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FEASIBILITY STUDY OF CONSTRUCTING A 1/10 SCALE APOLLO SERVICE MODULE SIMILITUDE MODEL

NASA CONTRACT NAS9-4567

PHASE I
FINAL REPORT

DECEMBER, 1965

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NATIONAL AERONAUTICS & SPACE ADMINISTRATION
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I. INTRODUCTION

The Apollo Spacecraft configuration is such that intense aerodynamic fluctuating pressures are generated around the service module at transonic Mach numbers during boost flight. An investigation is, therefore, required to experimentally describe the characteristics of this fluctuating pressure environment and the magnitude of the dynamic structural response that results.

The intent of NASA Contract NAS9-4567 is to provide a 1/10 dynamically scaled, geometrically similar, structural replica of the Apollo Service Module for use in this investigation. Construction of this model calls for advancement of the state-of-the-art in the miniaturization and production of materials and model fabrication. For this reason the project had been divided into two phases. Phase I is a feasibility study requiring the development of construction techniques for the fabrication of a true replica. Phase II calls for the construction and instrumentation of a model based on the techniques developed during the first phase.

Phase I was divided into three steps; preparation, fabrication and testing. This report describes the efforts in each of these steps and the results obtained which provide a basis for determining the feasibility of executing Phase II.

II. PREPARATION

The preparation period was utilized for familiarization with the Service Module structure, determination of manufacturing problem areas, planning methods of manufacture and evaluation of material sources.

Anticipated problem areas were: fabrication and procurement of miniature honeycomb sandwich materials such as core foil, facings and adhesives; fabrication of thin cross section structural members such as panel edgings and radial beams; and simulation of bonded, riveted and screw fastened joints.

The configurations decided upon for fabrication and test are representative samples of panel material, joint construction and radial beams. Fabrication of these specimens provided opportunity for technique development and the tests produced data for similitude evaluation.

III. FABRICATION

A. Honeycomb Core Material

Honeycomb core material as fine as that required in scaling the material used in the Service Module panels had never been produced before and presented a challenge in material rolling, bonding and handling. The major portion of honeycomb core material used in the full scale Apollo Service Module panels is of the following configuration.

Material: 5052-H39 aluminum alloy, .001 inch perforated foil
Cell Size: .25 inch hexagonal
Thickness: 1.000 inch nominal
Density : 2.3#/ft.^{3}

Briefly, this core material is produced by laminating sheets of foil pre-coated with strips of adhesive, which are properly spaced for the desired cell configuration, and curing under pressure and temperature. The laminated assembly is then cut to the desired thickness forming blocks which are colled HOBE's (honeycomb before expansion). The HOBE's are expanded to form a honeycomb cell configuration by drawing the outer laminates apart from each other, the adhesive strips effecting the node joints and the uncoated spans forming the sides of the cell. Producing aluminum honeycomb of this size is a matter of course to the honeycomb producing industry.

Scaling the Service Module honeycomb core material to a 1/10 size configuration requires:

Material:	5052-H39 aluminum alloy .0001 inch foil
Cell Size:	.025 inch hexagonal
Thickness:	.100 inch nominal
Density:	2.3#/ft.^{3}

The task of producing the material was accomplished in cooperation with Hexcel Products, Inc., Research and Development Group.

First efforts produced foil thicknesses of .00025 inch and .00013 inch in alloys 1145 and 5052-H39 respectively. With this material .070 inch cell configurations were fabricated in small sample sizes of approximately 2 square inches.

Alloy 5052-H39 foil rolled to .0001 inch gage was successfully produced overcoming the first major obstacle. This was fabricated into .025 inch hexagonal cell honeycomb of approximately 2 square inches area x .10 inch thick. The node bond adhesive used on these samples was Hexcelite 840 modified epoxy resin.

During the development of small cell honeycomb the following configurations were successfully produced in small pieces.

