

MECHANICAL COMPONENT SCREENING FOR SCANNER

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ABSTRACT

The critical mechanical components of the scan mirror mechanism are described and their evaluation and screening procedures are discussed. A bumper/damper unit is used in the design to reverse motion of the mirror and effect scan and retrace cycles. A wear evaluation was conducted on the bumper impact surfaces that established nylon 6-10 as an acceptable material. The elastomeric dampers were subjected to thermal vacuum tests for condensables and outgassing as well as parametric life tests. The flexure pivots that support the mirror were tested to establish a curve of stress plotted as a function of cycles to failure for rotational operation. The pivots met the life requirement of 150 000 000 cycles at a $\pm 2.9^\circ$ amplitude during fatigue testing. Screening procedures have been established for dampers and flexure pivots to obtain flight quality components.

INTRODUCTION

As the Earth Resources Technology Satellite travels along its polar orbit, a multispectral scanner (MSS) mirror mechanism scans a nominally symmetrical east-west swath reflecting an image of the earth surface into the scanner optical system. The image format is 185.2 by 185.2 kilometers (100 by 100 n. mi.) from the 926-kilometer (500 n. mi.) circular orbit. The assembly consists of the mirror and the devices that impart necessary rotation. The components that are subject to wear and fatigue are the flexure-pivot suspension and bumper/damper units that reverse the rotation at the limits of travel. The wear characteristics of the bumper pads and the fatigue characteristics of the dampers and flexure pivots were determined. Because redundancy cannot be used, screening procedures were established to identify potential early-life failures for the dampers and flexure pivots.

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SCAN MECHANISM DESCRIPTION

The relative position of the components that were evaluated in the MSS scan mirror assembly are shown in figure 1. The scan mirror mechanism includes the following components.

1. A nickel-plated beryllium mirror with reinforcing ribs
2. Two flexure pivots, which suspend the mirror and allow the mirror to rotate about the pivot axes
3. An aluminum support structure to which the mirror, the pivots, and the rest of the components are attached
4. An electromagnetic drive assembly with the coil mounted on the supporting structure and an armature mounted on the mirror
5. An electronic driver that energizes the coil over an appropriate portion of the retrace cycle
6. An optical switch that turns off the coil when the mirror reaches the center of the retrace cycle
7. Two bumper/damper units that act at each end of the scan interval to change the direction of rotation of the mirror and dissipate the energy of the impacts. Each unit (fig. 2) consists of a bumper cup with nylon impact surfaces, a spring suspended bumper, and two dampers. Two units are used so that the forces acting on the mirror produce a couple that will turn the mirror around with only a slight force on the flexure pivots. This feature minimizes rotation of the mirror about an axis perpendicular to the pivot axes (cross-scan jitter).

BUMPER MATERIAL

The four candidate materials chosen were linen-phenolic, per federal specification L-P-509 (Type 2, Grade L); low-moisture nylon, per L-P-410 (Composition 6-10); Teflon-filled polycarbonate (Lexan); and polyurethane from two suppliers. In the MSS application, the most important characteristics the material must have are good wear and impact resistance, suitability for adhesive bonding, and suitability for use in the space environment.

Nylon was selected as the bumper material. The bumper pads are bonded in place using an epoxy-polyamide film adhesive that has a shear strength of $27.58 \times 10^6 \text{ N/m}^2$ (4000 psi) and provides control of bond line thickness.

There were two test conditions used during the bumper-wear tests on a prototype bumper assembly configuration. The first condition was an impact velocity of 38.10 cm/sec (15 in/sec), which is comparable to that developed in the scan mirror mechanism. This is accomplished by introducing a 0.10-millimeter (0.004 inch) gap between the bumper and each of the impact surfaces on the bumper cup and by driving

the cup at a rate of 90 hertz. The second condition was an impact force in the shape of a half sinewave with a peak amplitude that was 20 percent greater than the force developed in the flight hardware. This is achieved by driving the cup 0.69 millimeter (± 0.027 inch, which includes the 0.004-inch gap) so that the bumper deflection that compresses the bumper spring and displaces the dampers is 0.58 millimeter (± 0.023 inch).

