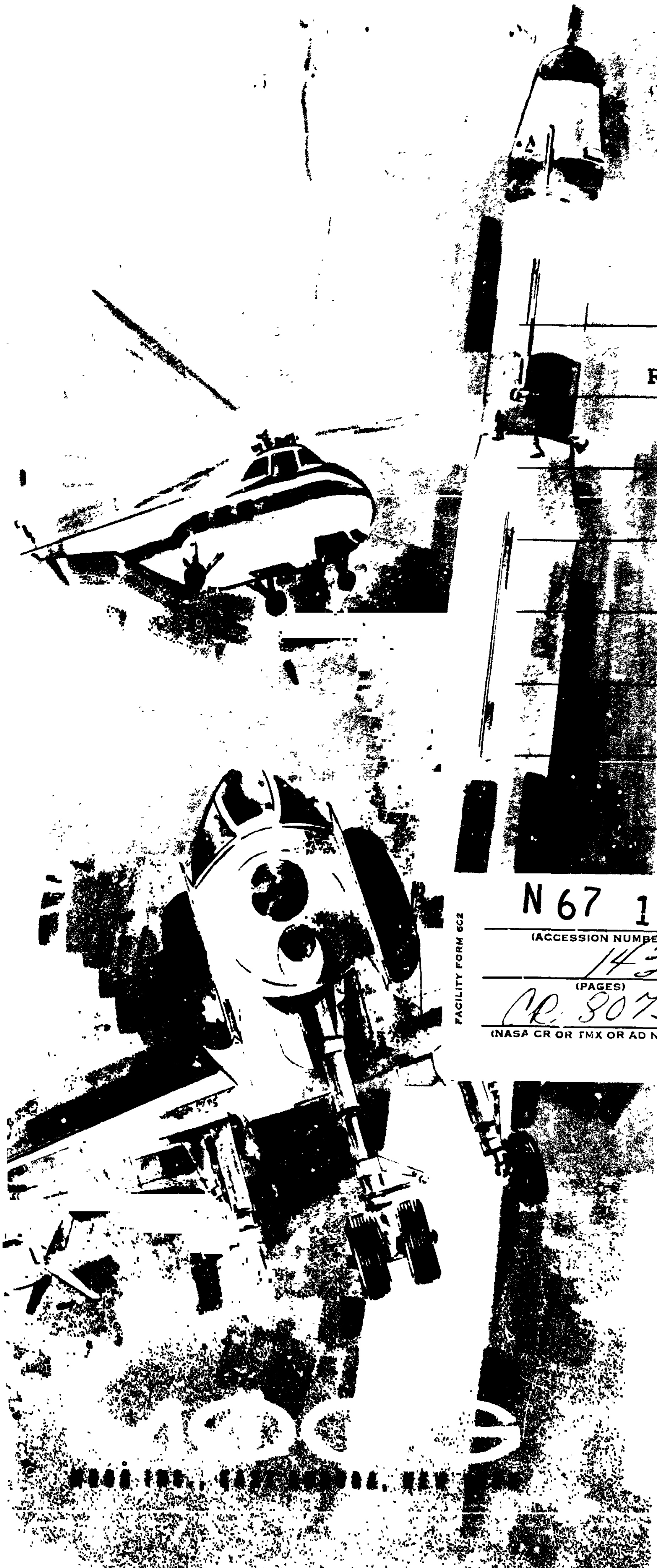
MR 1062



RELIABILITY ANALYSIS & PREDICTION

MOOG MODEL 17-200B

MECHANICAL FEEDBACK

SERVOACTUATOR GMSFC, NASA

PART NO. 50M35008,

CONTRACT NO. NAS8-18060

MOOG INC. **MR** 1062

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MECHANICAL FEEDBACK SERVOACTUATOR

GMSFC, NASA PART NO. 50M35008, Rev. B

CONTRACT NO. NAS8-18060

MOOG INC.
East Aurora, New York
Report No. MR 1062
Revision A

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REVISION RECORD

Rev.	Affected Pages	Brief Description of Revision	Date	Approval Signature
A	9-10	1. Added discussion of Failure Mode and Effects Analysis	10-25-66	[illegible]
	58-64	2. Added Tables II and III		
	65-71	3. Changed table numbers as follows: Table IV was II Table V was III Table VI was IV Table VII was V Table VIII was VI Table IX was VII Table X was VIII		
	37-57	4. Revised Table I to show failure effects in terms of piston position.		

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1.0 INTRODUCTION

This report presents the results of the reliability analysis performed on the Moog Model 17-200B Servoactuator. The analysis was performed by the Reliability Engineering Group of Moog Inc. and fulfills the requirements outlined in the Moog Reliability Program Plan, reference 3. The Reliability Analysis Program was initiated on March 1, 1965 and was terminated December 10, 1965. The program was delayed several months when the life cycle actuator specimen was not available to the Reliability Group. Execution of the actuator life cycle test program was deemed essential to help substantiate design reliability.

2.0 SCOPE

The reliability analysis was carried out on the production configuration of the 17-200 servoactuator. This configuration was modified during execution of the reliability program, however, all modifications and their affect upon reliability are accounted for in this analysis. Each design modification which became effective after the design review of 9/1/64 is discussed in Section 4.3 of this report.

The reliability analysis was divided into four major tasks. These tasks were: (1) margin of safety analysis, (2) review of failure experience, (3) failure mode and effects analysis, and (4) the reliability prediction. Each task is presented as a separate section of this report.

A careful review of each detail drawing provided the basis for the margin of safety analyses of critical design areas. These analyses were then used to determine structural failure modes.

All failure experience accumulated to date was reviewed and assessed for adequacy of corrective action and indication of potential failure modes. All testing associated wich such failure experience is discussed in this report.

In essence, the reliability analysis of the 17-200B actuator configuration represents an extension of the preliminary reliability analysis performed on the 'A' configuration. The detailed analysis described herein consists of an assessment of product reliability in its current configuration.

3.0 ACTUATOR DESCRIPTION

3.1 General

The 17-200B servoactuator basically consists of a forged body, a cylinder, a double-ended piston, a three stage flow control servovalve, and a mechanical feedback mechanism. The mechanical feedback mechanism regulates output of the servovalve to provide a desired piston position.

Reference 6 (Moog's Technical Proposal) provides a basic description of the 17-200 actuator including accessory components. Evolutionary design changes have occurred since publication of the referenced report, however, with regard to major design concepts, components, and functioning of the actuator assembly that report is still pertinent.

3.2 Actuator Configurations

The Model 17-200B actuator configuration represents the production configuration of the 17-200 actuator. The original 17-200 actuator incorporated a two stage servovalve Model 16-140A which appeared on the first two servoactuators. This two stage servovalve was then replaced with a three stage valve, Model 16-140C. The three stage valve has remained on all subsequent actuators shipped to GMSFC. All "B" configuration actuators incorporate the Model 16-140D servovalve.

The 17-200B configuration reflects the addition of several design changes from the "A" configuration. A summary of the major design changes incorporated on the 17-200B servoactuator are tabulated below:

1. Elimination of the current limiter assembly (P/N 063-41739)
2. Redesigned feedback spring (P/N 110-45185-045/055)
3. Redesigned cam follower bearing (P/N 120-44385)
4. New piston "o"-ring cap seal design
5. Potentiometer redesign (P/N 062-13999)

4.0 RELIABILITY ESTIMATES

4.1 General

The reliability estimates described below are at best educated "guesses" and no attempt has been made to assess confidence level.

Two estimates have been computed: the first represents the probability of successful operation in the flight environment. The second consists of the MTBF (mean time between failure) in the flight environment. This environment is specified as ten (10) minutes and/or 200 cycles of operation under the environment stipulated in paragraph 3.3.3 of reference 1.

4.2 Probability of Successful Operation

Reliability as expressed here consists of the maximum probability that each actuator will operate successfully in the flight environment defined previously from paragraph 5.5 of this report:

$$R_{max.} = 1 - P_r\{F\}$$

where: R = Reliability

$P_r\{F\}$ = Probability of failure in the flight environment

whereby: $R_{max.}$ = 0.9995

4.3 MTBF (Mean Time Between Failure)

From paragraph 5.5.1 of this report, the maximum attainable MTBF is 354 hours.

5.0 RELIABILITY ANALYSIS

5.1 General

The reliability analysis of the 17-200B servoactuator was concerned solely with potential failure of the servoactuator in the flight operating environment. Two primary causes of failure were considered, consisting of: (1) the possibility of a design inadequacy undetected because of inadequate analysis and/or evaluation tests, and (2) the possibility of an undetected quality defect which could result in fatigue and/or sudden failure. It was presumed that all other actuator malfunctions resulting from quality defects would be revealed prior to flight during pre-flight checkout tests.

5.2 Margin of Safety Analysis

5.2.1 Drawing Review

A review of all detail and assembly drawings was undertaken to identify actuator design features which were: (1) unique to the Model 17-200B, (2) similar to those of other servoactuators having previous failure experience, or (3) deemed critical relative to design maturity. Potential failure regions, indicated by this review, were documented in the Failure Mode Analysis (Table I) and all structural aspects of the servoactuator believed "marginal" were subjected to stress analysis (Appendix I).

5.2.2 Margin of Safety Approach

The margin of safety (MS) represents the ratio of excess strength to the required strength for a given structural component (reference 26). It was computed from:

$$MS = \frac{F}{f} - 1$$

where: F = allowable stress
f = operating stress

From the standpoint of reliability, if MS $\geq$ 1, the possibility of failure was considered to be negligible. If MS $\leq$ 1, the possibility of failure was admitted according to the formula:

$$P_r \{F\} = 0.01 \ (1 - MS)$$

The foregoing represents a gross approximation to accommodate the fact that strength distribution data for component materials is unavailable to Moog Inc.. In lieu of the foregoing, material properties as stipulated in MIL Handbook 5 were used. These properties are defined as the minimum strengths to be expected with at least a 99% conformance at a 95% confidence level. A discrete load distribution based upon the servoactuator life cycle requirement was used to evaluate stresses. If MS = 0, there was assumed to be a probability of failure. $P_r \{F\} = 0.01$ on the basis of 99% conformance. For MS $\geq$ 1. $P_r \{F\} = 0$

5.2.3 Calculations

All stress calculations are presented in Appendix I. They have been prepared in accordance with the analytical criteria defined in Section 5.2 of this report.

5.2.4 Summary - Margin of Safety Analysis

With the exception of the cylinder, all components analyzed were found to possess adequate margins of safety. Those components having a margin of safety less than one (MS < 1) were assigned failure probability numbers based upon the magnitude of the stress margin. These probability numbers are presented in Table III for the particular failure mode associated with the margin of safety calculation.

Stress calculations performed on the actuator cylinder indicated stress levels in excess of the material allowables at the cylinder to cylinder head juncture. These calculations were conducted using an internal cylinder pressure of 6000 psig on both sides of the piston. In order to substantiate these calculations, the decision was made to conduct a burst pressure test on the actuator. The burst test was performed on the life cycle specimen, S/N 35 in accordance with the qualification test requirements specified in NASA Specification 60B84500 paragraph 4.3.4.10. The piston rod was fully extended with a supply pressure of 6000 psig and a return pressure of 3000 psig maintained for five minutes. The actuator cylinder did not show signs of rupture or distortion during or after the test. The cylinder loading conditions used for the calculations is far more severe than the qualification test requirements. However, at the time the calculations were developed the qualification test procedure was not yet written.

5.3 Failure Experience on the 17-200B Servoactuator

5.3.1 Static Firing Failures

Several 17-200B servoactuators have failed in various ways during multi-engine firings. These failures prompted a series of design changes to achieve increased vibration capability. The redesigned subassemblies have been tested in various ways to insure design maturity. The failures incurred to date and the corrective action is presented below.

a. Feedback Spring Disengagement Failure

Disengagement of the lower feedback spring occurred during shutdown of the fifth multi-engine firing for which actuator S/N 10 had been used. A detail dimensional study of the parts and associated component testing revealed the cause of failure attributable to poor dimensional design of the pivots and seats together with lower-than-necessary vibration capability of the preload assembly. Several changes were made to correct this problem. These include:

(a) Increase engagement of pivots (from 0.065 inch to 0.110 inch minimum), depth of pivot cavities (from 0.065 inch to 0.170 inch), and engagement of feedback springs (from 0.070 inch to 0.175 inch). Collectively, these changes avoid essential loss of parts engagement with adverse tolerance condition which had existed with the original design.

(b) Reduce mass of the spring seats and pivots by change to titanium.

(c) Change the feedback spring design to increase the spring preload. This change increased the axial g capability of the assembly by approximately 200 g.

b. Cam Follower Bearing Failure

Two bearing failures occurred when the outer ring fractured during shutdown of multi-engine firings. A design mock-up of the cam follower assembly was made for vibration testing. Bearing failures identical to those experienced in the actuator were reproduced with this test configuration at an acceleration level just sufficient to cause lift-off of the assembly from the cam surface. It was clear that the bearing had essentially no capability to withstand impact loading caused by a high vibration level.

A "solid roller" type cam follower was then designed and successfully tested.

c. Cam Drive Shaft Braze Failure

Separation of a silver braze joint on the mechanical feedback cam drive shaft occurred during shutdown of the fifth multi-engine firing for which the actuator had been used. This failure was the result of an inadequate silver braze joint between the attach flange and drive tube member. The poor braze joint was found to be caused by inadequate diametral clearance of the mating pieces, such that braze material could not flow into the mating surface area, and inadequate heating of the joint caused by an improper induction heating coil.

Successful braze joints are now produced by the electron beam welding technique.

5.3.2 <u>Acceptance Test Failures</u>

a. Snubber Retainer Failure

On the first "B" model unit, during acceptance testing, the snubber retainers failed. A development program was immediately started to delete the snubbers from the design and still obtain stability during piston bottoming. This objective was accomplished by employing the piston face to cutoff the servovalve at the end of the stroke and providing a cross piston leakage port to prevent biasing of the pressure feedback network due to differential pressure across the bottomed piston.

b. Leakage Across Piston "O"-Ring Cap Seal

Leakage failures across the piston cap seal were occurring on several actuators during acceptance testing. To eliminate this problem a design change was incorporated which eliminated the cap and "o"-ring and replaced them with a new cap-quad ring design.

5.3.3 Potentiometer Evaluation Test Failure

After 100,000 cycles of the life cycle test intermittent noise was displayed by one of the test potentiometers. Since the noise characteristic could not be repeated at any particular stroke position, the test was completed before conducting a failure analysis. Similar failures occurred on several other potentiometers during acceptance testing and field checkout. This prompted a very thorough investigation into the cause of these failures. This investigation showed that during potentiometer assembly a tensile stress was placed on the flexible circuit board which caused the solder fillet to fracture and thus causing electrical discontinuity. At the request of NASA, the printed circuit construction was discontinued and a new design proposal is being reviewed.

5.4 Failure Mode and Effects Analysis

The failure modes and their effects upon servoactuator performance are tabulated by component in Table I. This analysis includes only those failure modes predicted to have a sufficient probability of occurrence derived from a review of :

1) the 17-200B evaluation test program,

2) the margin of safety analysis,

3) static firing test data,

4) acceptance test data, and

5) dominant failure modes encountered by other servoactuators during test and service useage.

The failure effects for each failure mode have been defined in terms of actual piston position.

All actuator piece parts which do not contribute to significant component failure modes are exempted because they fall into one of the following classes:

1) they are parts for which analyses or testing has assured adequate safety margins, or,

2) they are parts for which failure will not cause the actuator performance to be outside of the specification.

Tables II and III present a tabulation of all piece parts and their classification for exemption.

5.5 Reliability Prediction

Many methods are available for carrying out reliability studies. Prior to describing the method employed in this study, it is considered desirable to provide a definition of reliability as it applies to the mechanical device at hand.

5.5.1 Definition of Reliability

The classical definition of reliability, as set forth by AGREE [1], is stated as follows: Reliability is the probability that a device will perform a specified function without failure under given conditions for a specified period of time. This definition of reliability has lead to a predominantly statistical approach to reliability in the electronic field. This approach has not been particularly successful when applied to mechanical devices such as electrohydraulic servoactuators. By success is meant the actual achievement of design improvement as a result of reliability analyses.

A more suitable approach to reliability of mechanical devices is provided by R. J. Mc Crory [2] who defines reliability as a capability:

> "Reliability is the capability of a piece of equipment to perform its design function adequately for the intended period of time under the operating conditions to be encountered."

The foregoing definition of Reliability provided the basis for the reliability analysis of the 17-200B servoactuator. In this respect, the primary objective of the reliability analysis was to evaluate the capability of the actuator to withstand potential failures and to compare its relative sensitivity to failure to that of operational hardware. The Titan III servoactuator, Model 17-185, was selected as the basis for comparison with the 17-200B servoactuator configuration.

1 Advisory Group on Reliability of Electronic Equipment, Reliability of Military Electronic Equipment, U.S. Government Printing Office, Washington, D.C. 1957.

2 Elements of Realism in Mechanical Reliability, R. J. Mc Crory, ASME Design Engineering Conference, New York, New York, May 17-20, 1965.

5.5.2 Method of Analysis

The method of analysis employed for the Model 17-200B servoactuator was first used for a study of redundant servo-actuators and is described in reference 5. This method attempts to accommodate observed data typical of electro-hydraulic devices manufactured by Moog Inc.. For example, a complete study of 624 Titan II and III booster actuators (Moog Model 17-185) was carried out. A total of 69 or approximately 11% were returned to Moog with various defects revealed during three phases of pre-flight checkout tests at Martin Denver. A large number of these returned units possessed one or more defects which can be regarded as potential causes of failure in flight. A total of 79 defects were recorded, two which were catastrophic in nature. This data led to the presumption that many actuators were successfully flown which possessed defects or defective conditions which could have, but did not, result in flight failure. This presumption then lead to the question of a simple conditional probability; if a given defective condition is assumed to exist, what is the probability that it will lead to failure in flight.

A conditional probability of failure analysis was carried out for the Moog Model 17-185 servoactuator and is described in reference 5. If it is assumed that the Model 17-200B actuator will fail in a manner similar to the Model 17-185, than a relative probability of failure analysis can be carried out as described below.

5.5.3 Relative Probability of Failure Analysis

The analysis method used consists of an effort to compare the relative probabilities of failure of the 17-200B servoactuator to the Model 17-185 servoactuator. The 17-185 servoactuator underwent two years of prototype, evaluation, and certification testing and was also subjected to an extensive reliability analysis. A total of 200 have undergone flight tests without (known) failure.

In order to derive a reliability estimate for the 17-200B servoactuator, it was necessary to carry out the following tasks:

a. Compilation of a failure mode analysis, 17-200B servoactuator (Table I)

b. Compilation of known similar failures for the 17-185 servoactuator

c. Probability of the existence of a cause of the known similar failures for the 17-185 servoactuator (Table II)

d. Probability of the existence of a cause of failure for the 17-200B servoactuator (Table III)

e. Probability analysis of the component failures for the 17-200B servoactuator (Table III)

f. Computation of a reliability estimate for the flight regime.

The compilation of component failures and effects required a detailed review of dominant failure modes encountered by other servoactuators during test and service usage. The results of the 17-200B evaluation test program, the stress analysis, acceptance test data, and static firing test data were also used to complete this tabulation. These failures were regarded as the most likely to occur and those which must be analyzed for probability of occurrence. Compilation of similar known failure modes for the 17-185 actuator were then tabulated and the probability analysis of failures was carried out in the following manner.

The pre-flight regime was broken down into three operational regions consisting of:

a. Ground Checkout (GCO)

b. Static Firing (SF)

c. Count Down (CD)

For each of these regions the number of failure occurrences was tabulated for each failure mode. A probability of failure estimate was then prepared using the total number of failures for each failure mode. Probability of failure was a simple calculation of the ratio of failures to the total number of trials. Each pre-flight test sequence was considered a test trial For the 17-185 actuator, 973 test trials have been accumulated to date. All probability calculations derived in this report were based upon 1000 trials.

Probability of failure estimates were then prepared for all 17-200B potential failure modes which included those common to both the 17-185 actuator and those unique to the 17-200B actuator.

