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TO

AEC-NASA SPACE NUCLEAR PROPULSION OFFICE

LOWER THRUST STRUCTURE S/N 0000001
STATIC STRUCTURAL TEST EVALUATION REPORT

APRIL 1965

NERVA PROGRAM

CONTRACT SNP-1

MASTER

AEROJET-GENERAL CORPORATION

SACRAMENTO, CALIFORNIA

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REPORT NO. RN-S-0216

LOWER THRUST STRUCTURE S/N 0000001
STATIC STRUCTURAL TEST EVALUATION REPORT

ROCKET ENGINE OPERATIONS - NUCLEAR

NERVA PROGRAM APRIL 1965 CONTRACT SNP-1

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ABSTRACT

A static structural test was conducted on the NERVA E-Engine Lower Thrust Structure S/N 0000001 at Aerojet-General. The major test results and data evaluation are presented for 12 loading conditions which simulated boost and operating loads to which the structure would be subjected during flight or ground testing. The structure showed more than adequate capacity to withstand these loads.

for

W. D. Stinnett
Program Manager
REON

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I. INTRODUCTION

This document describes the static limit load test of an E-engine NERVA Lower Thrust Structure (LTS) and includes experimental findings and conclusions derived. The static test program was conducted in the Structural Test Laboratory at the Aerojet-General Corporation (AGC) Sacramento Plant beginning 20 June 1963 and terminating 27 August 1963.

Instrumentation and testing were performed in accordance with AGC Report No. 748N-TP.⁽¹⁾ The test program consisted of subjecting the LTS to 12 static load combinations which simulated maximum boost and operational loads.⁽²⁾ Testing was conducted in two Phases: during Phase 1, a stresscoat was utilized along with strain and deflection gages to establish the location of stress concentrations and the requirement for additional instrumentation for subsequent load combinations; during Phase 2, data for all strain gages and deflection transducers was recorded. The LTS load combinations were applied incrementally to 100% limit load.

-
- (1) Report No. 748N-TP, "Test Specifications for the Static Structural Test of the NERVA Lower Thrust Structure", 17 June 1963.
 - (2) AGC Report No. 9443-62-16, "NERVA TCA Structural Loads Document" was used as the basis for the test loads.

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II. TEST PROGRAM OBJECTIVES

The objectives of test 748N were as follows:

- A. Demonstrate the structural integrity of the LTS under static test loads.
- B. Determine axial and lateral stiffness of the structure.
- C. Provide information on the effect of thrust structure deflections on thrust misalignment.
- D. Provide information for design improvement.
- E. Permit a correlation of experimental and theoretical data.

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III. DESCRIPTION OF TEST

A. TEST ASSEMBLY

The lower thrust structure (S/N 0000001) shown in Figure 1 served as the static test unit. This unit, a full scale structure fabricated from CRES 347 and CRES 321 stainless steel at the AGC Sacramento Plant, consisted of a cylinder with a diameter of approximately 36 in. a length of 32 in. and a frustrum 18 in. deep. Thrust Vector Control (TVC) actuator supports, similar to those which provide pitch and yaw control for the NERVA engine, were attached to the upper end of the cylindrical portion at 0° and 270° positions. AGC drawing 280210 describes the test unit in detail.

The test unit was installed in a universal, self-reacting test fixture by bolting the pressure vessel closure simulator to the test fixture and then securing the LTS to this simulator with a three-sectioned, Marman-type clamp. The test fixture has the capability of individual or simultaneous application of axial, lateral, and TVC actuator test loads. The axial and lateral loads were applied to the LTS through a gimbal simulator attached to the gimbal housing. Using the swivel action of the gimbal simulator, the lateral loads were applied in tension at the 0°, 90°, 180° and 270° positions of the test unit. Axial and TVC actuator loads were applied both in tension and compression. Figure 2 presents the test setup and lists the tools used. Figure 3 is a photograph of the setup where the actuator supports and gimbal housing are being loaded.