The greater impact forces that were applied provide a conservative estimate of the wear rate that can be expected in the application. The technique that was used for measuring the amount of wear was to scribe a grid pattern on the nylon pads with the depth of the scribe mark being approximately 0.038 millimeter (0.0015 inch). A baseline measurement of the surface using a Talysurf machine was made before the initiation of the test. Subsequent measurements traversing the same paths were made at regular intervals for comparison to the baseline. The amount of wear was calculated by the change in the depth of the scribe lines. The test results for the nylon material are presented in figure 3, which shows the Talysurf traces of the surface contour at the beginning of the test (baseline) and after 142 000 000 cycles. Comparison of the traces show that the amount of the surface wear was less than 5.08×10^{-3} millimeters (0.2 mils, which is substantially less than the allowable wear of 50.8×10^{-3} millimeters (2 mils)). In addition to determining the material wear characteristics, this test program showed the integrity of the bond between the nylon and the aluminum bumper cup.

DAMPERS

Requirements

Two elastomeric dampers operate with a precompressed spring on each side of the scanning mechanism as shown in figure 2. These dampers dissipate energy by shear distortion. The dampers must displace approximately 0.48 millimeter (± 0.019 inch) during each cycle of the required life. The design goal for a pair of dampers originally was established at 0.350 N-sec/cm (0.20 lb-sec/in.) at an excitation frequency of 125 hertz. This frequency corresponds to the nominal 4-millisecond turnaround time on the scanner mechanism. Two types of damper designs finally were considered. The device that was developed by the contractor was a simple rectangular configuration using a silicone elastomeric material. The device from a commercial producer was cylindrical in shape and also used a silicone elastomer. The ultimate selection of the dampers required for the scanner mechanism was based on the results of thermal vacuum and cycling tests.

Evaluation Tests

Checks were made during the thermal vacuum test for condensable materials after 4 days at the real-time temperature and in a vacuum environment of 1.33×10^{-4} N/m² (10^{-6} torr). For a silicone damper to pass this test, a postcure at elevated temperature was required. From the results of tests on the Surveyor, it had been determined that silicone elastomers postcured at 449.80° K (350° F) showed the least change from the initial physical properties while still reducing the outgassing to an acceptable level. Tests were made on dampers as they were received and on dampers that had been

subjected to postcures at sea level. The dampers that were designed by the contractor were found to require 10 days at 449.80° K (350° F) while the vendor parts required 5 days at 449.80° K (350° F). After these bakeouts, both the parts designed by the contractor and the vendor parts had less than a 0.1-percent weight loss during the 4 days and had no oily deposits or condensables that could be found during the test.

Life tests were performed on dampers of both designs. An apparatus was designed that would simultaneously test six pairs of dampers. The life test was conducted with a cyclic shear displacement amplitude of 0.58 millimeter (± 0.023 inch). This displacement is approximately 20 percent greater than actually produced during impact in the scan mirror mechanism. Three pairs of commercial dampers and three pairs of the dampers designed by the contractor were tested in the condition in which they were received, and two additional pairs of both designs were tested after an elevated temperature postcure. Dampers were test cycled under the conditions of no prestress, a 0.76-millimeter (0.030 inch) compression prestress, and a 1.17-millimeter (0.046 inch) shear offset prestress. These mounting variations were evaluated to determine the effect they would have on the damper fatigue life. No fatigue failures occurred and the damping changes were comparable; therefore, no prestress was used in the system. The postcured dampers were harder and had higher damping constants. At equal displacements, the postcured dampers are receiving a higher stress under the test conditions than the dampers in the condition in which they were received. The actual values of the damping constant compared to the number of test cycles for two pairs of dampers from the commercial source that were postcured are shown in figure 4. The dampers maintained a constant damping value within 5 percent. As a result of the vacuum tests and life testing, the dampers of the cylindrical design were selected for use on the flight system.

Screening Criteria

Specifications were generated to provide the necessary controls and screening procedures. The manufacturer is required to produce parts from one controlled lot of materials. After receipt, the dampers are subjected to a postcure at 489.80° K (350° F). As a lot-acceptance test, 10 percent of the lot are tested for condensables and outgassing. Each device that is used in the system has the damping characteristics measured on a pair basis.

FLEXURE PIVOTS

Description

Two flexure pivots provide the suspension points for the scan mirror. These pivots must withstand the launch environment and subsequently operate at $\pm 2.9^\circ$ rotation for 150 000 000 cycles. The flexure pivots are commercially available devices. They are 90° symmetrical two-strip pivots, in which one of the strips has been divided for increased lateral rigidity. The stationary portion of the pivot is brazed to one cylindrical barrel, while the rotating portion of the pivot is brazed to the other cylindrical barrel. The flexing elements bend to allow rotation of one barrel with respect to the other. The flexure elements are stamped from close tolerance, cold-rolled spring stock of type 420 stainless steel. Flexure pivots do not have a fixed center of rotation.