For evaluation of reliability in the flight regime, the primary concern was the possibility that a defect (cause of failure) could exist which would lead to failure in flight. Thus, there are two areas of concern: (1) the probability that a servo-actuator possesses a "defect," and (2) the probability that this defective condition will lead to a failure in the flight regime. The foregoing can be expressed as the conditional probability:

$$P_r \{F\} = p(a)\, p(a - u)$$

where: $P_r \{F\}$ = probability of flight failure

$p(a)$ = probability of the existence of a cause of failure

$p(a - u)$ = the probability that the cause of failure will actually result in failure of the actuator

Although this approach, as described herein, will not stand up to rigorous mathematical proof, it is regarded as an appropriate means for estimating flight reliability. In this respect, the method enables the use of all observed failure data recorded to date. In addition, minimum and maximum reliability can be estimated with respect to the value of p (a - u), which can vary from 0 to 1.

Component Failures were designated by F_n, n = 1, 2, 3 . . . K, where K = the total number of dominant failure modes. Causes of failure were designated by a_n, n = 1, 2, 3 . . . M, where M represents the total number of failure causes. Thus for every F, there will be $\sum_{n=j}^{n=p} a_n$ causes of failure. The first step is compilation of the failure mode table in a form suitable for analysis.

The probability of the existence of a cause of failure p (a_n) was simply the number of occurrences of a given cause of failure divided by the number of trials (servoactuators subjected to test). If the flight regime is considered a sequential test, the population of flight actuators can be considered to possess causes of failure typical of the total sample, or $\sum_{n=1}^{M} a_n$ as revealed by previous tests.

Flight reliability is then dependent upon whether a specific cause of failure is of a nature that it will result in flight failure. This requires that for each cause of failure, the factor p (a - u) be determined for each failure cause, a_n. Although data is available for deriving such a factor for dominant causes of failure, this is not the case for many of the sporadic causes of failure. As a first approximation, it was decided to derive a conservative value for (a_n - u) which could apply to all failure causes. Of particular concern were potential failures emanating from contamination within and without the servoactuator. In order to arrive at an empirically based factor, failure experience relative to contamination within the solenoid valve assembly of the Moog Model 1721 servoactuator was reviewed. A total in excess of 3600 servoactuators have been manufactured and have accumulated approximately 500,000 flight operating hours in a temperature environment of 220° to 275° F.

One of the predominant causes of customer return with this servoactuator is residual contamination within the solenoid valve assembly which is undetected during production acceptance and pre-flight tests at Mc Donnell Aircraft Co.. Of some 3600 units manufactured, 42 have been returned for undetected "built-in" contamination which caused flight malfunction of the solenoid valve.

If we presume that every solenoid valve installed in the Model 1721 servoactuator will contain "built-in" contamination of varying degree, then the number of such cases where the contaminant is of a nature to cause flight failure is 1:86 or approximately 0.016. This figure was presumed to apply to all causes of failure.

In many cases, it was arbitrarily assumed that an undetected failure would automatically lead to flight failure. In order to account for the fact that some failure causes are specifically vibration and/or fatigue oriented, other factors were developed for fatigue failure characteristic of (1) vibration in the flight environment, and (2) life cycle in the flight environment. The factor for vibration was designated g_V. The factor for life cycling was designated g_C. These two factors were intended to represent the time and/or cycle sensitivity of the cause of failure.

The probability of a fatigue failure resulting from vibration at resonance, $p(g_V)$, was derived as follows. On the basis of survival alone, there is no way of analytically predicting failure during the flight regime, particularly when the vibration levels can only be described statistically. One method of guessing is to assume that a marginal flexure sleeve in a random vibration environment will adhere to patterns of failure which were encountered during extensive flexure sleeve evaluation tests at Moog. These tests indicate that the minimum life expectancy of a defective and/or marginal flexure sleeve, vibrating at resonance in a sinusoidal vibration environment (30 g peak), is five minutes. The five minute minimum life expectancy may be regarded as a mean of a normal distribution of time to failure in the flight environment. A standard deviation, σ of 90 seconds is assumed and the -3σ point is assumed to represent completion of static firing. Then, as illustrated in Figure 1, the probability of failure represents the shaded portion of the curve which, from tables, is $g_v = 0.423$ (approximately).

The probability of fatigue failure in a life cycle environment, $p(g_c)$ was derived as follows: Survival of a fatigue sensitive component through pre-flight usage (> 100 hours) does not, of course, obviate the possibility of failure during the subsequent 10 minutes of flight. In addition, there exists no method for predicting failure on the basis of survival alone. During pre-flight usage, however, there is considerable cyclic operation of the actuator and consequent stress cycles imposed upon fatigue sensitive components. On the basis of information provided by NASA, flight operation of the actuator requires a very small number of signals of high amplitude usually during the engine start regime. During most of the flight (90 to 95%) the servoactuator hovers about the null region. High amplitude signals ($\geq$ 50% of rated current) are generally required for failure of fatigue sensitive components.

In view of the foregoing, and the short duration of the flight environment (< 10 min.) relative to that for pre-flight (> 100 hours), a conservative estimate for all fatigue sensitive failures is that 1% will fail in the flight environment, provided that no failures have occurred prior to flight. Therefore, $p(g_c) = 0.01$.

5.5.4 Functional Schematic

A functional schematic in the form of a block diagram is presented on page 21 as Figure 2. This is followed by definitions of blocks and symbols and a description of major components assigned to each block. The grouping of components in each block can be further divided into a piece part level as indicated in the tabulation of component groups and parts on page 23.

5.5.5 Reliability Calculations

At the outset, the assumption is made that failure of the potentiometer does not affect flight reliability. The maximum probability of failure is then computed from Table III:

$$P_r\{F\}_{max.} = 0.0450$$

$$R_{min} = 1 - P_r\{F\}_{max.} = 0.9550$$

The minimum probability of failure is computed as:

$$P_r\{F\}_{min.} = 0.00047$$

$$R_{max} = 1 - P_r\{F\}_{min.} = 0.9995$$

Of significance here is the fact that the flight duration of the Titan III actuator. Model 17-185 is approximately 2 minutes. The specified duration of the 17-200B servoactuator is 10 minutes.

Since no known distribution exists for time-to-failure it is impossible to accommodate this divergence. It may be presumed that the conditional probability factors may be optimistic and hence the probability of failure should be higher. Since there is no method available for assessing the accuracy of the conditional probability factors. the assumption is made that the longer duration of the 17-200B flight environment has negligible influence.

5.5.5.1 MTBF (Mean Time Between Failure)

Minimum and maximum MTBF's may be computed as follows:

$$MTBF_{min} = \frac{167}{45} = 3.72 \text{ hrs.}$$

$$MTBF_{max} = \frac{16,700}{47} = 354 \text{ hrs.}$$

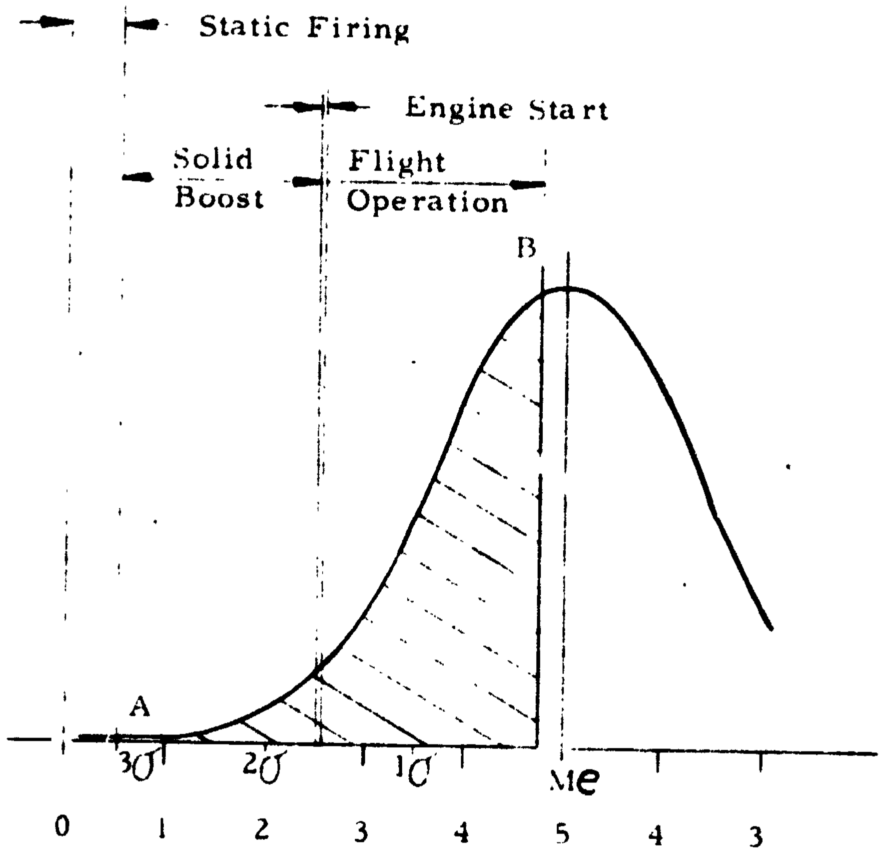
Assuming an exponential distribution of time-to-failure:

$$R_{min.} = e^{\frac{-t}{M_{min.}}} \approx 0.9560$$

where: $t = 0.167$ hrs.
$M_{min.} = MTBF_{min.}$

$$R_{max.} = e^{\frac{-t}{M_{max.}}} \approx 0.9995$$

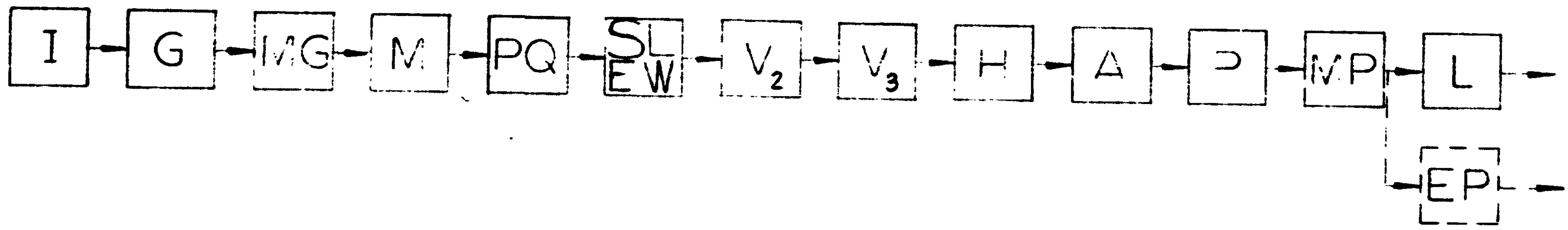
The significance here is the fact that the exponential assumption produces reliability estimates very close to those computed for the conditional probability method.



$$P_r\{f\} = P_r\{A \le x \le B\} = \frac{1}{2\pi\sigma} \int^{B} e^{\frac{-(x-u)^2}{2\sigma^2}} \, dx$$

From Tables $P_r\{f\} \simeq 0.43$

Figure 1
Normal Approximation for Estimating g_v



Functional Schematic
S-IC Servoactuator
Model 17-200B

Figure 2

SYMBOLS

FUNCTIONAL BLOCKS

- I — INPUT CURRENT CIRCUIT
- G — SERVOVALVE FIRST STAGE
- MG — MECHANICAL FEEDBACK, PISTON POSITION TO FIRST STAGE
- M — MECHANICAL FEEDBACK MECHANISM, PISTON POSITION
- PQ — LOAD PRESSURE FEEDBACK, FIRST STAGE
- SLEW — STATIC LOAD ERROR WASHOUT, FIRST STAGE
- V_2 — SERVOVALVE SECOND STAGE
- V_3 — SERVOVALVE THIRD STAGE
- H — ACCESSORY FLUID COMPONENTS
- A — ACTUATOR STRUCTURE
- P — PISTON ASSEMBLY
- MP — MECHANICAL FEEDBACK MECHANISM, PISTON
- L — LOAD

ELECTRICAL COMPONENTS

- EP — ELECTRICAL OUTPUT, PISTON POSITION

GROUPING OF COMPONENTS

I — INPUT CURRENT CIRCUIT

1. Electrical Connector (061-13496)

2. Servovalve Coil Assembly (060-29835-1)

G — SERVOVALVE FIRST STAGE

1. Torque Motor

   a. Polepiece - top and bottom (072-29841-3)
   b. Magnet (072-29842-1)
   c. Coil Assembly (060-29835-1)
   d. Armature-Flexure Sleeve-Flapper Assembly (029-41755-1)

2. Hydraulic Amplifier

   a. Inlet-Filter Orifice Assembly (020-26023-75)
   b. Nozzle Assemblies (070-41986-12-1)
   c. Body and Drain Orifice Assembly

MG — MECHANICAL FEEDBACK MECHANISM. PISTON POSITION TO FIRST STAGE

1. Feedback Spring Assembly

   a. Feedback Spring (110-45185-045/055)
   b. Spring Seat (111-44325)
   c. Pivot (111-44329)

GROUPING OF COMPONENTS

M — MECHANICAL FEEDBACK MECHANISM, PISTON POSITION

1. Cage Assembly (120-45292-1)

    a. Cage (120-45297)
    b. Cam Follower (120-44385)
    c. Leaf Spring (110-29719-1)
    d. Cage Loading Spring (110-29670-2)

PQ — LOAD PRESSURE FEEDBACK, FIRST STAGE

1. Summing Piston (130-29668-1)

2. Sleeve (121-21647-1)

3. Spring-Helical, Compression (110-29670-1)

SLEW — STATIC LOAD ERROR WASHOUT, FIRST STAGE

1. Slew Piston Assembly (111-29686)

    a. Piston (130-29690)
    b. Sleeve (051-29691)
    c. Helical Spring Compression (110-29688-1)

2. Slew Filter Orifice Assembly

V_2 — SERVOVALVE SECOND STAGE

1. Valve Body (031-42880)

2. Bushing and Spool Assembly (021-45336-1)

    a. Bushing (051-42678-1)
    b. Spool (052-41453)
    c. Spool Return Springs (110-41465-2)

GROUPING OF COMPONENTS

V3 SERVOVALVE THIRD STAGE

1. Valve Body (031-42880)
2. Body, Piston & Spool Assembly (030-41746)
   a. Spool (052-42776-1)

H ACCESSORY FLUID COMPONENTS

1. Filter Assembly (020-29672-1)
2. Prefiltration Valve
   a. Cap (049-13499)
   b. Sleeve (121-13811)
   c. Spool (049-13632)
3. Cylinder Bypass Valve
   a. Spool (052-13494)
   b. Knob (049-13465)
   c. Cap (049-13502)
4. Check Valve Assembly - Body (023-13725-1)
   a. Cap (049-11307)
   b. Spring (110-11351)
   c. Flapper (072-11308)
   d. Seat (111-11317)
5. Check Valve Assembly - Cylinder (023-13725-2)
6. Check Vent Assembly (023-12275)
   a. Diaphragm (083-12084-5)

GROUPING OF COMPONENTS

7. Inlet and Return Fittings

8. Test Ports

   a. Test Port Plug (073-20651-4CL)

9. Static Seals

A ACTUATOR STRUCTURE

1. Actuator Body Assembly (032-13875-3)

   a. Body (033-14009-1)
   b. Piston Rod Seal (080-24540-139)

2. Cylinder Assembly

   a. Cylinder (033-14018)
   b. Piston Rod Seal (080-24540-139)

3. Tailstock Assembly (121-13508)

   a. Bearing (121-13405)

P PISTON

1. Piston Assembly

   a. Piston (130-14013)
   b. Head Seal "O"-Ring (080-24540-142)
   c. "O"-Ring Cap (082-41693-447)

2. Piston Rod Seal (080-24540-139)

3. Rod End Assembly (121-13510)

   a. Bearing (121-13405)

GROUPING OF COMPONENTS

MP — MECHANICAL FEEDBACK MECHANISM, PISTON

1. Cam and Cam Guide Assembly (029-14010)

   a. Cam (120-14011)
   b. Cam Guide (023-13995)
   c. Cam Guide Tube (039-13991)

2. Potentiometer Extension (120-13507)

L — LOAD

1. Engine Inertia

2. Missile Structural Stiffness

3. Structural and Actuator Damping

EP — ELECTRICAL OUTPUT, PISTON POSITION

1. Potentiometer (062-13999)

   a. Case
   b. Pin (093-02454-4)
   c. Carriage and Shaft Assembly

      (1) Wipers
      (2) Shaft Bearings
      (3) Seals

   d. Potentiometer Element

Sheet 1 of 4

SER NO. 27 & SUBQ

MODEL: 16-140 D SERVOVALVE (SIC)

PARTS LIST

< Critical Parts

PART NO	NAME		NO REQ'D					REMARKS
0??-41744	Installation		X					REV A
??0-41745	Assembly	<	X					REV K
00?-42911	WIRING SCHEMATIC & COIL INSTALLATION		>					NOTE
030-41746-1	Body, Piston & Spool Assy			1				△
031-~~41747~~ 42880-1	Body, SERVOVALVE				1			030-4..88.. USED IN BODY SER NO 1-26
031J01769	Body-Forging	<				1		
130-29668-1	Piston-Summing	<			1			
121-29647-1	Sleeve-Summing Piston	<			2			
052-42776-1	Spool	<			1			△
049-29637-1	End Cap-Spool, RH			1				
049-29638-1	End Cap-Spool, LH			1				
080-24540-38	O-Ring			2				
090-06132-14C	Screw, Cap, Sch			10				
049-29636-3	End Cap-Filter (Press End)			1				
049-41748	Cap, Filter			1				
071-29671-1	Filter Inlet			1				
071-29695-1	Filter Support			1				
090-06132-14C	Screw, Cap, Sch			12				
080-24540-48	O-Ring			3				
110-29670-1	Spring Compression	<		2				
111-29644-1	Spring Seat, Summing Piston			4				
111-29646-1	Pivot-Adjustor			2				
111-28002-1	Pivot			2				
112-29649-1	Retaining Collar			2				
112-29648-1	Spring Cup			2				
090-29650-1	Screw, Retaining			2				
080-24540-30	O-Ring			2				
080-24540-7	O-Ring			2				
093-29693-1	Plug			3				
080-24540-84	O-Ring			3				
020-29672-1	Filter Assembly			2				
020-29673-75	Body & Orifice Assembly				2			
071-22286	Orifice					2		
071J01244	Orifice - Blank						2	
071-29642-1	Orifice Body					2		
071-29674-1	Filter				4			
071-29641-1	Filter Retainer - Upper				2			
071-29640-1	Filter Retainer - Lower				2			
071-29643-1	Retaining Screw - Filter			2				
080-24540-22	O-Ring			2				
?-24540-4	O-Ring			2				
?-24540-8	O-Ring			2				

REV LTR	E O NO	DATE	BY / CHK'D / PA	REV LTR	E O NO	DATE	BY / CHK'D / PA
	31176	1-27-65	L.A.K. / [illegible] / CM				
	31043	12 25 64	RS / KUREK / CM				
H	30990	12-2-64	RZ / KUREK /	S	32,144	5-21-65	RF / KUREK / CM
F	30728	10-29-64	RZ / KUREK / CM	R	EO 31405	3-8-65	RF / KUREK /

RELEASED PER E O. NO 30, 398 DATE: 7-24-64 28 MOOG SERVOCONTROLS, INC

SERVOVALVE (SIC)

PARTS LIST

MODEL: 16-140D Sn 27 & Subq.

Critical Parts

PART NO	NAME	NO. REQ'D				REMARKS
021-45336-1	1st & 2nd Stage Assembly	1				△3
031-45338-1	Body, Bushing & Spring Cup Assy.		1			
031-41452-1	Body Assembly			1		
031-41452-2	Body				1	
093-28472-D0635	Plug				1	
050-42685-1C-10	Bushing & Spool Assy.			1		
051-42678-1	Bushing				1	
052-41453	Spool				1	
112-45323	Cup, Spring			1		
112-45337-1	Spring Cup & Clinch Nut Assy.			1		
111-45322	Cup, Spring-Adjustor				1	
091-25527	Nut Clinch				1	
090-20054-AC6H4	Screw, Adjuster		1			
111M00351	Adjustor		1			
080-24540-2	O-Ring		1			
080-24540-22	O-Ring		2			
043-41447	End Cap		1			
043-41456	End Cap, Adjustor Side		1			
090-06130-9S	Screw		4			
090-06130-12S	Screw		4			
110-41465-2	Spring, Helical, Compr. Spool Return		2			
111-41449	Pivot		4			
111-41466	Seat, Spring, Spool Return		4			
071-41559	Plug O-Ring		3			
080-24540-1	O-Ring		3			
020-26023-75	Orifice Assembly		2			
071-26012	Body, Orifice			2		
071J01632	Body, Orifice-Blank				2	
071-22286	Orifice			2		
071J01244	Orifice-Blank				2	
071-26014	Filter		1			
071-41457	Retainer, Orifice		2			
080-24540-60	O-Ring (Orifice Body)		2			
080-24540-42	O-Ring (Filter Retainer)		2			
049-41455	Cover, Filter		2			
090-25606-10	Screw		4			

△3 021-41749 Used on SN 28, 40 & 49, to be replaced by 021-45336-1 upon return

REV LTR	EO NO	DATE	BY	CHK'D	PA	REV LTR	EO NO	DATE	BY	CHK'D	PA
AB	Retyped 32604	10/14/65	BH	PEC							

RELEASED PER E.O. NO 30,398 DATE 7/24/64

MOOG SERVOCONTROLS, INC.