Config.	Cell Size	Foil Thickness	Alloy	Density
1.	.025"	.0001"	5052H39	2.3#/ft^3
2.	.025"	.00013"	5052H39	3.0#/ft^3
3.	.025"	.00025"	1145	5.7#/ft^3
4.	.070"	.0001"	5052H39	.7#/ft^3
5.	.070"	.00013"	5052H39	.9#/ft^3
6.	.070"	.00025"	1145	1.6#/ft^3

Configurations 1 and 5 are shown in photograph #1 to the right and left respectively of the centered full scale honeycomb material. A 5/8 inch wide x 6 inch long sample of configuration 3 is pictured above the reference scale in photograph #1.

Fabrication of core panels for use in the model and for test specimens was the next step. Tooling was made for producing 4 inch W x 8 inch L core panels (L designates the ribbon direction). Here the second anticipated obstacle presented itself when node joints separated prior to full expansion, creating splits in the panels.

One such panel is pictured at the top of photograph #2. A stronger node joint was obviously necessary. An adhesive and foil cleaning study was made which produced excellent results. Using an improved adhesive, 4 inch W x 8 inch L panels were successfully expanded with consistently good node bonds. One of these panels is pictured at the bottom of photograph #2. The node bond separation in these panels (not more than one or two separations per piece) is equivalent to the quality of core material produced under normal production conditions. The peel strength of the node bonds varies from 20 to 28 grams per .10 inch width as compared to 5 to 15 grams per .10 inch width, which was typical of previous node bonds.

The honeycomb core material is trimmed by cutting with a razor. Extreme care in handling must be exercised to avoid crushing the foil.

Views of the .025 inch cell, .0001 inch foil, expanded core material are shown in photograph #3 (approx. 6.5 x) and photograph #4 (approx. 27.5 x). The foil appears wrinkled to a greater degree than in full scale material. Some of this may be attributed to edge crushing caused by indiscriminate handling. Wrinkling may also be caused by stresses introduced when the foil is stretched out preparatory to node bond adhesive application.

B. SANDWICH CONSTRUCTION

The external honeycomb sandwich panels used on the full size Apollo Service Module consist of aluminum honeycomb material as described in section A, face skins of .016 inch thick 7178-T6 aluminum sheet, and Bloomingdale HT-424 film face-to-core adhesive.

Three full scale sandwich panels of dimensions 10 inches W x 80 inches L x 1.000 inch T were fabricated using the materials listed above with one exception. Because of the availability, 7075-T6 Alclad aluminum alloy was used for face skins in place of 7178-T6. The ribbon direction runs along the panel length and since standard core material is available only 30 inches long in the ribbon direction, the core was butt spliced to obtain the 80 inch length. A panel is shown in the test set up of photograph #19.

Core and facing materials were cleaned in sulfuric acid-sodium chromate solutions as described in military specification MIL-A-9067. The core, film adhesive, and facings were assembled and bonded under pressure of approximately 10 psi and a temperature of 350^{o}F. per manufacturer's specification.

Four 1/10 scale sandwich panels 1 inch W x 8 inches L. x .10 inches thick were fabricated using .025 inch cell x .0001 inch 5052H39 foil core material, .0016 inch thick 2024-T81 face skins, and spray applied Hexcelite 830 Z modified adhesive, a nylon phenolic epoxy. Three of these panels are shown in the test set-up photograph #20. Photograph #5 shows a panel with one face skin peeled back. Note the core material which has pulled away and remains imbedded in the face-to-core adhesive indicating excellent bond strength.

Alloy 2024-T81 was used in place of 7178-T6 alloy for facings. This substitution was necessary because 7178-T6 alloy cannot be produced in a heat treated state in foil gages. It can, however, be provided in either the as-rolled or annealed condition. The ultimate tensile strength of 7178 as rolled is approximately 74% of 7178-T6.

Also considered for use was 5083 non-heat treatable alloy with an as-rolled ultimate tensile strength approximately 70% of the 7178-T6 values. The yield strength however drops to approximately 55% of 7178-T6.

Sample sheets of 2024-T81 foil were obtained from Alcoa's Davenport, Iowa, Works. This material has yield and ultimate tensile values 80% of the 7178-T6 values.

