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REPORT NO. RN-S-0338

TO

AEC-NASA SPACE NUCLEAR PROPULSION OFFICE

STRUCTURAL ANALYSIS OF THE
XE-1 PRESSURE VESSEL COMPONENTS AND ASSEMBLY

FEBRUARY 1967

NERVA PROGRAM

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ROCKET ENGINE OPERATIONS - NUCLEAR

FEBRUARY 1967

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STRUCTURAL ANALYSIS OF THE
XE-1 PRESSURE VESSEL COMPONENTS AND ASSEMBLY

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ABSTRACT

This document reports a structural analysis conducted on components of the XE pressure vessel assembly. These components are defined as the forward closure, cylinder, nozzle (including the reactor core-support cylinder), and bolts which connect the forward-closure and nozzle-jacket flanges to those of the cylinder. Analyses indicate all components have positive margins of safety and meet the requirements of the structures design criteria.

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I. SUMMARY

The XE-1 assembly is a downward-firing nuclear rocket technology engine.

The pressure vessel assembly consists of the forward closure, cylinder, nozzle (including the reactor core-support cylinder) and bolts for connecting the flanges of the forward closure and nozzle jacket to the cylinder flanges. The design of the assembly and its major structural components is documented by Aerojet-General engineering drawings, References 1a through 1f.

This report presents and summarizes a structural analysis of these components for the nuclear-firing condition. Analyses of the ambient-temperature proof hydrotest for the pressure vessel (with the nozzle replaced by a rigid closure plate) and for the nozzle (with the pressure vessel replaced by an adapter) are also included. The analyses indicate that all components have positive margins of safety and meet the requirements of the structures design criteria (see Reference 2).

The nozzle is included in this analysis to obtain its influence on the pressure vessel during operating condition. The nozzle hydrotest case was also included to provide a complete analysis for the nozzle as well as the pressure vessel. The minimum margins of safety for the various components are summarized below. The hydrotest condition, except in the case of the forward closure bolts, was found to be critical in each case.

<u>Component</u>	<u>Allowable Stress (psi)</u>	<u>Maximum Stress (psi)</u>	<u>M.S.</u>
Forward Closure	35,000	30,400	+0.00 *
Forward Closure Bolts	180,000	171,700	+0.05
Cylinder	35,000	28,200	+0.08 *
Nozzle	30,000	21,400	+0.22 *
Nozzle Bolts	180,000	161,300	+0.11

Margins of safety for various stations of the several components are summarized in Tables 1 and 2.

* Includes 1.15 safety factor

II. STRUCTURAL DESCRIPTION

The forward closure is basically a semi-ellipsoid of 2:1 ellipticity with a semi-major axis of 24.78 in. to the inside surface.

A trapezoidal cross-section flange at the equator provides a structure for bolting to the pressure-vessel cylinder flange. At a radius of 17.75 in., there is a smaller flange used in earlier models to transmit engine thrust to test facility structure. In the XE design, the flange has been retained as a convenient mounting platform for a nuclear detector cannister. Thrust is now transmitted from the pressure vessel to the lower thrust structure by 36 radially oriented shear bolts connecting the forward closure flange to the thrust structure.

At a radius of 22.38 in., there are 24 bosses for connecting various parts of the engine system to the pressure vessel. Seven of these ports (each of 2.50-in. inner diameter) are internally threaded for instrumentation connection. Twelve ports (also with 2.50-in. inner diameters) have Marman flanges for reactor control-rod guide-tube attachment. One internally threaded port is used to connect the coolant return from the neutron detector cannister. One flanged port of 3.26-in. inner diameter is for the diluent line connection. Three bosses are left blank.

The closure shell is 0.67-in. thick from the dome apex to the smaller 17.75-in. radius flange and 1.00-in. thick from there to the closure flange. The material used is 6061 aluminum alloy forging, heat-treated to the T6 condition.

The pressure vessel cylinder consists of a trapezoidal cross-section forward flange, a cylindrical shell, a torispherical shell, and a rectangular cross-section aft flange. The toroidal shell provides convergence of the vessel from the basic cylinder inner diameter of 49.75-in. to 42.00-in. at the

bolt-circle diameter of the aft flange. The basic cylinder thickness is 0.67-in., but 0.78-in. sections are provided to permit welding together of the three forged segments without loss of strength. As in the forward closure, the material is 6061 aluminum alloy forging heat-treated to the T6 condition.

The nozzle jacket consists of a rectangular cross-section flange for bolting to the pressure-vessel cylinder flange, convergent and divergent conical shells joined by a short toroidal shell at the nozzle throat, and a cylindrical shell with a trapezoidal cross-section flange at the forward end for reactor core support. The divergent nozzle-cone ends in a trapezoidal cross-section flange to which the nozzle fuel-torus is welded. Basic inner diameters of the core-support cylinder, nozzle throat and exit plane are 36.62, 9.32 and 28.75 in., respectively. Thickness of the core-support cylinder is 0.35-in. and of the divergent cone and toroidal throat, 0.57-in. In the convergent cone, the thickness tapers from 0.61-in. at the throat to approximately 1.00-in. at the bolting flange. Material of the nozzle jacket is annealed 347 corrosion resistant steel.

The inside surface of the jacket is completely covered by coolant tubes which are of semi-annular cross-section and are brazed to the jacket. Although the tubes contribute somewhat to the meridional strength and stiffness of the shell, the tubes are generally of such slight thickness that their contribution to nozzle hoop stiffness is negligible. Therefore (also because the computer program used requires completely axisymmetrical structure), the effect of the tubes has not been included in the analysis. The tubes themselves, however, are evaluated separately, in a noncomputerized analysis, for primary stress and cyclic life.

The closure is connected to the cylinder by 120 5/8-in. reduced shank bolts at a bolt-circle diameter of 52.00-in. The nozzle jacket is connected to the cylinder by 96 5/8-in. cooled, reduced shank bolts at a bolt-circle diameter of 42.00 in. Bolts are of high-strength A286 material.

III. DISCUSSION OF PRESSURE VESSEL LOADS AND ENVIRONMENT

The primary load exerted on the pressure vessel is internal pressure. In addition, in the firing condition there is a reaction equal and opposite to the thrust at the forward-closure connection to the lower thrust structure. There are also reactor loads applied to the core-support flange and forward-closure joint. These loads are illustrated by the free-body diagrams of Figures 1 and 2. Figure 3 shows the pressure profile in the nozzle chamber.

In addition to the primary mechanical loads, the entire pressure-vessel assembly is subjected to thermal loading due to the complete pressure vessel wall being "scrubbed" by cryogenic fluid while being subjected to nuclear radiation. Thermal gradients in both the radial (across the thickness) and axial directions result. A complete description of these gradients, which were used as input in the stress analysis, is provided by the heat transfer analysis, Reference 7.

Some additional secondary loadings (e.g., the line loads at the diluent line or hot bleed port attachment) are treated in other reports.

Figure 1
Free-Body Diagrams for Hydrotest Conditions

Figure 2
 Free-Body Diagram for Nuclear Firing Test Condition

Figure 3
Nozzle Pressure Profile

IV. DISCUSSION OF METHOD OF ANALYSIS

The XE pressure vessel is analyzed with a computerized finite-element method which determines displacements and stresses within axisymmetric solids (Reference 8). The continuous body (closure or cylinder) is replaced by a system of elements with triangular or quadrilateral cross sections. The points at the intersecting sides of an element are called nodal points and are positioned by radial and axial coordinates. Displacements are calculated for the nodal points and stresses are calculated for the elements.