SERVOVALVE (SIC)

☒ PARTS LIST

MODEL: 16-140D SN 27 & Subq.

◁ Critical Parts

PART NO	NAME	◁	NO REQ'D				REMARKS
029-41755-1	Arm, Flex, Sl & FBW Assy.	◁		1			
072-41750	Flapper	◁			1		
072-41654	Armature				1		
070-41751-1	Flexure Sleeve	◁			1		
110-29678-075/085	Wire, Feedback, Spool	◁			1		
110-29679-290/310	Wire, Feedback, Sum, Piston	◁			1		SN 28, 30 & 31 used 110-29679-310/340
080-24540-77	O-Ring - Flexure Sleeve			1			
090-07587-9C	Screw - Flexure Sleeve			2			
092-04548	Washer, Lock			2			
072-45217-9	Polepiece & Stop Assembly			1			SN 27 thru 46, 48, 49 & 50 used 072-42858-9
072-29841-3	Polepiece				1		
072J01785	Polepiece-Investment Casting					1	
072-41538-9	Stop, Armature				2		
072-29841-3	Polepiece			1			
072J01785	Polepiece-Investment Casting				1		
072-29842-1	Magnet, Permanent			2			
102-29847	Spacer - Motor			2			Thickness A/R
060-29835-1	Coil Assembly			2			
060-29831-1	Form, Coil				2		
090-06141-32S	Screw - Motor			4			
070-41986-12-1	Assembly, Nozzle			2			
070-41984-1	Body, Nozzle				2		
070J01929	Body, Nozzle Blank					2	
070-06766-12	Nozzle Tip				2		
070J01072	Tip, Nozzle - Blank					2	
061-13496	Conn. Recp. Elect (PT07H-8-4P)			1			Solder per EM270 Type I
064-06089-10	Tubing, Teflon (Approx 18" lg.)			A/R			
064-06089-15	Tubing, Teflon (4 pcs Approx 1" lg)			A/R			
094-20120C20	Lockwire			A/R			
080-24540-78	O-Ring (Nozzle Block)		4				
080-24540-5	O-Ring (Nozzle Block)		1				
090-06132-38S	Screw Mounting		4				
049-44118	Cover		1				
090-06141-10C	Screw		2				

⚠ Alt. 031-41764-2 Body, Piston & Spool Assy. using 031-42880-2 Body in place of 031-42880-1 Body, and 052-42776-2 Spool in place of 052-42776-1 Spool.

LTR.	E.O. NO	DATE	BY	CHK'D	P.A
AB	Retyped 32604	10/14/65	BH	PEC	

RELEASED PER E.O. NO.: 30,398 DATE: 7/24/64

MOOG SERVOCONTROLS, INC

SERVOVALVE (SIC)

PARTS LIST

MODEL: 16-140D

Denotes Critical Parts

	PART NO	NAME	NO REQ'D			REMARKS
AA	120-45293	Cage & Follower Assembly	1			SER NO
AA	120-45292-1	Cage Assembly		1		USED 120-45235
	120-41802	Cage End			1	
AA	120-45297	Yoke, Spring Cage			1	
	120-41804	Extension, Cam Follower			1	
	091-29763-1	Nut, Adjustor		1		
	110-29719-1	Leaf Spring		1		
	110J01788	Spring, Leaf Stamping			1	
	110-29720-1	Leaf Spring		1		
	110J01787	Spring, Leaf, Stamping			1	
	120-44388	Cam Follower Assembly		1		
	120-44386	Shaft			1	
	120-44385	Roller, Cam Follower			1	
	121-44387	Clevis			1	
	090-06130-10C	Screw Cap, Sch. (Leaf Spring)	4			
	092-29724-1	Retainer	2			
Z	092J01911	RETAINER - STAMP		2		
	110-45185-045/055	Spring Helical Compr. (Feedback)	2			
	110-29670-2	Spring, Compr. (Cage Loading)	1			2
	111-44326-1	Pivot, (Lower)	1			
	090-29728-1	Screw Adjustor	1			
	111-44325	Seat, Spring	4			
	111-44327	Pivot Flapper	1			
	111-44329	Pivot	1			
	103-29818-1	Bracket Assembly	1			
	103-42827	Bracket-Cam HSG Support		1		(Replaces 103-29729-1)
	103J01895	Brkt.-Cam HSG Spt. Invest. Cast.			1	
	093-29814-1	Dowel (Bracket to Cam Guide)		1		
	090-06129-16C	Screw Cap, Sch., (Brkt. to Body)	4			
	094-41371	Ring, Retaining	4			
	073-13459-1	Union	4			
	080-24540-54	O-Ring (Union)	8			
	073-29711-1	Union	4			
	080-24540-7	O-Ring	8			
	103-24927	Clamp, Cable	1			
	103-41103-1	Clamp, Cable	1			
	092-41104-1	Washer	1			
	090-24951-8C	Screw, Button Hd.	2			
	074-20382	Nameplate	1			
	090-06204	Drive Screw	2			
	MS20995C20	Lockwire	A/R			
		Potting Compound 2651	A/R			Emerson-Cumings

2 On Ser. No's. 27 thru 73; Select (-1) or (-2) to obtain min. clearance between Pivot shank dia. & its mating hole of Cage Assembly.

REV LTR	E O NO	DATE	BY / CHK'D / PA	REV LTR	E O NO	DATE	BY / CHK'D / PA
AA	32541	9-24-65	RF				
Z	30,576A	9-22-65	RF				
	Retyped						
W	32,448	8/27/65	BH				

RELEASED PER E O NO 30,398 DATE 7/24/65

MOOG SERVOCONTROLS, INC.

PARTS LIST

SERVOACTUATOR

MODEL 17 200B SN 25& Subq

Critical Parts

PART NO	NAME	NO REQ					REMARKS
001 14007	Installation	X					[illegible]
007-41787	Schematic	X					REV A
079 42124	Container, Shipping	X					
010-14005 1	Actuator Assembly	1					REV H
032 13860 4	Body & Spool Assembly		1				
052-14097	Spool (Prefiltration)			1			
029 42253	Spool & Knob Assembly			1			
052-13494	Spool (Cyl. By-pass)				1		
049-13465	Knob				1		
093 02454-6	Roll Pin (ESNA No. 79 013-078-0687)				1		
032-13875 5	Body Assembly			1			
094 29145	Insert Fastener				18		Rosan SR 258L
071 14036	Orifice Bushing				1		
033-14009-1	Body Actuator				1		
033 13279-1	Forging-Body Actuator					1	
080-24540-21	O-ring (Spool, Cyl By pass)		2				
049-13502	Cap, Cyl By-pass		1				
049-13504	Cap, Cyl By-pass		1				
110-13503	Spring, Detent		1				
090-06127	Screw, Cap, Sch		3				
092 05115	Washer, #8		4				
049-45516	CAP, PREFILTRATION, [illegible]		1				[illegible]
049-13479	Cap, Prefiltration		1				
080 24540-79	O-ring (Cap & Spool)		[illegible]				
082-[illegible]-1368	Cap O-ring O.D.		[illegible]				
049-13500	Cap, Prefiltration Valve		1				
121-13811	Sleeve, Prefiltration		1				
049 13632	Button, Spool, Prefiltration		1				
092-07110-1	Washer, Flat		8				
090 06129-20C	Screw, Cap, Sch.		6				
080-4250[illegible]	SEAL, QUAD RING (ROD SEAL)		1				
082-2[illegible]	Cap O ring		1				
102-14152	Spacer		1				
091-14133D425	Nut		1				
081 14155	Washer		1				
[illegible]	O-RING (ROD SEAL)		1				
082-[illegible]	CAP, O-RING I.D.		1				

REV LTR	E.O. NO	DATE	BY	CHK'D	P.A.	REV LTR	E.O. NO	DATE	BY	CHK'D	P.A.
M	[illegible]	[illegible]	[illegible]	[illegible]		V	[illegible]	[illegible]	CH	[illegible]	
L	32,150	6-28-65	RL	[illegible]		T	[illegible]	[illegible]	CH	[illegible]	
G	Retyped 31,316	2/18/65	JC	[illegible]							

RELEASED PER E.O. NO 30,351 DATE 7/24/64 32 MOOG SERVOCONTROLS, INC.

PARTS : LIST

SERVOACTUATOR

MODEL: 17-200B SN 25 & Subq

◁ Critical Parts

PART NO	NAME	NO REQ'D				REMARKS
121 13436	Liner-Bearing, Piston	1				
080 24540-152	O-ring (Cyl. to Body)	1				
082-20036-274	Ring-Back-up	1				MS 28774-274
080-24540-109	O-ring (Cam Giude)	2				
080-24540 112	O-ring (Slew Piston)	4				
023-13725-1	Check Valve Assy (Act.Body)	1				
049-11307	Cap		1			
110-11351	Spring		1			
072 11308	Flapper		1			
111-11317	Seat		1			
090-13684	Screw (Mach., Fillister Hd.)	1				
103-13454	Trunnion	2				
090 13498	Screw, Cap, Sch, Hd	8				
071-13365-1	Element-Filter	1				
080-24540-117	O-ring (Filter, Small End)	2				
082-13736	Ring-Back-up Filter	1				
080 24540 141	O-ring (Filter, Large End)	1				
082-20036-142	Ring-Back up (Filter, Large End)	1				MS 28774 142
091-13526	Nut-Filter Retainer	1				
082-20036-131	Ring-Back up	1				MS 28774-131
111-29686	Pivot-Spring(Slew Piston Assy)	8				
111-29687	Seat -Spring	8				
110 29688-1	Spring Helical, Compr ◁	4				
050-41131	Piston & Sleeve Assy	2				
130-29690	Piston		2			
051 29691	Sleeve		2			
049-29689	Cap-Spring	2				
094-29692	Retainer	2				
029-14010	Cam & Cam Guide Assembly	1				
120-14011	Cam		1			
023-13995	Cam Guide		1			
039-13991	Tube, Cam Guide			1		
039J01768	Cam Guide-Blank				1	
103-13992	Bracket, Cam Guide			1		
103J01901	Bracket, Cam Guide Invest. Cast.				1	
071-13993	Ring, Cam Guide			1		

REV LTR	E O NO	DATE	BY / CHK'D / P A	REV LTR	E O NO	DATE	BY / CHK'D / P A
C	Retyped 31,316	2/18/65	JC / RCS				

RELEASED PER E O NO 30,351 DATE 7/24/64

MOOG SERVOCONTROLS, INC.

PARTS LIST

SERVOACTUATOR

MODEL: 17-200B SN 25 & Subq

◁ Critical Parts

PART NO.	NAME		NO REQ'D			REMARKS
080-24540-154	O-ring (Cam)		4			
082-13505	Scraper Ring (Cam)		4			
121-13439	Liner-Bearing, Cam Guide		1			
073-45246	Cam Drive Tube Assembly		1			
073-13450	Tube, Cam, Drive			1		
073-45244	Collar, Cam Drive			1		
073-45245	Fitting, Cam Drive			1		
073J01894	Fitting, Cam Drive-Invest. Cast.				1	
	~~Silver Alloy(QQ-S-5665 Aly?S BAG-1)~~				~~A/R~~	
	~~Flux(Handy & Harman)~~				~~A/R~~	
090-06130-12C	Screw, Cap, Sch.		4			
130-14013	Rod-Piston	◁	1			
130J01763	Rod-Piston-Turned Blank	◁		1		
130J01668	Rod-Piston-Forging	◁			1	
080-24540-142	O-ring(Piston)		1			
082-41693-447	Cap O-ring O.D.		1			
~~102-14014~~	~~Plate, Snubber~~		~~2~~			
~~094-14015~~	~~Ring, Retaining~~		~~3~~			
080-24540-143	O-ring(Cam Drive to Piston)		1			
091-13472	Nut-Jam		1			
~~110-14004~~	~~Spring-Snubber~~		~~4~~			
062-14016	Pot and Extension Assembly	◁	1			
062-13999	Potentiometer	◁		1		
120-13507	Extension, Potentiometer			1		
093-02454-4	Roll Pin(ESNA-79-018-078-0312)			1		
080-24540-36	O-ring			1		
121-13578	Ring-Bearing			1		
080-24540-23	O-ring(Potentiometer Extension)		1			
080-24540-125	O-ring(Potentiometer)		1			
090-06276-9C	Screw, Cap, Sch. (Potentiometer)		8			
023-12275	Check Vent Assembly		2			
083-42983	Diaphragm Assembly			2		
083-12084-5	Diaphragm				2	083J01938
090-12244	Screw-Vent				2	
102-12226	Spacer			2		
083-12245	Seal, Washer			2		

REV LTR	E.O. NO.	DATE	BY / CHK'D	P.A.
K	31,475	5-3-6?	?F / Kurek	
H	32,019	?-4-?	?P / Kurek	
G	Retyped 31,316	2/18/65	JC / RCS	

REV LTR	E.O. NO.	DATE	BY / CHK'D	P.A.
N	32,480	9-14-65	RF / Kurek	
L	32,160	?-23-65	? / Kurek	

RELEASED PER E.O. NO 30,351 DATE 7/24/64

MOOG SERVOCONTROLS, INC

PARTS LIST

SERVOACTUATOR

MODEL: 17-200B SN 25 & Subq

◁ Critical Parts

	PART NO.	NAME		NO REQ'D			REMARKS
	033-14018	Cylinder, Actuator	◁	1			
	033-14003	Forging-Cylinder	◁		1		
	073-13459-1	Union(Cyl to Act. Body)		1			
	080-24540-54	O-ring(Cyl to Act. Body)		2			
	121-13436	Liner-Bearing(Cyl)		1			
L	080-42900-242-2	SEAL, QUAD RING(CYL TO PISTON ROD)		1			
L	082-26308-2	Cap, O-ring		1			
	131-09084-37	Ring-Scraper		1			
	073-29711	Union		1			
	080-24540-7	O-ring		2			
	023-13725-2	Check Valve Assembly(Cylinder)		1			
	049-11307	Cap			1		
	110-11351	Spring			1		
	072-11308	Flapper			1		
	111-11317	Seat			1		
	080-24540-54	O-ring		1			
	049-14023	Cap		1			
	090-06129-16C	Screw, Cap, Sch.		4			
	092-07110-1	Washer		4			
	090-29911-69	Screw, Cap, Sch.		17			MS24678-69
	090-13684	Screw, Machine Fil Hd		1			
	101-14109	Adaptor, Heat Shield		1			
	121-13508	Bearing Assembly	◁	1			
	121-13405	Bearing, Spherical	◁		1		
	121-13509	Bearing Fitting, Body End	◁		1		
	121-13441	Bearing Forging-Fitting	◁			1	
	090-29911-61	Screw, Cap, Sch.		2			MS24678-61
	090-06129-8C	Screw, Cap, Sch.		1			
	121-13510	Bearing Assy-Rod End	◁	1			
	121-13405	Bearing, Spherical	◁		1		
	121-13491	Bearing Fitting, Rod End	◁		1		
	121-13433	Bearing Fitting-Forging	◁			1	
	094-13511	Lock, Rod End		1			
	091-13512	Nut, Rod End		1			
	080-24540-139	O-RING (ROD SEAL)		1			
	082-13354-242	CAP O-RING I.D.		1			

REV LTR	E.O. NO.	DATE	BY / CHK'D / P.A.
M	32,160A	7-28-65	RF / KUREK /
L	32,160	6-23-65	RF / KUREK /
G	Retyped 31,316	2/18/65	JC / RCS /

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MOOG SERVOCONTROLS, INC.

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PARTS LIST

SERVOACTUATOR

MODEL: 17-200B SN 25 & Subq

◁ Critical Parts

PART NO	NAME	NO REQ'D		REMARKS
131-14102	Plate Vernier	1		
090-06132-14C	Screw, Cap, Sch.	2		
092-06091	Washer, No. 10	2		
131-14101	Scale	1		
103-13513	Bracket, Scale	1		
090-13646	Screw, Captive, Sch.	2		
090-06130-24C	Screw, Cap, Sch.	1		
092-06115	Washer, #8	1		
023-12722	Valve-Bleeder	1		
073-20651-4CL	Plug, Bleeder(Test Port)	4		AN814-4CL
080-24540-69	O-ring(Plug & Valve)	5		
074M00437	Nameplate	1		
090-06204	Drive Screw	2		AN535-00-2
131-14099	Sleeve Mid stroke Lock	1		
131-14100	Forging-Mid-stroke Lock		2	
090-13608	Screw, Captive, Sch.	4		
103-13996	Clamp, Mid-stroke Lock	1		
090-14026	Screw, Captive Sch.	16		
010-41745	Servo Valve Assy, Mod 16-140D ◁	1		
080-24540-20	O-ring(Elec. Conn)	1		
090-29911-62	Screw(Valve to Body)	4		MS24678-62
090-06129-44C	Screw(Valve to Guide)	2		
051-13713	Guide Bushing	1		
051-13768	Ferrule & Tube Assy	1		
051-13717	Ferrule		1	
073-13721	Tube Dust Cover		1	
049-13518	Cover	1		
049J01786	Cover, Casting		1	
080-24540-155	O-ring(Cover)	1		
090-06132-20C	Screw, Cap, Sch.	15		
092-06091	Washer, No. 10	15		
090-13597	Screw(Seelskrew)	1		
049-13618	Cover, Shipping	2		
080-24540-116	O-ring(Cover)	2		
092-07110-1	Washer, Flat	8		
090-06129-16C	Screw, Cap, Sch.	8		
094-20120-C20	Lockwire	A/R		MS20995 C20

REV. LTR.	E.O. NO.	DATE	BY	CHK'D	P.A.	REV LTR	E.O. NO	DATE	BY	CHK'D	P.A.
J	32,034	4-[illegible]-65	RF	[illegible]							
G	Retyped 31,316	2/18/65	JC	[illegible]							

RELEASED PER E.O. NO. 30,351 DATE 7/24/64 36

MOOG SERVOCONTROLS, INC.

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: I INPUT CURRENT CIRCUIT

Part: Electrical Connector (061-13496)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
1	Loss of insulation resistance causing short.	No output from torque motor.	Actuator fails to respond to input current signals; piston returns to null position

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: I INPUT CURRENT CIRCUIT

Part: Servovalve Coil Assembly (060-29835-1)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
2	Open in one coil	Slight reduction in gain of torque motor	Slight reduction in positional accuracy capability.
3	Open in both coils	No output from torque motor	Actuator fails to respond to input current signals; piston returns to null position

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: G SERVOVALVE FIRST STAGE

Part: **Assembly (010-41747)**

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
4	**Null bias out-of-tolerance, random and/or indeterminable shift**	**Asymmetrical flow at neutral position**	**Out-of-tolerance null offset of piston.**

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: G SERVOVALVE FIRST STAGE

Part: Torque Motor (029-41755-1)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
5	Fatigue failure of flexure sleeve	Excessive null bias of servo-valve with neutral signal applied	Out-of-tolerance null offset of piston
		Armature goes hard-over with application of signal	Piston position is uncontrollable; will move to hardover position in response to any input signal.
		Excessive leakage into actuator end-housing assembly and loss of system fluid	Loss of control of piston or subsequent limit cycle oscillation; piston will overshoot (step response) and oscillate
6	Asymmetry of air gaps, improper setup, and/or drift	Improper and/or erratic flow	Improper piston position vs input signal;

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: G SERVOVALVE FIRST STAGE

Part Hydraulic Amplifier (070-41986-12-1)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
7	Partial clogging of one nozzle, contamination.	Inadequate pressure applied to one side of second stage spool	Out-of-tolerance null offset of piston
8	Complete clogging of nozzle	Catastrophic null offset	Piston moves to "hard-over" position.
9	Plugged inlet orifice	Improper and/or erratic flow from servovalve	Improper and/or erratic piston position vs input signal
10	Plugged drain orifice	No control from servovalve	Actuator fails to respond to input current signals; piston remains in position acquired at time of failure but will drift under external loading due to valve internal leakage.