The 2024-T81 was determined the most suitable substitute.

The core and facings were cleaned per military specification MIL-A-9067 as done with full scale materials with the following exception. Deviation was made in the use of the solution prescribed for cleaning facings, which proved too strong causing .0003 inch (19%) removal of stock in 15 minutes of immersion. For this reason the facings are cleaned in the weaker core cleaning solution.

The face-to-core adhesive is spray applied on the facing surface in thin coats (each allowed to dry) to a total thickness of .0015 inch to .002 inch, 1/10 the thickness of the full scale adhesive. The coated parts are then pre-cured at 230^{o}F for 10 minutes prior to assembly and final cure. This step is necessary in order to evaporate excess adhesive solvents which would otherwise be driven off during the cure cycle. The .0001 inch core foil is not perforated as is the full scale foil, therefore does not afford a ready path of escape for excessive gases evolving out of the adhesive solvent during final cure. Not taking this precaution can result in cell node bond failure (node blow) or the creation of paths of separation in the face-to-core bond line during the cure cycle.

The use of adhesive thicknesses greater than .002 inch can also produce an excess volatile content. The assembly is cured at 350°F for 2 hours under a pressure of 10 to 12 psi.

A number of small sandwich specimens were made with various adhesives in determining the one best suited. Some are listed here.

1. A glass cloth impregnated with Hexcelite 840 modified epoxy. Samples were made with 2 mil fabric. Impregnating the cloth was tedious and time consuming but processing could be improved. The bond obtained appeared adequate but was not comparable to that obtained with spray applications. Assembly of small parts is more difficult with the use of film adhesive and the danger of slippage during assembly and cure is introduced.
2. A flexible unsupported film was also used in small sandwich samples. The use of this adhesive was not pursued to any extent because of its low density and the bond did not approach that of the spray-applied adhesive. Assembly problems encountered with the cloth film would also apply here.

3. Hexcelite 840 modified was spray applied in a variety of thicknesses and pre-cure cycles to determine optimum application. The bonds developed were consistently better than bonds obtained with the other adhesives. Hexcelite 830 modified epoxy was later substituted for the 840 because of higher strength and better filleting. Two advantages realized in spraying the adhesive are that surfaces coated soon after cleaning are protected from contamination during the interval between cleaning, final assembly and cure, and the process is more expedient.

C. Panel Edging

Panel edgings on the full scale Apollo Service Module are predominantly extruded 7075-T6 Aluminum Alloy. The edgings are in the form of "Z" and "h" cross sections to which core and facing materials are bonded. Due to the limited quantities required for a 1/10 scale model, machine and chemical milling were determined the preferred method of fabrication rather than extrusion which would entail high initial tooling costs.

Straight lengths of "Z" closeouts and the L members were machined from 1/4 inch 7075-T651 Aluminum plate. Cross sectional dimensions were held .020 inch oversize to reduce machining variations due to deflection of thin sections. The curved legs of the L members were rolled to shape after machine milling.

The curved end closeouts were turned on a lathe from 7075-T6 Aluminum plate to within .010 inch of final cross sectional size. Lathe work allows machining to thinner sections than possible in milling operations.

Chemical milling operations were performed in a heated mil-etch alkaline solution followed by a clear water rinse, a neutralizing bath and a final clear water rinse. The chemical milling process is explained in detail in the report section on radial beams.

A partially assembled panel frame is shown in photograph #6. Tooling used for cutting, bending, fitting, drilling and riveting operations are shown in photograph #7 along with a number of edge components. Fit and drilling operations are performed after chemical milling.

D. Corner Joint Assembly

A representative corner joint-sandwich assembly was fabricated in both full and 1/10 scales. The joint selected consists of a square sandwich panel bordered on two adjacent sides by an "h" section closeout and a "Z" section closeout which ultimately fastens to a radial beam in the final assembly with screws and nuts.

A departure from genuine full scale materials was necessary in both assemblies due to difficulties in material procurement.