The pressure vessel is analyzed for two conditions: (1) hydrotest; and (2) static nuclear firing test. The hydrotest condition is discussed first.

The XE pressure vessel is proof hydrotested to 740 psig in the attitude and configuration shown in Figure 4.

Figure 4

Pressure Vessel Proof Test Configuration and Attitude

The aft ring of the cylinder is bolted to a rigid hydrotest plate and is considered fixed. The pressure vessel components are theoretically separated at the bolted interfaces and each component is analyzed separately. The interactions are accommodated by applying discontinuity moments and shears, as well as axial loads, to the bolted face of each component so the compatibility of the entire structure is maintained. Since the aft cylinder closure is considered rigid in the hydrotest condition, zero radial and axial displacement of the ring is specified. The discontinuities must be determined to obtain bolt loads.

The forward joint compatibility must also be established. The applied axial loads and discontinuities are shown in Figure 5.

Figure 5

Forward Joint Free-Body Diagram

The values of the discontinuity moment, M_1 , and shear, Q_1 , are determined by applying chamber pressure, unit moment, and unit shear to both the closure and the cylinder to obtain influence coefficients for each loading condition for each structural component. From these six computer runs, equations for rotation and radial deflection are written for each pressure vessel component. These equations (in terms of the unknown discontinuities, influence coefficients, and pressure) are of the following form:

1. Cylinder radial deflection at the cylinder to closure interface:

$$\omega_{\text{cyl.}} = K_1 M_1 + K_2 Q_1 + K_3.$$

2. Cylinder rotation at the interface:

$$\theta_{\text{cyl.}} = K_4 M_1 + K_5 Q_1 + K_6.$$

3. Closure radial deflection at the interface:

$$\omega_{\text{clos.}} = K_7 M_1 + K_8 Q_1 + K_9.$$

4. Closure rotation at the interface:

$$\theta_{\text{clos.}} = K_{10} M_1 + K_{11} Q_1 + K_{12}.$$

Where:

ω	=	radial deflection,
θ	=	rotation,
Q_1	=	discontinuity shear,
M_1	=	discontinuity moment,
K's	=	influence coefficients,
N_1	=	bolt ejection load, and
$N_{2,3}$	=	shell membrane loads.

The unknown discontinuities are then determined by equating the radial deflection and rotation of the cylinder to the radial deflection and rotation of the closure:

$$(i.e., \omega_{cylinder} = \omega_{closure}),$$

$$K_1 M_1 + K_2 Q_1 + K_3 = K_7 M_1 + K_8 Q_1 + K_9; \text{ and } (\theta_{cylinder} = \theta_{closure}),$$

$$K_4 M_1 + K_5 Q_1 + K_6 = K_{10} M_1 + K_{11} Q_1 + K_{12}.$$

Simultaneous solution of the two equations above yields values for M_1 and Q_1 .

The moment and shear discontinuities and pressure loading are now applied to the pressure vessel components for a complete solution.

The method of analysis for the nozzle hydrotest condition is the same as described above. The adapter is designed to simulate the pressure vessel and, therefore, will have the same influence on the nozzle as that of the pressure vessel cylinder.

The static nuclear firing condition is slightly more involved since it requires consideration of a varying thermal environment of the pressure vessel structure which causes varying material properties of the structural components, as well as additional loading. The analysis for this condition is more extensive than that of the hydrotest condition, but the method is the same.

V. CRITERIA

Factors of safety used in the analysis are from Basic Loads and Structural Design Criteria for the XE-1 Engine, Reference 2. The factors are 1.15 and 1.25 on the yield and ultimate strengths, respectively. These values are essentially in agreement with SNPO-C criteria, Reference 9, which, in turn, are based on Section III of the ASME Boiler and Pressure Vessel Code, Reference 10. In the operating condition, limit load (i.e., M.E.O.P.) is taken as 1.05 times nominal pressure. It should be noted, however, that all the firing condition stress graphs are based on nominal pressure.

Analytical treatment with special techniques (e.g., the evaluation of cyclic damage due to thermal stress "ratcheting") follows the criteria of Reference 9.

The basis for the selection of consistent and conservative material properties is described in The Material Properties Data Book, Reference 11, from which all allowables and other mechanical properties were taken.

VI. ANALYSIS

A. STRUCTURAL MODELS

1. Pressure Vessel Hydrotest Condition

a. Structural Model

The structural model for the hydrotest condition consists of the forward closure and cylindrical sections of the pressure vessel. As discussed in Section III, the vessel is separated at the forward joint and free-bodied as shown in Figure 6 for input into the computerized finite element analysis.

Pressure Vessel Final Hydrotest Free-Body

The above free-body diagram shows the case for the final computer run with previously determined discontinuities applied.

b. Example of Geometric Model

An example of the geometric models used is shown in Figure 7, which represents in full scale the flange portion of the forward

NOTE: PLAIN NUMBERS INDICATE NODAL POINTS
WHILE THE UNDERLINED NUMBERS IDENTIFY
ELEMENTS.

Figure 7
Example of Geometric Model

closure. The plain numbers indicate nodal points, while underlined numbers identify elements. The complete geometric model is presented as Figures 8 and 9 (pages 16 and 17) which are the computer plots of the geometry to scale.

c. Example of Computer Input

The nodal points are positioned by R and Z coordinates. Temperatures and loads may be applied at nodal points while pressures are applied over element surfaces. Figure 10 presents an example for an imaginary element (No. 1) defined by nodal points 1 through 4.

Figure 10

Example of Computer Input

Figure 8
Forward Closure Geometry

Figure 9
Cylinder Geometry

2. Nozzle Hydrotest Condition

The final free-body diagram for this condition is shown in Figure 11.

Figure 11

Nozzle Final Hydrotest Free-Body

Since the adapter is designed to simulate the pressure vessel, the nozzle will be made theoretically compatible with the pressure vessel cylinder as discussed in Section IV. The finite element geometric model for the nozzle is shown in Figure 12.

3. Static Nuclear Firing Condition

The final free-body diagram for this condition includes the nozzle as shown in Figure 13.

Figure 13

Final Nuclear Firing Test Free-Body

Figure 12
Nozzle Geometry

The finite element geometric model for the forward closure and cylindrical sections is the same as that of the pressure-vessel hydrotest condition. The nozzle shell finite-element geometric model is the same as that of the nozzle hydrotest condition.

B. LOADS

1. Pressure Vessel Hydrotest Condition

The load applied to the pressure vessel in the ambient-temperature proof hydrotest condition is an internal single-zone chamber pressure of 740 psig, Reference 3.

2. Nozzle Hydrotest Condition

The load applied to the nozzle in the ambient-temperature proof hydrotest condition is an internal single zone pressure of 810 psig, (Reference 1h).

3. Static Nuclear Firing Condition

The loads during firing (as shown in Figure 2) are internal chamber and nozzle pressures, thrust reaction, reactor loads, and induced thermal loads from the environment reported in Reference 7. The engine thrust is reacted at the lower-thrust-structure attachment to the forward-closure ring.