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: MG MECHANICAL FEEDBACK MECHANISM, PISTON POSITION TO FIRST STAGE

Part Feedback Springs (110-45185-045/055

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
11	"Surging" at resonance under vibration and loss of one spring	Loss of mechanical feedback gain	Piston position is uncontrollable; piston moves to hardover position
12	"Surging" at resonance under vibration	Spurious signals supplied to torque motor	Erratic piston position vs input command signal;

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: M. MECHANICAL FEEDBACK MECHANISM, PISTON POSITION

Part Cage Preload Spring (110-29670-2)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
13	"Surging" at resonance under vibration	Possible lift off of cam follower, piston in near or full retract position	Erratic piston position vs input command signal;

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: PQ LOAD PRESSURE FEEDBACK, FIRST STAGE

Part SAS Summing Piston (130-29668-1)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
14	Excessive friction and inadequate response to pressure feedback	Inadequate "damping" provided at resonance	Unstable piston position under external load.

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group SLEW STATIC LOAD ERROR WASHOUT, FIRST STAGE

Part Slew Piston Assemblies (111-29686)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
15	Excessive friction in both capacitance pistons	Inadequate slew compensation	Incorrect position under external loading.
16	**Piston seizes in bushing, misalignment**		NOTE: If one piston incurs excessive friction, capacitance of remaining piston is sufficiently high such that failure would probably have negligible influence upon system performance.

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: $\boxed{V_3}$ SERVOVALVE THIRD STAGE

Part Body, Piston, & Spool Assembly (030-41746)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
17	Seizure of spool in bushing		
	a. at null position	No control flow from servo-valve	No response to command input signal; piston remains in null position. Piston remains in whatever command position existed prior to seizure.
	b. at any position other than null	No control flow from servo-valve	Piston moves to hard-over position in one direction.

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: H ACCESSORY FLUID COMPONENTS

Part Static Seals

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
18	**External leakage; seepage from static seals**	No effect on functioning of components	**Excessive external leakage from hydraulic enclosures; could result in loss of system supply pressure and subsequent loss of piston positional control.**

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: $\boxed{A}$ ACTUATOR STRUCTURE

Part Actuator Body (033-14009-1)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
19	**Fatigue fracture at point of stress concentration**	Loss of load carrying capability Loss of hydraulic system pressure	Actuator is inoperable and piston is free to move under any external load.

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: $\boxed{A}$ ACTUATOR STRUCTURE

Part Dynamic Seals, Piston Rod ("o"-ring 080-24540-139)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
20	Excessive leakage, actuator body seals	Loss of hydraulic system pressure.	Piston position is uncontrollable; piston moves to hardover position.

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: $\boxed{A}$ ACTUATOR STRUCTURE

Part **Actuator Cylinder (033-14018)**

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
21	Fatigue fracture at point of stress concentration	Loss of load carrying capability Loss of hydraulic system pressure	Actuator is inoperable and piston is free to move under any external load.

TABLE I

POTENTIAL FAILURE MODES & EFEECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: $\boxed{\Delta}$ ACTUATOR STRUCTURE

Part Tailstock Bearing (121-13405)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
22	Excessive radial looseness	No effect on function	Degradation in dynamic response and positioning accuracy

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: P PISTON

Part Piston Rod Liners (121-13436)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
23	**Excessive wear**	Excessive leakage from piston rod seal May result in misalignment of piston rod and excessive wear of rod seals	External leakage at piston rod seal with possible loss of hydraulic system supply pressure and fire hazard. Piston position may become uncontrollable.

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: P PISTON

Part Piston Rod End Bearing Assembly (121-13510)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
24	Fracture at threaded shank	Loss of load carrying capability	Actuator is inoperable and the piston is free to move under any external load.
25	Excessive radial looseness	No effect on function	Degradation in dynamic response and positioning accuracy (slight null offset).

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-2[illegible]B SERVOACTUATOR

Functional Group: MP MECHANICAL FEEDBACK MECHANISM, PISTON

Part Cam & Cam Guide Assembly (029-14010)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
26	Sudden binding of cam in cam guide	Sudden fracture of cam drive tube and/or the parts which fasten the tube to the cam.	Piston position is uncontrollable; piston moves to hardover position in response to any input signal (open loop operation)
27	Excessive friction, cam in cam guide	Fatigue fracture of cam drive tube and/or the parts which fasten the tube to the cam.	Piston position is uncontrollable; piston moves to hardover position in response to any input signal (open loop operation)

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: MP MECHANICAL FEEDBACK MECHANISM, PISTON

Part Potentiometer & Extension Assembly (120-13507)

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
28	Sensitivity of extension shaft to vibration; excessive amplitude at resonance	Ultimate fatigue failure of extension shaft and/or connections Spurious motion of wiper carriage	Erratic piston position telemetry signal. Excessive noise in piston position telemetering signal.

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: EP ELECTRICAL OUTPUT, PISTON POSITION

Part No.	Potentiometer (062-13999) Potential Failure	Effect on Function	Effect on Actuator Performance
29	Open circuit, element	No voltage across element	No piston position telemetering signal
30	Open circuit, wiper and/or fracture of wiper arm	No voltage output from wiper circuit	No piston position telemetering signal
31	Excessive wear of wiper arm and/or element	Excessive and erratic contact resistance, wiper arm with element	Excessive noise in telemetering signal
32	Degradation in electrical parameters, linearity, resolution	Improper voltage output vs wiper position	Improper telemetering signal
33	Sensitivity to vibration, resonance of case, element, and wiper carriage	Spurious output dependent upon amplitude at resonance	Excessive noise in telemetering signal

TABLE I

POTENTIAL FAILURE MODES & EFFECTS, MODEL 17-200B SERVOACTUATOR

Functional Group: EP ELECTRICAL OUTPUT, PISTON POSITION

Part Connector

No.	Potential Failure	Effect on Function	Effect on Actuator Performance
34	**Open circuit at terminals and/or in wiring**	**No output from wiper** **No excitation voltage to element**	**No piston position telemetering signal.**

TABLE II

Summary of components which are exempted from contribution to significant failure modes because they are parts for which analysis or testing has assured adequate safety margins.

Part Number	Part Name
031-42880-1	Body, Servovalve
121-29647-1	Sleeve-Summing Piston
052-42776-1	Spool
049-29637-1	End Cap-Spool, RH
049-29638-1	End Cap-Spool, LH
049-29636-3	End Cap-Filter (press End)
049-41748	Cap, Filter
071-29671-1	Filter, Inlet
110-29670-1	Spring, Compression
090-29650-1	Screw, Retaining
020-29672-1	Filter Assembly
071-29674-1	Filter
071-29643-1	Retaining Screw - Filter
021-45336-1	1st & 2nd Stage Assembly
031-45338-1	Body, Bushing & Spring Cup Assy.
031-41452-1	Body Assembly
031-41452-2	Body
093-28472-D0635	Plug
050-42685-1C-10	Bushing & Spool Assy.
051-42678-1	Bushing
052-41453	Spool
043-41447	End Cap
043-41456	End Cap, Adjustor Side
110-41465-2	Spring, Helical, Compr. Spool Return
020-26023-75	Orifice Assembly
071-26012	Body, Orifice
071-22286	Orifice
071-26014	Filter
049-41455	Cover, Filter
090-25606-10	Screw

TABLE II (cont'd.)

Part Number	Part Name
072-41750	Flapper
072-41654	Armature
110-29678-075/085	Wire, Feedback, Spool
110-29679-290/310	Wire, Feedback, Su. Piston
072-29842-1	Magnet, Permanent
120-45293	Cage & Follower Assembly
120-45292-1	Cage Assembly
120-41802	Cage End
120-45297	Yoke, Spring Cage
120-41804	Extension, Cam Follower
091-29733-1	Nut, Adjustor
110-29719-1	Leaf Spring
110-29720-1	Leaf Spring
120-44388	Cam Follower Assembly
120-44386	Shaft
120-44385	Roller, Cam Follower
121-44387	Clevis
090-06130-10C	Screw Cap, Sch. (Leaf Spring)
111-44326-1	Pivot, (Lower)
090-29728-1	Screw Adjustor
111-44327	Pivot Flapper
111-44329	Pivot
103-29818-1	Bracket Assembly
103-42837	Bracket-Cam HSG Support
093-29814-1	Dowel (Bracket to Cam Guide)
094-41371	Ring, Retaining
	potting Compound 2651
010-14008-1	Actuator Assembly
032-13860-4	Body & Spool Assembly
052-14097	Spool (Prefiltration)
029-42253	Spool & Knob Assembly
052-13494	Spool (Cyl. By-pass)
032-13875-3	Body Assembly
049-13502	Cap, Cyl By-Pass
049-13504	Cap, Cyl By-Pass
049-13499	Cap, Prefiltration
049-13500	Cap, Prefiltration Valve

TABLE II (cont' d.)

Part Number	Part Name
121-13436	Liner-Bearing, Piston
082-20036-274	Ring-Back-up
071-13365-1	Element-Filter
091-13526	Nut-Filter Retainer
110-29688-1	Spring-Helical, Compr
050-41131	Piston & Sleeve Assy
130-29690	Piston
049-29689	Cap-Spring
094-29692	Retainer
073-45246	Cam Drive Tube Assembly
073-13450	Tube, Cam Drive
130-14013	Rod-Piston
091-13472	Nut-Jam
073-13459-1	Union (Cyl to Act. Body)
080-42900-242-2	Seal, Quad. Ring(Cyl to Piston Rod)
131-09084-37	Ring-Scraper
073-29711	Union
049-14023	Cap
121-13509	Bearing Fitting, Body End
090-29911-61	Screw, Cap, Sch.
090-06129-8C	Screw, Cap, Sch.
121-13405	Bearing, Spherical
121-13491	Bearing Fitting, Rod End
094-13511	Lock, Rod End
091-13512	Nut, Rod End
090-29911-62	Screw (Valve to Body)
090-06129-44C	Screw (Valve to Guide)

TABLE III

Summary of components which are exempted from contribution to significant failure modes because they are parts for which failure will not cause the actuator performance to be outside of the specification.

Part Number	Part Name
090-06132-14C	Screw, Cap, Sch
071-29695-1	Filter Support
090-06132-14C	Screw, Cap, Sch
111-29644-1	Spring Seat, Summing Piston
111-29646-1	Pivot-Adjustor
111-28002-1	Pivot
112-29649-1	Retaining Collar
112-29648-1	Spring Cup
093-29693-1	Plug
071-29642-1	Orifice Body
071-29641-1	Filter Retainer - Upper
071-29640-1	Filter Retainer - Lower
112-45323	Cup, Spring
112-45337-1	Spring Cup & Clinch Nut Assy.
111-45322	Cup, Spring-Adjustor
091-25527	Nut Clinch
090-20054-AC6-H4	Screw, Adjustor
111M00351	Adjustor
090-06130-9S	Screw
090-06130-12S	Screw
111-41449	Pivot
111-41466	Seat, Spring, Spool Return
071-41457	Retainer, Orifice
090-07587-9C	Screw - Flexure Sleeve
092-04548	Washer, Lock
072-45217-9	Polepiece & Stop Assembly
072-29841-3	Polepiece
072-41538-9	Stop, Armature
072-29841-3	Piston
102-29847	Spacer - Motor
060-29831-1	Form, Coil
090-06141-32S	Screw - Motor

TABLE III (cont' d.)

Part Number	Part Name
064-06089-10	Tubing, Teflon (Approx 18"lg.)
064-06089-15	Tubing, Teflon (4 pcs Approx 1"lg)
094-20120C20	Lockwire
090-06132-38S	Screw Mounting
049-44118	Cover
090-06141-10C	Screw
092-29724-1	Retainer
111-44325	Seat, Spring
090-06129-16C	Screw Cap, Sch., (Brkt. to Body)
073-13459-1	Union
073-29711-1	Union
103-24927	Clamp, Cable
103-41103-1	Clamp, Cable
092-41104-1	Washer
090-24951-8C	Screw, Button Hd.
074-20382	Nameplate
090-06204	Drive Screw
MS0995C20	Lockwire
049-13465	Knob
093-02454-6	Roll Pin(ESNA No. 79-018-078-0687)
094-29115	Insert Fastener
071-14036	Orifice Bushing
110-13503	Spring, Detent
090-06127	Screw, Cap, Sch
092-06115	Washer, #8
082-29969-0-1368	Cap -O-ring O. D.
121-13811	Sleeve, Prefiltration
049-13632	Button, Spool, Prefiltration
092-07110-1	Washer, Flat
090-06129-20C	Screw, Cap, Sch
102-14132	Spacer
091-14133D428	Nut
081-14135	Washer
023-13725-1	Check Valve Assy (Act. Body)
049-11307	Cap
110-11351	Spring
072-11308	Flapper

TABLE III

Part Number	Part Name
111-11317	Seat
090-13684	Screw (Mach. , Fillister Hd.)
103-13454	Trunnion
090-13498	Screw, Cap, Sch, Hd
082-13736	Ring-Back-up Filter
082-20036-142	Ring-Back-up (Filter, Large End
082-20036-131	Ring-Back-up
111-29687	Seat-Spring
051-29691	Sleeve
103-13992	Bracket, Cam Guide
071-13993	Ring, Cam Guide
082-13505	Scraper Ring (Cam)
121-13439	Liner-Bearing, Cam Guide
073-45244	Collar, Cam Drive
073-45245	Fitting, Cam Drive
090-06130-12C	Screw, Cap, Sch
093-02454-4	Roll Pin (ESNA-79-018-078-0312)
121-13578	Ring-Bearing
090-06276-9C	Screw, Cap. Sch. (Potentiometer)
023-12275	Check Vent Assembly
083-42983	Diaphragm Assembly
083-12084-5	Diaphragm
090-12244	Screw-Vent
102-12226	Spacer
083-12245	Seal, Washer
023-13725-2	Check Valve Assembly (Cylinder)
049-11307	Cap
110-11351	Spring
072-11308	Flapper
111-11317	Seat
090-06129-16C	Screw, Cap, Sch.
092-07110-1	Washer
090-29911-69	Screw, Cap, Sch.
090-13684	Screw, Machine Fil Hd
101-14109	Adaptor, Heat Shield

TABLE III (cont'd.)

Part Number	Part Name
131-14102	Plate Vernier
090-06132-14C	Screw, Cap. Sch.
092-06091	Washer, No. 10
131-14101	Scale
103113513	Bracket, Scale
090-13646	Screw, Captive, Sch.
090-06130-24C	Screw, Cap. Sch.
092-06115	Washer, #3
023-12722	Valve-Bleeder
073-20651-4CL	Plug, Bleeder (Test Port)
074M00437	Nameplate
090-06204	Drive Screw
131-14099	Sleeve Mid-stroke Lock
090-13608	Screw, Captive, Sch.
103-13996	Clamp, Mid-stroke Lock
090-14026	Screw, Captive Sch.
051-13713	Guide-Bushing
051-13768	Ferrule & Tube Assy.
051-13717	Ferrule
073-13721	Tube Dust Cover
049-13518	Cover
090-06132-20C	Screw, Cap, Sch.
092-06091	Washer, No. 10
090-13597	Screw (Seelskrew)
049-13618	Cover, Shipping
092-07130-1	Washer, Flat
094-20120-C20	Lockwire

TABLE IV
PROBABILITY OF THE EXISTENCE OF A CAUSE OF FAILURE
TITAN III SERVOACTUATOR
MODEL 17-185

Group	No.	Number of Preflight Failures			P_r (a) x 10^{-3}
		GCO	SF	CD	
I	1	1	-	1	2
	2	-	-	-	-
	3	1	-	-	1
G	4	3	-	-	3
	5	-	-	-	-
	6	2	-	4	6
	7	2	-	-	2
	8	2	-	-	2
	9	2	-	-	2
	10	-	-	-	-
MG	11	-	-	-	-
	12	-	-	-	-

TABLE IV
PROBABILITY OF THE EXISTENCE OF A CAUSE OF FAILURE
TITAN III SERVOACTUATOR
MODEL 17-185

Group	No.	Number of Preflight Failures			P_r (a) x 10^{-3}
		GCO	SF	CD	
M	13	-	-	-	-
PQ	14	N. A.	N. A.	N. A.	-
SLEW	15	N. A.	N. A.	N. A.	-
	16	N. A.	N. A.	N. A.	-
V_3	17	-	-	2	2
H	18	5	1	-	6
A	19	-	-	-	-
	20	-	-	-	-
	21	-	-	-	-
	22	-	-	-	-

TABLE IV
PROBABILITY OF THE EXISTENCE OF A CAUSE OF FAILURE
TITAN III SERVOACTUATOR
MODEL 17-165

Group	No.	Number of Preflight Failures			P_r (a) x 10^{-3}
		GCO	SF	CD	
P	23	9	-	1	10
	24	-	-	-	-
	25	-	-	-	-
MP	26	-	-	-	-
	27	-	-	-	-
	28	-	-	-	-
EP	29	-	-	-	-
	30	-	-	-	-
	31	-	-	-	-
	32	-	-	-	-
	33	-	-	-	-
	34	-	-	-	-

TABLE V

PROBABILITY OF FAILURE
MODEL 17-200B SERVOACTUATOR

Group	No.	$P_r (a) \times 10^{-3}$	Flight Prediction	
			$P_r (A_f)$	Value $\times 10^{-5}$
MG	11	Negl.	- - -	---
	12	Negl.	- - -	---
M	13	Negl.	- - -	---
PQ	14	.5	$P (a_{14}) P (a_{14} - u)$	.8
SLEW	15	Negl.	- - -	---
	16	Negl.	- - -	---
V_3	17	2	$P (a_{17}) P (a_{17} - u)$	3 2
H	18	6	$P (a_{18}) P (a_{18} - u)$	9.6
A	19	1	$P (a_{19}) P (a_{19} - u) G_c$	.02
	20	1	$P (a_{20}) P (a_{20} - u) G_c$	.02

TABLE V

PROBABILITY OF FAILURE
MODEL 17-200B SERVOACTUATOR

Group	No.	P_r (a) x 10^{-3}	Flight Prediction. P_r (A_f)	Value x 10^{-5}
I	1	2	$P(a_1) P(a_1 - u)$	3.2
	2	1	$[P(a_2) P(a_2 - u)]^2$	---
	3	1	$[P(a_3) P(a_3 - u)]^2$	---
G	4	3	$P(a_4) P(a_4 - u)$	4.8
	5	1	$P(a_5) G_c$	1.0
	6	6	$P(a_6) P(a_6 - u)$	9.6
	7	2	$P(a_7) P(a_7 - u)$	3.2
	8	2	$P(a_8) P(a_8 - u)$	3.2
	9	2	$P(a_9) P(a_9 - u)$	3.2
	10	..5	$P(a_{10}) P(a_{10} - u) G_v$	.34

TABLE V

TABLE V

PROBABILITY OF FAILURE
MODEL 17-200B SERVOACTUATOR

Group	No.	$P_r(a) \times 10^{-3}$	Flight Prediction $P_r(A_f)$	Value $\times 10^{-5}$
A (cont'd.)	21	1	$P(a_{21}) P(a_{21}-u) G_c$	.02
	22	1	$P(a_{22}) P(a_{22}-u)$	1.6
P	23	10	$P(a_{23}) P(a_{23}-u) G_c$	.16
	24	.5	$P(a_{24}) P(a_{24}-u) G_v$	.34
	25	1	$P(a_{25}) P(a_{25}-u)$	1.6
MP	26	.5	$P(a_{26}) P(a_{26}-u)$	.8
	27	Negl.	---	---
	28	2	$P(a_{28}) P(a_{28}-u) G_v$	1.35
EP	29	1	$P(a_{29}) P(a_{29}-u)$	1.6
	30	1	$P(a_{30}) P(a_{30}-u)$	1.6
	31	.5	$P(a_{31}) P(a_{31}-u)$	.8

TABLE V

PROBABILITY OF FAILURE
MODEL 17-200B SERVOACTUATOR

Group	No.	$P_r(a) \times 10^{-3}$	Flight Prediction	
			$P_r(A_f)$	Value $\times 10^{-5}$
EP (cont'd.)	32	.5	$P(a_{32}) P(a_{32} - u)$	.8
	33	2	$P(a_{33}) P(a_{33} - u) G_v$	1.35
	34	1	$\left[P(a_{34}) P(a_{34} - u)\right]^2$	---
			TOTAL	54.20

APPENDIX I

MARGIN OF SAFETY ANALYSIS

STRESS CALCULATIONS

DEFINITION OF SYMBOLS

A	area, in^2
A_1	amplification factor
a	outside radius, in.
a_1	outside diameter of cylinder flange, in.
b	inside radius, in.
b_1	bolt circle diameter of cylinder flange, in.
C	distance from neutral axis to fibre of maximum stress, in.
C_1	end fixity coefficient
D_1	flexural rigidity of head
D_2	flexural rigidity of cylinder
d	mean diameter of cylinder, in.
E	modulus of elasticity, psi.
e	eccentricity, in.
F	load, lb.
F_e	endurance limit, psi.
F_{su}	ultimate shear, psi.
F_{tu}	ultimate tensile stress, psi.
F_{ty}	yield tensile stress, psi.
f	stress, psi.
f_b	bending stress, psi.
F_{cy}	yield compressive stress, psi
F_{sy}	yield shear stress, psi

DEFINITION OF SYMBOLS

f_r	radial stress, psi.
f_s	shearing stress, psi
f_t, f_λ	tangential stress, psi
f_x	longitudinal stress, psi
F_c	compressive stress
g	acceleration of gravity, ft. /sec.^{2} - 32.2
h	cylinder flange thickness, in.
I	moment of inertia, in.^{4}
K_r	stress multiplication factor
K_s	stress concentration factor
K_t	spring rate, lb. /in.
K. E.	kinetic energy, in. -lb.
L	length, in.
M	bending moment, in. -lb.
M_1	axial bending moment per unit length of circumference at inner edge, in. -lb. /in.
M_2	axial bending moment per unit length of circumference on head at junction, in. -lb. /in.
M_o	bending moment per unit length of circumference exerted by head on cylinder, in. -lb. /in.
M_r	radial bending moment per unit length of circumference in head, in. -lb. /in.