A genuine full size assembly would consist of:

COMPONENT	DESCRIPTION
Core	.001 inch perforated 5052-H39 aluminum foil .25 inch cell, 1.00 inch nominal thickness, 2.3#/ft^3 density and .003 inch perforated 5052-H39 aluminum foil, .187 inch cell, 1.00 inch nominal thickness, 8.1#/ft.^{3} density
Facings	.016 inch, 7178-T6 aluminum alloy sheet
Closeouts	7075-T6 Aluminum alloy extrusions
Angle Bracket	7075-T6 Aluminum alloy extrusions
Doublers	7075-T6 Aluminum alloy sheet
Gusset	7075-T6 Aluminum alloy sheet
Rivets	Commercial aluminum blind rivets
Adhesive	Face-to-core: Bloomingdale HT-424 film Closeout-to-core: Thixotropic epoxy

The full scale specimen fabricated consists of:

COMPONENT	DESCRIPTION
Core	Same as genuine full scale except no 8.1#/ft.^{3} density material
Facings	.016 inch, 7075-T6 Alclad aluminum alloy sheet
Closeouts	6061-T6 Aluminum Alloy bar
Angle Bracket	6061-T6 Aluminum Alloy bar
Doublers	6061-T6 Aluminum Alloy sheet
Gusset	6061-T6 Aluminum Alloy sheet
Rivets	6061-T6 Aluminum Alloy rod
Adhesive	Face-to-core: same as genuine full scale Closeout-to-core: Thixotropic mixture of 6140 resin, zonite 41 hardener and celite filler Doublers and Gusset: Epibond 1210 epoxy paste

The 1/10 scale counterpart consists of:

COMPONENT	DESCRIPTION
Core	.0001 inch 5052-H39 Aluminum foil, .025 inch cell, .097 inch thick, 2.3#/ft.^{3} density
Facings	.0016 inch 2024-T81 aluminum foil
Closeouts	7075-T6 aluminum alloy plate
Angle Bracket	7075-T6 aluminum alloy plate
Doublers	7075-T6 aluminum alloy sheet
Gusset	7075-T6 aluminum alloy sheet
Rivets	6061-T6 aluminum rod
Adhesives	Face-to-core: Hexcelite 840 modified, .002 inch thick spray applied Closeout-to-core and Doublers: Chemlok 304 epoxy paste

The full scale faying surface were immediately coated with HT-424 primer. Joints were liberally coated with thixotropic paste and the parts assembled. Bonding was done between hot platens under pressure of approximately 10 psi and at 350^{o}F for two hours. Doublers and gussets were bonded with Epibond 1210 in a secondary operation.

The 1/10 scale faying surfaces were sprayed with a .002 inch coating of Hexcelite 840 adhesive. Hexcelite 830 adhesive will be used in future applications since it fillets better and makes a stronger bond. The joints were meticulously coated with Chemlok 304 paste and the parts assembled.

Bonding was done in a small oven at 350°F for two hours under a dead weight load of approximately 5 psi. Doublers and gussets were bonded in a secondary operation. The fabricated assemblies are pictured in photographs #8 and #21. Photograph #9 shows closeouts, rivets and bracket prior to riveting.

Densified core material was not incorporated at the closeouts in either assembly. The 2.3#/ft.^{3} material is carried to the edges. Since the two specimens are for a relative comparison only, the variation is considered of slight consequence. The final model panels will include densified edge core material.

E. Screw Fasteners

Fasteners used for attaching panel sectors to radial beams on the full scale Service Module are 1/4 inch diameter - 20 threads per inch screws (NAS 1134C9). Flanged nuts (MF-1031-4) are anchored to the beam flanges with rivets.

The 1/10 scale model fasteners used were .60 mm (.0236 inches) diameter - 169 threads per inch. The nuts were cemented onto the beam flanges for anchoring. The screws were obtained from watch makers stock. The maximum stock length obtained (.075 inches) was just sufficiently adequate to engage 2-1/2 threads in the nuts. This is the calculated minimum engagement for full screw strength.

