevered off the engine mount, this creates an aft-end-down moment for the engine. This is desirable because the aft end of the engine is directed towards the ground as the engine moves forward. Striking the ground is a good way to safely disperse the engine's kinetic energy.



Stress: Load Case 29
Five Point Mount

Conclusion
Analysis shows the 5-point truss is more robust and has lower stress than either of the 4-point trusses. Because of its superior performance, the 5-point engine mount was selected for the baseline configuration. Benefits of this design include:

- Bending loads on the dynafocal tube and truss elements are greatly reduced.
- Stress levels for critical flight load cases stay below the material yield point.
- The engine mount survives 40 g crash loads with standard size tubing.

One disadvantage is the additional cabin structure required to support loads at the fifth mounting point. The estimated weight for the added structure is 2 pounds. This must be balanced against the extra weight required to reinforce a less efficient truss structure to meet the crash load criteria. Overall, the 5-point mount appears to be weight competitive.

Postscript
Increasing the dynafocal tube diameter can reduce peak stress levels even further. The original tubing was .875” O.D. x .065 wall, but 1.0” O.D. x .058 wall tubing is stronger and the weight penalty is less than one ounce. Stress results for this option are presented below.

FEA Results for FIVE-POINT Engine Mount with 1.0” Dynafocal Tube:

Load Case	Max Tension Stress (psi)	Max Compression Stress (psi)
17	24,789	33,682
18	14,104	20,212
27	30,570	62,717
29	33,362	136,540

Stress was reduced anywhere from 4% to 19% relative to the .875” dynafocal tube. Even this small reduction may improve fatigue life. Lower initial stress also provides larger fail-safe margins should the engine mount ever be damaged or in case of propeller blade loss. For these reasons, the baseline configuration will specify 1.0” O.D. x .058 wall tubing.