The amount of the change of center of rotation during rotational displacement of one sleeve with respect to the other is termed centershift. The flexure pivots being used are 15.9 millimeters (5/8 inch) in diameter and of the $\pm 15^\circ$ design. The flexing strips are 0.34 millimeter (0.0135 inch) thick.

Vibration Tests

The two important tests that these parts must pass before being included in any flight system are vibration and cycling life testing. The pivots used on MSS have passed successfully tests that simulate the launch environment. Vibration tests were conducted on an early engineering model and later on the systems with the final design. The early vibration testing is critical with a flexure-mounted system, as launch excitations close to the natural frequency of the pivot/mirror system can result in extremely high gravity loads. There is negligible inherent damping in flexural pivots, and this results in high transmissibilities of input vibration levels. The mechanism was designed originally with 12.7-millimeter (1/2 inch) diameter pivots, and had to be redesigned to accept stiffer 15.9-millimeter (5/8 inch) diameter pivots in order to pass this test.

Life Tests

To demonstrate compliance with the system cycling life requirements, it was necessary to determine the fatigue-life characteristics of the flexure pivots. A test program to produce a curve of stress as a function of cycles to failure for the parts was performed (fig. 5). The equivalent angle is included on the right of the figure for the $\pm 15^\circ$ pivot design, and also the MSS operation required is noted.

The expression used to calculate the stress for pure rotation is

$$S = \frac{Et}{2L} \Theta$$

where S = stress, E = material modulus of elasticity (1.99×10^{11} N/m² (29×10^6 psi) for 420 CRES), t = flexure thickness, Θ = angle through which the pivot is rotated (in radians), and L = effective flexure length.

The test grouping is typical of fatigue testing. Some scatter in results is expected because of the inhomogeneous nature of metals and the tolerances in manufacturing the parts. These particular pivots closely follow the life curve that is predicted by the manufacturer data. Also, previous life tests of different size pivots (ref. 1) indicate that the MSS design requirements are below the endurance limit of the pivot material.

Screening Requirements

In addition to meeting life and environmental requirements, the pivots must meet critical parameters regarding mirror support. Two pivots from each manufacturer lot are subjected to destructive static-load test and to a metallurgical examination. Each

pivot is visually examined with both a conventional microscope and a small diameter borescope for surface defects. Pivots are examined for evidence of burrs, nicks, or scratches that might reduce the strength or life of the part. Each pivot is subjected to a dye-penetrant inspection for evidence of surface and subsurface cracks. All flight pivots are subjected to a radial load of 978.56 newtons (220 pounds) to test for defective braze joints.

To ensure minimum stress and satisfactory mirror alinement under scan conditions, each pivot is tested for center-shift and torsional spring rate at $\pm 3^\circ$ rotation. Then, pivots are paired so that they have closely matched torsional spring rates, and, when rotated, the center shifts are the same magnitude and in the same direction. Matched pairs of pivots that passed all visual and physical tests were used in the flight units.

CONCLUDING REMARKS

To achieve state-of-the-art performance of a complex instrument, consideration of each detailed element is required. This must be followed by evaluation and screening of the critical ones before final selection and use. The identification and removal of potential early-life failures is an important factor in meeting overall system performance requirements. The qualification and screening of critical materials and mechanical devices such as dampers and flexure pivots is particularly important in applications such as the multispectral scanner, because redundancy cannot be used as readily for mechanical devices as it is for electronics.

REFERENCE

1. Olson, J. L.; The Evaluation of Flexural Pivots to Meet Critical Performance and Life Requirements. Rep. 70-DE-76, ASME, May, 1970.