Several screws were tensile tested and the ultimate strength found to be 105,000 psi. The average torquing limit was determined to be 3 inch ounces.

Nuts were made from .030 inch thick steel sheet, drilled, tapped and cut into .050 inch squares.

Suppliers have been contacted and can produce screws and nuts in quantity to required specifications.

F. Radial Beams

Two 1/10 scale radial beams have been fabricated in a manner similar to that in which full scale beams are processed. The procedure is a milling machine process followed by a chemical milling process. The first of the two beams has been manufactured to completion while the second stands ready for chemical milling.

1. Machining.

Difficulties encountered in machining the first beam were rectified in the second beam machining with refined processing techniques. A description of the procedure follows.

The beam material, 1/4 inch thick x 7 inches wide x 19 inches long, 7075-T651 aluminum plate, was Blanchard ground to .224 inch thick. This was doweled and fastened with twenty-four ¼ inch cap screws about its perimeter to a heatable aluminum platen mounted on a vertical milling machine table. The dowel pins were referenced and the first side machined using stub end mills and keyway cutters. In the machining operation, .012 inch of excess material was left on each surface for a total of .024 inch additional stock to be removed later by chemical milling. The shear webs,which finish to

.0018 inch were left a nominal .026 inch thick. Thinner cross sections were found to increase the tendency to "oil can" during the machining process.

The piece was then sprayed with a coat of shellac and heated to approximately 135°F in preparation for receiving a stiffening material named Rigidax, Type WS. This material is necessary to further prevent web "oil canning" and deflection of other thin sections while machining the second side. The use of shellac and heat were found necessary to obtain intimate contact and adhesion between the aluminum and Rigidax for maximum rigidity. Molten Rigidax is poured into the machined recesses of the beam. The assembly is allowed to cool, the Rigidax solidifying into a rigid block, forming the necessary support to the thin sections. The excess Rigidax is machined flush and the piece turned over for machining the second side. The dowel pins are again referenced for registration with the first side. The second side was machined and the peripheral supporting frame cut away. See photographs #10 and #11. The Rigidax, which is soluble under hot tap water, was removed from the first side of the beam. The shellac was cleaned off and the part de-burred. The beam was then ready for inspection and preparation for chemical milling.

An improvement to be yet incorporated in the machining process is the addition of an adhesive between the platen and plate stock while machining each side. This adhesive will assist in holding the plate flat during stock removal. The platen will be heated to soften the adhesive for part release.

The first beam was machined without benefit of the heated platen. The piece, along with the aluminum plate, was removed from the milling machine table and oven heated. The part, having been shellacked, was then filled with Rigidax and replaced on the milling machine table. This procedure, as well as being awkward and time consuming, introduced distortions and the possibility of alignment error.

The stock used on the first beam was not Blanchard ground and required a skim cut to obtain desired thickness. The use of ground stock eliminated bowing caused by this cut.

The web thickness of the first beam varied from flanges to center by .002 to .005 inch. Since final web thickness is .0018 inch, the variation had to be corrected prior to chemical milling. This was done by

tedious hand scraping and sanding until a reasonably uniform thickness was obtained. The second beam proved considerably more uniform with web thicknesses varying no more than .001 to .002 inch thus reducing the amount of hand working required. This is a consequence of the improved machining techniques which were incorporated.

Both beams, upon removal from the machining fixture, were bowed in two directions. Across the 5.5 inch width the bow was approximately .010 inch and along the 15 inch length, approximately .030 inch. This is a result of machining stresses which were later automatically relieved in chemical milling, leaving the part exceptionally straight.

2. Chemical Milling

A chemical milling operation was set up to produce thin aluminum cross-sections in the radial beams and other parts which would be virtually impossible to obtain by conventional machining methods. The setup consisted of a heated alkaline etch solution, rinse water and deoxidizer bath. The alkaline solution used was Wyandotte mil-etch mixed 1 lb. per gallon of water and heated to 180^{o}F. for an etch rate of approximately .0004 inch per minute per surface. The deoxidizer solution was 1 lb. of Wyandotte 2487 per gallon of water.

