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DEVELOPMENT OF A NEW SEAL FOR USE ON LARGE OPENINGS OF PRESSURIZED SPACECRAFT

(Center Director's Discretionary Fund Project Number 92-11)

By B. Weddendorf

Structures and Dynamics Laboratory
Science and Engineering Directorate

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This report presents the results of the Center Director's Discretionary Fund project "Development of a New Seal for Use on Large Openings of Pressurized Spacecraft." The goal of this project was to design, build, and test an example of the seal invented by the author for use on Space Station *Freedom* and patented in 1991. The seal features a metallic spring core and replaceable elastomeric sealing elements. The metallic spring is designed to retain the sealing force of the elastomeric element against both sides of face seal gland for any specified amount of waviness or separation of the glands. A seal able to tolerate at least 1.3 mm (0.05 in) of flange distortion or separation and a test fixture of this seal which allowed direct comparison testing of O-rings were built. These designs were tested to compare leakage at different amounts of flange deflection. Results of the testing show the development seal exceeded its requirement to seal 1.3 mm of flange separation by 1 mm. This compared with the O-ring leakage increasing dramatically at 0.5 mm of separation. The development seal also leaked at a lower rate than the O-ring seals in all tests.

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TECHNICAL MEMORANDUM

DEVELOPMENT OF A NEW SEAL FOR USE ON LARGE OPENINGS OF PRESSURIZED SPACECRAFT

INTRODUCTION

This report presents the results of the Center Director's Discretionary Fund project development of a new seal for use on large openings of pressurized spacecraft. The goal of this project was to design, build, and test an example of the seal invented by the author for use on Space Station *Freedom* (S.S. *Freedom*). This seal was patented October 1991 as U.S. Patent No. 5,052,152 "Double Face Sealing Device." The double face sealing device (DFSD or the seal) is intended to provide redundant long-term sealing of large openings in lightweight pressure vessels which have structural deflections at the sealing flanges too large to seal by conventional methods. This seal is for use in face sealing applications only, for example, around a hatch, a window, or between modules of a spacecraft. The seal is best suited for low pressure applications, such as atmospheric to vacuum sealing. It is ideal for use in aircraft and surface ship doors and hatchways, as well as large vacuum chamber doors. The seal features a central metal plate with cantilever springs projecting radially inward and outward, as can be seen in figure 1. This plate can be made up of segments around the circumference of the seal. The seal is not limited to round applications, but will work on any opening with radiused corners. The plates are secured to one of the sealing flanges. The sealing action is accomplished by two replaceable elastomeric elements, each one continuous. The elements are shaped to snap over the cantilever springs of the seal and are positively retained by a slight preload against the flange by the spring. When the seal is activated by bringing the opposing flange into contact with the elastomeric elements, the cantilever springs deflect, storing most of the sealing energy, as can be seen in figure 1. Before the springs are over stressed by this deflection, the sealing flange contacts the metal plate through which all additional loads will travel. The seal is redundant, as either element alone would form a sufficient seal. The seal may be tailored to seal flanges with any amount of warpage, deflection, or separation, provided there is enough space to fit sufficiently long cantilever springs. The sealing force required is lower than in an elastomeric seal, and the metallic springs serve to provide a more uniform stress state in the elastomer than occurs in a purely elastomeric seal. The uniformity of the stress state should enhance the performance of the seal over long periods. This is because the metal springs linear response and greater travel minimizes and compensates for viscous flow of the elastomer which occurs over time as a function of the local stress. The added resilience of the metal makes the seal less sensitive to extremes of temperature and more responsive to dynamic flange movement at cold temperatures and after long time periods.

SEAL DESIGN AND FABRICATION

The application chosen for the DFSD was a modification of the S.S. *Freedom* square hatch, as shown in figure 2. The hatch is an example of a lightweight, unevenly loaded sealed structure. Its square shape creates uneven loading upon pressurization. The hatch, however, was so large that the test fixture required would have been difficult to move, requiring a crane or forklift to be used to switch out plates between each test. For this reason, the hatch configuration was modified to remove the straight sections and leave only the four corners. The corners form a circular seal of a manageable size for testing, while still retaining the same cross-sectional size and curvature as the S.S. *Freedom* hatch. The S.S. *Freedom* hatch baseline seal design consisted of a pair of O-rings. This design was duplicated in circular form for comparison testing with the DFSD. The flange width for the hatch seal limited the length of the DFSD spring legs, which limits the amount of deflection the seal can handle. This application would prove if the DFSD could be installed into the same amount of space as dual O-rings.

Once the application was decided, the next step was to determine the cross section of the DFSD to fit within the space limitations of the S.S. *Freedom* hatch seal. The design goal was to allow the largest amount of flange distortion and separation possible. This translated directly to travel of the springs. Computer finite element modeling (FEM) was used to achieve a near constant stress design for the critical and highly stressed springs. This was done using existing personal computer FEM capability in the branch, and later checked and revised using the ALGOR™ FEM package purchased for this project. The replaceable elastomeric elements were designed to work in conjunction with the metal springs. The cross section of these parts was designed to snap over the turned out ends of the metal springs for positive retention. The elastomeric elements must be sized to function with the spring as a system, with particular consideration to the thickness of the elastomer squeezed between the tip of the spring leg and the flange. This thickness must be sufficient to allow the elastomer to bridge the gaps in the spring between the separate plates. The thickness and its tolerance then must be considered in the sizing of the spring plate. Both the inside and outside elastomeric parts have the same cross section (fig. 3), so they could be made with the same extrusion die. This technique was chosen over molded construction to keep cost down and to allow for different seal lengths if needed in the future. The extruded shape requires a bonded joint to achieve a continuous seal. The material chosen for the DFSD seal elements and the O-rings was fluorocarbon elastomer per MIL-R-83248 (Viton 747™) with a hardness of 75 Shore A. This was the material baselined for the S.S. *Freedom* hatch O-rings. The overall seal was sized and toleranced so that the maximum deflection of the cantilever spring legs would never exceed a deflection resulting in a stress which was 90 percent of the yield stress. This can be done in any application of the DFSD because the central plate upon which the cantilever springs are mounted takes all the clamping load once the flanges contact it on both sides. The thickness of this block, therefore, can be tailored to limit the deflection of the springs at any desired level. The radial width of the block was minimized in this design to maximize the length of the cantilever springs. Reduction of this width is limited by the retention screws. These screws are needed to hold the seal plate down to one of the flanges, into a set of tapped holes. This set of tapped holes is the only feature on the sealing flanges required by the DFSD. The DFSD spring plates were cut into four pieces to demonstrate the feasibility of larger segmented plate designs (figs. 4 and 5). This division of the spring plates also allows for the future use of the hardware as corner pieces of a large square opening seal by simply adding straight segments between the corners (fig. 6). Additionally, the seal springs were separated by thin cuts to allow them to act independently in tracking flange distortion. The material chosen for the spring plates was CRES 17-4 PH per AMS 5604 in the H900 condition. This material has high strength, allowing for high stresses and therefore larger deflections. As with any spring design, high stresses are inevitable as a result of large deflections, so a high strength material is required.

A test fixture was designed to accommodate both the DFSD and the comparison S.S. *Freedom* hatch O-rings (figs. 7 and 8). This fixture allows both seals to be tested with the same test facility lines and instruments. To allow testing both seals in this manner, the test fixture is made up of a single fixed back plate which is ported for pressurization and instrumentation lines, and removable face plates for both types of seals to be tested. Three ports are provided in the back plate, two near the center and one through the flange between the seals (fig. 7). The plates are joined with eight axial bolts through tabs around the periphery. A spacer (fig. 9) can be installed around each bolt between the tabs to separate the plates at that point, creating a controlled flange distortion. Different length spacers are used to achieve the necessary flange deflection. The back plate is supported by a tubular steel stand by two of the bolting lugs. The stand accommodates either seal plate by supporting it in a similar manner and acts as a guide when assembling the fixture (fig. 10).

The metal spring elements of the seal were fabricated under contract by a local machine shop. The four corner pieces formed a circle when placed together, with enough room for an electro discharge machining (EDM) cut between segments. Because they formed a continuous circle, that is the way they were machined, with most of the machining done on a turret lathe. The spring elements were rough machined to slightly above drawing dimensions before final heat treating to the H925 condition. The final step was EDM cutting of the separation notches in the spring legs. These parts were quite expensive to machine, and, in talking to the shop manager, the very small and delicate cross section chosen was mostly responsible for the cost. The notches also contributed to the cost, and it is not known if they are necessary. These parts ran several months over schedule in the shop, but the quality of the spring and the elastomer was very high, and they were accepted based on visual inspection. The test fixtures were machined from 6061 aluminum plate. Another local machine shop did this job under contract, and the parts were easily made on schedule. The O-ring plate had an unacceptable finish in the base of the grooves and had to be returned for rework.