DEFINITION OF SYMBOLS

M_t	tangential bending moment per unit of radius in head, in. -lb. /in.
M_x	longitudinal bending moment per unit length of circumference in cylinder, in. -lb. /in.
MS	margin of safety
m	$\frac{1}{\mu}$
N	number of bolts in flange attachment
N_1	radial force per unit length of circumference on midplane of head at junction, lb. /in.
N_o	axial force per unit length of circumference acting on cylinder positive when tension, lb. /in.
P	load, lb.
p	internal pressure, psi
p_B	burst pressure, psi.
p_p	proof pressure, psi.
Q_o	shear force per unit length of circumference exerted by head on cylinder, lb. /in.
R	load, lb.
r	mean radius of cylinder, in.
r_a	outer radius, in.
r_i	inner radius, in.
t	thickness, in.
t_1	thickness of cylinder head, in.

DEFINITION OF SYMBOLS

t_2	thickness of cylinder, in.
V	velocity, in. /sec.
W	actuator weight, lb.
W_1	weight of driven mass, lb.
W_3	load, lb.
W_o	radial displacement of cylinder at juncture, positive inward, in.
W'_o	rotation or slope of cylinder at juncture
X_1	maximum air gap between armature and polepiece stop, in.
Z	section modulus, in. ^{3}
$\frac{dw}{dr}$	rotation at the edge of head, $r = \frac{d}{2}$
δ	radial displacement of midplane of head, positive outward, in.
δ_1	radial displacement of surface of head acted upon by pressure, positive outward, in.
δ_2	deflection of cam follower, in.
δ_{tp}	dimensional change due to temperature differential, in.
Δ	deflection, in.
μ	poisson's ratio
ζ	hyperbolic function = $(\sin \beta x) e^{-\beta x}$
ρ	$\sqrt{\frac{I}{A}}$ - radius of gyration, in.
ρ'	coefficient of thermal expansion, in. /in. /° F.
θ	angle of rotation, radians
λ	hyperbolic function = $(\cos \beta x + \sin \beta x) e^{-\beta x}$

MARGIN OF SAFETY

Stress analyses of the major structural elements of the Model 17-200 servoactuator were carried out as Task 1 of the reliability analysis. The results of each stress analyses is presented as margin of safety (MS). As discussed in reference 26, margin of safety represents the ratio of excess strength to the required strength and was calculated as follows:

$$MS = \frac{F}{f} - 1$$

where: F = allowable stress

f = operating stress

MS = margin of safety

DESIGN CRITERIA

a. Pressure Rating

Pressures	Supply Pressure	Return Pressure
Rated pressure	2000 ± 200	20 to 100
Proof	3300	1000
Burst	6000	2000

b. Pressure Design Criteria

yield pressure $= p_y = p_p$
ultimate pressure $= p_b$

$MS \geq 1$ yield and ultimate

c. Structural Design Criteria

yield load $= P_y = 72{,}000$ lb.
ultimate load $= P_u = p_o A_p$

$MS \geq 1$ yield and ultimate

d. Fatigue Design Criteria

Stresses will be calculated on the basis of maximum operating pressure and must be:

$f_f \leq$ endurance limit of the material

THERMAL ENVIRONMENT

Elevated temperature (275° F) material properties were used in all calculations. These properties were taken from MIL-Handbook-5 and represent the minimum strength to be expected for the material.

TABLE VI

SUMMARY OF MINIMUM MARGINS OF SAFETY

Part Name	Moog Part Number	Minimum MS
Cylinder	033-14018	-.16
Piston Head	130-14013	.15
Piston Shaft	130-14013	.88
Actuator Body	033-14009	1.65
Rod End	121-13510	.99
Tailstock	121-13508	.84
Flexure Sleeve	070-41751	2.80

1.0 ACTUATOR CYLINDER P/N 033-14018

1.1 Discussion

The actuator cylinder, with integral head, is forged from 4340 steel. The cylinder is bolted to the actuator body through an external flange. The cylinder head is designed as a flat circular plate with a circular hole at the center. For this analysis, the head is assumed fully restrained at the inner edge with the external edge considered partially restrained at the juncture with the cylinder. The head thickness at the relief radius is considered constant for the entire head. The actual thickness of the head is sufficient to make stresses calculated from the assumed thickness conservative.

1.2 Loading

Figure 3 shows the loads acting on the cylinder. The loading consists of a uniform pressure p which, acting alone, produces a uniform expansion of the cylinder; a bending moment per unit length of circumference M_o; a shear force per unit length of circumference Q_o; and an axial force per unit length N_o. The axial force N_o is produced by the pressure acting on the head which tends to stretch the cylinder. The shear force Q_o and bending moment M_o are produced by the restraint exerted by the head in preventing the expansion of the cylinder under pressure.

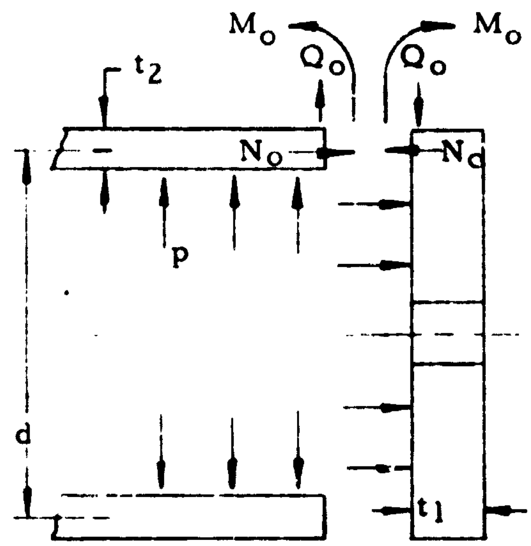


Figure 3

Figure 3 also shows the loads applied to the head which is regarded as a thin elastic plate, fixed at the inner edge, carrying a uniformly distributed load. The equilibrium of forces axially determines N_o.

The internal pressure loads used to determine stress levels are:

$P_{yield} = P_p = 3300$ psi

$P_{ultimate} = P_B = 6000$ psi

ACTUATOR CYLINDER

1.3 Material Allowables

Material: 4340 steel (H. T. R_c 34 to 38)

F_{ty} = 130,000 psi at 80° F; 123,500 psi at 275° F

F_{tu} = 155,000 psi at 80° F; 147,200 psi at 275° F

F_{cy} = 130,000 psi at 80° F; 123,500 psi at 275° F

F_{su} = 97,800 psi at 80° F; 92,800 psi at 275° F

F_{sy} = 82,200 psi at 80° F; 78,000 psi at 275° F

F_e = 77,500 psi

ACTUATOR CYLINDER

1.4 Stress Calculations

1.4.1 Head Cylinder End

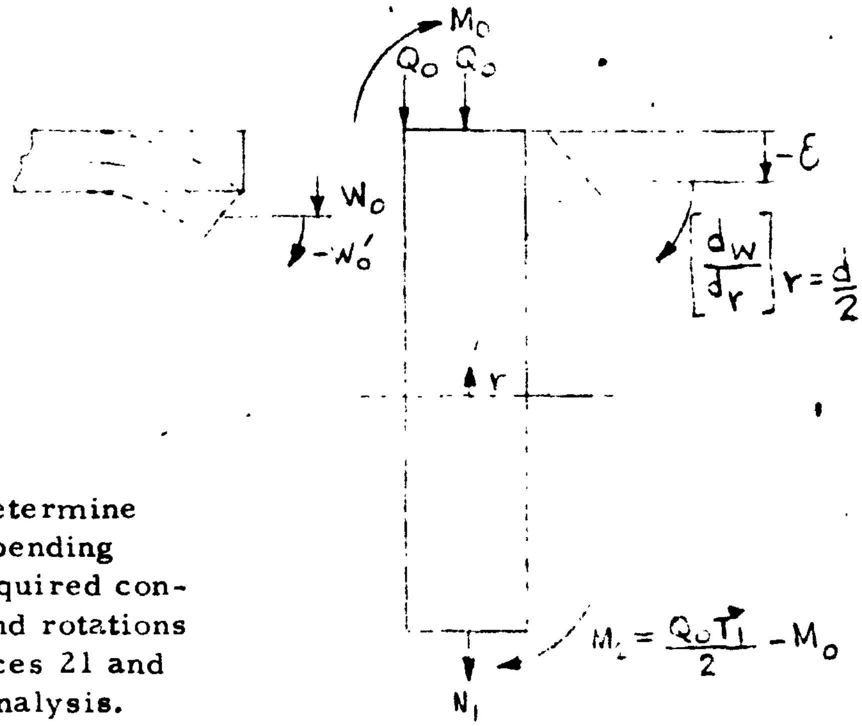


Figure 4

It is first necessary to determine the shear force Q_o, and bending moment M_o, from the required continuity of displacement and rotations at the junction. References 21 and 22 will be used for this analysis.

From the theory of a cylindrical shell, the radial displacement (positive inward) W_o and the rotation at the junction are (reference 21, page 393):

$$W_o = -\frac{1}{2\beta^3 D_2}(\beta M_o + Q_o) - \frac{Pd^2}{4Et_2}\left(1-\frac{\mu}{2}\right)$$

$$-W_o' = -\frac{1}{2\beta^2 D_2}(2\beta M_o + Q_o)$$

$$\text{where } \beta = \left[\frac{12(1-\mu^2)}{(dt_2)^2}\right]^{1/4} = 0.952$$

$$D_2 = \frac{Et_2^3}{12(1-\mu^2)} = 0.1216 \times 10^6$$

$$\mu = .3$$

$$d = 9.8$$

$$E = 28.5 \times 10^{-6}$$

$$t_2 = .36$$

ACTUATOR CYLINDER

The equations for W_o and $-W_o$ include the effect of N_o. From equilibrium of forces in an axial direction $N_o = \frac{pd}{4}$

$$W_o = -\frac{(.952\,M_o + Q_o)}{2(.952)^3\,(.1216 \times 10^6)} - \frac{(9.8)^2\,p\,(1 - \frac{3}{2})}{4(28.5 \times 10^6)(.36)}$$

$$W_o = -1.993 \times 10^6 p - 4.54 \times 10^6 M_o - 4.77 \times 10^{-6} Q_o$$

$$-W_o' = -\frac{[2(.952)\,M_o + Q_o]}{2(.952)^2\,(.1216 \times 10^6)}$$

$$-W_o' = -8.64 \times 10^{-6} M_o - 4.54 \times 10^{-6} Q_o$$

The head, treated as a thin elastic plate with a fixed inner edge, deflects under pressure and bending moment at the outer edge. To determine the rotation at the edge, $r = d/2$, it is necessary to use the method of superposition as outlined in reference 21, pages 61 through 67. Using this method, it is necessary to superimpose on the rotation at the edge obtained for the plate without a fixed inner edge the rotation produced by the bending moments and shear forces shown in Figure 5.

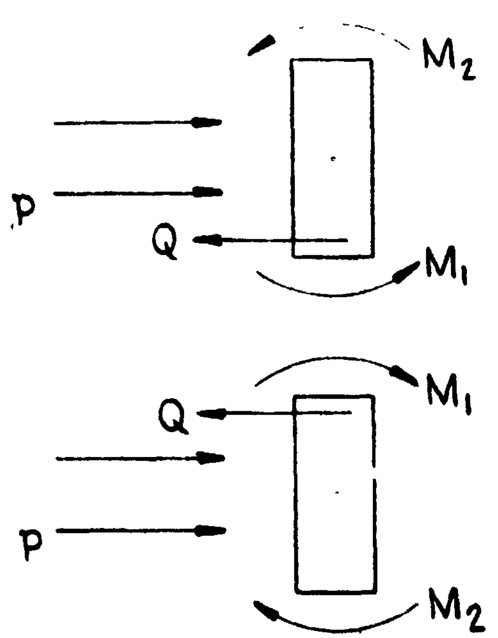
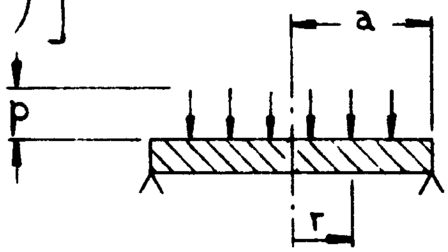


Figure 5

ACTUATOR CYLINDER

For the case of a uniformly loaded circular plate with supported edges (reference 21, page 61 - 62):

$$\left[\frac{dw}{dr}\right]_1 = \frac{rp}{16 D_1}\left[r^2 - a^2\left(\frac{3+\mu}{1+\mu}\right)\right]$$



where $E = 28.5 \times 10^6$

$t_1 = .97$

$\mu = .3$

$a = r = d/2 = 4.9$

$$D_1 = \frac{E t_1^3}{12(1-\mu^2)} = 1.954 \times 10^6$$

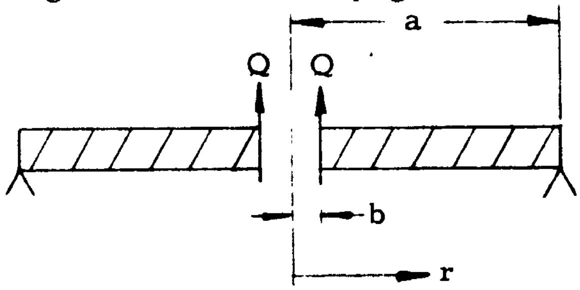
at $r = d/2$

$$\left[\frac{dw}{dr}\right]_1 = \frac{4.9\,p}{16(1.954 \times 10^6)}\left[\overline{4.9}^2 - \overline{4.9}^2\left(\frac{3.3}{1.3}\right)\right]$$

$$\left[\frac{dw}{dr}\right]_1 = -5.79 \times 10^{-6}\,p$$

ACTUATOR CYLINDER

For the case of a plate with a shearing force Q distributed along the inner edge (reference 21, pages 64 - 65):



$$\left[\frac{dw}{dr}\right]_2 = \frac{b^2\, r\, p}{8D_1}\left\{(1 - 2\ln\frac{r}{a}) + (\frac{1-\mu}{1+\mu}) - \frac{2b^2}{a^2-b^2}\ln\frac{b}{a} - \frac{2}{r^2}\,\frac{(1+\mu)}{(1-\mu)}\,\frac{a^2 b^2}{a^2-b^2}\ln\frac{b}{a}\right\}$$

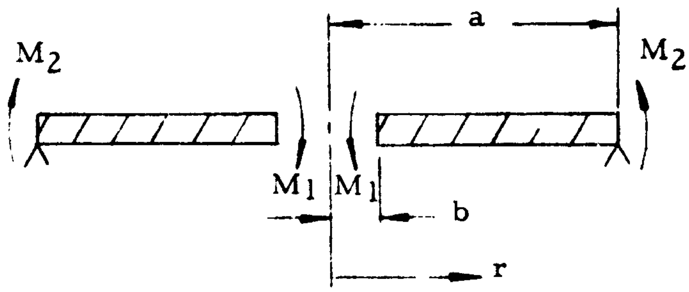
at $r = \frac{d}{2}$, $b = 2.375$ in.:

$$\left[\frac{dw}{dr}\right]_2 = \frac{(2.375)^2(4.9)p}{8\,(1.954\times 10^6)}\left\{1 + (\frac{.7}{1.3} - \frac{2\,(2.375)^2(-.724)}{(\overline{4.9}^2 - \overline{2.375}^2)} - \frac{2}{(4.9)^2}\left[(\frac{1.3}{.7})\,\frac{(4.9)^2(2.375)^2}{(\overline{4.9}^2 - \overline{2.375}^2)}\,(-.724)\right]\right\}$$

$$\left[\frac{dw}{dr}\right]_2 = 4.98\times 10^{-6}\ p$$

ACTUATOR CYLINDER

For the case of a circular plate with the moments M_1 and M_2 uniformly distributed along the inner and outer boundaries, respectively, (reference 21, pages 63 and 64):



$$\left[\frac{dw}{dr}\right]_3 = -\frac{1}{D_1(a^2 - b^2)}\left\{\frac{r(a^2 M_2 + b^2 M_1)}{1+\mu} + \frac{a^2 b^2 (M_2 + M_1)}{r(1-\mu)}\right\}$$

at $r = d/2$:

$$\left[\frac{dw}{dr}\right]_3 = \frac{-1}{(1.954 \times 10^6)(\overline{4.9}^2 - \overline{2.375}^2)}\left\{\frac{4.9\left[\overline{4.9}^2 M_2 + \overline{2.375}^2 M_1\right]}{1.3} + \frac{(4.9)^2 (2.375)^2 (M_2 + M_1)}{4.9(.7)}\right\}$$

$$\therefore \left[\frac{dw}{dr}\right]_3 = -1.695 \times 10^{-6} M_1 - 3.62 \times 10^{-6} M_2$$

ACTUATOR CYLINDER

From reference 21, page 66:

$$M_1 = \frac{b^2 p}{8\left[(1+\ )\frac{a^2}{b^2} + 1 - \mu\right]} \left\{ 2(1-\mu)\left(\frac{a^2}{b^2} - 1\right) + 4(1+\mu)\frac{a^2}{b^2} \ln \frac{a}{b} \right\}$$

$$M_1 = \frac{(2.375)^2 p}{8\left[\frac{(1.3)(4.9)^2}{(2.375)^2} + .7\right]} \left\{ 2(.7)\left[\frac{(4.9)^2}{(2.375)^2} - 1\right] + 4(1.3)\frac{(4.9)^2}{(2.375)^2}(.72) \right\}$$

$$M_1 = 2.31\ p$$

$$M_2 = Q_o \frac{t_1}{2} - M_o$$

$$M_2 = .485\ Q_o - M_o$$

$$\left[\frac{dw}{dr}\right]_3 = -3.91 \times 10^{-6} p - 1.757 \times 10^{-6} Q_o + 3.62 \times 10^{-6} M_o$$

$$\left[\frac{dw}{dr}\right]_{r=\frac{d}{2}} = \sum \left[\frac{dw}{dr}\right]_n$$

$$\left[\frac{dw}{dr}\right]_{r=\frac{d}{2}} = -4.72 \times 10^{-6} p - 1.757 \times 10^{-6} Q_o + 3.62 \times 10^{-6} M_o$$