Figure 1

Lower Thrust Structure S/N 0000001

Figure 3

Typical Test Setup for Loading Gimbal Housing
and Actuator Supports

B. INSTRUMENTATION

A total of 256 instrument channels were used in measuring strain and deflection on the test unit. Strain gages were mounted in the following areas: on the LTS frustrum and cylinder at 135°; on both sets of actuator support tubes; on the gimbal housing rib at 0°; in the frustrum access hole near 180°; in the cylinder access holes at 90°, 180°, and 333°; on the bulkhead stiffener at 0°; on the frustrum-cylinder joint at 0°; on the LTS - Pressure Vessel (PV) clamp at 135° and 270°. Deflection gages were positioned to monitor the simulated gimbal and the following additional places: both actuator support yokes; the frustrum-cylinder flange at 0°, 90°, 180°, and 270°; the inside surface of the LTS-PV joint at 135°; the outside surface of the LTS-PV joint at 0°, 90°, 180°, and 270°; the LTS-PV clamp at 0° and 135°. Stresscoat was applied around the base of the LTS, on all ribs of the gimbal housing, on a 90° pie-section around the frustrum access hole at 195°, around all cylinder access holes, and around the frustrum-cylinder joint.

Figures 4 through 8, show three typical strain gage locations. Locations of all strain gages and deflection transducers are shown in Figures 9 and 10, respectively.

Figure 4

Strain Gages Installed at $\Theta = 135^\circ$

Figure 5

Interior Strain Gages Installed at $\Theta = 135^\circ$

Figure 6

Strain Gages Installed on TVC Actuator Supports

Figure 7

Strain Gages Installed Adjacent Cylinder Cutout
with Doubter-Type Reinforcement

Figure 8

Strain Gages Installed Adjacent Cylinder Cutout
with Rim-Type Reinforcement

Figure 9

Strain Gage Locations

GAGE NO.	AXIS	POLARITY
1	X	+ DOWN
2	Y	+ CLOCKWISE
3	Z	+ OUT
4	X	+ DOWN
5	Y	+ OUT
6	Z	+ CLOCKWISE
7	X	+ DOWN
8	Y	+ CLOCKWISE
9	Z	+ OUT
10	X	+ DOWN
11	Y	+ OUT
12	Z	+ CLOCKWISE
13	X	+ UP
14	Y	+ CLOCKWISE
15	Z	+ OUT
16	X	+ UP
17	Y	+ OUT
18	Z	+ CLOCKWISE
19	X	+ UP
20	Y	+ CLOCKWISE
21	Z	+ OUT
22	X	+ UP
23	Y	+ OUT
24	Z	+ CLOCKWISE
25	X	+ UP
26	Y	+ CLOCKWISE
27	Z	+ OUT
28	X	+ UP

GAGE NO.	AXIS	POLARITY
29	Y	+ OUT
30	Z	+ CLOCKWISE
31	X	+ UP
32	Y	+ CLOCKWISE
33	Z	+ OUT
34	X	+ UP
35	Y	+ OUT
36	Z	+ CLOCKWISE
37	X	+ UP
38	Y	+ CLOCKWISE
39	Z	+ OUT
40	X	+ UP
41	Y	+ CLOCKWISE
42	Z	+ OUT
43	X	+ UP
44	Z	+ OUT - 180° POS
45	Y	+ OUT - 270° POS
46		+ OUT
47		+ OUT
48		+ OUT
49		+ OUT
50		+ OUT
51		+ OUT
52		JOINT SECTION
53		+ DOWN
54		+ DOWN
55		+ DOWN
56		+ DOWN

TAB BLOCK II	X	Y	Z	Ø
13	14	15	0°	
16	17	18	90°	
19	20	21	180°	
22	23	24	270°

TAB BLOCK I	X	Y	Z	Ø
1	2	3	0°	
4	5	6	90°	
7	8	9	180°	
10	11	12	270°

NOTES
1 * DENOTES RECTILINEAR TYPE GAGE

Figure 10

Deflection Transducer Locations

C. TEST OPERATIONS

The following events summarize the LTS test operations:

1. Pre-test dimensions of the LTS were obtained and recorded. Considerable mismatch was noticed in the welded joint above the LTS-PV flange. Figure 11 is a photograph of a cast of this area.
2. Strain and internal deflection gages were mounted on the test unit.
3. The test unit and associated tooling were assembled and secured to the test fixture. The bolts on the LTS-PV clamp were tightened in increments to 1200-inch-pounds torque and the strains, representing clamp preload, were recorded.
4. External deflection transducers were mounted.
5. Stresscoat was applied to the unit. All instrumentation, after being connected to the recorders, was zeroed, balanced and ranged.
6. The Case-1 load combination was applied incrementally to 100% limit-load. Instrumentation recordings were made after each load increment, with the stresscoat inspected after each increment above 60%. This run constituted Phase 1 of the test.
7. Additional strain gages were installed and the loads for Cases 2, 10, 11, 12, 3, 4, 5, and 6 were applied. Additional strain gages were affixed to the unit and the remaining Cases (Cases 7, 8, and 9) were run. Loading Cases 2 through 12 constituted Phase 2.
8. The test unit was taken out of the test fixture, cleaned (all instrumentation being removed) and a post-test inspection was conducted.