SEAL TESTING

Testing of the DFSD began with verification of the test hardware by assembly fit check and visual inspection. Because the DFSD was a completely new design, details like the retention of the elastomeric elements and ease of assembly and installation were unknown. Assembly of the DFSD revealed that the seal is easy to assemble and use, the parts fit perfectly, and the elastomer retention is quite positive (fig. 11). The next step in testing was a series of leak tests at different amounts of flange warpage and separation. This was to determine how much each type of seal could tolerate without a sharp rise in leakage. This testing was conducted by the Experiments and Components Test Branch of the Systems Analysis and Integration Laboratory in building 4708. The test method chosen was pressure decay. Pressure decay testing involved pressurizing the interior of the seal cavity with approximately 15 lb/in^2 gauge missile grade air through one of the central ports and sampling pressure transducers ported to the other central port and the port between the seals with a PC-based data acquisition system (fig. 12). The PC collected three data points every second from the transducers. The pressure drop between two points in time gives the leak rate, which can be determined directly from these data. The leak rate in lb/in^2 gauge/second was converted into standard cubic centimeter/minute (sccm) leak rate by application of the ideal gas law with the volume of the test fixture. The tests began by pressurizing the fixture to roughly 103 kPa (15 lb/in^2 gauge), and then allowing the seals to leak the pressure down. Each test was run for approximately 15 min, or until about 34 kPa (5 lb/in^2 gauge) remained in the fixture. The test setup was verified by establishing a baseline leak rate for each seal by testing them fully compressed. This was followed by tests of both kinds of seals with spacers of varying thickness used to separate or bend the flanges. The tests were conducted with three trails for each condition. The test selected to represent the three is one which agrees closely with at least one of the other two tests. Figure 13 gives the leakage test data.

The tests showed that the new seal works very well. The new seal tolerated more than three times as much joint warpage before increasing leakage as the S.S. *Freedom* hatch design O-rings used as a control. The leak rate for the DFSD is lower than that of the O-ring seals in all tests. The DFSD began leaking faster only when the flange was warped 2.3 mm (0.090 in) which was 1 mm greater than the requirement. The comparison O-rings leaked at 0.76 mm (0.03 in) warpage and at 0.5 mm (0.020 in) separation.

Structural testing of the seal was conducted by the Structural Test Branch of the Structures and Dynamics Laboratory. The spring rate and maximum stress of the metal seal springs were determined to verify the finite element analysis done for design. Testing was done in a SATEC tensile testing machine. Three types of tests were run. The first was to determine the spring rate of the seal. This was done by placing the seal test fixture horizontally in the SATEC machine and loading it in compression. Four of the fixture clamping bolts were installed loosely to act as guides. Four linear displacement transducers were installed around the outside of the fixture, measuring the deflection of the seal. The data collected were load versus deflection. This test setup can be seen in figures 14 and 15. The second test was to determine the maximum stress in the seal springs. This was accomplished by placing six uniaxial strain gauges on the seal springs. These were installed in three sets of two gauges. Each set of two measured the stress on opposing spring legs, and each set was placed to measure the stress at a different location along the spring. Gauges one and two measured stress near the center of the spring. Gauges three and four were placed near the root of the spring. Gauges five and six were near the tip. The placement of the gauges can be seen in figure 16. The overall test setup can be seen in figure 17. Data collected for this test were load, deflection, and strain of the seal springs. The third type of test was to determine if the bolting torque used in the leak testing was sufficient to fully compress the seal. The data collected were strain at two levels of bolt torque. For this test, the displacement transducers were removed so that all eight bolts could be used to bolt the fixture together. Results of the tests are as follows:

Test 1: Maximum load when seal is fully compressed = 29,804 N (6,700 lb)

Maximum deflection of seal = 1.727 mm (0.068 in)

Test 2: Maximum load when seal is fully compressed = 29,804 N (6,700 lb)

Maximum deflection of seal = 1.727 mm (0.068 in)

Maximum strain = 3,000 microstrain

Maximum stress = 620 kN/m^2 (90,000 lb/in^2)

Minimum strain = 2,850 microstrain

Test 3: At 40.6 N-m (30 ft-lb)

Maximum strain = 3,400 microstrain

Maximum stress = 702 kN/m^2 (102,000 lb/in^2)

Minimum strain = 3,103 microstrain

At 81.2 N-m (60 ft-lb)

Maximum strain = 3,412 microstrain

Maximum stress = 704 kN/m^2 (102,360 lb/in^2)

Minimum strain = 3,093 microstrain.

The test results show the seal to have a required load of 29,804 N (6,700 lb) to fully compress. This load over the average circumference of the seal is 7,237 N/m (41 lb/in) per seal. The stroke distance of the seal was less than expected. This was partially due to the horizontal test setup in which the seal bore the weight of the top plate of the test fixture and a large and heavy spherical bearing. The combined weight of the plate and bearing was about 828 N (186 lb). Taking this extra force into account, the spring rate is about 4.31×10^6 N/m^2 (624 lb/in^2) for each seal in the double seal. The curve of load versus deflection is very linear, showing a crisp transition to horizontal when the flanges contact the central plate. This curve can be seen in figure 18. The stresses in the three parts of the spring agreed well, proving the design was near constant stress as predicted by the finite element model. The bolt torque of 40.6 N-m (30 ft-lb) was sufficient to fully compress the seal. This was evident when the torque was doubled without causing much of a change in the strain in the seal spring.

CONCLUSIONS

This project proved that the DFSD is a viable seal for a typical low pressure face sealing application. The seal was successfully designed and built to fit in the same amount of space allowed for double O-rings in the hatch flange design proposed for S.S. *Freedom*. The DFSD outperformed the O-rings in all leakage tests and exceeded the design requirements for tolerance of flange warpage and separation. Further, the DFSD was shown to have a low required sealing force and a linear deflection versus load curve in structural testing. The seal also proved to have a predictable and controllable stress in the spring elements. Functional testing showed that the seal met all requirements: it is easy to install and remove and provides positive retention of the elastomeric elements. All of this testing confirms that the DFSD works as it is intended, and can therefore solve difficult sealing problems in a variety of applications.

Manufacturing the DFSD for this application proved difficult, however. This was because of the delicate cross section (fig. 19). Increasing the size of the flanges and the DFSD cross section is an obvious answer to this problem, if space permits. Another solution for lowering the cost of the DFSD may be the use of formed metal bellows technology to create the seal spring elements. These springs would be formed by one bellows segment, with the distal ends facing the opposite flanges. The center of the single bellows segment would be electron beam spot welded to the central metal plate. These springs and central plate could form a similar shape to the machined cross section used in this project without any difficult machining. An example of this type of design is shown in figure 20. Development of a DFSD design using this technology will be left to industrial users as it is outside the scope of this project.

The DFSD is patented, and the technology is available for commercial license. Firms considering commercial application of the DFSD should contact the MSFC Chief Intellectual Property Counsel, MSFC, AL 35812.