ACTUATOR CYLINDER

The effect of a uniform tension in the midplane o. the plate of amount N_1 is to produce a radial displacement of amount:

$$\delta = \frac{1}{E} (1 - \mu) \frac{N_1}{T_1} r$$

This is the displacement of the center line of the head. The displacement of the edge abutting the cylinder is:

$$\delta_1 = \delta + \frac{t_1}{2} \left[\frac{dw}{dr}\right]_{r = \frac{d}{2}}$$

$$N_1 = -Q_o$$

$$\delta_1 = \frac{-.7 (4.9) Q_o}{28.5 \times 10^{-6}(.97)} + \frac{.97}{2} \left[-4.72 \times 10^{-6} p - 1.757 \times 10^{-6} Q_o + 3.62 \times 10^{-6} M_o\right]$$

$$\delta_1 = -2.29 \times 10^{-6} p - .949 \times 10^{-6} Q_o + 1.757 \times 10^{-6} M_o$$

Figure 2 shows the details at the junction and indicates the positive sense of W_o, - W_o', δ, and $\left[\frac{dw}{dr}\right]_{r = \frac{d}{2}}$ The conditions to be satisfied at the junction are:

$$W_o = -\delta_1 = -\delta - \frac{t_1}{2} \left[\frac{dw}{dr}\right]_{r = \frac{d}{2}}$$

$$-1.993 p - 4.54 M_o - 4.77 Q_o = 2.29 p + .949 Q_o - 1.757 M_o$$

$$2.783 M_o + 5.719 Q_o = -4.28 p$$

$$M_o + 2.055 Q_o = -1.537 p$$

ACTUATOR CYLINDER

and

$$-W'_o = \left[\frac{dw}{dr}\right]_{r = \frac{d}{2}}$$

$$-8.64\ M_o - 4.54\ Q_o = -4.72\ p - 1.757\ Q_o + 3.62\ M_o$$

$$-12.26\ M_o - 2.783\ Q_o = -4.72\ p$$

$$-M_o - .227\ Q_o = -.385\ p$$

$$Q_o = -1.05\ p$$

$$M_o = .62\ p$$

ACTUATOR CYLINDER

1.4.1.1. Stress in the Cylinder

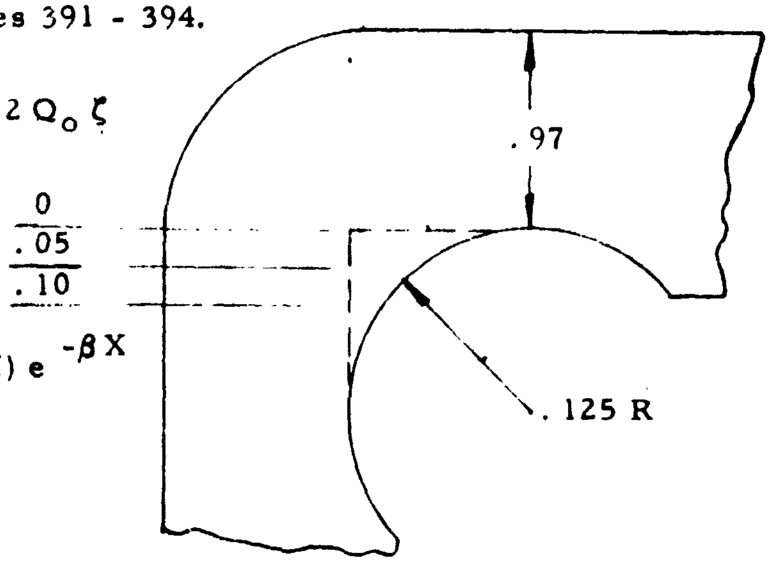
From reference 15, pages 391 - 394.

$$M_x = \frac{1}{2\beta} \quad 2\beta M_o \lambda + 2 Q_o \zeta$$

$$M_\zeta = \mu M_x$$

$$\lambda = (\cos \beta X + \sin \beta X)\, e^{-\beta X}$$

$$\zeta = (\sin \beta X)\, e^{-\beta X}$$



X	βX	λ	ζ	M_x	M_λ	t_2	f_x	f_λ
.05	.0476	.998	.0454	.569	.171	.387	29.13	13.18
.10	.0952	.990	.087	.518	.155	.36	30.81	13.99
.15	.1438	.982	.1242	.472	.142	.36	28.66	13.38
.20	.1904	.967	.1573	.425	.127	.36	26.48	12.69

ACTUATOR CYLINDER

The maximum combined longitudinal stress occurs at 0. 1 inches from the juncture of head and cylinder.

$$f_x = \frac{6 M_x}{t_2{}^2} + \frac{pR}{2 t_2}$$

$$f_x = \frac{6(.518)\,p}{(.36)^2} + \frac{4.9\,p}{2(.36)} = 30.81\,p$$

The maximum combined longitudinal yield stress for $p_y = 3300$ psi is:

$$(f_x)_y = 30.81\,p_y = 30.81\,(3300) = 101,800 \text{ psi}$$

The maximum combined longitudinal ultimate stress for p_u = 6000 psi is:

$$(f_x)_u = 30.81\,p_u = 30.81\,(6000) = 184,860 \text{ psi}$$

The maximum combined longitudinal operating stress for p_o = 2200 psi is:

$$(f_x)_o = 30.81\,(2200) = 67,800 \text{ psi}$$

The margin of safety for combined longitudinal stress

$$\text{yield MS} = \left[\frac{130,000}{101,800} - 1\right] = \underline{\underline{.28}}$$

$$\text{ultimate MS} = \left[\frac{155,000}{184,860} - 1\right] = \underline{\underline{-.16}}$$

$$\text{fatigue limit MS} = \frac{77,500}{67,800} = 1.14$$

The maximum combined tangential stress occurs at . 1 inches from the juncture of head and cylinder:

$$f_\lambda = \frac{pR}{t_2} + \frac{6 M_\lambda}{t_2{}^2}$$

$$f_\lambda = \frac{4.9\,p}{(.36)} + \frac{6(.155)\,p}{(.36)^2} = 13.99\,p$$

ACTUATOR CYLINDER

The maximum combined tangential yield stress for p_y = 3300 psi is:

$$(f_\lambda)_y = 13.99\ (3300) = 46,200 \text{ psi}$$

The maximum combined tangential ultimate stress for p_u = 6000 psi is:

$$(f_\lambda)_y = 13.99\ (6000) = 83,940 \text{ psi}$$

The maximum combined longitudinal operating stress for p_o = 2200 psi is:

$$(f_x)_o = 13.99\ (2200) = 30,800 \text{ psi}$$

The margin of safety for maximum combined tangential stress:

$$\text{yield MS} = \left[\frac{130,000}{46,200} - 1\right] = \underline{1.81}$$

$$\text{ultimate MS} = \left[\frac{155,000}{83,940} - 1\right] = \underline{.85}$$

The radial shear stress at the juncture of head and cylinder is:

$$f_s = \frac{Q_o}{t_2} = \frac{-1.05\ p}{.36} = 2.92\ p$$

The yield and ultimate radial shear stresses are:

$$(f_s)_y = 2.92\ p_y = 2.92\ (3300) = 9,630 \text{ psi}$$

$$(f_s)_u = 2.92\ p_u = 2.92\ (6000) = 17,520 \text{ psi}$$

The margin of safety for the radial shear stress is:

$$\text{yield MS} = \left[\frac{78,000}{9,630} - 1\right] = \underline{\text{Large}}$$

$$\text{ultimate MS} = \left[\frac{92,800}{17,520} - 1\right] = \underline{\text{Large}}$$

ACTUATOR CYLINDER

1.4.1.2 Stresses in the Head

The maximum bending moment at the midplane and outer edge of the head is assumed to occur at the tangent point of the relief radius or at the point of minimum thickness.

$$M_R = M_2 = \frac{Q_o t_1}{2} - M_o$$

$$M_R = \frac{-1.05\,p(.97)}{2} - .62\,p = -1.13\,p \ \frac{\text{in. -lb.}}{\text{in.}}$$

$$M_t = .3\,M_R = -3.39\,p \ \frac{\text{in. -lb.}}{\text{in.}}$$

Radial Tension $= N_1 = Q_o = 1.05$ p lb./in.

Normal Shear $= N_o = \frac{pd}{4} = \frac{9.8\,p}{4} = 2.45$ p lb./in.

The maximum combined radial stress

$$f_R = \frac{6\,M_R}{(t_1)^2} + \frac{N_1}{t_1}$$

$$f_R = \frac{6(-1.13)\,p}{(.97)^2} + \frac{1.05\,p}{.97} = -5.91\,p$$

The maximum combined radial yield stress for p_y = 3300 psi.

$$f_{Ry} = -5.91\,(3300) = 19,500 \text{ psi}$$

The maximum combined radial ultimate stress for p_u = 6000 psi.

$$f_{Ru} = -5.91\,(6000) = 35,460 \text{ psi}$$

ACTUATOR CYLINDER

The margin of safety for the combined radial stress

$$\text{yield MS} = \left[\frac{130,000}{19,500} - 1\right] = \underline{\text{Large}}$$

$$\text{ultimate MS} = \left[\frac{155,000}{35,460} - 1\right] = \underline{3.37}$$

The maximum combined tangential stress:

$$f_t = .3\, f_R = .3\,(-5.91) = -1.773\ p$$

The maximum combined tangential yield stress for p_y = 3300 psi

$$f_{ty} = -1.77\,(3300) = 5,840 \text{ psi}$$

The maximum combined tangential ultimate stress for p_u = 6000 psi

$$f_{tu} = -1.77\,(6000) = 10,620 \text{ psi}$$

The margin of safety for the combined tangential stress:

$$\text{yield MS} = \left[\frac{130,000}{5,840} - 1\right] = \underline{\text{Large}}$$

$$\text{ultimate MS} = \left[\frac{155,000}{10,620} - 1\right] = \underline{\text{Large}}$$

The normal shear stress is:

$$f_s = \frac{N_o}{t_1} = \frac{2.45\ p}{.97} = 2.53\ p$$

ACTUATOR CYLINDER

The normal shear yield stress for $p_y = 3300$ psi

$$(f_s)_y = 2.53\,(3300) = 8,350 \text{ psi}$$

The normal shear ultimate stress for $p_u = 6000$ psi

$$(f_s)_u = 2.53\,(6000) = 15,180 \text{ psi}$$

The margin of safety for the normal shear stress

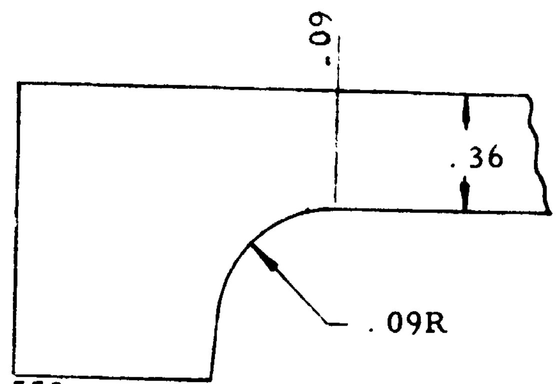
$$\text{yield MS} = \left[\frac{130,000}{8,350} - 1\right] = \underline{\text{Large}}$$

$$\text{ultimate MS} = \left[\frac{155,000}{15,180} - 1\right] = \underline{\text{Large}}$$

ACTUATOR CYLINDER

1.4.2 Cylinder Flanged End

The flanged end of the cylinder is bolted to the actuator body. The flange is also set in a recess machined in the actuator body which restrains the flange from rotating. This gives the cylinder the effect of being built in. The edge bending moment and shear are as shown in reference 21, page 399.



$$M_o = \frac{p}{2\beta^2} = \frac{p}{2\,(.952)^2} = .553\ p$$

$$V_o = \frac{p}{\beta} = \frac{p}{.958} = -1.05\ p$$

The combined longitudinal stress and the combined tangential stress at .09 inches from the edge of the flange, $t_3 = .36''$, $M_x = .461\,p$, $M_\lambda = .138\,p$;

$$f_x = \frac{pR}{2t_3} + \frac{6M_x}{t_3{}^2} = \frac{4.9\ p}{2\,(.36)} + \frac{6\,(.461\ p)}{(.36)^2}$$

$$f_x = 28.11\ p$$

$$f_\lambda = \frac{pR}{t} + \frac{6M_\lambda}{t^2} = \frac{4.9\ p}{.36} + \frac{6\ .138\ p}{(.42)^2}$$

$$f_\lambda = 20.72\ p$$

ACTUATOR CYLINDER

The maximum combined tangential yield stress for p_y = 3300 psi is:

$$(f_x)_y \quad = \quad 28.11\ (3300) \quad = \quad 92,800 \text{ psi}$$

The maximum combined tangential ultimate stress for p_u = 6000 psi is:

$$(f_x)_u \quad = \quad 28.11\ (6000) \quad = \quad 168,500 \text{ psi}$$

The margin of safety for the combined tangential stress is:

$$\text{yield MS} \quad = \quad \left[\frac{130,000}{92,800} - 1 \right] \quad = \quad \underline{.4}$$

$$\text{ultimate MS} \quad = \quad \left[\frac{155,000}{168,500} - 1 \right] \quad = \quad \underline{-.08}$$

The maximum shear stress:

$$f_s \quad = \quad \frac{Q_o}{t} \quad = \quad \frac{1.05\ p}{.36} \quad = \quad 2.91\ p$$

The maximum shear yield stress for p_y = 3300 psi

$$(f_s)_y \quad = \quad 2.91\ (3300) \quad = \quad 9,600 \text{ psi}$$

The maximum shear ultimate stress for p_u = 6000 psi

$$(f_s)_u \quad = \quad 2.91\ (6000) \quad = \quad 17,460 \text{ psi}$$

The margin of safety for the maximum shear stress:

$$\text{yield MS} \quad = \quad \left[\frac{78,000}{9,600} - 1 \right] \quad = \quad \underline{\text{Large}}$$

$$\text{ultimate MS} \quad = \quad \left[\frac{92,800}{17,460} - 1 \right] \quad = \quad \underline{\text{Large}}$$

ACTUATOR CYLINDER

The cylinder is on the borderline between a long and a short cylinder. If the cylinder is considered to be in the short range the bending moments at one end cannot be considered separate of the conditions at the opposite end. Then considering the cylinder as short from reference 21, page 402.

$$M_o = \frac{p}{2\beta^2} \left[\frac{\sinh 2\alpha - \sin 2\alpha}{\sinh 2\alpha + \sin 2\alpha}\right]$$

where $\alpha = \frac{\beta L}{2\beta^2} = 5.85$

$\sinh 2\alpha$ = Large

$\sin 2\alpha = .75471$

$$M_o = \frac{p}{2(.952)^2} \left[\frac{\sinh 2\alpha - .75471}{\sinh 2\alpha + .75471}\right] = \frac{p}{2(.952)^2} \quad R$$

As the value in the brackets (R) approaches unity, it indicates that the short cylinder effects can be neglected.

1.4.2.1 Tensile Stress in Flange Attachment

Bolts

N = 17 bolts

$P = \pi R^2 p = \pi (4.9)^2 p = 75.4\, p$ Total Load on flange bolts

$P_t = \frac{P}{N} = \frac{75.4\, p}{17} = 4.43\, p$ Tensile Load per bolt

$(P_t)_y = 4.43\,(3300) = 14,600$ lb.

$(P_t)_u = 4.43\,(6000) = 26,600$ lb.

ACTUATOR CYLINDER

The allowable strengths for the MS bolt (MIL-B-7838) at 275° F are:

$F_y = 26,900$ lb.

$F_u = 41,400$ lb.

The margins of safety are.

$$\text{yield MS} = \left[\frac{26,900}{14,600} - 1\right] = \underline{.84}$$

$$\text{ultimate MS} = \left[\frac{41,400}{26,600} - 1\right] = \underline{.56}$$

2.0 PISTON ACTUATOR - HEAD P/N 130-14013

$a = 4.73$

$t = 1.16$

$b = 2$

$m = \frac{1}{\mu} = \frac{1}{3} = 3.33$

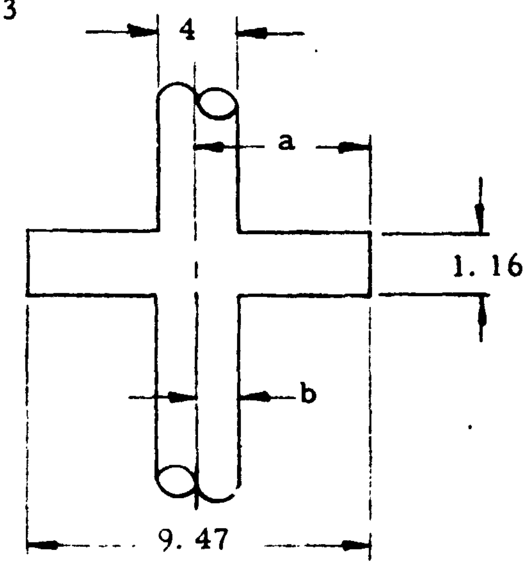


Figure 6

2.1 Internal Pressure Loads

$p_y = 2200 - 20 = 2180$ psi

$p_u = 6000 - 2000 = 4,000$ psi

2.2 Material Allowables (Reference 11):

Material - 4340 steel (R_c 30-34)

F_{ty} = 113,700 psi at 80° F; 108,000 psi at 275° F

F_{tu} = 138,000 psi at 80° F; 131,000 psi at 275° F

F_{cu} = 138,000 psi at 80° F; 131,000 psi at 275° F

F_{sy} = 73,100 psi at 80° F; 69,400 psi at 275° F

F_{su} = 87,000 psi at 80° F; 82,600 psi at 275° F

PISTON ACTUATOR - HEAD

2.3 Stress Calculations

To determine the bending moment at the inner edge of the head, use the formula for flat plates reference 17. Table X, Case 21

$$M_1 = \frac{3\,p}{24}\left[\frac{4\,a^4(m+1)\ln\frac{a}{b} - a^4(m+3) + b^4(m-1) + 4\,a^2\,b^2}{a^2(m+1) + b^2(m-1)}\right]$$

Substitution of the above values in this equation gives:

$$M_1 = 5.49\ p\ \text{in./lb.}$$

$$f_b = \frac{6\,M_1}{t^2} = \frac{6\,(5.49\ p)}{(1.16)^2} = 28.4\ p$$

The bending stress for $p_y = 2180$ psi is:

$$(f_b)_y = 28.4\,(2180) = 61{,}900\ \text{psi}$$

The bending stress for $p_u = 4000$ psi is:

$$(f_b)_u = 28.4\,(4000) = 113{,}700\ \text{psi}$$

The margins of safety are:

$$\text{yield MS} = \left[\frac{108{,}000}{61{,}900} - 1\right] = \underline{.75}$$

$$\text{ultimate MS} = \left[\frac{131{,}000}{113{,}700} - 1\right] = \underline{.15}$$

$$\text{fatigue limit MS} = \frac{65{,}500}{61{,}900} = 1.06$$

PISTON ACTUATOR - HEAD

Shear stress at inner edge of head:

$$\text{Shear Force} = F_s = p\,A$$

$$F_s = \frac{\pi}{4}\; 9.47^2 - 4^2\; p$$

$$F_s = 57.9\; p$$

$$\text{Shear Area} = A_s = \pi\, D\, t = \pi\,(4)\,(1.7)$$

$$A_s = 21.4 \text{ in.}^2$$

For a shaft in tension determine the stress concentration factor at the fillet using reference 24, page 67, figure 58:

$$D = 9.47$$

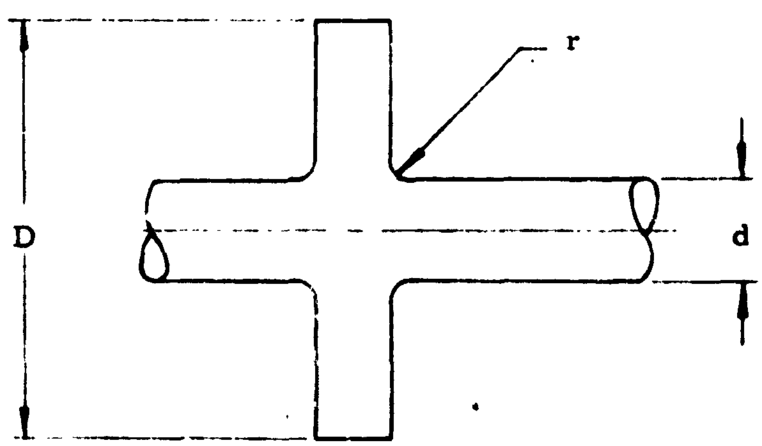
$$d = 4$$

$$r = .25$$

$$\frac{r}{d} = \frac{.25}{4} = .0625$$

$$\frac{D}{d} = \frac{9.47}{4} = 2.37$$

$$K_s = 2.6$$



PISTON ACTUATOR - HEAD

Yield shear stress for p_y = 2180 psi is:

$$(f_s)_y = \frac{F_s K_t}{A_s} = \frac{57.9\,(2180)(2.6)}{21.4} = 15,350 \text{ psi}$$