Figure 11

Weld Mismatch of Cylinder and Flange Ring

IV. EXPERIMENTAL RESULTS

A. LOADS

The LTS was subjected to 12 static limit load combinations at room temperature. Planned, and actual, test loads are shown in Figure 12. Generally, Case 9 was the most critical load application. Loads for this Case consisted of 126 kips axial tension, .98 kip and .95 kip shear forces at the gimbal simulator, and 28.22 kips and 28.49 kips compression forces at the pitch and yaw actuators, respectively.

PLANNED LOADS					
CASE NO.	F (KIPS)	V_p (KIPS)	V (KIPS)	A_p (KIPS)	A (KIPS)
1	81.5	0	0	0	0
2	-33.6	1.2	0	35.0	0
3	33.6	-1.2	0	-35.0	0
4	-33.6	0	1.2	0	35.0
5	33.6	0	-1.2	0	-35.0
6	47.9	1.2	0	35.0	0
7	115.1	-1.2	0	-35.0	0
8	34.1	0.85	0.85	24.7	24.7
9	128.9	-0.85	-0.85	-24.7	-24.7
10	-51.8	0	0	0	0
11	-85.4	1.2	0	35.0	0
12	-18.2	-1.2	0	-35.0	0

ACTUAL LOADS					
CASE NO	F (KIPS)	V_p (KIPS)	V_y (KIPS)	A_p (KIPS)	A (KIPS)
1	83.03	0	0	0	0
2	-38.62	1.2	0	39.21	0
3	34.40	-.92	0	-35.80	0
4	-36.70	0	1.02	0	37.30
5	35.10	0	-1.27	0	-38.39
6	54.30	1.37	0	38.59	0
7	124.10	-1.12	0	-39.35	0
8	30.65	1.07	.84	28.43	25.41
9	126.00	-.98	-.95	-28.22	-28.49
10	-52.10	0	0	0	0
11	-88.77	1.27	0	39.06	0
12	-20.20	-1.02	0	-35.90	0

CASE LOADING SEQUENCE WAS 1, 2, 10, 11, 12, 3, 4, 5, 6, 7, 8 AND 9.

Figure 12

Planned and Actual Test Loads

B. STRAIN DISTRIBUTION

Stress values of the cylinder and frustrum at 100% limit load for the Cases found most critical are presented in Tables 1 and 2. To illustrate their nature and magnitude, plots of the stress values are shown in Figures 13 through 28.

Strain results for several Cases on the pitch and yaw actuator supports (taken from gages located at mid-span) are recorded in Table 3. These values have been further reduced to axial stresses and loads. Table 4 presents a comparison of axial and bending stresses for both supports.

Installation of the three segment Marman-type clamp resulted in strains which have been reduced to stresses and are presented in Table 5. Stresses from Case 9 are shown for magnitude and comparison with installation stresses.

TABLE 1

MEASURED UTS BIAXIAL STRESSES AT MAXIMUM LOAD

Gage Location	Case No.	Outer Surface Stresses, psi		Inner Surface Stresses, psi	
		Meridional	Hoop	Meridional	Hoop
105	9	32,580	21,570	-10,370	10,870
106	9	44,180	27,070	-31,900	4,080
107	9	25,960	23,220	11,030	18,390
108	9	-7,550	-6,035	31,820	8,665
109	9	17,810	-750	17,400	1,740
110	9	30,300	4,080	13,790	-4,100
111	9	22,480	-6,580	21,510	-9,580
112	9	-160	-20,640	33,570	-1,160
113	9	23,600	-7,580	-870	-17,020
114	9	21,820	-1,480	-	-
115	9	26,500	1,800	19,600	-900
116	9	26,000	1,500	22,440	80
117	9	29,470	-1,180	31,400	1,600
118	9	28,550	-8,080	32,570	-3,180
119	9	20,390	-3,080	39,400	3,400
120	9	39,620	13,080	14,180	1,150
128	9	26,310	-2,580	28,000	2,300
129	6	30,240	-3,580	33,940	-1,230
130	6	4,040	-23,570	41,170	6,390
131	6	35,420	6,370	-9,420	-12,660
132	9	14,860	-4,590	19,090	-3,840
133	9	20,090	-500	18,180	-4,640
134	9	23,200	25,000	20,240	-3,090

Notes: 1. Stresses were determined using computer program 14039, "Inelastic Biaxial Strain Gage Data Reduction". Stress-strain control parameters were $E = 29 \times 10^6$ psi, $\mu = 0.3$ and an average .2% offset yield strength = 38,000 psi.