C. MECHANICAL PROPERTIES

Material properties for the various components at required temperatures are taken from Reference 11, except as noted. Tensile strengths at ambient temperature are as follows:

Component	Material	Spec	Allowable Tensile Strengths at 70°F	
			F _{ty} PSI	F _{tu} PSI
Forward Closure and Cylinder	6061-T6	QQ-A-367	35,000	42,000
Nozzle	CRES 347 Annealed	QQ-S-763	30,000	75,000
Closure Bolts	A286	AMS 5735	180,000 (Ref 1e & 1f)	200,000 (Ref 1e & 1f)

D. ANALYTICAL RESULTS

1. Pressure Vessel Hydrotest Condition

a. Stresses and Deformation

The results of the influence computer runs yield the following displacement equations for the forward joint:

(1) Cylinder radial deflection at the cylinder to closure interface;

$$(10)^6 \omega_{cyl.} = 4.872 M_1 - 19.803 Q_1 - 10,402$$

(2) Cylinder rotation at the interface:

$$(10)^6 \theta_{cyl.} = -2.441 M_1 + 4.711 Q_1 + 14,460.$$

(3) Closure radial deflection at the interface;

$$(10)^6 \omega_{cyl.} = 3.725 M_1 + 15.348 Q_1 - 50,014.$$

(4) Closure rotation at the interface;

$$(10)^6 \theta_{\text{clos.}} = 1.848 M_1 + 3.623 Q_1 - 8,713.$$

Simultaneous solution of the equations (obtained by equating deflections and rotations) yields the following values for the discontinuities:

$$\begin{aligned} M_1 &= 5,736 \text{ in.-lb/in.}, \text{ and} \\ Q_1 &= 1,314 \text{ lb/in.} \end{aligned}$$

The results of the influence computer runs yield the following displacement equations for the aft joint:

(1) Cylinder radial deflection at the aft interface;

$$(10)^6 (\omega_{\text{cyl.}}) = 4.488 M_2 - 14.678 Q_2 + 102,572.$$

(2) Cylinder rotation at the aft interface;

$$(10)^6 (\theta_{\text{cyl.}}) = 2.798 M_2 - 4.260 Q_2 + 31,227.$$

Since both rotation and deflection are considered zero, the above two equations are set equal to zero and the solution yields the following values for the discontinuities:

$$M_2 = -975 \text{ in.-lb/in.}, Q_2 = 6690 \text{ lb/in.}$$

The above discontinuities (when applied to the pressure vessel components) yield a complete solution. Hoop and meridional stresses for the inside and outside surfaces of the pressure vessel components are shown in graphical form in Figures 14 through 17.

Figure 14
Forward Closure Hydrotest Meridional Stress

Figure 15
Forward Closure Hydrotest Hoop Stress

Figure 16
Cylinder Hydrotest Meridional Stress

Figure 17

b. Bolt Loads

(1) Forward Joint

The total external bolt load is due to ejection load, plus added bolt tension caused by discontinuity moment:

$$P_{EXT.} = P_{EJ.} + P_M,$$
$$P_{EJ.} = \frac{p_c \pi (R_{seal})^2}{N} = \frac{(740) \pi (26.796)^2}{120},$$

$$P_{EJ.} = 13,910 \text{ lb/bolt.}$$

$$P_M = \frac{M}{e} \times \frac{2 \pi R_{BC}}{N} = \frac{5736}{1.13} \times \frac{2\pi (26.0)}{120},$$

$$P_M = 6,920 \text{ lb/bolt.}$$

Total external bolt load is:

$$P_{EXT.} = 13,910 + 6,920 = 20,830 \text{ lb/bolt,}$$

Preload, $P_{i_{max}} = 33,800 \text{ lb}$, Reference (12), due to 150 ft-lb pre-torque, Reference (3).

The preload of 33,800 lb/bolt has been obtained by 200 ft-lb of torque. See Reference (12) for preload versus torque relations. However, due to a change to Teflon washers, it is found that only 150 ft-lb of torque is required to obtain the 33,800 lb/bolt preload. Therefore, the pre-torque requirement has been lowered.

Total bolt load:

$$\begin{aligned} P_B &= P_i + K P_E = 33,800 + 0.151 * (20,830), \\ &= 36,950 \text{ lb/bolt.} \end{aligned}$$

Bolt Minimum Cross Sectional
Area (Reference lg):

$$A = \pi (0.525)^2 / 4 = 0.2165 \text{ in.}^2$$

Bolt Maximum Stress:

$$\sigma = P_B / A = 36,950 / 0.2165 = 170,800, \text{ psi}$$

2. Nozzle Hydrotest Condition

a. Stresses and Deformations

The results of the influence computer runs yield the following displacement equations for the nozzle joint:

(1) Adapter radial deflection at the adapter-to-nozzle interface;

$$(10)^6 (\omega_{\text{adp.}}) = 4.488 M_2 - 14.678 Q_2 + 112,265.$$

*See Appendix A

(2) Adapter rotation at the adapter-to-nozzle interface;

$$(10)^6 (\theta_{\text{adp.}}) = 2.798 M_2 - 4.260 Q_2 + 34,178.$$

(3) Nozzle radial deflection at the adapter-to-nozzle interface;

$$(10)^6 (\omega_{\text{noz.}}) = 0.524 M_2 + 1.621 Q_2 - 9,304.$$

(4) Nozzle rotation at the adapter-to-nozzle interface;

$$(10)^6 (\theta_{\text{noz.}}) = -0.559 M_2 - 0.484 Q_2 + 5,544.$$

Simultaneous solution of the equations (obtained by equating deflections and rotations) yields the following values for the discontinuities:

$$M_2 = -192 \text{ in-lb/in.} \quad Q_2 = 7412 \text{ lb/in.}$$

The above discontinuities when applied to the nozzle yield a complete solution. Hoop and meridional stresses for the inside and outside surfaces of the nozzle shell are shown in graphical form in Figures 18 and 19.

b. Bolt Loads

The total external bolt load is due to ejection load, plus added bolt tension because of discontinuity moment:

$$\begin{aligned} P_{\text{EXT}} &= \frac{p_c \pi (R_{\text{seal}})^2}{N} + \frac{M_2}{e} \left[\frac{(2\pi R_{\text{B.C.}})}{N} \right], \\ &= \frac{810\pi (21.667)^2}{96} + \frac{192}{1.0} \left[\frac{2\pi (21.0)}{96} \right], \end{aligned}$$

Figure 18
Nozzle Hydrotest Meridional Stress

Figure 19
Nozzle Hydrotest Hoop Stress

$$P_{EXT.} = 12,444 + 264 = 12,708 \text{ lb/bolt.}$$

Preload, $P_i = 34,000 \text{ lb}$, Reference (12), due to 2420 in-lb torque, Reference (1h).

$$\begin{aligned} \text{Total bolt load, } P_b &= P_i + KP_{EXT.} = 34,000 + 0.074 * (12,708), \\ &= 34,940 \text{ lb/bolt.} \end{aligned}$$

$$\text{Maximum Bolt Stress, } \sigma_b = P_b/A = 34,940/.2165 = 161,300 \text{ psi.}$$

3. Maximum Stresses and Minimum Margins of Safety for Pressure Vessel and Nozzle Proof Hydrotest Conditions

Table 1 lists the allowable and maximum stresses for the forward closure, pressure vessel cylinder, forward closure bolts, nozzle, and nozzle bolts. The minimum margin of safety for each component is also given in Table 1.