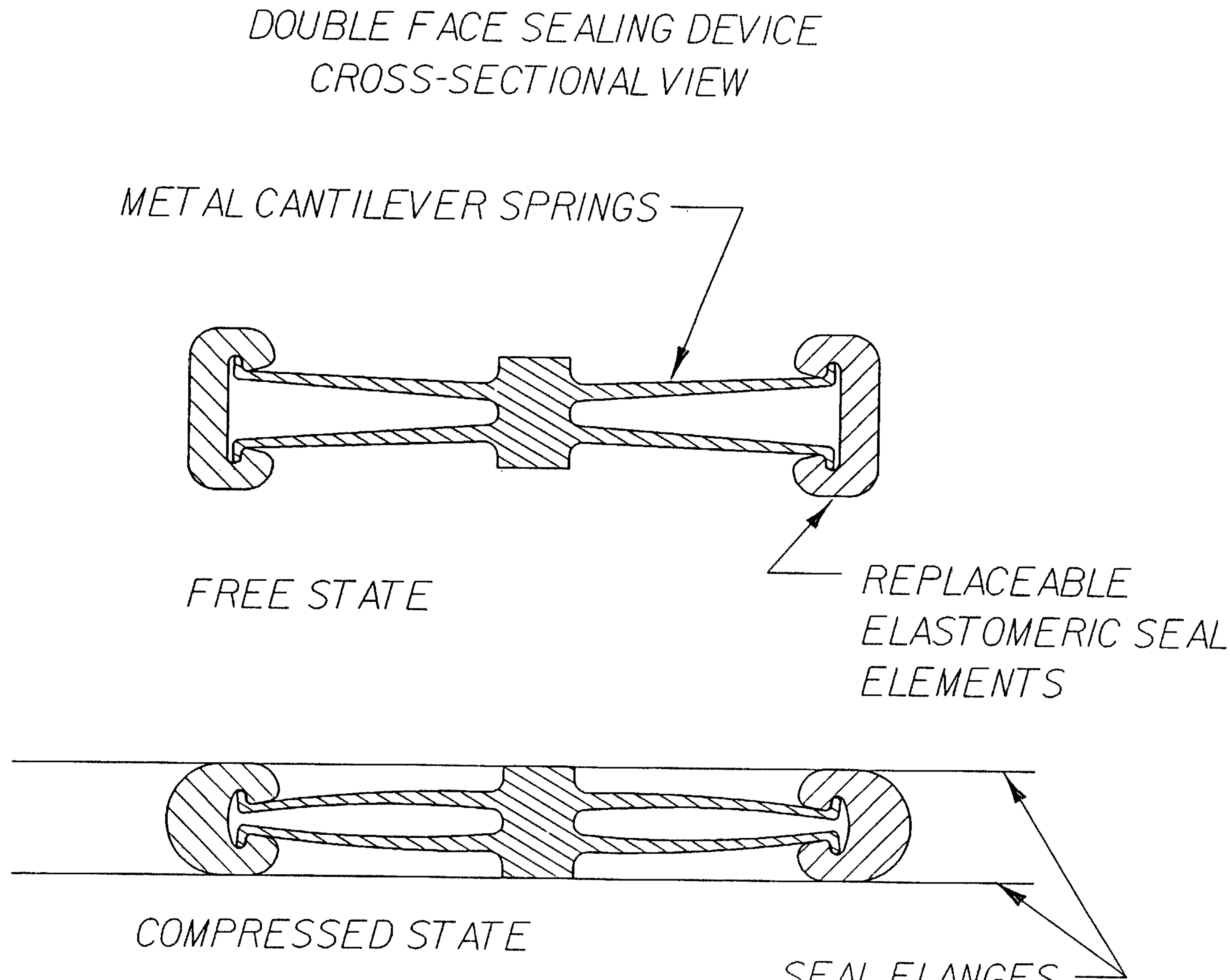

Figure 1. DFSD cross section.

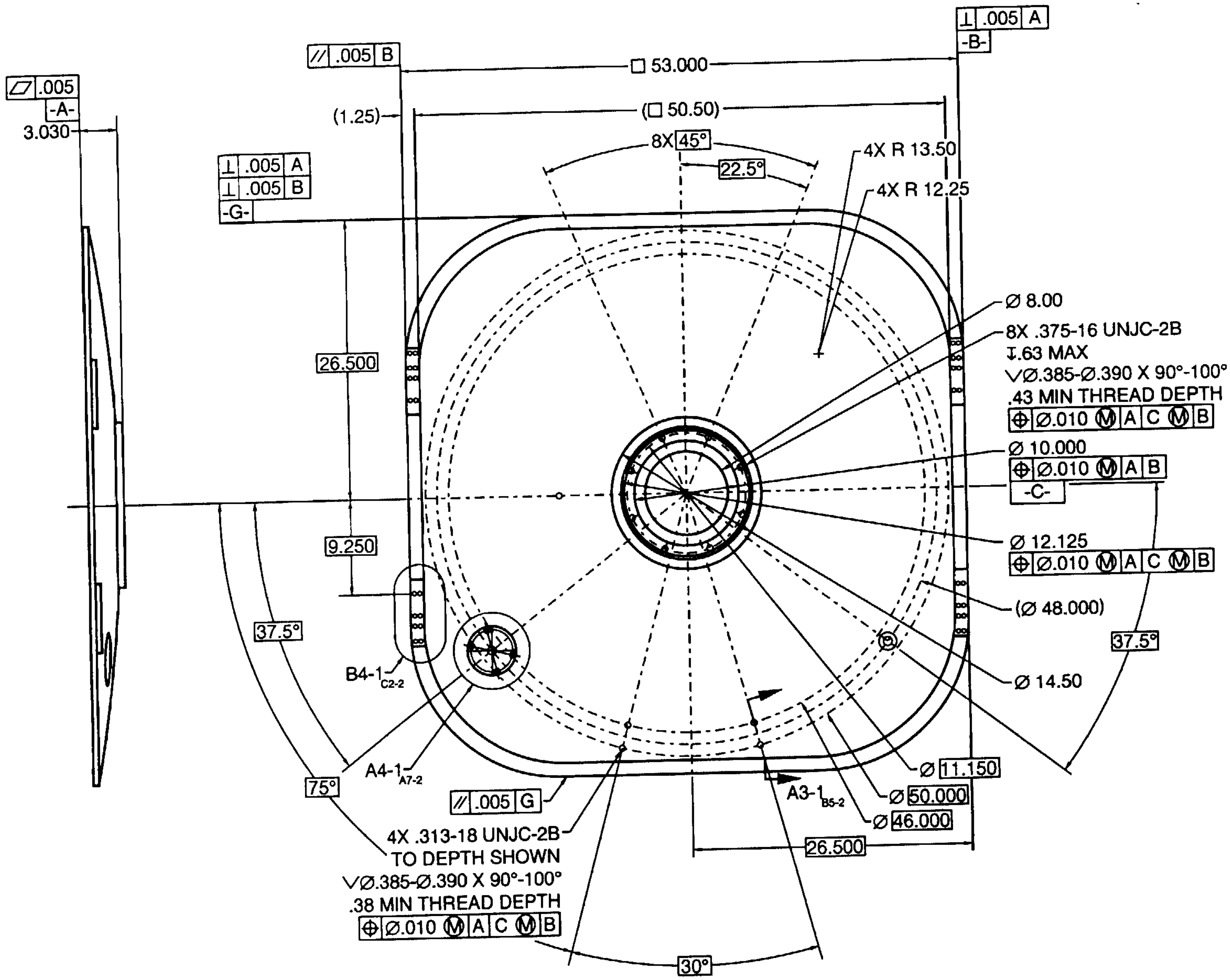

Figure 2. S.S. *Freedom* hatch.