2. See Figure 9 for strain gage locations.

TABLE 2

MEASURED UNIAXIAL STRESSES ADJACENT REINFORCED CUTOUTS

Gage	Case	Stress, psi
135	9	35,350
136	9	18,850
137	9	15,660
138	9	15,810
139	9	16,100
140	9	31,600
141	9	32,450
142	9	31,440
143	9	29,860
145	6	31,830
146	6	37,140
147	6	23,410
151	9	-38,160
152	9	-35,980
153	9	-35,540
154	9	-34,450

- Note: 1. Stresses were determined using computer program 14039, "Inelastic Biaxial Strain Gage Data Reduction". Stress-strain control parameters were $E = 29 \times 10^6$ psi and an average .2% offset yield strength = 38,000 psi.
2. See Figure 9 for strain gage locations.

Figure 13

Cylinder to PV Flange Ring Meridional Stresses at $\Theta = 135^\circ$

Figure 14

Cylinder to PV Flange Ring Hoop Stresses at $\Theta = 135^\circ$

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NOTES: 1. CASE 9, 100% TEST LOAD
2. SEE TABLE 1 FOR STRESSES

Figure 15

Cylinder to Frustrum -
Cylinder Flange Meridional Stresses at Theta = 135°

NOTES: 1. CASE 9, 100% TEST LOAD
2. SEE TABLE 1 FOR STRESSES

Figure 16

Cylinder to Frustrum -
Cylinder Flange Hoop Stresses at Theta = 135°

NOTES: 1. CASE 6, 100% TEST LOAD
2. SEE TABLE 1 FOR STRESSES

Figure 17

Cylinder to Frustrum -
Cylinder Flange Meridional Stresses at $\Theta = 0^\circ$

NOTES: 1. CASE 6, 100% TEST LOAD
2. SEE TABLE 1 FOR STRESSES

Figure 18

Cylinder to Frustrum -
Cylinder Flange Hoop Stresses at $\Theta = 0^\circ$

Figure 19

Frustrum to Frustrum -
Cylinder Flange Meridional Stresses at $\Theta = 135^\circ$