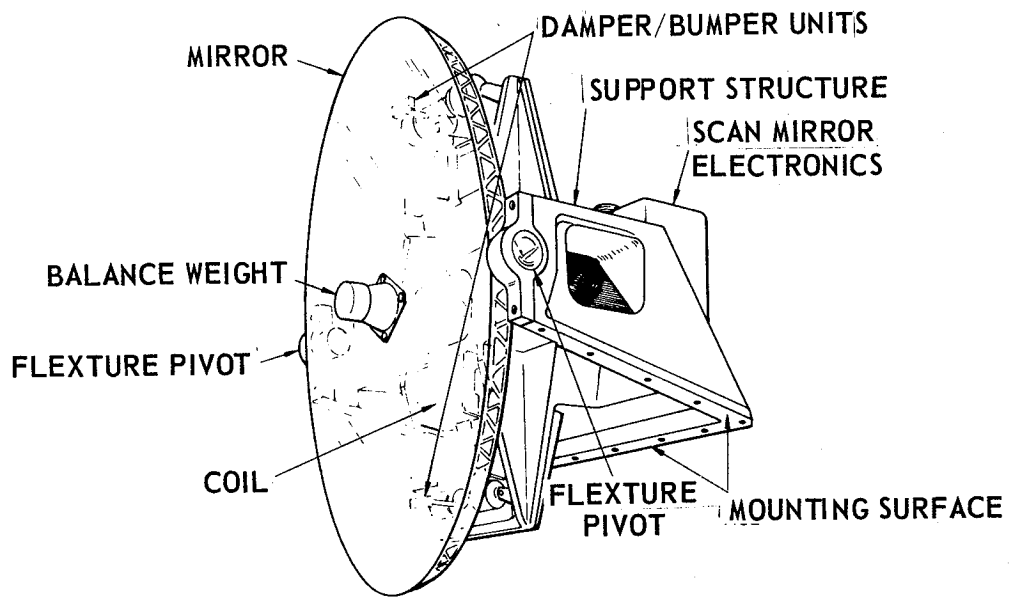


Figure 1. - Scan mirror mechanism.