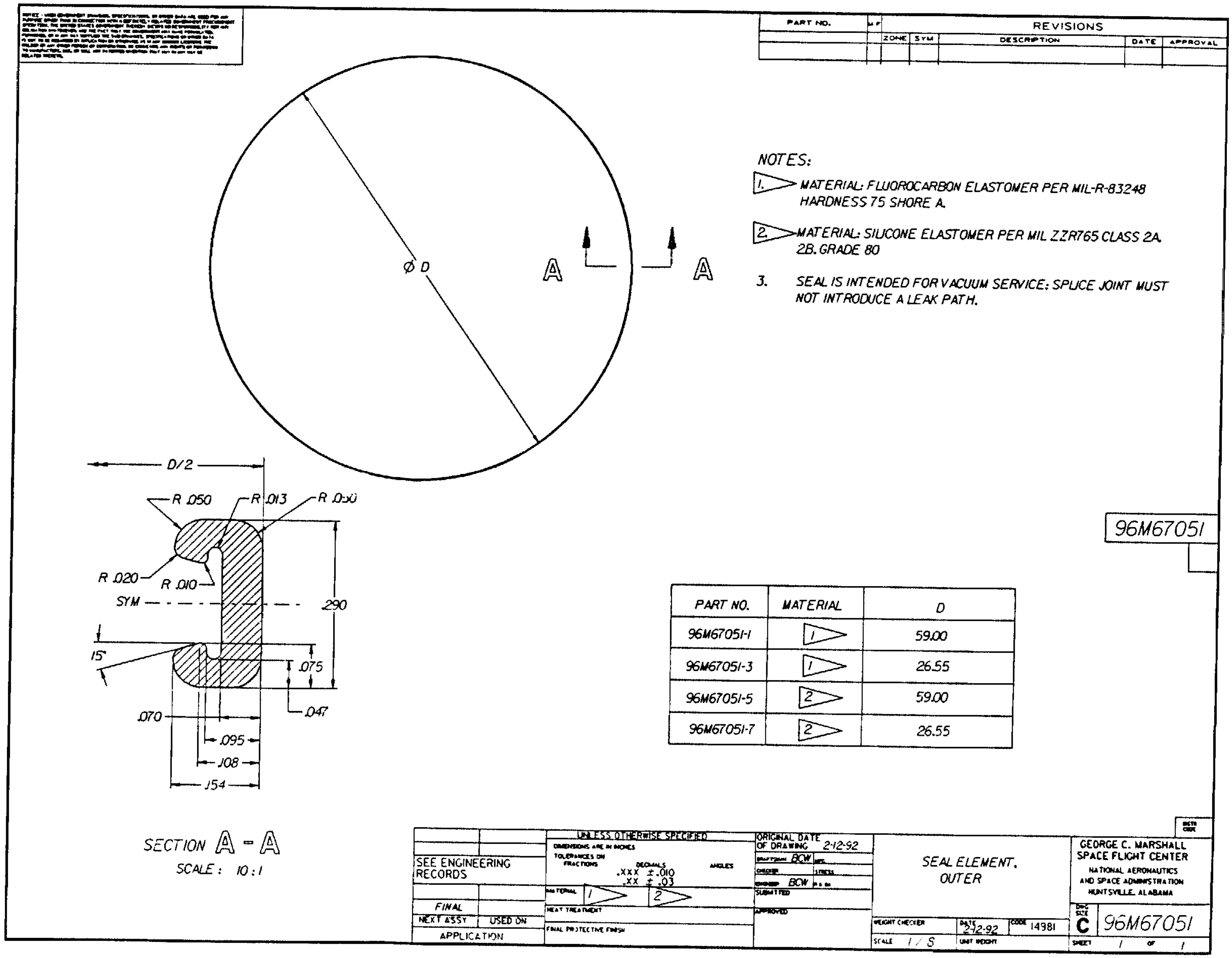

Figure 3. Elastomeric element drawing.

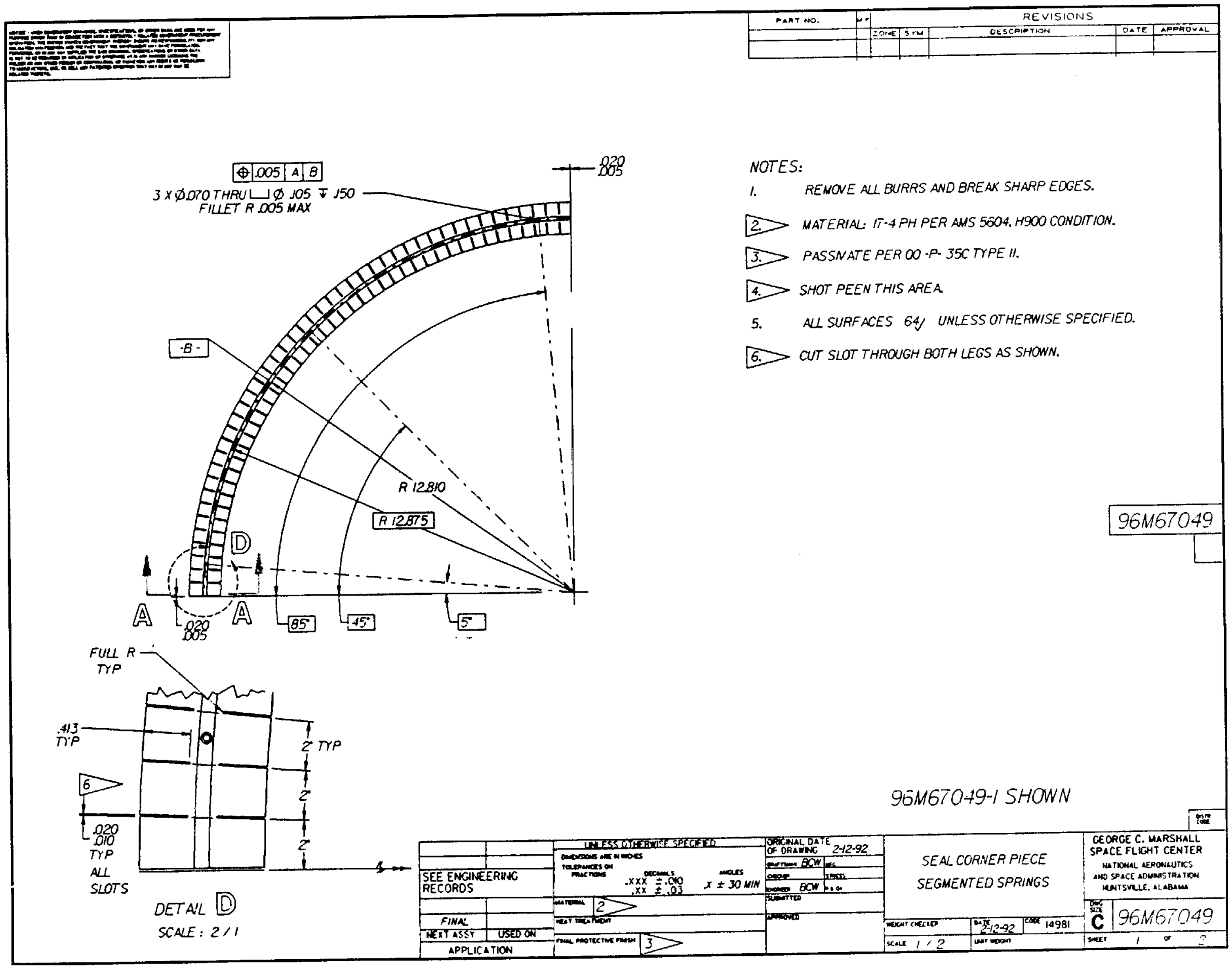

Figure 4. DFSD spring plate corner section.

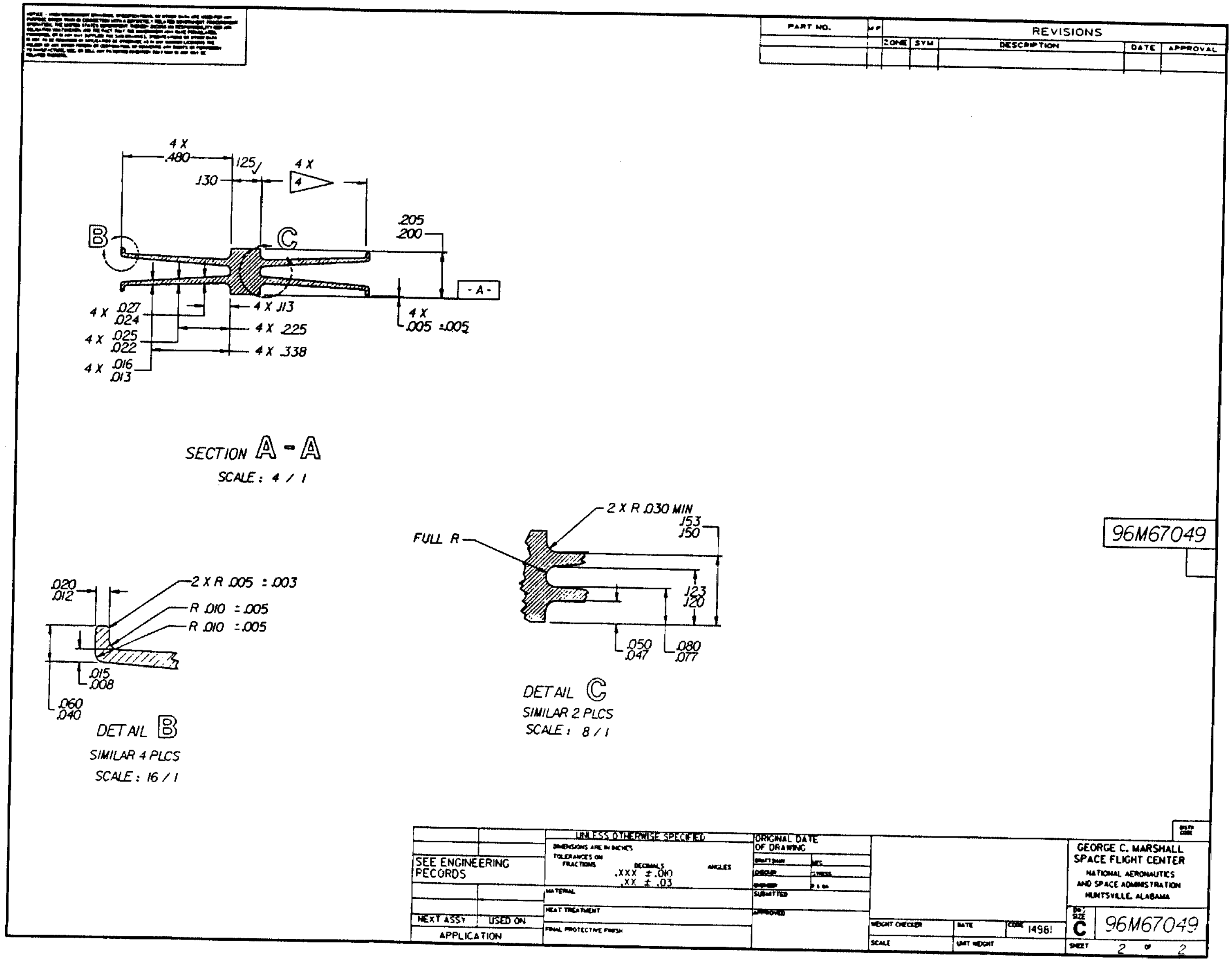

Figure 5. DFSD spring plate corner section.