Figure 20

Frustrum to Frustrum -
Cylinder Flange Hoop Stresses at $\Theta = 135^\circ$

NOTES: 1. CASE 9, 100% TEST LOAD
2. SEE TABLE 1 FOR STRESSES

Figure 21

Frustrum to Frustrum -
Cylinder Flange Meridional Stresses at $\Theta = 0^\circ$

NOTES: 1. CASE 9, 100% TEST LOAD
2. SEE TABLE 1 FOR STRESSES

Figure 22

Frustrum to Frustrum -
Cylinder Flange Meridional Stresses at $\Theta = 0^\circ$

NOTES: 1. CASE 9, 100% TEST LOAD
2. SEE TABLE 1 FOR STRESSES

Figure 23

Frustrum to Gimbal Housing Meridional Stresses at $\Theta = 135^\circ$

NOTES: 1. CASE 9, 100% TEST LOAD
2. SEE TABLE 1 FOR STRESSES

Figure 24

Frustrum to Gimbal Housing Hoop Stresses at $\Theta = 135^\circ$

Figure 25

Doubler Reinforced Cylinder Cutout Tangential Stresses

Figure 26

Rim Reinforced Cylinder Oblong Cutout Tangential Stresses

Figure 27

Rim Reinforced Cylinder Circular Cutout Tangential Stresses

Figure 28

Rim Reinforced Frustrum Circular Cutout Tangential Stresses

TABLE 3

TVC ACTUATOR SUPPORT AXIAL LOADS AND STRESSES

Case No.	Member	Area in ²	AVE Strain micro-in./in	Axial Stress psi	Axial Load lbs
2, 2	A	.675	682	19,800	13,350
	B	.539	726	21,050	11,350
	C	.405	-850	-24,600	-9,980
	D	.491	732	21,200	10,420
	E	.405	-787	-22,800	-9,240
	F	.539	667	19,350	10,420
	G	.675	720	20,900	14,090
3, 3	A	.675	-658	-19,100	-12,880
	B	.539	-698	-20,240	-10,910
	C	.405	936	27,200	10,990
	D	.491	-645	-18,700	-9,180
	E	.405	828	24,000	9,730
	F	.539	-641	-18,600	-10,020
	G	.675	-660	-19,150	-12,920
7, 4	A	.675	-786	-22,800	-15,400
	B	.539	-805	-23,350	-12,600
	C	.405	1,009	29,300	11,860
	D	.491	-691	-20,000	9,820
	E	.405	951	27,600	11,200
	F	.539	-684	-19,850	-10,700
	G	.675	-765	-22,200	-15,000
11, 5	A	.675	691	20,000	13,500
	B	.539	726	21,000	11,320
	C	.405	-988	-28,650	-11,600
	D	.491	744	21,600	10,600

TABLE 3 (cont.)

Case No.	Member	Area in ²	AVE Strain $\frac{1}{4}$ micro-in/in	Axial Stress psi	Axial Load lbs
	E	.405	-929	26,950	-10,900
	F	.539	720	20,900	11,280
	G	.675	760	22,050	14,900
4 $\frac{1}{6}$	J	.675	724	21,000	14,170
	K	.539	736	22,400	11,500
	L	.405	-781	-22,640	-9,170
	N	.491	723	21,000	10,300
	P $\frac{1}{9}$	.405	-853	-24,800	-10,050
	Q	.539	651	18,900	10,180
	R	.675	638	18,500	12,490
5 $\frac{1}{7}$	J	.675	-623	-18,100	-12,200
	K	.539	-681	-19,800	-10,650
	L	.405	975	28,240	11,450
	N	.491	-718	-20,800	-10,220
	P $\frac{1}{9}$	.405	887	25,800	10,430
	Q	.539	-689	-20,000	-10,770
	R $\frac{1}{10}$	.675	748	-21,700	-14,650

Notes: $\frac{1}{4}$ For location at measured strains see Figure 9. AVE strain $\epsilon_{AVE} = \frac{1}{4} (\epsilon_1 + \epsilon_2 + \epsilon_3 + \epsilon_4)$.

$\frac{2}{1}$ Pitch actuator load $A_p = 39,210$ lbs

$\frac{3}{1}$ Pitch actuator load $A_p = -35,800$ lbs

$\frac{4}{1}$ Pitch actuator load $A_p = -39,350$ lbs

$\frac{5}{1}$ Pitch actuator load $A_p = 39,060$ lbs

$\frac{6}{1}$ Yaw actuator load $A_y = 37,300$ lbs

$\frac{7}{1}$ Yaw actuator load $A_y = -38,390$ lbs

TABLE 3 (cont.)

8. $E = 29 \times 10^6$ psi

Axial stress = $E \epsilon_{AVE}$

Axial load = $E \epsilon_{AVE} A$

9. $\epsilon_{AVE} = \frac{1}{2}(\epsilon_1 + \epsilon_3)$ as ϵ_4 was faulty

10. $\epsilon_{AVE} = \frac{1}{2}(\epsilon_1 + \epsilon_3)$ as ϵ_2 was faulty

TABLE 4

COMPARISON OF TVC ACTUATOR SUPPORT AXIAL AND BENDING STRESSES

Pitch Actuator Support

Member	Case 2			Case 7
	σ_a ksi	σ_b ksi	σ_b/σ_a	σ_a ksi	σ_b ksi	σ_b/σ_a
A	19.80	3.48	.18	-22.8	3.02	.13
B	21.05	3.24	.15	-23.35	2.10	.09
C	-24.60	2.76	.11	29.30	3.26	.11
D	21.20	3.28	.15	-20.00	1.68	.08
E	-22.80	5.30	.23	27.60	3.78	.14
F	19.35	2.82	.15	-19.85	2.78	.14
G	20.90	1.06	.05	-22.20	1.60	.07

Yaw Actuator Support

	Case 4			Case 5
	σ_a ksi	σ_b ksi	σ_b/σ_a	σ_a ksi	σ_b ksi	σ_b/σ_a
J	21.00	5.28	.25	-18.10	7.30	.40
K	22.40	5.80	.26	-19.80	7.90	.40
L	-22.64	4.50	.20	28.24	4.20	.15
N	21.00	5.86	.28	-20.80	9.00	.43
P	-24.80	5.16	.21	25.80	2.40	.09
Q	18.90	5.68	.30	-20.00	7.70	.39
R	18.50	5.22	.28	-21.70	8.32	.38

Notes: $A_p = 39.21$ kips
$A_p = -39.35$ kips
$A_y = 37.30$ kips
$A_y = -38.39$ kips
See Figure 9 for member and strain gage locations.