4. Static Nuclear Firing Condition

a. Stresses and Deformations

The results of the influence computer runs yield the following displacement equations for the forward and aft joints (in each of the bracketed pairs of values for the constant terms, the upper value represents the effect of pressure, the lower represents the effect of thermal loading):

(1) Cylinder radial deflection at the cylinder-to-closure interface;

*See Appendix A

TABLE 1

Allowable and Maximum Stress: Minimum Margin of Safety
Proof Hydrotest Conditions

<u>Item</u>	<u>Ident</u>	<u>Material</u>	<u>Temp,</u> <u>°F</u>	<u>Allowable</u> <u>Stress, psi</u> <u>Ref 11</u> <u>Except as Noted</u>	<u>Max Stress</u> <u>Value, Ref,</u> <u>psi Page No</u>	<u>M.S. =</u> <u>$\frac{F_{ty}}{1.15\sigma_{max}}$</u>	<u>-1</u>
Forward Closure	Ref 1b	6061-T6	70	35,000	30,400 23	+0.00	
Forward Closure Bolts	Ref 1g	A286	70	(Ref 1g) 180,000	170,800 28	+0.05*	
P.V. Cylinder	Ref 1c	6061-T6	70	35,000	28,200 26	+0.08	
Nozzle	Ref 1d	CRES 347 Annealed	70	30,000	21,400 31	+0.22	
Nozzle Bolts	Ref 1i	A286	70	(Ref 1i) 180,000	161,300 32	+0.11*	

*Does not include 1.15 safety factor.

$$(10)^6 \omega_{cyl} = 4.480 M_1 - 18.211 Q_1 + \begin{Bmatrix} -7,002 \\ -94,532 \end{Bmatrix} \begin{matrix} \text{Mech} \\ \text{Thermal} \end{matrix} .$$

(2) Cylinder rotation at the cylinder-to-closure interface;

$$(10)^6 \theta_{cyl} = -2.246 M_1 + 4.332 Q_1 + \begin{Bmatrix} +12,682 \\ +207 \end{Bmatrix} .$$

(3) Closure radial deflection at the cylinder-to-closure interface;

$$(10)^6 \omega_{clos} = 3.435 M_1 + 14.152 Q_1 + \begin{Bmatrix} -44,624 \\ -93,280 \end{Bmatrix} .$$

(4) Closure rotation at the cylinder-to-closure interface;

$$(10)^6 \theta_{clos} = 1.704 M_1 + 3.342 Q_1 + \begin{Bmatrix} -8,256 \\ -268 \end{Bmatrix} .$$

(5) Cylinder radial deflection at the cylinder-to-nozzle interface;

$$(10)^6 \omega_{cyl} = 4.071 M_2 - 13.590 Q_2 + \begin{Bmatrix} +78,860 \\ -80,200 \end{Bmatrix} .$$

(6) Cylinder rotation at the cylinder-to-nozzle interface;

$$(10)^6 \theta_{cyl} = 2.531 M_2 - 3.955 Q_2 + \begin{Bmatrix} +22,056 \\ -1,311 \end{Bmatrix} .$$

(7) Nozzle radial deflection at the cylinder-to-nozzle interface;

$$(10)^6 \omega_{\text{noz}} = 0.476 M_2 + 1.506 Q_2 + \begin{Bmatrix} -7,134 \\ -53,104 \end{Bmatrix}.$$

(8) Nozzle rotation at the cylinder-to-nozzle interface;

$$(10)^6 \theta_{\text{noz}} = -0.508 M_2 - 0.452 Q_2 + \begin{Bmatrix} +4,226 \\ +4,790 \end{Bmatrix}.$$

Cylinder and closure displacements and rotations, and cylinder and nozzle displacements and rotations, above, are equated. Solution of the systems of simultaneous equations yields the following values for the discontinuities:

	Mechanical	Thermal
M_1 (in-lb/in)	5636	111
Q_1 (lb/in)	1345	-35
M_2 (in-lb/in)	964	-84
Q_2 (lb/in)	5926	-1815

Hoop and meridional stresses for the inside and outside surfaces of the pressure vessel components and nozzle shell are shown in Figures 20 through 33. It should be noted that the stress profiles are based on nominal stress whereas the M.S. calculation takes the limit load (or M.E.O.P.) as 1.05 times nominal pressure.

Figure 20
Forward Closure Nuclear Firing Test Total Meridional Stress

Figure 21
Forward Closure Nuclear Firing Test Total Hoop Stress

Figure 22
Cylinder Nuclear Firing Test Mechanical Meridional Stress

Figure 23

Cylinder Nuclear Firing Test Thermal Meridional Stress

Figure 24
Cylinder Nuclear Firing Test Total Meridional Stress

Figure 25
Cylinder Nuclear Firing Test Mechanical Hoop Stress
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Figure 26

Cylinder Nuclear Firing Test Thermal Hoop Stress

Figure 27
Cylinder Nuclear Firing Test Total Hoop Stress

Figure 28
Nozzle Nuclear Firing Test Mechanical Meridional Stress

Figure 29
Nozzle Nuclear Firing Test Thermal Meridional Stress

Figure 30
Nozzle Nuclear Firing Test Total Meridional Stress

Figure 31
Nozzle Nuclear Firing Test Mechanical Hoop Stress

Figure 32
Nozzle Nuclear Firing Test Thermal Hoop Stress

Figure 33
Nozzle Nuclear Firing Test Total Hoop Stress

b. Bolt Loads

The total external bolt-load is due to ejection load, plus added bolt tension because of discontinuity moments. The discontinuity moment is due to mechanical plus thermal discontinuity.

(1) Forward Joint

(a) Mechanical External Bolt Load

$$\begin{aligned} P_{\text{EXT}_{\text{MECH}}} &= \frac{p \pi (R_{\text{seal}})^2}{N} + \frac{M_{\text{LM}}}{e} \left(\frac{2\pi R_{\text{B.C.}}}{N} \right) - \frac{\text{Thrust}}{N} \\ &= \frac{691\pi(26.796)^2}{120} + \frac{5636}{1.13} \left[\frac{2\pi(26.0)}{120} \right] - \\ &\quad \frac{54,945}{120} \\ &= 12,990 + 6,790 - 458, \\ &= 19,322 \text{ lb/bolt.} \end{aligned}$$

(b) Thermal External Bolt Load

$$P_{EXT. Therm} = \frac{M_{IT}}{e} \left(\frac{2\pi R_{B.C.}}{N} \right) = \frac{111}{1.13} \times \frac{2\pi(26.0)}{120} = 134 \text{ lb/bolt}$$

(c) Total Bolt Load

$$\text{Total External Load} = P_{ext. mech.} + P_{ext. therm)}$$

$$= 19,322 + 134 = 19,465 \text{ lb/bolt.}$$

See Appendix A for the following method of bolt load analysis:

	$T(\text{Ref. 7})$ °F	ΔT °F	$E(\text{Ref. 11})$ psi	$\alpha(\text{Ref. 11})$ in/in-°F
Flange	-235	-305	$10.95(10)^6$	$11.0(10)^{-6}$
Bolt	-7	-77	$29.05(10)^6$	$8.6(10)^{-6}$

$$(\Delta P_e + \Delta P_c) = \text{total external bolt load} = 19,456 \text{ lb/bolt.}$$

Change in flange strain:

$$\Delta \epsilon_F = \frac{E_B A_B (\alpha_B \Delta T_B - \alpha_F \Delta T_F) + (\Delta P_e + \Delta P_c)}{E_B A_B + E_F A_F}$$

$$= \frac{29.05(.2165) [8.6 (-77) - 11 (-305)] + 19,456}{(29.05)(10)^6 (.2165) + 10.95(10)^6 (3.28)*}$$

$$\Delta \epsilon_F = \frac{36,356}{42.2} (10)^{-6} = 862 \mu \text{ in/in.}$$

Change in flange load at bolt-circle diameter:

$$\Delta P_F = \epsilon_F E_F A_F = 862 (10.95)(3.28) = (30,980) \text{ lb.}$$

* See Appendix A

Nominal preload is 31,400 lb, therefore, sealing will be achieved.

Change in bolt strain:

$$\Delta \epsilon_B = \frac{(\Delta P_e + \Delta P_c) - \Delta P_F}{E_B A_B}$$

$$\Delta \epsilon_B = \frac{19,456 - 30,980}{6.3(10)^6} = \frac{-11,524}{6.3(10)^6} = -1830 \mu \text{ in/in.}$$

Change in bolt load:

$$\Delta P_B = \Delta \epsilon_B E_B A_B = -1830 (6.3) = -11,524 \text{ lb.}$$

Total Bolt Load:

$$P_B = P_i + \Delta P_B,$$

$$P_i = 37,200 \text{ lb/bolt (Reference 12, due to pre-torque of 220 ft lb, (Reference 1a),}$$

$$P_B = 37,200 + (-11,524) = 25,676 \text{ lb/bolt.}$$

Bolt Stress: maximum stress is due to ambient temperature preload, or

$$\sigma = \frac{37,200}{.2165} = 171,700 \text{ psi.}$$

(2) Aft Joint

(a) Mechanical External Bolt Load

$$P_{\text{EXT. MECH.}} = \text{Ejection Load } (P_{ej}) + \text{Bending Load } (P_m),$$

$$\begin{aligned}
 P_{ej} &= \frac{1}{N} \{ p_{\text{fwd head}} \pi (R_{\text{F.S.}})^2 - \text{Thrust} - P_{\text{FF}} \\
 &\quad - p_c \pi [(R_{\text{F.S.}})^2 - (R_{\text{A.S.}})^2] \}, \\
 &= \frac{1,558,718 - 54,945 - 58,800 - 733 \pi [(26,796)^2 - (21.667)^2]}{96},
 \end{aligned}$$

$$P_{ej} = \frac{872,577}{96} = 9,090 \text{ lb/bolt.}$$

$$P_m = \frac{M_{2M}(2\pi R_{\text{B.C.}})}{eN} = \frac{964}{1.0} \left[\frac{2\pi(21.0)}{96} \right] = 1326 \text{ lb/bolt.}$$

$$P_{\text{EXT. MECH.}} = 9090 + 1326 = 10,416 \text{ lb/bolt.}$$

(b) Thermal External Bolt Load

$$P_{\text{EXT. Therm}} = \frac{M_{2T}}{e} \left(\frac{2\pi R_{\text{B.C.}}}{N} \right),$$

$$= \frac{-84}{1.0} \left[\frac{2\pi(21.0)}{96} \right] = -130 \text{ lb/bolt.}$$

(c) Total Bolt Load

Total external bolt load $(\Delta P_c + \Delta P_c)$

$$= P_{\text{EXT}} = 10,416 - 130 = 10,286 \text{ lb/bolt.}$$

	T	ΔT	E	α
	$^{\circ}\text{F}$	$^{\circ}\text{F}$	psi	in./in. $^{\circ}\text{F}$
Flange	-300	-370	$31.9(10)^6$	$7.35(10)^{-6}$
Bolt	-360	-430	$29.6(10)^6$	$7.0(10)^{-6}$

Change in flange strain:

$$\Delta \epsilon_F = \frac{E_B A_B (\alpha_B \Delta T_B - \alpha_F \Delta T_F) + (\Delta P_e + \Delta P_c)}{E_B A_B + E_F A_F^*}$$

$$= \frac{29.6(10)^6 (0.2165) [7(10)^{-6} (-430) - 7.35(10)^{-6} (-370)] + 10,286}{29.6(10)^6 (0.2165) + 31.9(10)^6 (2.72)^*}$$

$$\Delta \epsilon_F = 90.5 \mu \text{ in./in.}$$

Change in flange load:

$$\Delta P_F = \Delta \epsilon_F E_F A_F = (90.5) (10)^{-6} (31.9) (10)^6 (2.72),$$

$$\Delta P_F = 7826 \text{ lb.}$$

Change in bolt strain:

$$\Delta \epsilon_B = \frac{(\Delta P_e + \Delta P_c) - \Delta P_F}{E_B A_B} = \frac{10,286 - (7826)}{(6.41)(10)^6},$$

$$\Delta \epsilon_B = 384 \mu \text{ in./in.}$$

Change in bolt load:

$$\Delta P_B = \Delta \epsilon_B E_B A_B,$$

$$= 384(10)^{-6} (6.41) (10)^6 = 2460 \text{ lb/bolt.}$$

* See Appendix A

Total bolt load

$$P_B = P_i + \Delta P_B,$$

$$P_i = 35,500 \text{ lb/bolt (Reference 12) due to maximum pre-torque of 210 ft/lb (Reference 1a),}$$

$$\therefore P_B = 35,500 + 2,460 = 37,960 \text{ lb/bolt}$$

Bolt stress during operation:

$$\sigma = \frac{37,960}{0.2165} = 175,000 \text{ psi;}$$

$$\text{M.S.} = \frac{225,000}{1.05(1.15)(175,000)} - 1 = + 0.06.$$

Bolt stress due to preload:

$$\sigma = \frac{35,500}{0.2165} = 164,000 \text{ psi;}$$

$$\text{M.S.} = \frac{180,000}{164,000} - 1 = + 0.10$$

5. Maximum Stresses and Minimum Margins of Safety for Nuclear Firing Condition

Table 2 lists the allowable and maximum stresses for the forward closure, pressure vessel cylinder, forward closure bolts, nozzle, and nozzle bolts. The minimum margin of safety for each component is also given in Table 2.

6. Nozzle Fatigue Life

a. Nozzle Jacket

The maximum nozzle shell stress occurs in nozzle element #393 (see figure 11).