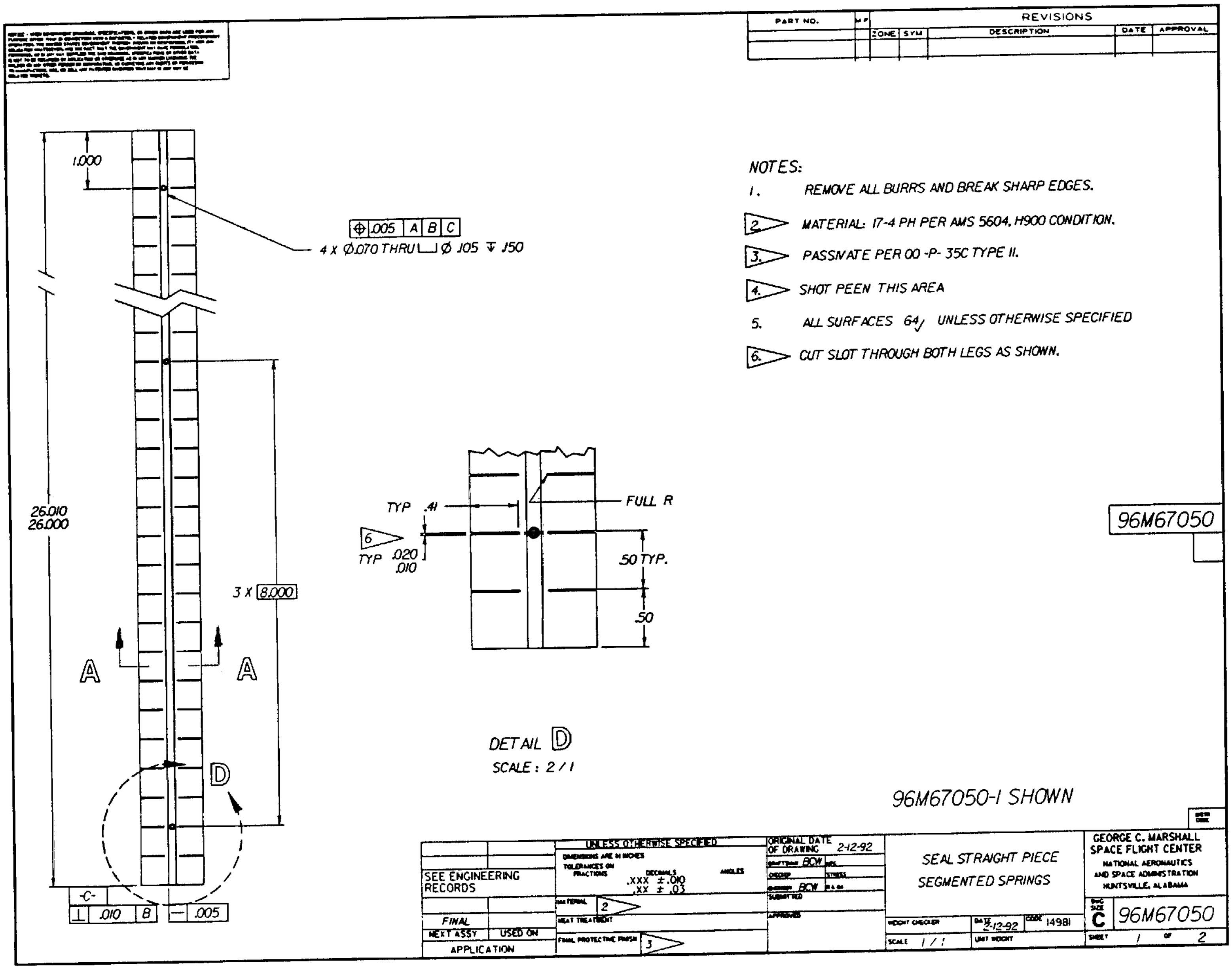

Figure 6. Straight spring plate section.

Figure 7. Front of leak test fixture.

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Figure 8. Back of leak test fixture.

Figure 9. Side view of leak test setup.

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Figure 10. Assembly of leak test fixture.

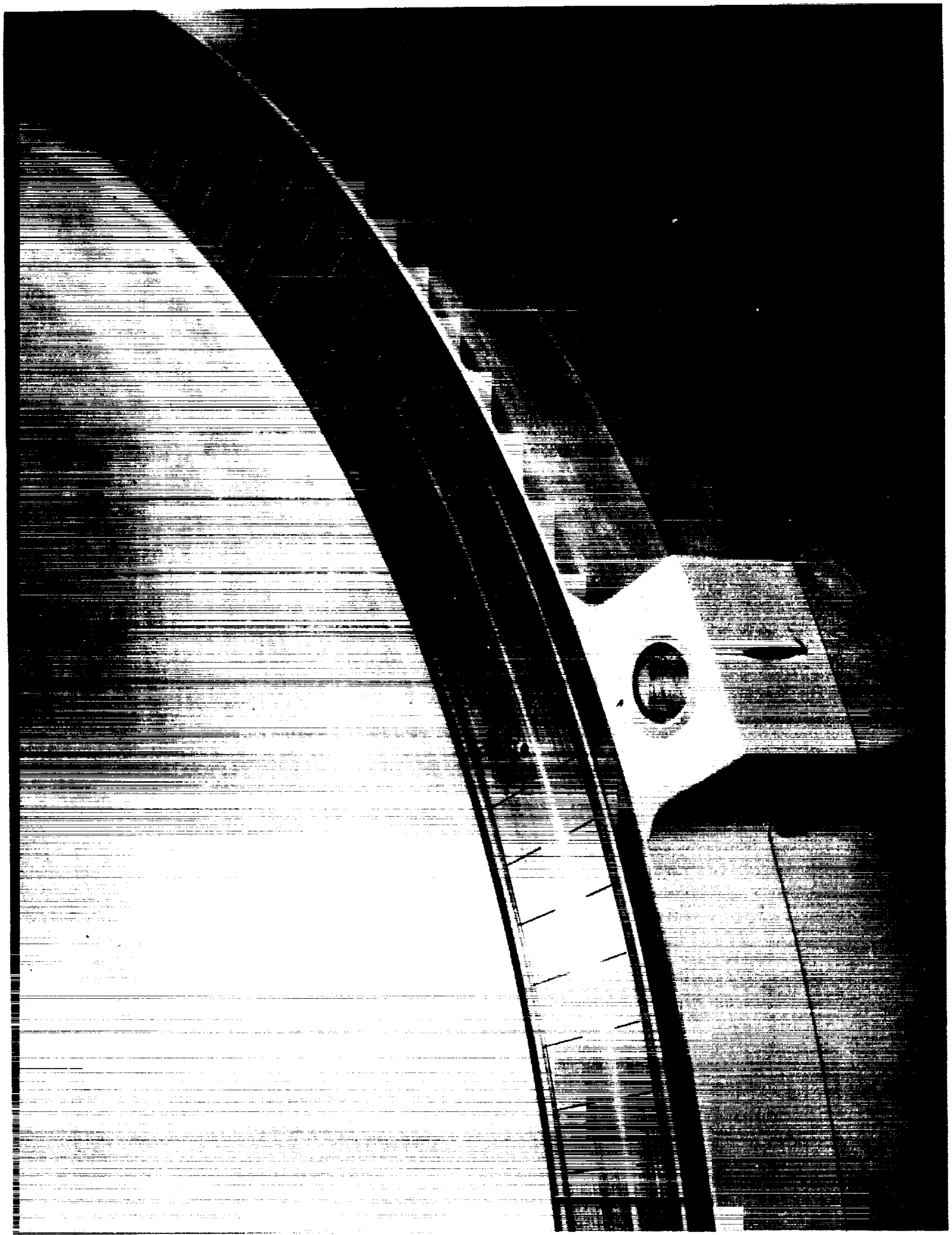

Figure 11. Closeup of DFSD assembled on flange.