TABLE 5

SUMMARY OF CLAMP STRESSES

A. Strain Gage Pattern at $\theta = 135^\circ$

Gage No.	Axial Stress, psi		Circumferential Stress, psi	
	Installation	Case 9	Installation	Case 9
121	-8,770	-4,750	7,530	5,420
122	-7,660	-3,540	7,450	510
123	-7,200	-5,500	6,060	-5,170
124	-9,150	-11,100	130	-8,870

B. Strain Gage Pattern Adjacent Clamp Connection

125	-540	-5,140	26,200	510
126	-3,410	-7,200	17,200	90
127	-3,980	-7,950	8,400	-800

- Notes: 1. Stresses were determined using computer program 14039, "Inelastic Biaxial Strain Gage Data Reduction". Stress-strain control parameters were $E = 29 \times 10^6$ psi, $\mu = 0.3$ and an average .2% offset yield strength = 38,000 psi.
2. See Figure 9 for strain gage locations.

C. DEFLECTION

Table 6 is the tabulated values pertaining to the frustrum and actuator frames of deflection at 100° test load for the 12 Cases. Figure 29 shows schematic deflected shapes of the LTS considering Case 9 and 11 test loads.

Data at the LTS-PV joint was limited to Cases 6, 7, 8, and 9 because of instrumentation problems. The results have been reduced to flange rotations in separation across the interface. A tabulation of data and a sketch are shown in Figure 30.

TABLE 6

FRUSTRUM AND TVC ACTUATOR SUPPORT DEFLECTIONS
AT MAXIMUM TEST LOAD

LOAD CONDITION						
GAGE NO.	1	2	3	4	5	6
1	.035	.031	-.025	△	.005	.094
2	-	-	-	.009	-.010	-
3	-	-.002	△	-	-	.010
7	.028	-.008	.011	△	-.001	.015
8	-	-	-	.022	-.010	-
9	-	-.017	-.014	-	-	-.025
10	.031	.005	.003	.011	-.002	.030
11	-	-	-	.010	-.002	-
12	-	-.011	-.012	-	-	.031
37	-	-	-	.003	-.011	-
39	-	-	-	-.001	Out	-
40	-	.063	-.062	-	-	.018
42	-	-.088	-.067	-	-	Out
43	.046	-.007	.006	△	.002	△.038
44	-	.026	-.016	-	-	.031
45	-	-	-	+.039	-.038	-

GAGE NO.	7	8	9	10	11	12
1	.004	.062	.017	-.024	.009	-.043
2	-	.016	-.009	-	-	-
3	.003	-.007	.007	-	-.002	.003
7	.048	.020	.044	-.012	-.091/△	.005
8	-	.041	-.028	-	-	-
9	.016	-.016	.009	-	-.018	.016
10	.038/△	.025	.014	-.028	-.019	-.021
11	-	.017	-.002	-	-	-
12	Faulty	.021	-.008	-	△	-.020
37	-	.008	-.004	-	-	-
39	-	-.003	△	-	-	-
40	-.006	.012	-.003	-	.055	.053
42	.114	-.101	.066	-	-.092	.075
43	△	.045/△	.051	-.028	-.031	-.024
44	-.019	.030	-.006	-	.029	-.019
45	-	.038	-.041	-	-	-.019

NOTES:

△ MEASURED DEFLECTIONS WERE ZERO THROUGHOUT TEST RUN. DATA MAY BE FAULTY.

△ DATA WENT NON-LINEAR BETWEEN 80 to 100% TEST LOAD. PROJECTED VALUE IS -.020.

△ PROJECTED VALUES SHOWN. DATA INDICATES SLIP OCCURRED BETWEEN 60 - 80% OF TEST LOAD

SEE FIGURE 30 FOR DEFLECTION DATA AT PV-LTS JOINT.