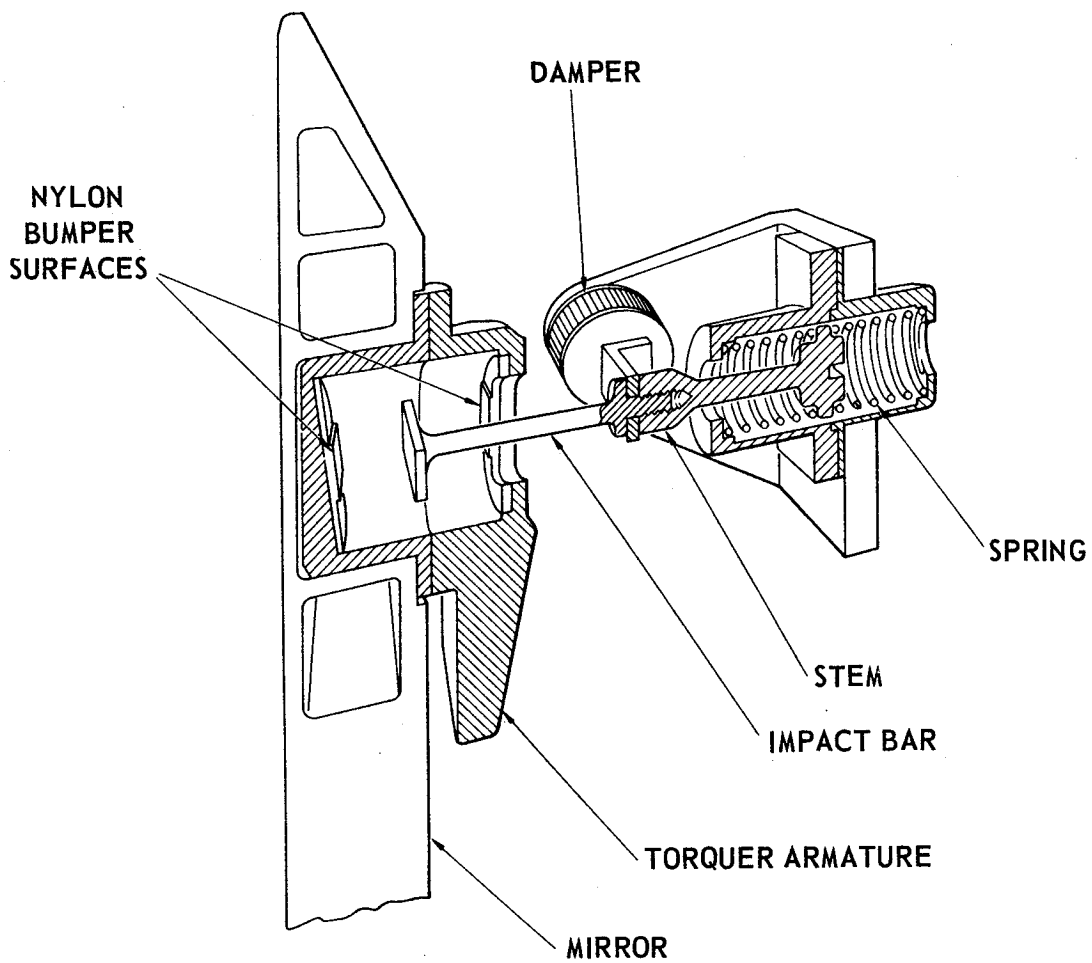
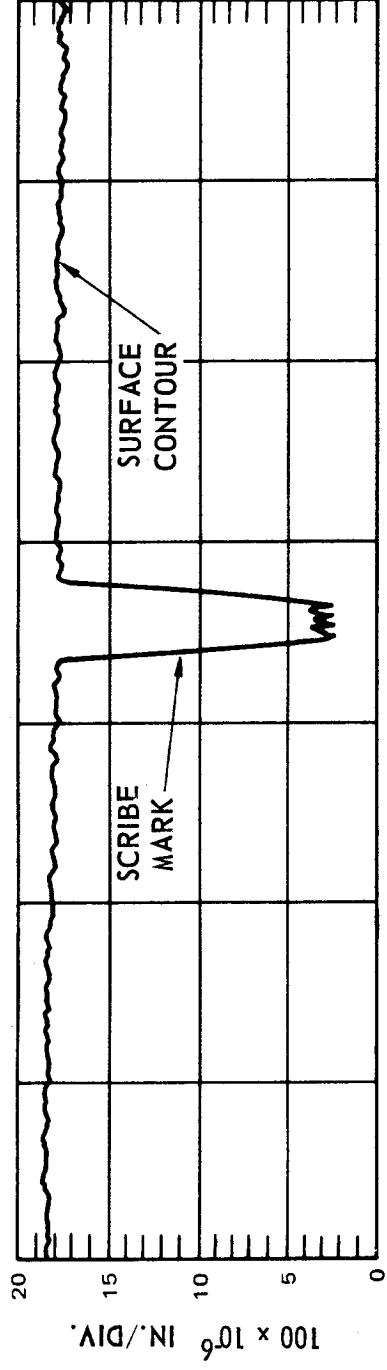
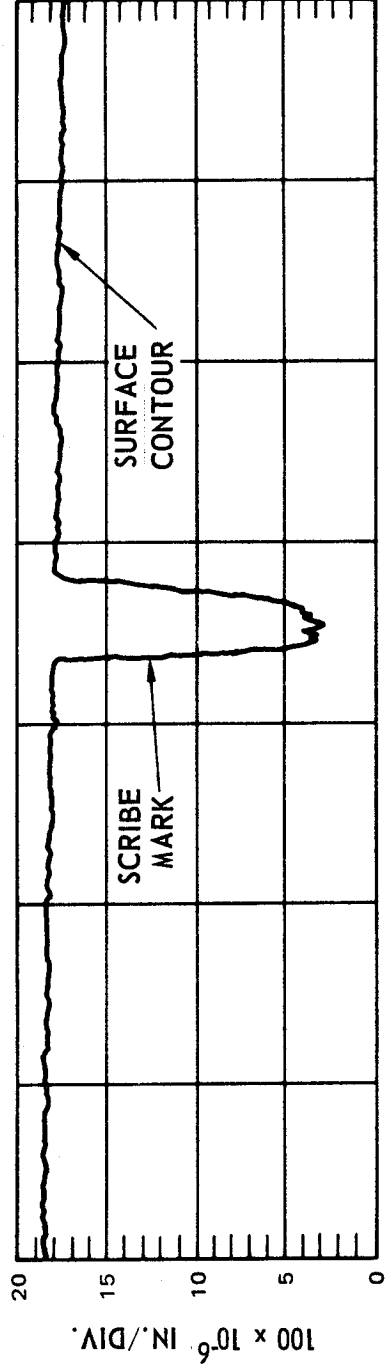


Figure 2. - Bumper/damper unit.