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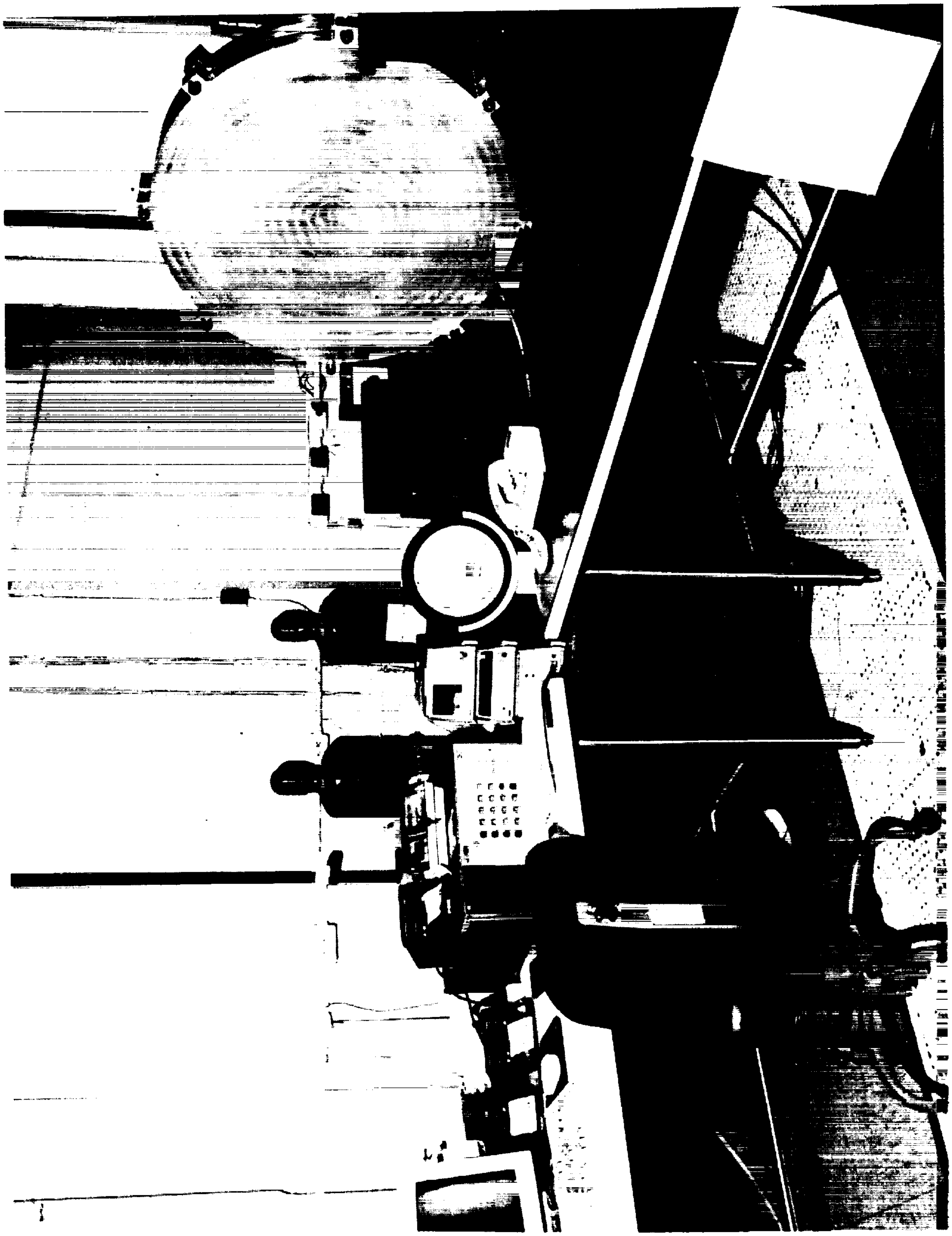

Figure 12. Leak test setup.

DFSD Leakage vs. Flange Warpage

Warpage, in	Time, sec	Pressure, psi	Leak Rate, sccm
0.000	881	0.063	3.77
0.010	960	0.059	3.24
0.015	824	0.077	4.40
0.020	940	0.066	3.71
0.030	790	0.053	3.54
0.040	940	0.036	2.02
0.050	840	0.022	1.38
0.060	868	0.033	2.01
0.070	856	0.022	1.36
0.080	874	0.023	1.39
0.090	980	0.779	41.95

DFSD leakage vs. Flange S eparation

Separation, in	Time, sec	Pressure, psi	Leak Rate, sccm
0.020	982	0.071	3.82
0.050	938	0.065	3.66

O-Ring Seal Leakage vs. Flange Deflection

Warpage, in	Time, sec	Pressure, psi	Leak Rate, sccm
0.000	897	0.086	4.46
0.010	897	0.122	6.33
0.015	919	0.132	6.68
0.020	840	0.105	5.82
0.030	208	0.512	114.56

O-Ring Seal Leakage vs. Flange Separation

Separation, in	Time, sec	Pressure, psi	Leak Rate, sccm
0.020	419	5.980	664.21

Figure 13. Table of leak test data.

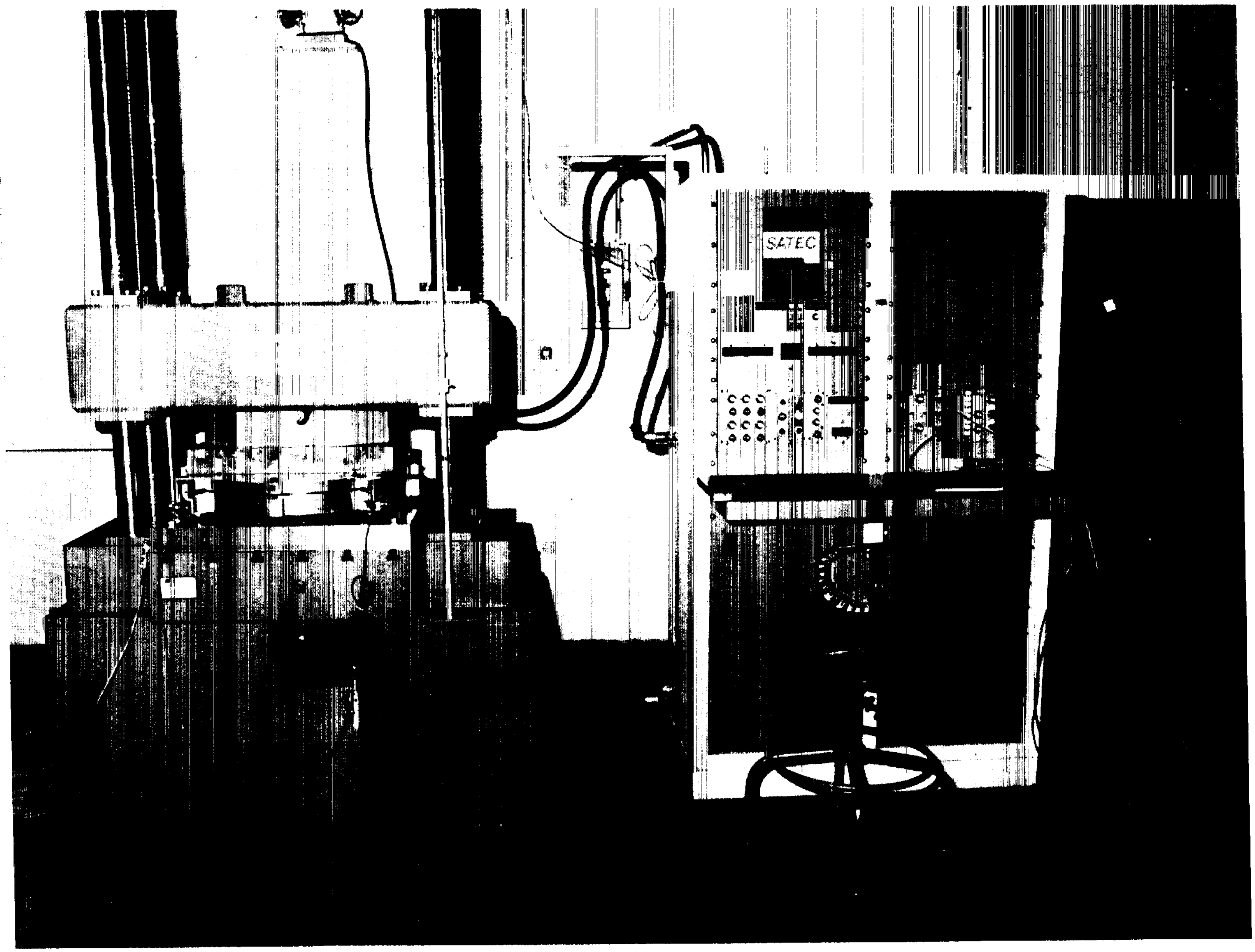

Figure 14. Structural test setup.