The Principal Stresses are:

TABLE 2

Allowable and Maximum Stress: Minimum Margin of Safety
Nuclear Firing Condition

<u>Item</u>	<u>Ident</u>	<u>Material</u>	<u>Temp, °F</u>	<u>Allowable Stress, psi Ref 11 Except as Noted</u>	<u>Max Stress</u>		<u>M.S. = F_{ty} (1.05*) 1.15σ⁻¹</u>	
					<u>Value, psi</u>	<u>Ref Page No.</u>		
Forward Closure	Ref 1b	6061-T6	-235	41,000	28,600	36	+0.18	
Forward Closure Bolts	Ref 1e	A286	15	(Ref 1g) 180,000	171,700	52	+0.05**	
P.V. Cylinder	Ref 1c	6061-T6	-226	40,500	28,000	41	+0.19	
Nozzle Bolts	Ref 1f	A286	70	(Ref 13) 180,000	164,000	55	+0.10**	
Nozzle	Ref 1d	CRES 347 Annealed	35	31,000	15,400		+0.66	

* The 1.05 factor is used to convert nominal pressure to M.E.O.P.
The maximum stress levels in the preceding column are based on nominal pressures.

** These values do not include the 1.15 safety factor nor the 1.05 factor.
Maximum stresses are from the preload condition.

$$\sigma_1 = \sigma_m = \sigma_{ij_{\text{Thermal}}} + \sigma_{ij_{\text{Mech.}}},$$

$$= 75,000 + 5,460,$$

$$\therefore \sigma_1 = 80,460 \text{ psi.}$$

$$\sigma_2 = \sigma_{jk_T} + \sigma_{jk_M} = 3,650 + (-50),$$

$$\therefore \sigma_2 = 3,600 \text{ psi.}$$

$$\sigma_3 = \sigma_h = \sigma_{T_T} + \sigma_{T_M} = 76,213 + 15,274,$$

$$\therefore \sigma_3 = 91,487 \text{ psi.}$$

Stress differences are:

$$s_{12} = \sigma_1 - \sigma_2 = 80,460 - 3,600 = 76,860 = s_{r12}.$$

$$s_{23} = \sigma_2 - \sigma_3 = 3,600 - 91,487 = |-87,887| = s_{r23}.$$

$$s_{31} = \sigma_3 - \sigma_1 = 91,487 - 80,460 = 11,027 = s_{r31}.$$

The alternating stress intensity is:

$$s_{\text{alt. } ij} = 0.5 s_{ri j}.$$

$$s_{\text{alt}_{12}} = 0.5 (76,860) = 38,430 \text{ psi.}$$

$$s_{\text{alt}_{23}} = 0.5 (87,887) = 43,944 \text{ psi.}$$

$$s_{\text{alt}_{31}} = 0.5 (11,027) = 5,514 \text{ psi.}$$

$$S_{alt} = \max S_{alt\ ij} = S_{alt\ 23} = 43,944 \text{ psi.}$$

The cyclic life is from Figure N-415 (B) of Reference 10:

$$\text{When } S_{alt} = 43,944 \text{ psi,}$$

$$N = 35,000 \text{ cycles.}$$

However, based on experience with various tests, the nozzle U-tubes govern the cyclic life of the nozzle. The fatigue life of the U-tubes is evaluated in the following section.

b. Nozzle U Tubes

As in any regeneratively cooled nozzle, the XE-1 coolant-propellant tubes are exposed to severe thermal gradients. One tube-wall surface is in contact with the cryogenic propellant while the other is in contact with the hot gas from the reactor.

Extensive experience with nozzle leak tests, proof tests, chemical simulation firing tests, and nuclear firings leaves no doubt as to the structural integrity of the U-tubes. However, precise analytical prediction of tube stresses, margins of safety, and the number of cycles to failure is quite difficult. The three basic ingredients of stress analysis (applied loads, geometry, and material properties) vary with the three dimensions (with respect to engine center line): radial, circumferential, and axial. In addition, the applied loads and material properties are a function of time.

Because of the conservatism in the proof test, and because large margins can readily be shown on static primary (pressure) stress, no detailed analysis of the static case is presented here. Instead, a simplified model is constructed and evaluated for fatigue life under thermal

"ratcheting": i.e., alternating between thermal tensile and thermal compressive stresses beyond the elastic limit in the cooldown and heating phases of the nuclear firing cycle, respectively. The tube crown is the critical portion for this investigation.

Since the tubes contribute very little to the overall structural continuity in the hoop direction, a composite strip of jacket and tube is taken as a beam, or bar, structure and investigated for fatigue stresses from thermal gradients at several possibly critical locations in the nozzle. The above sketch shows the simplified model. Symbols used here and in the following analysis are:

Avg	Average
α	Thermal Coefficient of Expansion
Δ	Difference in
σ	Stress
c	Chamber (nozzle)
E	Modulus of Elasticity
F_{ty}	Allowable Tensile Yield Stress
G	Gas Side

H	Hoop
J	Jacket
p	Pressure
R	Radius
T	Temperature. As subscript, Tube
Tot.	Total

7. Stresses In The Tube Crown

a. Primary Meridional Stress

$$\sigma_p = \frac{\Delta p R_{GT}}{2t_T}$$

For conservatism, the outer radius of the tube and the minimum thickness allowed by the tolerance are used.

- (1) Thermal, Meridional Stress Due To Tube Crown Being At Different Average Temperature Than Nozzle Jacket

$$\sigma_{TH_{T1}} = \frac{E_T \alpha_T}{2} (T_{Avg_T} - T_{Avg_J}).$$

- (2) Thermal, Meridional Stress Due To Temperature Gradient Through Thickness Of Crown

$$\sigma_{TH_{T2}} = \frac{E_T}{2} \alpha_T (T_{GT} - T_{LT}).$$

The above stresses are calculated and summed in Table 3. The pressures and temperatures used are for the nuclear steady-state. As stated earlier, these parameters vary with time: i.e., from one phase in

Table 3

Analysis of XE-1 Nozzle U-Tubes
for Cyclic Life and Margins of Safety on Primary Stress

①	②	③	④	⑤	⑥	⑦	⑧	⑨	⑩	⑪	⑫	⑬	⑭	⑮	⑯	⑰	⑱	⑲	⑳
STA. IN.	P _T PSI	P _C PSI	ΔP _T PSI	R _{G_T} IN.	σ _{H_T} KSI	T _{G_T} °F	T _{L_T} °F	T _{AVG_T} °F	F _{TY_T} KSI	M.S. _T (3)	T _{AVG_J} °F	ΔT _{T-J} °F	E _T PSI(10) ⁻⁶	α _T IN/IN-°F(10) ⁶	σ _{TH_{T1}} KSI	ΔT _{G_T-L_T} °F	σ _{TH_{T2}} KSI	σ _{TOT_T} KSI	CYCLES TO FAILURE
(1)	(1)	(1)	② - ③	(1)	$\frac{④ \times ⑤ (10)^{-3}}{0.012}$	(1)	(1)	$\frac{⑦ + ⑧}{2}$	(2)	$\frac{⑩}{1.15 ⑥}^{-1}$	(4)	⑨ - ⑫	(2)	(2)	$\frac{⑭ ⑮ ⑯}{2}$	⑦ - ⑧	$\frac{⑭ ⑮ ⑯}{2}$	$\frac{⑥}{2} + ⑰ + ⑱$	(5)
0	774	560	214	.299	5.33	1065	790	928	20.5	+2.19	-267	1195	23.2	10.00	-138.62	275	-31.90	-167.86	160
10	787	560	227	.295	5.58	860	630	745	21.5	+2.35	-258	1003	24.3	9.65	-117.60	230	-26.97	-141.78	
15	790	558	232	.237	4.58	843	600	722	21.8	+3.14	-59	781	24.5	9.60	-91.85	243	-28.58	-118.14	
20	785	545	240	.150	3.00	970	645	808	21.0	+5.09	-168	976	24.0	9.70	-113.61	325	-37.83	-149.94	400
25.5 (THROAT)	732	385	347	.071	2.05	1600	800	1200	19.2	+7.14	-243	1443	21.4	10.50	-162.12	800	-89.88	-250.98	30
35	874	40	834	.110	7.64	1315	995	1155	19.5	+1.22	-259	1414	21.8	10.40	-160.29	320	-36.28	-192.75	50
45	894	12	882	.166	12.20	1030	895	963	20.0	+0.42	-283	1246	23.0	10.10	-144.72	135	-15.68	-154.30	150
56.5 (EX. PLANE)	900	5	895	.227	16.93	850	780	815	21.0	+0.08	-250	1065	23.9	9.70	-123.45	70	-8.11	-123.10	