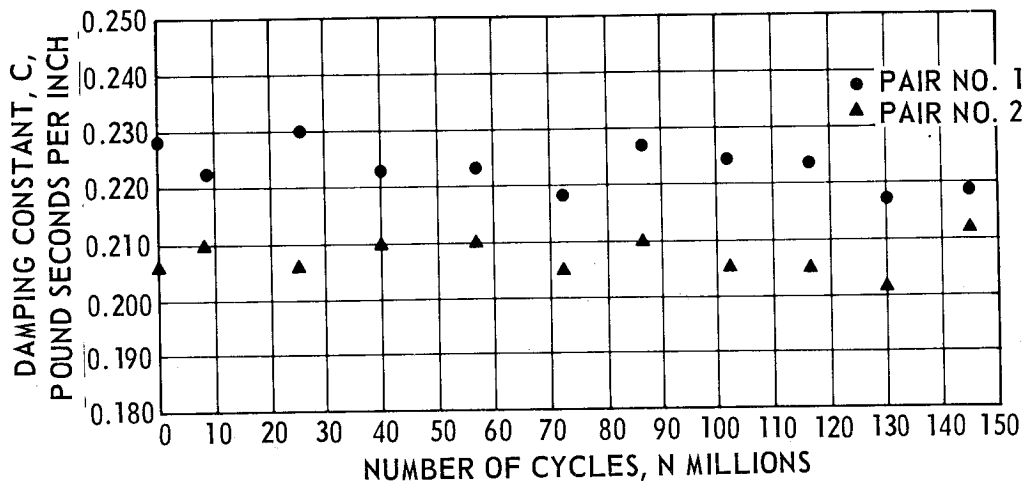


(a) Baseline.



(b) After 142 000 000.

Figure 3. - Bumper surface wear.



NOTE: DAMPING TEST FREQUENCY = 144

Figure 4. - Damping as a function of life.

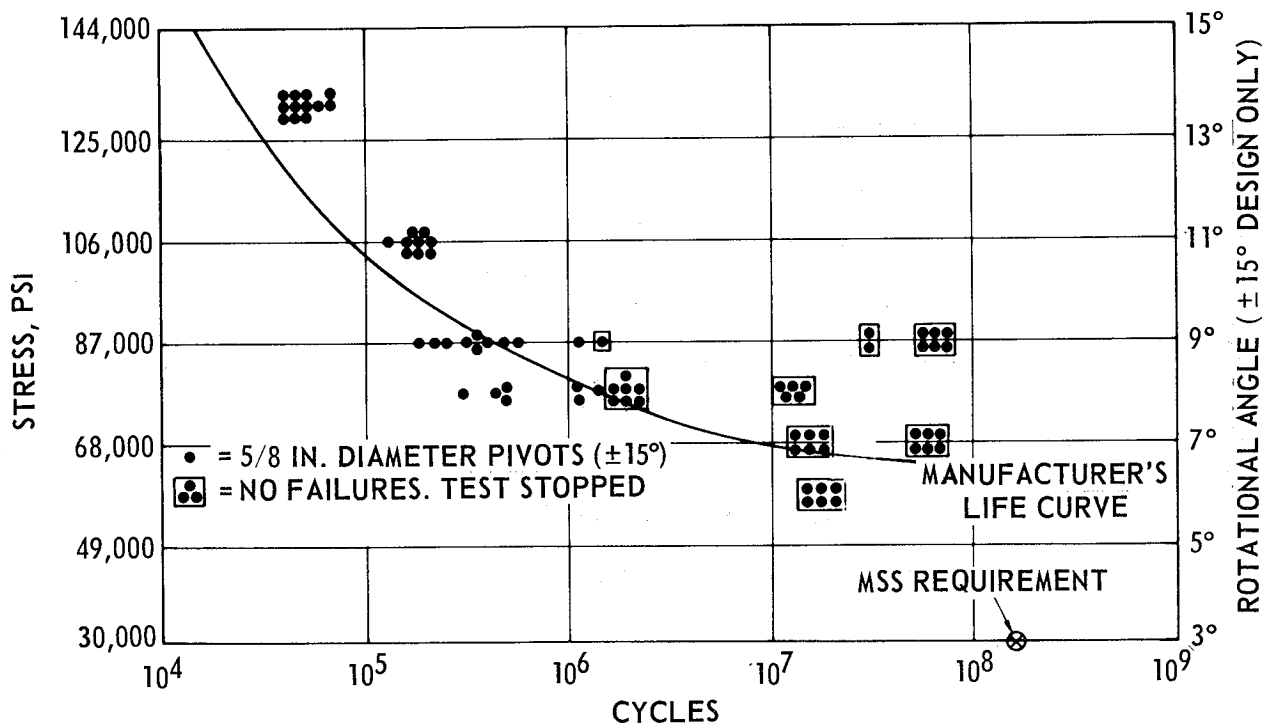


Figure 5. - Flexure pivot fatigue test results.