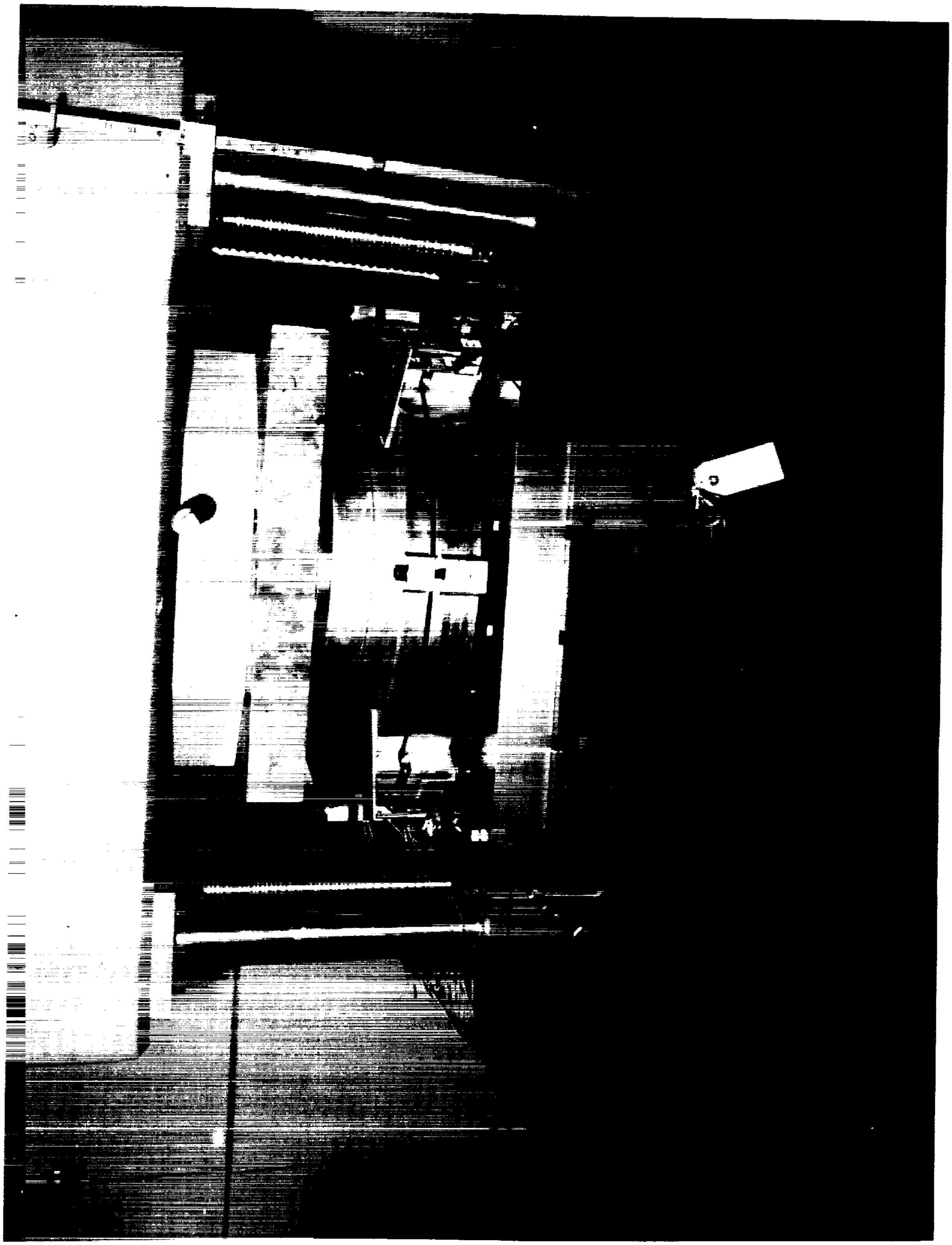

Figure 15. Deflection versus load test.

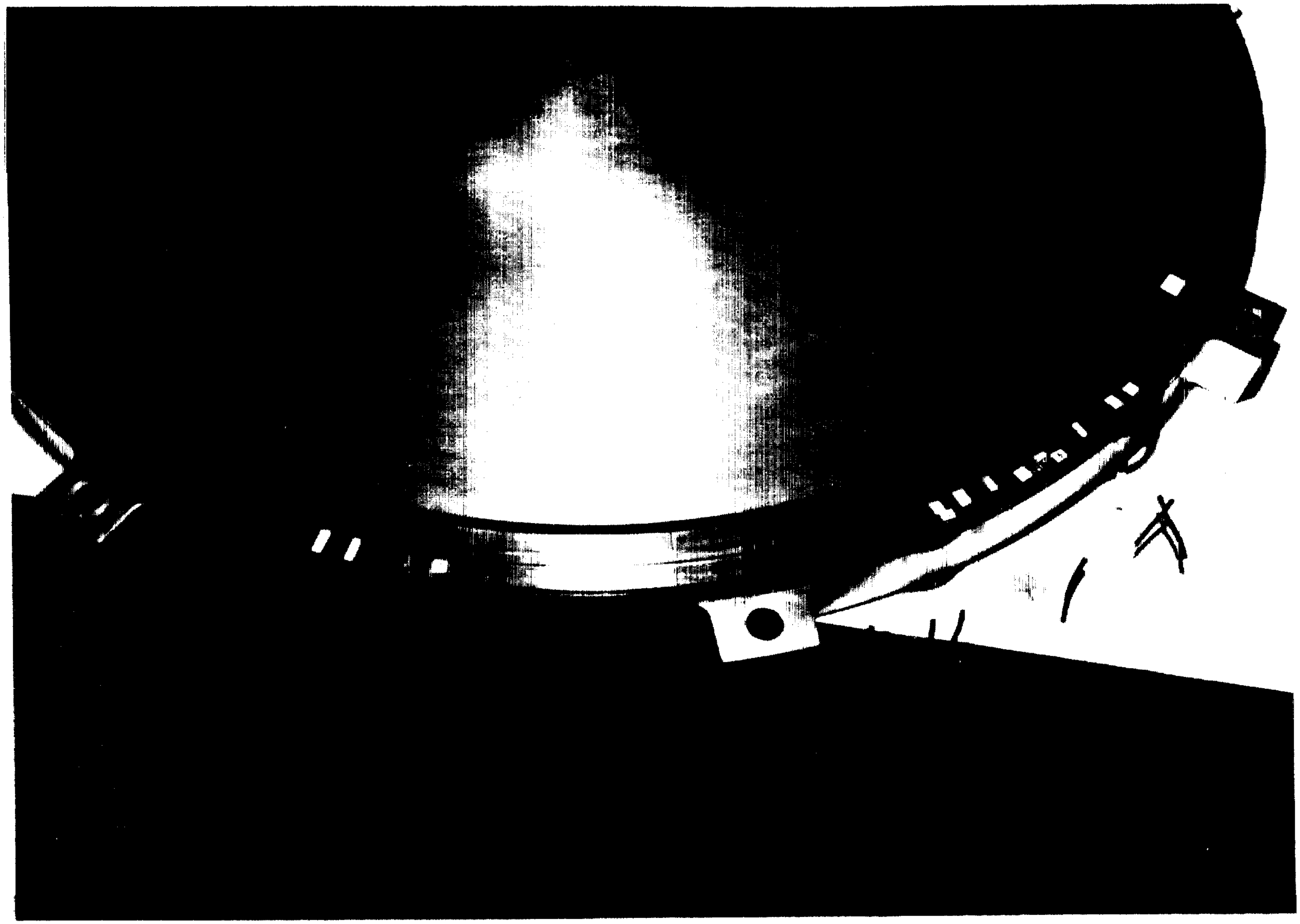

Figure 16. Strain gauge locations.