NOTES: 1. SEE TABLE 6 FOR DEFLECTION VALUES
2. LTS SHOWN IN PITCH PLANE

Figure 29

Representative Deflected Shapes under Loading

135° THETA ORIENTATION

CASE NO.	β_L RAD	β_P RAD	β RAD	s^1 IN
6	.0040	-.0007	.0047	.0019
7	.0116	-.0006	.0122	.0073
8	.0011	0	.0011	.0009'
9.	.0074	-.0001	.0075	.0102

- NOTES: 1. INSIDE SURFACE SEPARATION USING GAGE 52 AND ASSUMING PIVOT ABOUT FLANGE TOES
 2. DEFLECTION INSTRUMENTS WERE FAULTY ON OTHER RUNS

Figure 30

LTS-PV Joint Rotation

D. STRESSCOAT

The stresscoat testing during Phase I proved somewhat unsuccessful. Initially no cracking was visible since the threshold strain of the stresscoat lacquer was higher than the strains experienced under load. Attempts were then made to lower the initial threshold value by applying cold water and CO₂ gas. Statiflux and dye-etchant were applied, but the stresscoat patterns obtained could not be evaluated in any quantitative degree because of the highly varied sensitivity caused by the applied water and CO₂. However, principal tensile strain directions were determined which aided in locating additional strain instrumentation.

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V. EVALUATION AND CONCLUSIONS

A. DISCUSSION

1. Structural Integrity

Results of all loading conditions indicated that the lower thrust structure successfully withstood the static loads applied to simulated expected NERVA boost and operational flight loads. The maximum stress level at any measured point occurred at the weld joint between the cylinder and the PV flange ring. At station 106 the meridional stress of 44,180 psi on the outer surface (see Figure 13) were recorded, primarily as a result of high bending stresses caused, significantly, by the weld mismatch and by distortion which ranged from .030 in. to .100 in. (Figure 11 shows this mismatch condition, which is considered undesirable). It is significant to note that, in spite of this inelastic condition, the permanent strain of 615 micro-inches/inch was considerably less than the limit value of .2% permanent deformation.

Stress distribution around a reinforced hole, as shown in Figure 25, is similar to that expected in a flat plate under uniaxial tension loading. Stress at the edge of this heavily reinforced hole was approximately twice as great as that in the skin. For the circular shaped, reinforced, holes shown in Figures 27 and 28, the stresses in the skin and reinforcement were high and were of the same order of magnitude. The greatest strain on the part was measured at location 151 on the inside edge of the hole (Figure 25). This 3300 micro-inches/inch strain resulted in an uniaxial stress of 39,160 psi and a permanent strain of 1840 micro-inches/inch, which is slightly less than the limit value of .2%. Although, compared to the doubler type, the circular-rim type reinforcement was more effective in reducing the stress concentration factor at the hole, both types satisfied the yield design criteria.

The stresses at the weld joining of the cylinder to the frustrum-cylinder flange (shown in Figure 17) were also significant when the pitch actuator load was tension (Case 6). At location 130 the meridional stress on the inner surface

was 41,170 psi, divided about equally between membrane and bending stresses. Permanent strain recorded was 350 micro-inches/inch, which again is well within the allowable permanent set. It should be noted that mismatch and distortion was visible at this joint.

The stresses at the frustrum-to-gimbal-housing weld joint (as shown in Figure 23) consisted of both high membrane and bending stresses. At location 119 a meridional stress of 39,400 psi was indicated with permanent set of 955 micro-inches/inch on the inner surface. On the other side of the weld joint, at location 120, a meridional stress of 39,620 psi with a permanent set of 504 micro-inches/inch was shown indicating a reversal of stress. The resulting high bending stresses were caused by the lap joint weld design, weld mismatch and distortion. As noted in discussing other weld areas, the inelastic strain was still within the allowable limit, even with the defects.

From Table 3 it can be deduced that the axial stresses of all seven tubular members of the actuator supports were generally of the same magnitude. The members connected to the frustrum from the actuator yoke were always maximum and of opposite sign compared to the other members. Load symmetry between actuator supports was within the same order-of-magnitude as symmetry within the same support. Maximum axial stress was 29,300 psi, which was tension stress on member C during Case 7. At midspan, the bending was reasonably uniform, as shown in Table 4. On support member C the bending stress was about 3,300 psi or 11% of the membrane stress. In no case was there a greater combined stress. As the inelastic strain was small (100 micro-inches/inch), the structural adequacy of the actuator supports was proven.