Ultimate shear stress for p_u = 4000 psi is:

$$(f_s)_u = \frac{F_s K_t}{A_s} = \frac{57.9\,(4000)(2.6)}{21.4} = 28,100 \text{ psi}$$

Margin of safety:

$$\text{yield MS} = \left[\frac{69,400}{15,350} - 1\right] = \underline{3.53}$$

$$\text{ultimate MS} = \left[\frac{82,600}{28,100} - 1\right] = \underline{1.94}$$

3.0 PISTON ACTUATOR SHAFT P/N 130-14013

3.1 Sketch

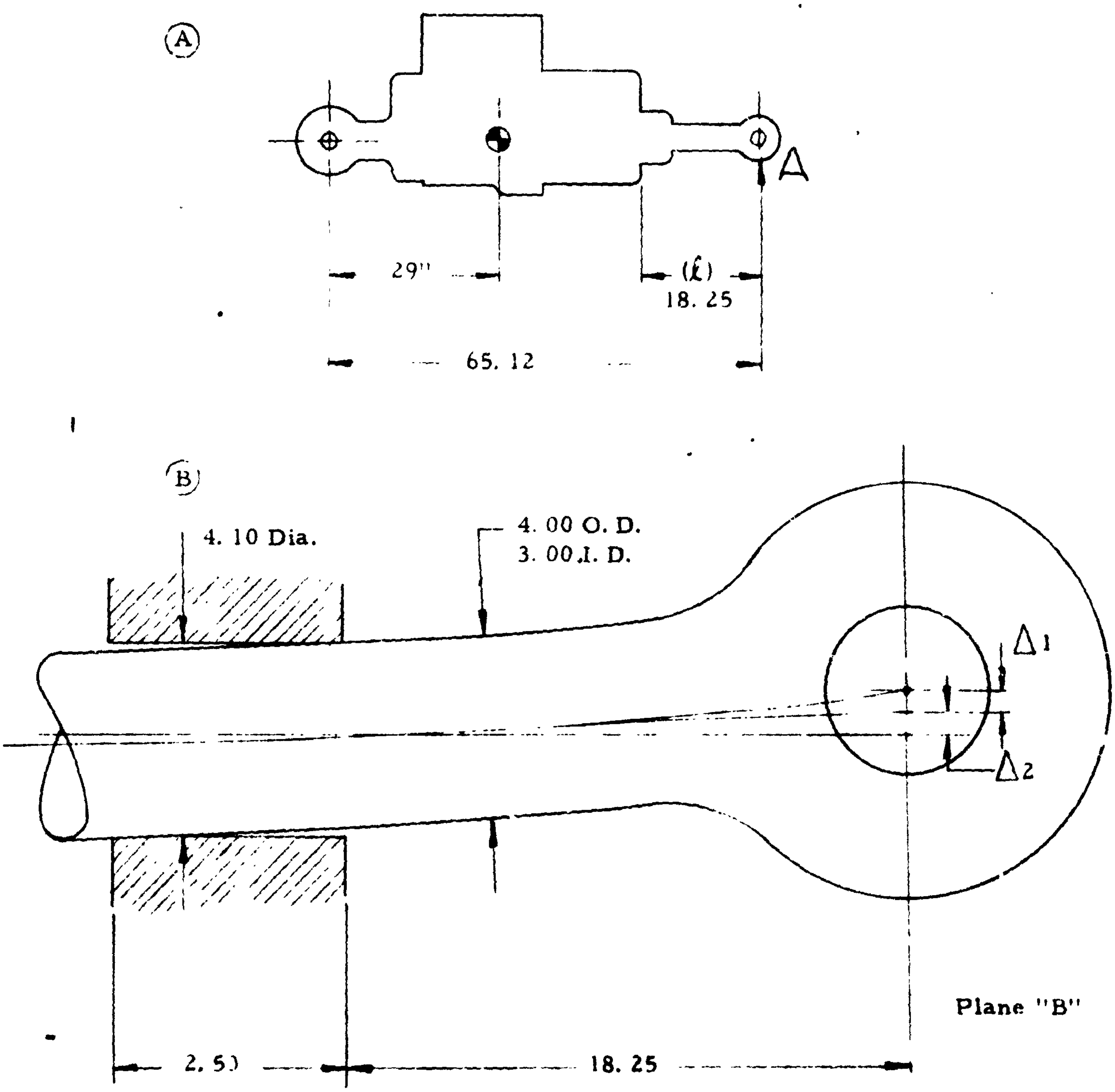


Figure 7

PISTON ACTUATOR SHAFT

3.2 Discussion

The piston is assumed in the extend position. The piston rod is considered as a cantilever supported in its forward bearing. The bucking stresses are considered under combined column and vibrational loads. The column length is assumed to extend from the ℄ of the rod end to the front face of the head. The tubular cross-section is treated as if it was constant.

3.3 Detail Loads

yield load = 72,000 lb.

ultimate load = $A_p \, p_o$

vibration level -10.4 g

max. actuator weight 320 lb.

3.4 Material Allowables

See Section

3.5 Calculated Stresses

3.5.1 Combined stress in Plane B due to combined bending and tension (or compression)

Solving first for the reaction at "A"

Referring to Sketch A:

$$R_A \, (65.12) = (29)\,(W)\,(g)\,(A_1)$$

g	=	vibration level in g's	=	10.4 g's
A_1	=	amplification factor	=	5
W	=	actuator weight	=	320 lbs.

PISTON ACTUATOR SHAFT

$$R_A = \frac{29(320)(10.4)(5)}{65.12}$$

$$R_A = 7,410$$

$$R_B = 10.4\,(5)(320) - 7410 = 9,240 \text{ lb.}$$

Tension Stress $f_t = P/A$

$$A = \text{Area} = \pi/4\,(4^2 - 3^2) = 5.5 \text{ in.}^2$$

yield $P = 72,000$ lb.

ultimate $P = 2200\,(57.9) = 127,000$ lb.

$$\text{yield } f_t = \frac{72,000}{5.5} = 13,100 \text{ psi}$$

$$\text{ultimate } f_t = \frac{127,000}{5.5} = 23,100 \text{ psi}$$

Bending Stress $f_{b1} = \dfrac{R_A \ell c}{I}$

$$\ell = 18.25 \text{ in.}$$

$$c = 412 = 2 \text{ in.}$$

$$I = \pi/64\,(4^4 - 3^4) = 8.59 \text{ in.}^4$$

$$\text{yield } f_{b1} = \frac{7410\,(18.25)\,(2)}{8.59} = 31,500 \text{ psi}$$

ultimate f_{b1} = yield f_{b1}

Deflection of the shaft due to bending

PISTON ACTUATOR SHAFT

$$\Delta_1 = \frac{R_A \ell^3}{3EI}$$

ℓ = 18.25 in.

E = 28.5 (10^6)

I = 8.59 in.^{4}

yield R_A = 7410 lb.

yield $$\Delta_1 = \frac{(7410)(18.25)^3}{(3)(28.5)(10^6)(8.59)} = .0613 \text{ in.}$$

Because the rod is not firmly built in at its bearing an additional deflection due to bearing clearance is present. The bearing clearance is determined as follows:

Minimum Piston Diameter = $D_p = D_1 + \delta_{tp}$

where D_1 = 3.997

$\delta_{tp} = D_1 \rho'_1 \Delta t$

δ_{tp} = change in diameter due to thermal expansion

ρ'_1 6.3×10^{-6} in/in/°F coefficient of thermal expansion for 4340 steel

Δt (275-75) = 200°F total temperature change

δ_{tp} $3.997\ (6.3 \times 10^{-6})\ (200) = .00503$

D_p 4.002"

TABLE VII
STRESS SUMMARY
CYLINDER P/N 033-41311

Location of Stress	Type of Stress	Magnitude of Stress, psi	Material Allowable Stress, psi	Margin of Safety
Cylinder-Head End	Max. Combined Longitudinal			
	Yield	101,800	130,000	.28
	Ultimate	184,860	155,000	-.16
	Max. Combined Tangential			
	Yield	46,200	130,000	1.81
	Ultimate	83,940	155,000	.85
	Radial Shear			
	Yield	9,630	78,000	Large
	Ultimate	17,520	92,800	Large
Head	Max. Combined Radial			
	Yield	19,500	130,000	Large
	Ultimate	35,460	155,000	3.37
	Max. Combined Tangential			
	Yield	5,840	130,000	Large
	Ultimate	10,620	155,000	Large
	Normal Shear			
	Yield	8,350	130,000	Large
	Ultimate	15,180	155,000	Large
Cylinder-Flanged End	Max. Combined Tangential			
	Yield	92,800	130,000	.40
	Ultimate	168,500	155,000	-.08

TABLE V (cont'd.)
STRESS SUMMARY
CYLINDER P/N 333-41311

Location of Stress	Type of Stress	Magnitude of Stress, psi	Material Allowable Stress, psi	Margin of Safety
	Max. Shear			
	Yield	9,600	78,000	Large
	Ultimate	17,460	92,800	Large
Flange Attachment Bolts	**Tensile Load**			
	Yield	14,600	26,900	.84
	Ultimate	26,600	41,400	.56

PISTON ACTUATOR SHAFT

Maximum Bearing Diameter $= D_B = D_2 + \delta_{tb}$

where: $D'_2 = 4.097$

$D_2 = D'_2 - t_r = 4.097 - .0926 = 4.004$

t_r = max. rulon thickness

$\delta_{tb} = D_2 \rho'_2 \Delta t$

$\delta_{tb} = 4.004 (6.3 \times 10^{-6}) = .005''$

$D_B = 4.009$

The Maximum Clearance is:

$D_3 = D_B - D_p = 4.009 - 4.002$

$D_3 = .007$

For simplicity it is conservatively assumed that the deflection is a straight line ratio. Refer to the sketch on page

$\Delta_2 = .007/2.5 (20.75) = .058$ in.

Bending stress due to the eccentrically applied bending load assuming that the additional shaft deflection due to this load is less than 10%:

$$f_{b2} = \frac{P(\Delta_1 + \Delta_2) C}{I}$$

$\Delta_1 = .0613$ in.

$\Delta_2 = .058$ in.

$C = 2$ in.

$I = 8.59$ in.^{4}

PISTON ACTUATOR SHAFT

yield $P = 72{,}000$ lb.

yield $f_{b2} = \dfrac{(72{,}000)\,(.119)\,(2)}{8.59} = 1{,}995$ psi

Total Combined Stress $f = f_t + f_{b1} + f_{b2}$

yield $f = 13{,}100 + 31{,}500 + 1{,}995 = 46{,}595$ psi

ultimate $f = 23{,}100 + 31{,}500 + 1{,}995 = 56{,}595$ psi

Critical stress for the rod in bending

$$F_{cr} = F_{cy} - \frac{18.36}{C_1}\left(\frac{L}{\rho}\right)^2$$ (Ref. 5, pg. 2.10, 2.1 & Ref. 9, pg. 5-54)

$F_{cy} = 108{,}000$ psi (compressive yield)

$L = 21.12$ in. (length of column in compression)

$\rho = \sqrt{I/A} = 1.25$ (radius of gyration)

$I = 8.59$ in.^{4}

$A = 5.5$ in.^{2}

$C_1 = 2.86$ (end fixity coefficient)

Determination of column classification (long or short)

$$\frac{L}{\rho\sqrt{C_1}} = \frac{21.12}{1.25\sqrt{2.86}} = 10 < 65 \quad \therefore \text{ short column}$$

PISTON ACTUATOR SHAFT

$$F_{cr} = 108,000 - \frac{(18.36)(21.12)^2}{(2.86)(1.25)^2} = 106,170 \text{ psi}$$

Margin of Safety

$$\text{yield MS} = \left[\frac{106,170}{46,595} - 1 \right] = \underline{1.28}$$

$$\text{ultimate MS} = \left[\frac{106,170}{56,595} - 1 \right] \quad \underline{.88}$$

A second iteration considering additional moment due to the beam column effect is not necessary because of the high margin of safety.

4.0 ACTUATOR BODY, P/N 033-14009

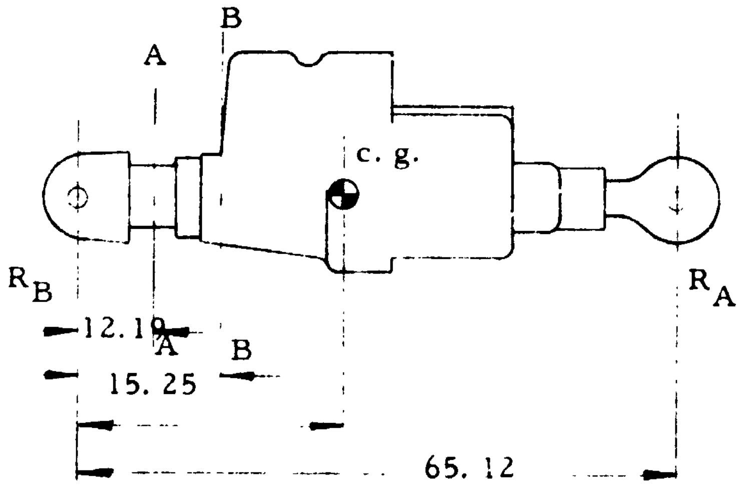


Figure 8

4.1 Discussion

The piston is assumed in the extend position. The actuator is treated as a simply supported beam with the "suspect" sections of the body stressed in bending.

4.2 Detail Loads

vibration level - 10.4 g's

max. actuator weight = 320 lbs.

yield load = 72,000 lbs.

ultimate load = 127,000 lbs.

TABLE VIII

STRESS SUMMARY
ACTUATOR BODY P/N 033-14009

Location of Stress	Type of Stress	Magnitude of Stress psi	Material Allowable Stress, psi	Margin of Safety
Plane A	Tensile			
	yield	5,720	-	-
	ultimate	10,100	-	-
	Bending			
	yield	10,550	-	-
	ultimate	10,550	-	-
	Combined tensile and bending			
	yield	16,270	47,700	1.93
	ultimate	20,650	54,600	1.65
Plane B	Tensile			
	yield	3,970	-	-
	ultimate	7,000	-	-
	Bending			
	yield	8,610	-	-
	ultimate	8,610	-	-
	Combined tensile and bending			
	yield	12,580	47,700	2.79
	ultimate	15,610	54,600	2.5

4.3 Material Allowables

Material 7079-T62 and

F_{tu} = 71,000 psi at 80° F and 54,600 psi at 275° F

F_{ty} = 62,000 psi at 80° F and 47,700 psi at 275° F

4.4 Calculated Stresses

4.4.1 Combined Stress in Plane A Due to Combined Bending and Tension

Tension Stress

$$f_t = \frac{P}{A}$$

$$A = \text{Area} = \frac{\pi}{4}\left[\bar{5}^2 - \bar{3}^2\right] = 12.58 \text{ in.}^2$$

yield P = 72,000 lb.

ultimate P = 127,000 lb.

$$\text{yield } f_t = \frac{72,000}{12.58} = 5,720 \text{ psi}$$

$$\text{ultimate } f_t = \frac{127,000}{12.58} = 10,100 \text{ psi}$$

Bending Stress

$$f_b = \frac{R_B \ell C}{I}$$

$$C = \frac{5}{2} = 2.5''$$

$$I = \frac{\pi}{64}\left[\bar{5}^4 - \bar{3}^4\right] = 26.7 \text{ in.}^4$$

$$\ell = 12.19$$

$$\text{yield } f_b = \frac{9240\ (12.19)\ (2.5)}{26.7} = 10,550 \text{ psi}$$

ultimate f_b = yield f_b

ACTUATOR BODY, P/N 033-14009

Total Combined Stress

$$f = f_t + f_b$$

yield f = 5720 + 10,550 = 16,270 psi

ultimate = 10,100 + 10,550 = 20.650 psi

Margin of Safety

$$\text{yield MS} = \left[\frac{47,700}{16,270} - 1\right] = 1.93$$

$$\text{ultimate MS} = \left[\frac{54,600}{20,650} - 1\right] = 1.65$$

4.4.2 Combined Stress in Plane B Due to Combined Bending and Tension

Tension Stress

$$f_t = \frac{P}{A}$$

$$A = \frac{\pi}{4}\left[\overline{5.66}^2 - \overline{3}^2\right] = 18.15 \text{ in.}^2$$

$$\text{yield} \quad f_t = \frac{72,000}{18.15} = 3,970 \text{ psi}$$

$$\text{ultimate} \quad f_t = \frac{127,000}{18.15} = 7,000 \text{ psi}$$

Bending Stress

$$f_b = \frac{R_B \ell C}{I}$$

$$\ell = 15.25''$$

$$C = 2.83''$$

$$I = \frac{\pi}{64}\left[\overline{5.66}^4 - \overline{3}^4\right] = 46.3 \text{ in.}^4$$

$$\text{yield} \quad f_b = \frac{9240\,(15.25)\,(2.83)}{46.3} = 8,610 \text{ psi}$$

$$\text{ultimate} \quad f_b = \text{yield } f_b$$

ACTUATOR BODY, P/N 033-14009

Total Combined Stress

$$f = f_t + f_b$$

$$\text{yield } f = 3,970 + 8,610 = 12,580 \text{ psi}$$

$$\text{ultimate } f = 7,000 + 8,610 = 15,610 \text{ psi}$$

Margin of Safety

$$\text{yield MS} = \left[\frac{47,700}{12,580} - 1\right] = 2.79$$

$$\text{ultimate MS} = \left[\frac{54,600}{15,610} - 1\right] = 2.5$$

5.0 ROD END, P/N 121-13510

5.1 Sketch

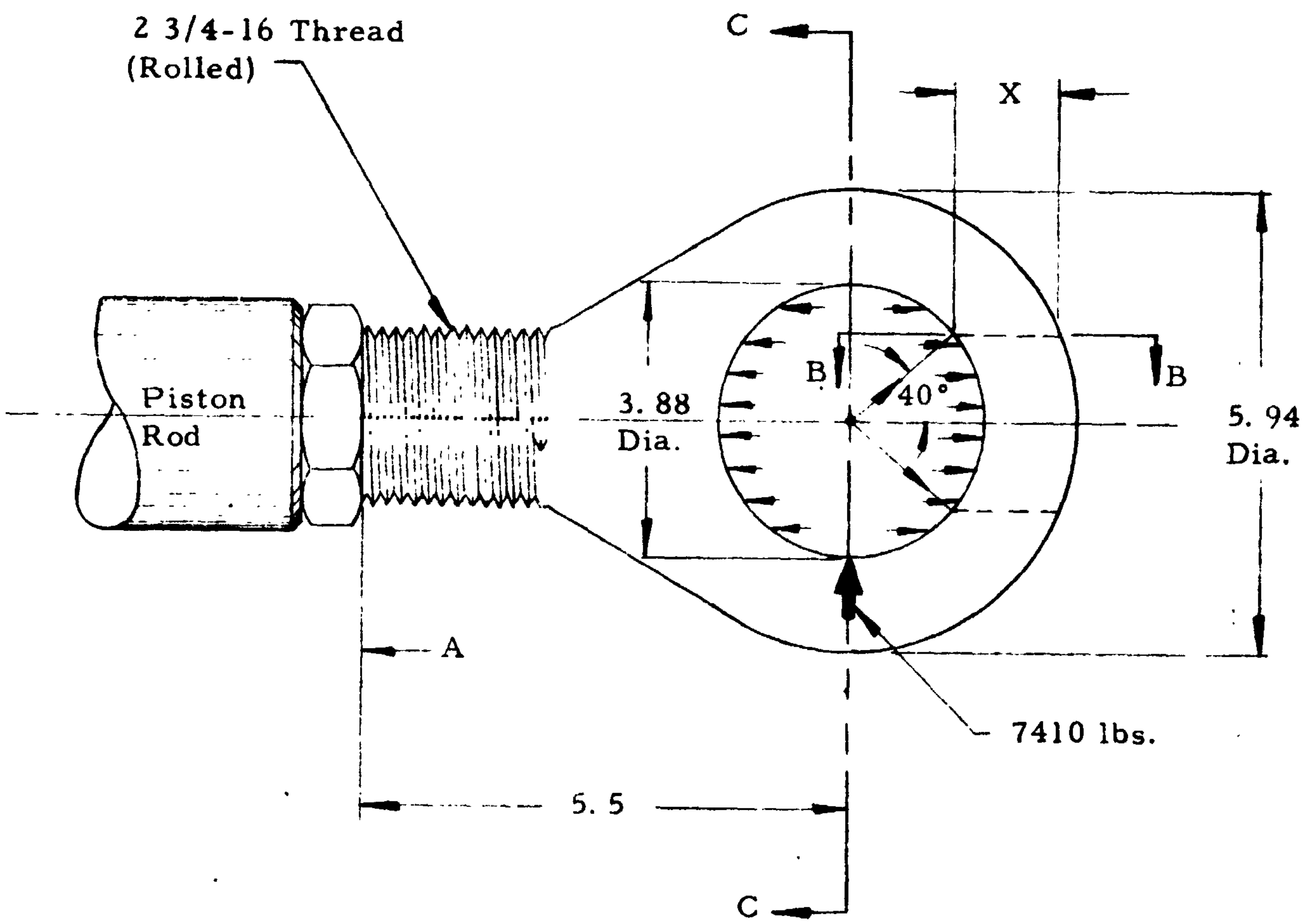


Figure 9

TABLE IX

STRESS SUMMARY
ROD END P/N 121-13510

Location of Stress	Type of Stress	Magnitude of Stress psi	Material Allowable Stress, psi	Margin of Safety
Plane A	Tensile			
	yield	15,650	-	-
	ultimate	27,600	-	-
	Bending			
	yield	29.300	-	-
	ultimate	29.300	-	-
	Combined tensile and bending			
	yield	44,950	93,300	1.08
	ultimate	56,900	121,500	1.13
Plane B	Shear			
	ultimate	40,700	80,800	.99
Section CC	Tensile			
	yield	30.800	93,300	2.04
	ultimate	54,400	121,500	1.23
Rod Eye and Bearing	Bearing			
	ultimate	21,800	243,000	Large

ROD END

5.2 Discussion

The rod end is considered as a cantilevered member from the point of exit from the rod nut. It is being analyzed in the actuator piston extend position. The reaction load of 7410 pounds is derived in the section entitled Piston. In addition it is assumed that the rod end is in its extreme extend position of adjustment.