Figure 17. Strain and deflection versus load test.

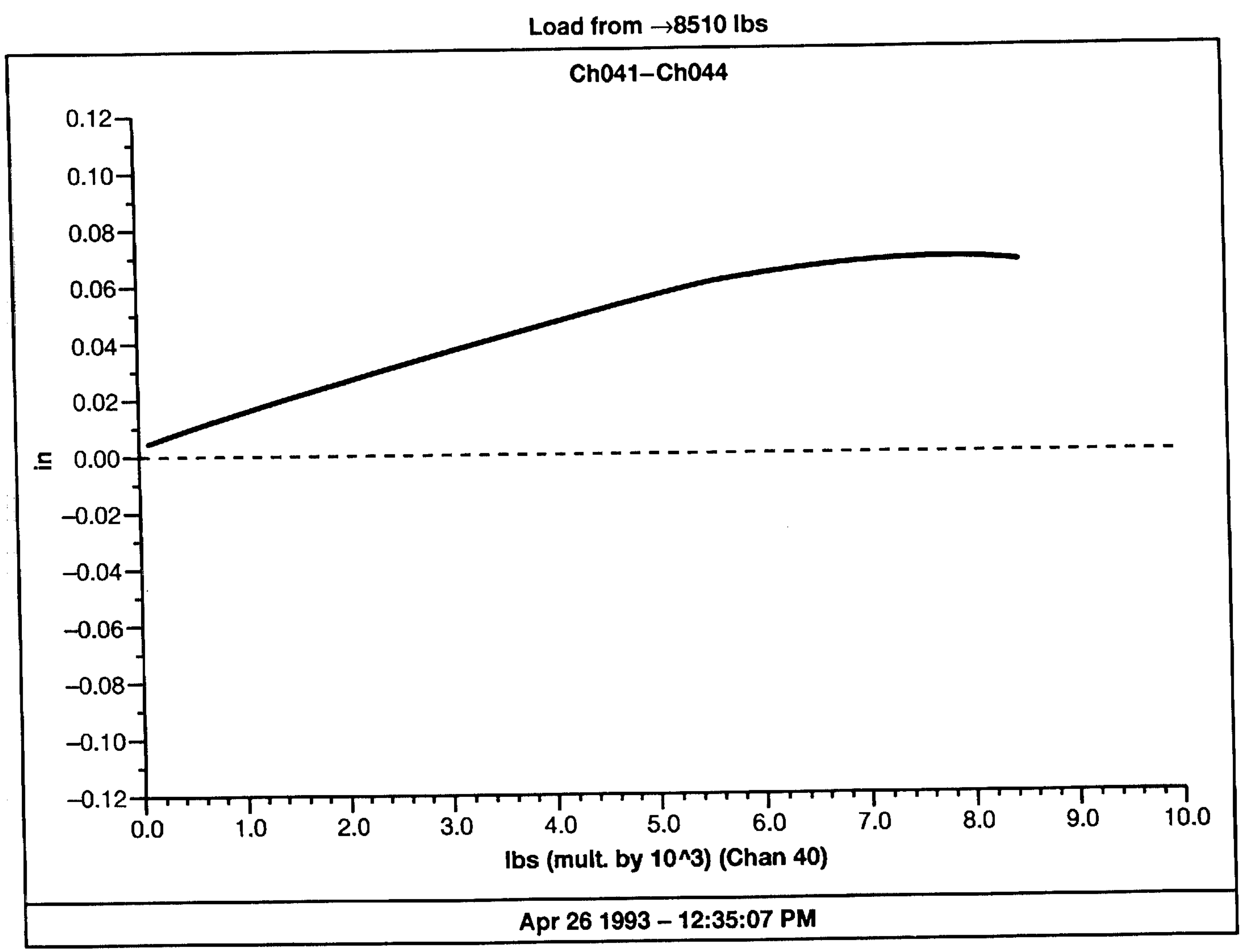

Figure 18. Deflection versus load chart.

Figure 19. DFSD cross section.

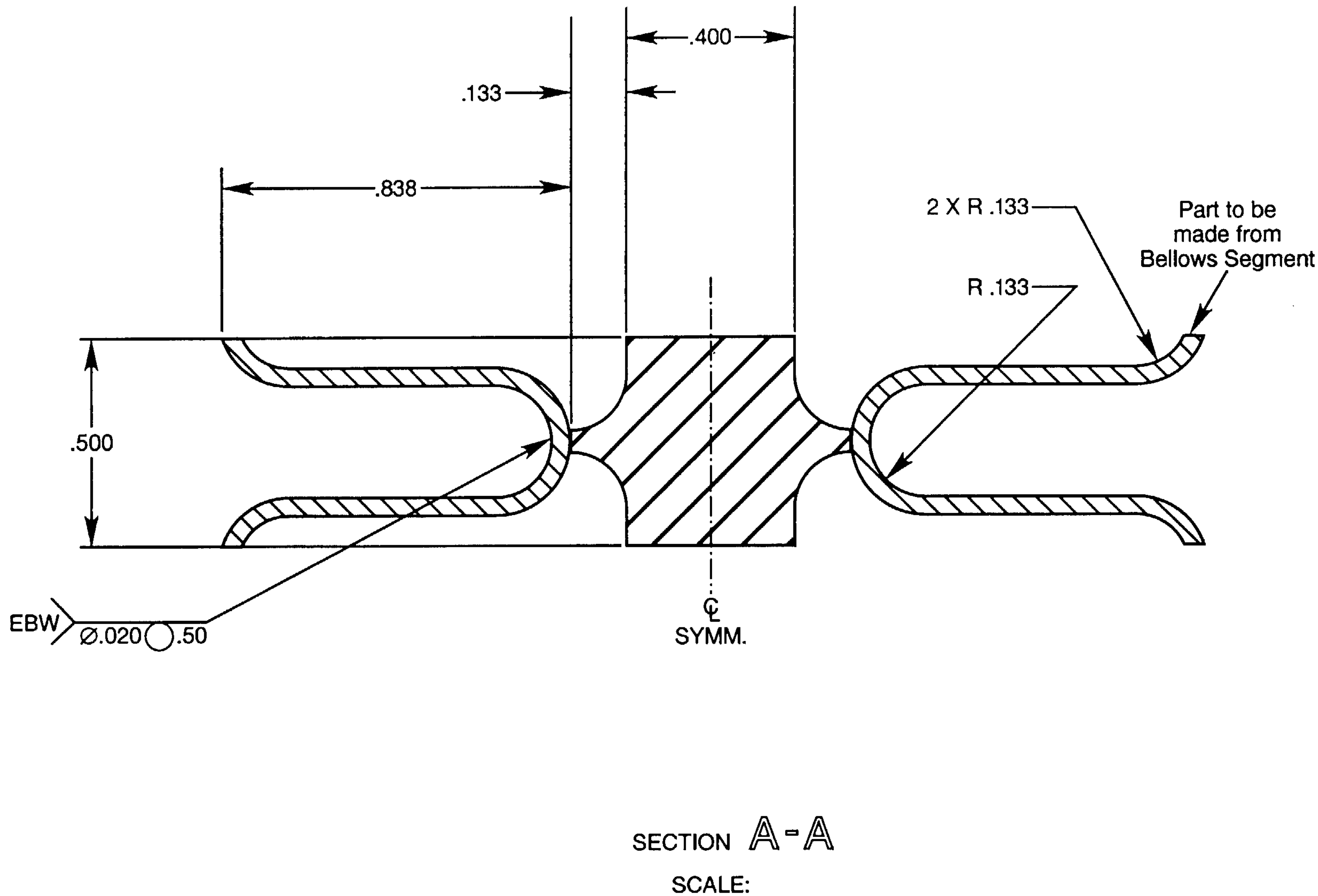

Figure 20. Bellows construction DFSD cross section.

APPROVAL

DEVELOPMENT OF A NEW SEAL FOR USE ON LARGE OPENINGS OF PRESSURIZED SPACECRAFT

By B. Weddendorf

The information in this report has been reviewed for technical content. Review of any information concerning Department of Defense or nuclear energy activities or programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

J.C. Blair

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Director, Structures and Dynamics Laboratory

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