- (1) IN. FROM CORE SUPPORT FLANGE, REF. 6
 (2) REF. 11
 (3) BASED ON PRIMARY (MEMBRANE) STRESS ONLY
 (4) REF. 7
 (5) REF. 14

the cycle to another (e.g., from cooldown to steady-state firing). However, the total stress value computed here for steady state is conservatively taken as the value of alternating stress in the fatigue evaluation.

The nozzle throat was found to be the critical section with greatest stress (251 KSI) and highest average tube wall temperature (1200°F). The ASME Code, Reference 10, and the SNPO-C criteria, Reference 9, do not contain a stress-cycle curve for metal temperature beyond 800°F. The graph in Reference 9 has been modified to accommodate higher temperatures in another analysis, Reference (14). Using this modified graph and the above temperature and stress values results in an indicated tube cyclic life: $N = 30$ cycles.

APPENDIX A

METHOD OF ANALYSIS FOR XE-1 PVARA BOLT LOADS

I. OUTLINE

A. Compute bolt load required to react "ejection" force

$$(\Sigma F_z = 0, \text{ i.e., } p\pi R_s^2 = P_{B_E} N).$$

B. Compute bolt load required to insure joint compatibility; i.e., to transmit the flange interface moment without differential rotation of the flange faces. This moment is obtained by computer solution considering both pressure and thermal loading. The bolt-force component due to moment is obtained by considering the moment to be transmitted by "heel-and-toe" couple action, $(M = P_{B_c} \times a_c)$.

C. The above method of treating the interface moment assumes a rigid structure for the bolt: i.e., no bolt stretch is considered possible. In actuality the condition of compatibility (equal rotation of the flange faces because no gapping at the bolt circle) must be met by a bolt-flange preload.

D. Add bolt "ejection" and moment load components from A and B above.

E. Compute flange load loss during firing considering the interaction of preload, total "external" load (from D) and differential thermal contraction of bolt and flange. Follow method derived in Section II.

F. For design, make bolt tensile preload equal to or greater than calculated in E above to prevent gapping at the bolt circle at maximum "external" load.

G. Calculate bolt strain corresponding to selected preload from F above, ($\epsilon = \frac{P}{A}$). Select pre-torque corresponding to this prestrain. Use Reference 12.^E

H. Calculate bolt margin of safety using preload determined in F above.

II. COMPUTATION OF NET BOLT LOADS IN FLANGED JOINTS CONSIDERING THERMAL LOAD, MECHANICAL EXTERNAL LOAD, AND PRELOAD

A. INTRODUCTION:

Although the method derived here may be considered general and applicable to any joint with elastic bolts and flanges, it was derived specifically to determine preload change in NERVA XE pressure-vessel closure joints during nuclear firing. Prior to firing, the condition of the joint is one of ambient temperature with the only load being pre-tension in the bolt, pre-compression in the flange. At firing, as pressure and temperature, respectively, increase and decrease to steady-state values, the joint is subjected simultaneously to "external" load and differential thermal contraction of bolt and flange.

The "external" load, consisting of both "straight ejection" force and "prying" action due to eccentricity of the flanges, tends, of course, to increase the bolt load. The thermal effect, on the other hand, tends to increase or decrease the bolt load. In the forward closure joint, for example, the aluminum flange with lower temperature and greater coefficient of expansion than the A286 bolt tends to shrink more than the bolt. The net effect is a loss in bolt load as shown in the example of Section II-D.

B. DISCUSSION OF METHOD OF ANALYSIS

Beginning at an initial condition of given preload and temperature

and proceeding to a new condition with "external" load and different temperatures results in two unknown joint quantities: bolt tensile load and joint compression load. The two principles used to provide the necessary equations are:

1. The necessity of equilibrium of axial forces on the flange, and

2. The fact that stretch (additional) of the bolt must equal growth (compression relief) of the flange. Note that in certain cases, including our example "stretch" and "growth" may be negative.

$$(1) \sum F_X = 0:$$

$$(2) \Delta L_B = \Delta L_F$$

$$P_B = (P_e + P_c) + P_F$$

C. ANALYSIS

1. Equilibrium of Forces

Condition 1 (Preload Only): $P_{B_1} = (P_{e_1} + P_{c_1}) + P_{F_1}$

$$P_i = 0 + P_i$$

$$(P_i = \text{Preload})$$

Condition 2 (External Load and Different Temperatures):

$$P_{B_2} = (P_{e_2} + P_{c_2}) + P_{F_2}$$

$$(P_{B_1} + \Delta P_B) = (P_{e_1} + \Delta P_e + P_{c_1} + \Delta P_c) + (P_{F_1} - \Delta P_F)$$

$$(+ \Delta P_B) = (0 + \Delta P_e + 0 + \Delta P_c) + (- \Delta P_F)$$

$$\Delta P_B = \Delta P_e + \Delta P_c - \Delta P_F$$

$$\Delta \epsilon_B^E A_B = \Delta P_e + \Delta P_c - \Delta \epsilon_F^E A_F$$

$$\Delta \epsilon_B = \frac{\Delta P_e + \Delta P_c - \Delta \epsilon_F^E A_F}{E_B A_B}$$

2. Compatibility of Displacements of Bolt and Flange

$$\Delta L_B = \Delta L_F$$

$$\alpha_B \Delta T_B L + \Delta \epsilon_B L = \alpha_F \Delta T_F L + \Delta \epsilon_F L$$

$$\alpha_B \Delta T_B + \frac{\Delta P_e + \Delta P_c - \Delta \epsilon_F E_F A_F}{E_B A_B} = \alpha_F \Delta T_F + \Delta \epsilon_F$$

$$\begin{aligned} E_B A_B \alpha_B \Delta T_B + \Delta P_e + \Delta P_c - \Delta \epsilon_F E_F A_F \\ = E_B A_B \alpha_F \Delta T_F + \Delta \epsilon_F E_B A_B \end{aligned}$$

$$\begin{aligned} E_B A_B (\alpha_B \Delta T_B - \alpha_F \Delta T_F) + \Delta P_e + \Delta P_c \\ = \Delta \epsilon_F (E_B A_B + E_F A_F) \end{aligned}$$

$$\Delta \epsilon_F = \frac{E_B A_B (\alpha_B \Delta T_B - \alpha_F \Delta T_F) + \Delta P_e + \Delta P_c}{E_B A_B + E_F A_F}$$

Solve above equation for $\Delta \epsilon_F$ and then compute flange load loss (gain),

$$\Delta P_F = \Delta \epsilon_F E_F A_F .$$

Substitute value for $\Delta \epsilon_F$ into (1) above and solve for $\Delta \epsilon_B$ and ΔP_B .