2. Axial and Lateral Stiffness

LTS axial spring constants computed from deflection data in Table 6 are shown in Table 7. Load Cases 1 and 10 were used as only a single load was applied. Overall value was shown to be between 1.8 to 1.9×10^6 lbs/in. Because of complex multiple loading in the remainder of the Cases, lateral LTS and actuator support spring constants were not computed.

TABLE 7

AXIAL SPRING CONSTANT

OVERALL SPRING CONSTANT

$$K_{TOT} = \frac{F_1}{S_{TOT}}$$

CASE 1

$$K_{TOT} = \frac{83.3 \times 10^3}{.046} = 1.8 \times 10^6 \text{ LB/IN}$$

$$\text{WHERE } S_{TOT} = S_{43}$$

CASE 10

$$K_{TOT} = \frac{-52.1 \times 10^3}{.028} = 1.9 \times 10^6 \text{ LB/IN}$$

CYLINDER SPRING CONSTANT

$$K_{CYL} = \frac{F_1}{S_{CYL}}$$

CASE 1

$$K_{CYL} = \frac{83.3 \times 10^3}{.031} = 2.7 \times 10^6 \text{ LB/IN}$$

$$\text{WHERE } S_{CYL} = \frac{1}{3} (S_1 + S_7 + S_{10})$$

CONE SPRING CONSTANT

$$K_{CONE} = \frac{F_1}{S_{CONE}}$$

$$\text{CASE 1 } K_{CONE} = \frac{83.3 \times 10^3}{.015} = 5.5 \times 10^6 \text{ LB/IN}$$

$$\text{CASE 10 } K_{CONE} = \frac{52.1 \times 10^3}{-.007} = 7.4 \times 10^6 \text{ LB/IN}$$

$$\text{WHERE } S_{CONE} = S_{TOT} - S_{CYL}$$

NOTES: 1. K_{CYL} AND K_{CONE} SHOULD BE CONSIDERED AS ONLY APPROXIMATE. A CORRECTION FOR LOCAL RIM ROTATION WAS NOT APPLIED BECAUSE OF INSUFFICIENT DATA.

2. SEE TABLE 6 AND FIGURE 12 FOR DEFLECTION DATA AND APPLIED LOADS, RESPECTIVELY.

3. Thrust Alignment

The deflection of the structure (Table 6) for the 12 loading conditions can be used in evaluating the effect of this engine component on possible thrust misalignment.

4. Design Improvement

Although the LTS withstood all loads satisfactorily, it would be desirable to control the mismatch and distortion at the weld joints, thereby minimizing discontinuity stresses and possible permanent deformation. Also, butt joints should be used in place of lap joints to eliminate the built-in moment. To accomplish these steps improved tooling should be considered.

Sizing of the actuator support members was satisfactory as the loads were well distributed. As noted in the data tabulation of Table 6 and in the schematic LTS distortion shown in Figure 29 of the frustrum-cylinder flange is significantly influenced by the actuator support loads. Because of a lack of rotation data at the location it is not possible to completely evaluate the influence of this juncture. Stiffening of the juncture should be considered if reduction of actuator yoke deformation is desired.

With a stress concentration factor of 1 the use of rim-type reinforcements around the circular holes proved quite satisfactory and should be extended to the other holes. As distortion occurred locally during welding, improved tooling and stress-relieving is desirable.

5. Theoretical Analysis

A considerable amount of experimental data was obtained for various static loading conditions simulating expected NERVA flight loads. A comparison with theoretical data was not made since - because of program redirection - the analyses for the testing conditions were not completed.

B. CONCLUSIONS

The major objective, to demonstrate the structural adequacy of the lower thrust structure, was successfully accomplished. With improved fabrication to minimize the defects discussed the structure should be capable of withstanding a 10% to 20% increase over those critical loads imposed during this program.

Axial stiffness of the cylinder and frustrum assembly was determined. However, lateral stiffness of this assembly and stiffness, both axial and lateral, of the actuator supports were not determined because of load complexity.

Based on the experimental data, the part was evaluated for design and fabrication improvements and suggestions were made.

The attainment of LTS deflection data for thrust misalignment analyses was completed.

A correlation with theoretical analysis was not performed, as an analysis of the test conditions was not available.