5.3 Detail Loads

yield load - 72,000 lb.

ultimate load - 127,000 lb.

vibration reaction load - 7410 lb.

5.4 Material Allowables

Material - 410 stainless steel (R_c 26-32)

F_{tu} = 128,000 at 80° F and 121,500 psi at 275° F
F_{ty} = 98,200 at 80° F and 93,300 psi at 275° F
F_{bu} = 178,300 at 80° F and C/D ratio 1.5, 169,500 psi at 275° F
F_{su} = 80,800 at 80° F and 76,800 psi at 275° F
F_{brg_μ} = 256,000 at 80° F and 243,000 at 275° F

5.5 Calculated Stresses

5.5.1 Combined Stress in Plane A Due to Combined Bending and Tension (or Compression)

Tensile Stress $f_t = P/A$

ROD END

$$A = \frac{\pi}{4}(2.42)^2 = 4.6 \text{ in.}^2$$

yield P = 72,000 lbs.

ultimate P = 127,000 lb.

$$\text{yield } f_t = \frac{72,000}{(4.6)} = 15,650 \text{ psi}$$

$$\text{ultimate } f_t = \frac{127,000}{(4.6)} = 27,600 \text{ psi}$$

Bending Stress $f_b = \dfrac{P\ell C}{I}$

ℓ = 5.5 in.

C = 1.21

$$I = \frac{\pi}{64}(2.42)^4 = 1.68 \text{ in.}^4$$

yield P = 7,410

ultimate P = 7,410

$$\text{yield } f_b = \frac{(7410)(5.5)(1.21)}{1.68} = 29,300 \text{ psi}$$

ultimate f_b = 29,300 psi

Combined Stress = $f = f_t + f_b$

yield f = 15,650 + 29,300 = 44,950 psi

ultimate f = 27,600 + 29,300 = 56,900 psi

ROD END

Margin of Safety

$$\text{yield MS} = \left[\frac{93,300}{44,950} - 1\right] = \underline{1.08}$$

$$\text{ultimate MS} = \left[\frac{121,500}{56,900} - 1\right] = \underline{1.13}$$

5.5.2 Shear Stress at Plane B Due to Eye Loading

The effect of the vibration load is omitted since it is small compared to the column load. This method of analysis is conservative since it was developed for loosely fitting pins and in this case the bearing is pressed into the eye.

Shear Stress $f_s = \dfrac{P}{2 \text{ X } T}$

$$X = r_a\left[\sqrt{1 - \left(\frac{r_i}{r_a}\right)^2} \sin 40° - \frac{r_i}{r_a} \cos 40°\right]$$

$$= 1.04 \text{ in.}$$

r_a = 2.97 in.

r_i = 1.94 in.

T = 1.5 in.

P = 127,000 lb.

$$f_s = \frac{127,000}{2(1.04)(1.5)} = 40,700 \text{ psi}$$

Margin of Safety

$$\text{ultimate MS} = \left[\frac{80,800}{40,700} - 1\right] = \underline{.99}$$

ROD END

5.5.3 The Tensile Stress Through Section CC

This analysis treats the hoop of the eye as a thick walled cylinder subjected to a uniform internal radial pressure. The pressure is assumed to be equal to the column load divided by the projected bearing area. This simplification of the analysis is presented to back-up the preceding calculations for shear. Because the bearing is pressed into the eye, the load is distributed over the entire semi-circular section of the eye very much like an internal pressure. The discrepancies that exist between this treatment of the stress and the actual condition are in the direction of safety. (Ref. 17, Table XIII, Case No. 27)

Tensile Stress $f_t = \dfrac{P}{2\, r_i\, t} \left[\dfrac{r_a^2 + r_i^2}{r_a^2 - r_i^2} \right]$

r_a = 2.97 in.

r_i = 1.94 in.

T = 1.5 in.

yield P= 72,000 lb.

ultimate P= 127,000 lb.

$$\text{yield } f_t = \frac{72{,}000}{(2)(1.94)(1.5)} \left[\frac{(2.97)^2 + (1.94)^2}{(2.97)^2 - (1.94)^2} \right]$$

yield f_t = 30,800 psi

$$\text{ultimate } f_t = \frac{127{,}000}{(2)(1.94)(1.5)} \left[\frac{(2.97)^2 + (1.94)^2}{(2.97)^2 - (1.94)^2} \right]$$

ultimate f_t = 54,400 psi

Margin of Safety

$$\text{yield MS} = \left[\frac{93{,}300}{30{,}800} - 1 \right] = \underline{2.04}$$

ROD END

$$\text{ultimate MS} = \left[\frac{121{,}500}{54{,}400} - 1\right] = \underline{1.23}$$

5.5.4 Bearing Stress Existing at Interface of Rod Eye and the Bearing

Bearing Stress $f_{BR} = \dfrac{P}{2\ r_i\ T}$

r_i = 1.94
T = 1.5
P = 127,000

$$f_{BR} = \frac{127{,}000}{(2)(1.94)(1.5)} = 21{,}800 \text{ psi}$$

Margin of Safety assuming $e/D = 1.5$

$$\text{ultimate bearing MS} = \left[\frac{243{,}000}{21{,}800} - 1\right] = \underline{\text{Large}}$$

6.0 TAILSTOCK P/N 121-13508

6.1 Sketch

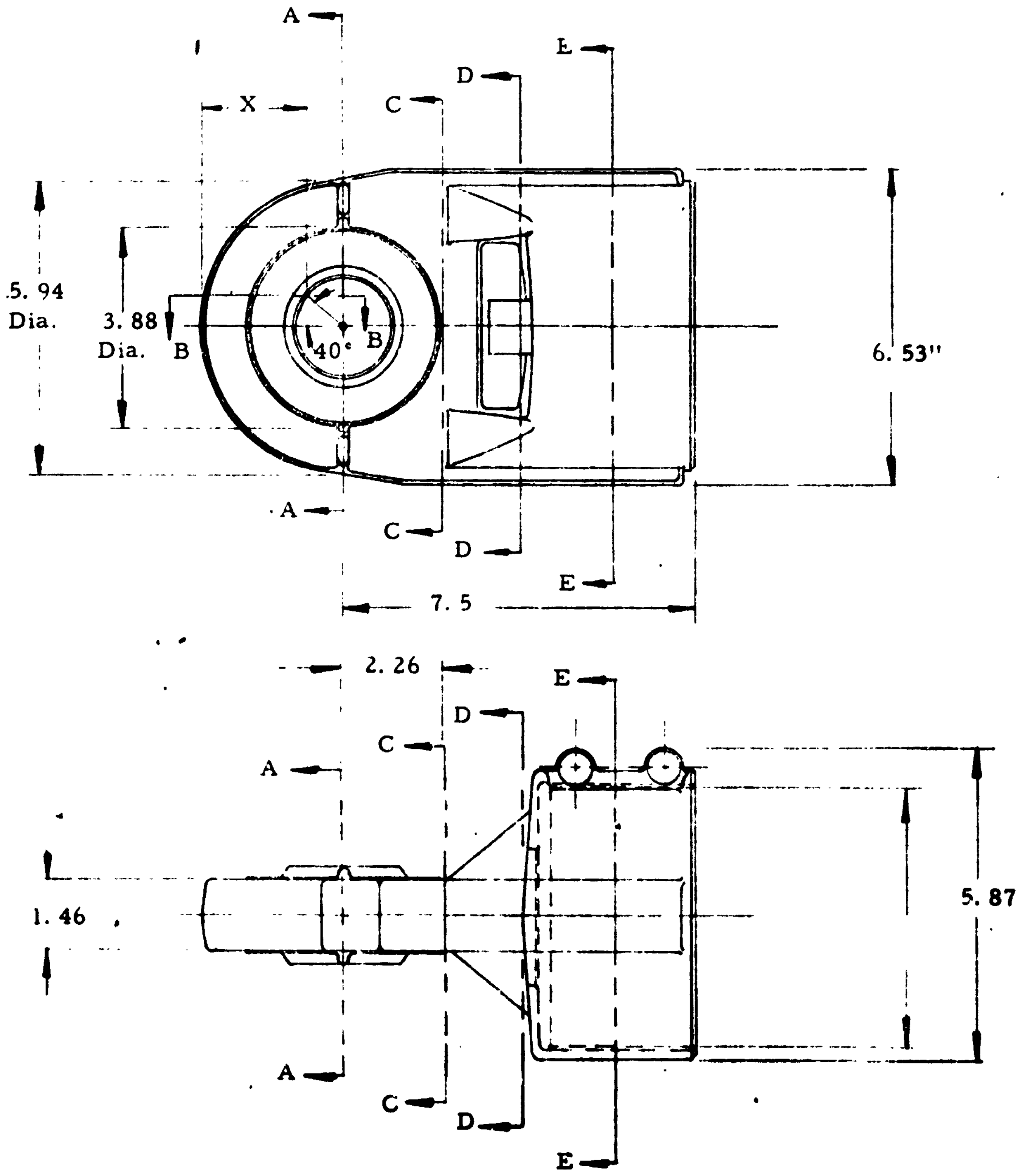


Figure 10

TABLE X
STRESS SUMMARY
TAILSTOCK P/N 121-13508

Location of Stress	Type of Stress	Magnitude of Stress psi	Material Allowable Stress, psi	Margin of Safety
Plane B	Shear			
	ultimate	65,200	76,800	.84
Section AA	Tensile			
	yield	31,600	93,300	1.95
	ultimate	55,700	121,500	1.18
Plane CC	Combined Bending & Tension			
	yield	9,550	93,300	large
	ultimate	15,320	121,500	large

TAILSTOCK

6.2 Discussion

The tailstock is treated as a load carrying member rigidly attached to the actuator body. The deflection of the tailstock due to R_B is neglected. Margins of safety throughout the body of the tailstock are fairly high to allow for the fact that actual stress distribution in the unit is not as simple as the analysis assumes, and to insure adequate stiffness of the servoactuator as a whole.

6.3 Detail Loads

yield load = 72,000 lb.

ultimate load = 127,000 lb.

reaction at B - R_B = 9240 lb.

6.4 Material Allowables

Material - 410 stainless steel (R_C 26 - 32)

See Section 7

6.5 Calculated Stress

6.5.1 Shear Stress in Plane B Due to Eye Loading

The effect of the vibration load is omitted since it is small compared to the column load. This method of analysis is conservative since it was developed for loosely fitted pins and in this case the bearing is pressed into the eye (Ref.   page 261).

Shear Stress $f_s = \dfrac{P}{2 \times T}$

TAILSTOCK

$$X = r_a \left[\sqrt{1 - \left(\frac{r_i}{r_a}\right)^2} \sin 40^\circ - \frac{r_i}{r_a} \cos 40^\circ \right]$$

$$= 1.04 \text{ in.}$$

r_a = 2.97
r_i = 1.94 in.
t = 1.46 in.
P = 127,000 lb.

$$f_s = \frac{127,000}{(2)(1.04)(1.46)} = 41,800 \text{ psi}$$

Margin of Safety

$$\text{ultimate MS} = \frac{76,800}{41,800} - 1 = \underline{.84}$$

6.5.2 The Tensile Stress Through Section AA

See for discussion

Tensile Stress $f_t = \dfrac{P}{2\ r_i\ t} \left[\dfrac{r_a^2 + r_i^2}{r_a^2 - r_i^2} \right]$

r_a = 2.97 in.
r_i = 1.94 in.
t = 1.46 in.
yield P = 72,000 lb.
ultimate P = 127,000 lb.

$$\text{yield } f_t = \frac{72,000}{(2)(1.94)(1.46)} \left[\frac{(2.97)^2 + (1.94)^2}{(2.97)^2 - (1.94)^2} \right]$$

TAILSTOCK

yield f_t = 31,600 psi

$$\text{ultimate } f_t = \frac{127,000}{(2)(1.94)(1.46)} \left[\frac{(2.97)^2 + (1.94)^2}{(2.97)^2 - (1.94)^2}\right]$$

ultimate f_t = 55,700 psi

Margin of Safety

$$\text{yield margin} \left[\frac{93,300}{31,600} - 1\right] = \underline{1.95}$$

$$\text{ultimate margin} \left[\frac{121,500}{55,700} - 1\right] = \underline{1.18}$$

6.5.3 Combined Stress in Plane CC Due to Combined Bending and Tension (or Compression)

Tensile Stress $f_t = P/A$

$A = Ht = 6.53\ (1.46) = 9.56\ \text{in.}^2$

H = 6.53 in. (height of section

t = 1.46 in. (thickness of section)

yield P = 72,000 lb.

ultimate P = 127,000 lb.

$$\text{yield } f_t = \frac{72,000}{9.56} = 7,530 \text{ psi}$$

$$\text{ultimate } f_t = \frac{127,000}{9.56} = 13,300 \text{ psi}$$

TAILSTOCK

Bending Stress $f_b = \frac{P \ell C}{I}$

ℓ = 2.26 in.

I = $\frac{1}{12}$ t H^3 = 33.8 $in.^4$

H = 6.53 in.

t = 1.46 in.

yield P = 9240 lb.

yield $f_b = \frac{(9240)(2.26)(3.26)}{33.8} = 2,020$ psi

ultimate f_b = 2,020 psi

Combined yield stress f = 7530 + 2020 = 9,550 psi

Combined ultimate stress f = 13300 + 2020 = 15,320 psi

Margin of Safety

yield MS = $\left[\frac{93,300}{9.550} - 1\right]$ = Large

ultimate MS = $\left[\frac{121,500}{15,320} - 1\right]$ = Large

7.0 FLEXURE SLEEVE P/N 070-41751

7.1 Discussion

The flexure sleeve is part of the first stage assembly and has as its function; (1) to provide a seal between the high pressure hydraulic supply and the torque motor, and (2) to provide the connecting link between the electrical input to the first stage and the hydraulic output. An input signal produces a torque unbalance on the servovalve torque motor. As torque is applied to the armature, the armature pivots about the flexure sleeve support, and the flapper is displaced between the nozzle assemblies. This change in flapper-to-nozzle spacing creates a nozzle differential pressure which displaces the servovalve spool.

7.2 Loads

The flexure sleeve is analyzed for an ultimate internal burst pressure of p_u = 2000 psi. In addition to this, stresses are calculated for the combined affect of bending the internal chamber pressure. Maximum bending stresses occur when the armature pivots about the flexure sleeve and strikes the polepiece stops (see sketch). When in this position the maximum internal first stage chamber pressure is p_1 = 1100 psi.

7.3 Material Allowables (reference 5)

Material: 17-4 Ph Cres. (H. T. to R_c 40-49)

F_{tu} = 182,000 psi at 80° F; 171,000 psi at 275° F

F_{ty} = 163,000 psi at 80° F; 152,000 psi at 275° F

F_{cy} = 182,000 psi at 80° F; 152,000 psi at 275° F

F_{su} = 115,000 psi at 80° F; 105,000 psi at 275° F

FLEXURE SLEEVE

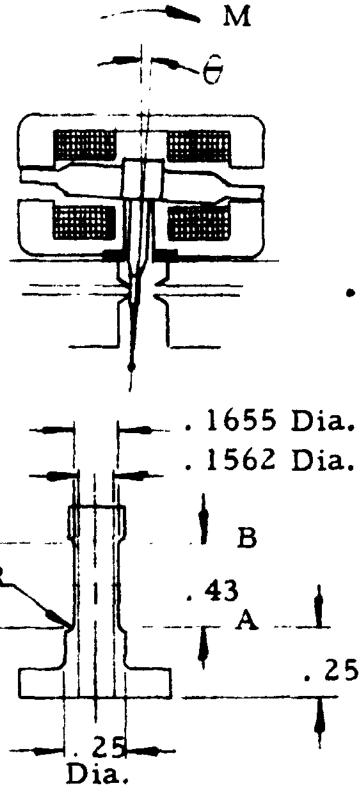


Figure 11

7.4 Stress Calculations

Assume the flexure sleeve conforms to a cantilever beam with an end couple. Consider the length of the beam to be that slender section of the tube denoted by ℓ = .43" in the sketch. Referring to reference 11, Table III, Case No. 9:

$$\theta = \frac{M\ell}{EI}$$

$$\theta = \frac{X_1}{r}$$ where X_1 = Maximum air gap between armature and polepiece stop = .005"

r = Distance from ℄ of armature to point where armature strikes the polepiece stop = .90"

$$\theta = \frac{.005}{.90} \quad .00556 \text{ radians}$$

ℓ = length of beam = .43"

$$I = \frac{\pi}{64}\left[d_o^{\,4} - d_i^{\,4}\right] = \frac{\pi}{64}\left[.\overline{1655}^4 - .\overline{1562}^4\right] = 7.55 \times 10^{-6} \text{ in.}^4$$

$$M = \frac{\theta EI}{\ell} = \frac{.00556\,(28.5 \times 10^6)\,(7.55 \times 10^{-6})}{.43}$$

M = 2.75 in.-lb.

FLEXURE SLEEVE

The maximum bending stress due to the combined affect of the end couple M and the internal pressure p = 1100 psi is:

$$f_b = \frac{MC}{I} + \frac{pR}{2t}$$

$$f_b = \frac{2.75\,(.0827)}{7.55 \times 10^{-6}} + \frac{1100\,(.0827)}{2\,(.0046)}$$

$$f_b = 39,960 \text{ psi}$$

The margin of safety for the maximum combined bending stress is:

$$\text{yield MS} = \left[\frac{152,000}{39,960} - 1\right] = \underline{2.80}$$

The maximum hoop stress in the flexure sleeve due to an internal pressure of p_u = 6000 psi is:

$$f_h = \frac{pD}{2t} = \frac{2000\,(.1655)}{2\,(.0046)} = 36,400 \text{ psi}$$

The margin of safety is:

$$MS = \left[\frac{171,000}{36,400} - 1\right] = \underline{3.70}$$