D. EXAMPLE

Take a forward closure joint as an example. Use measured temperatures, where available, to determine mechanical properties and temperature changes:

	T $^{\circ}F$	ΔT $^{\circ}F$	E psi	α in./in.- $^{\circ}F$
Flange	-159	-239	10.8	11.55
Bolt	- 16	- 96	29.1	8.6

For total "external" load, take $\Delta P_c + \Delta P_e = 35.1 \times p = 35.1 \times 740$,
 $= 26,000$ lb/bolt.

Change in flange strain:

$$\begin{aligned}\Delta \epsilon_F &= \frac{E_B A_B (\alpha_B \Delta T_B - \alpha_F \Delta T_F) + (\Delta P_e + \Delta P_c)}{E_B A_B + E_F A_F^*}, \\ &= \frac{29.1(10)^6 (.2165) [(8.6(10)^{-6}(-96) - 11.55 (10)^{-6} (-239))] + 26,000}{29.1(10)^6 (0.2165) + 3.28^* (10.8)(10)^{-6}}, \\ &= \frac{29.1 (.2165) (-1935) + 26,000}{6.3 (10)^6 + 35.4 (10)^6} = 915 (10)^{-6} \mu.\end{aligned}$$

Loss of Flange Preload:

$$\Delta P_F = \Delta \epsilon_F E_F A_F = 915 (10)^{-6} 10.8(10)^6 3.28 = 32,400 \text{ lb.}$$

Change in Bolt Strain:

$$\begin{aligned}\Delta \epsilon_B &= \frac{(\Delta P_e + \Delta P_c) - \Delta \epsilon_F E_F A_F}{E_B A_B}, \\ &= \frac{26,000 - 32,400}{6.3 (10)^6} = -1016 (10)^{-6} \text{ in./in.}\end{aligned}$$

*See Section III of this appendix for determination of A_F .

Change in Bolt Load:

$$\Delta P_B = \Delta \epsilon_B E_B A_B = -1016 (10)^{-6} 6.3(10)^6 = -6400 \text{ lb.}$$

Total Bolt Load:

$$P_B = P_i + \Delta P_B = 32,400 - 6400 = 26,000 \text{ lb.}$$

III. DETERMINATION OF EFFECTIVE COMPRESSION AREA

A. INTRODUCTION

Perhaps the most difficult point in the application of a joint interaction analysis is the accurate determination of the effective area of the compression part of the joint. The "tension spring" (bolt) stiffness is usually well defined by the shank cross-sectional area times the modulus. The "compression spring" (flange) stiffness, however, is generally indeterminate because it is not possible to determine how much of the flange total face-area is effective in transmitting bolt load. Arbitrary amounts of area (e.g., that under the bolt head or washer, Reference (15)) are often taken, but experience indicates this method usually results in an excessive bolt-stiffness parameter and bolt load.

A source (Reference (16)) has been found which tabulates smaller bolt stiffness parameters and generally yields better agreement with test results.

The theory of this source must be based on larger compression spring constants, and, therefore, on larger effective compression areas. Bolt stiffness parameter:

$$K_l = \frac{A_B E_B}{A_B E_B + A_F E_F} \cdot$$

This stiffness parameter may be used in the preload-external load interaction equation for the case of no temperature change (i.e., ambient temperature hydrotest case):

$$P_B = P_i + K_1 P_e .$$

The above bolt stiffness parameter and equation are not used directly in the foregoing analysis, but are explained here to clarify and justify the use of the bolt stiffness data from Reference (16) for the calculation of effective compression area.

B. FORWARD CLOSURE BOLTS SPRING CONSTANT CALCULATION

$$\text{Flange } L = 1.75 \text{ in.} \quad \text{Bolt } \underline{\text{Nom.}}^* D = 5/8 \text{ in.}$$

$$L/D = \frac{1.75}{.625} = 2.8.$$

Stiffness parameter of 5/8-in. bolt filling hole (Reference (16)):

$$K'_1 = 0.264 - 0.8 (0.264 - 0.208) = 0.219 ,$$

$$K'_1 = \frac{A'_B E_B}{A'_B E_B + A'_F E_F} ,$$

$$0.219 = \frac{\pi/4 (0.625)^2 30(10)^6}{\pi/4 (0.625)^2 30(10)^6 + A'_F 10(10)^6} ,$$

$$0.219 = \frac{9.21 (10)^6}{9.21 (10)^6 + A'_F 10(10)^6} ,$$

$$2.02 + 2.19 A'_F = 9.21 ,$$

*5/8-in. shank of bolt in example is undercut to 0.525-in. (References 1e & 1g)

$$A_F = \frac{9.21 - 2.02}{2.19} = 3.28 \text{ in.}^2$$

True stiffness parameter of undercut bolt:

$$\begin{aligned} K_1 &= \frac{\pi/4 (.525)^2 29.1(10)^6}{6.3(10)^6 + 3.28(10.8)(10)^6} = \frac{6.3(10)^6}{6.3(10)^6 + 35.4 (10)^6} , \\ &= \frac{6.3(10)^6}{6.3(10)^6 + 35.4 (10)^6} = .151^{**} . \end{aligned}$$

C. NOZZLE BOLT SPRING CONSTANT CALCULATION

Use only nozzle flange as compressed flange portion.

Flange length, $L = 1.5 \text{ in.}$

$$D = 5/8\text{-in.}, \text{ and } L/D = \frac{1.5}{.625} = 2.40$$

Stiffness parameter for 5/8-in. bolt filling hole

$$K_1' = .1115 - .4 (.1115 - .0842) = .1006 \text{ (Reference 16).}$$

$$\text{Since } K_1' = \frac{A_B' E_B}{A_B' E_B + A_F E_F} ,$$

$$0.1006 = \frac{9.21}{9.21 + 30 A_F} , \text{ and}$$

$$A_F = 2.72 \text{ in.}^2$$

True stiffness parameter of undercut bolt:

$$K_1 = \frac{(0.2165) (29.1) (10)^6}{6.3(10)^6 + 2.72 (29.1)(10)^6} = 0.074 .$$

** Using area under washer, $A = 1.078 \text{ in.}^2$, $K_1 = 0.35$

REFERENCES

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- 1b. AGC Drawing Number 1113856-9 (Closure)
- 1c. AGC Drawing Number 1113855-9 (Cylinder)
- 1d. AGC Drawing Number 706988-29 (Nozzle)
- 1e. AGC Drawing Number 702999-3 (Closure Bolts)
- 1f. AGC Drawing Number 1117511 (Nozzle Bolts)
- 1g. AGC Drawing Number 286664 (Closure Bolts-Hydrotest)
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- 1i. AGC Drawing Number 265233 (Nozzle Bolt-Hydrotest)
2. AGC Report No. RN-TM-0159, Basic Loads and Structural Design Criteria for the XE-1 Engine
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