Inspection of the beam prior to chemical milling and recording the dimensions is essential for stock removal control. With this information, a chemical milling schedule was established.

The entire beam was covered with three spray coats of Miccroshield stop-off lacquer. Thorough drying of each coat is necessary. Beam sections found in inspection to be over-thick were then stripped of stop-off with a reducing solvent and etched to a predetermined "stage" dimension established in the chemical milling schedule. As each "stage" dimension was reached, other areas were stripped and etched until all cross sections were over-size the prescribed amount. The beam, totally stripped of maskant, was then chemical milled until a shear panel thickness of .006 to .008 inch was reached. The panels were carefully wet sanded by hand removing the peaks from the surface of each panel, approximately .0007 inch per surface. This operation reduces the risk of etching pin holes through the panel during the last mil-etch step to .0018 inch thick.

A maximum etch period of 5 minutes at any one time had been determined. Longer periods produce increased smut build up which causes inefficient and uneven stock removal and becomes difficult to remove in the deoxidizer solution. Both alkaline and deoxidizer dips are always followed by a clear water rinse.

Pin holes did appear in four beam stringers, all on the same side, which are attributed to inadequate maskant protection on these difficult to reach and observe regions. Improved application methods and precautionary inspections will be taken to prevent this from occurring on future parts.

The completed beam is pictured in photograph #12. Note the grain running in the direction in which the plate was rolled. Chemical milling of heat-treatable alloys produces a more pronounced texture than non-heat treatable aluminum alloys. Allowing the piece to stand in the de-oxidizer bath, thereby etching some of the alloying elements, aids in reducing these "high spots". Also noted, end grains etch slightly faster and become pronounced.

IV. TESTING

A. Honeycomb Core Material

All testing of core specimens was done on a Tinius Olsen Electromatic universal testing machine. The deflections were measured by measuring the head travel with a Tinius Olsen Deflectometer. Special test fixtures had to be made for the flexure test, the plate shear test, and the edge wise compression test. These fixtures with the various test specimens are shown in photograph #13.

The testing procedures were based on the applicable ASTM specification but some changes had to be made because of the small size of the samples.

1. Flexure Test

This testing procedure was based on ASTM Specification C 393-62. A special test fixture was fabricated with a span length of two inches and is shown in photograph #14.

The core shear stress was calculated with the following equation:

$$S = \frac{P}{(h+c)\,b}$$

Where S = Core shear stress (psi)
P = Load (pounds)
h = Sandwich thickness (inches)
c = Core thickness (inches)
b = Sandwich width (inches)

Core shear modulus could not be calculated from this test because the flexural stiffness of the sandwich was too low. Facings thicker than the .0016" facings used on these sandwiches would be required to increase the

flexural stiffness; it is estimated that a facing thickness of .006" to .008" is necessary.

The core shear modulus is given by the formula:

$$G = \frac{P_S \, a_S \, c}{2W_S \, (h+c)^2 b \left(1 - \frac{11 P_S a_S^3}{768 w_S D b}\right)}$$

Where G = Core shear modulus

P_S = Total load, applied at 2 points located a distance $a_S/4$ from each reaction

w_S = Midspan deflection P_S/w_S - slope of load deflection curve

a_S = Span length

c = Core thickness

h = Sandwich thickness

b = Sandwich width

D = Flexural stiffness of the sandwich

$$D = \frac{E \, (h^3 - c^3)}{12 \, L}$$

E = Modulus of elasticity of the facings

L = $1 - v^2$, v is Poisson's ratio of the facings ($1 - v^2 = .89$ for aluminum)

For valid results the expression $\frac{11 P_S a_S^3}{768 w_S D b}$ has to be between .4 and .6. For the samples tested this expression varied from .9 to greater than 1. As can be seen from the formula this value can be decreased by increasing the value of D. This can be easily accomplished by using thicker facings.

The results of this test are shown in Table I.

2. Plate Shear Test

The procedures for this test were based on ASTM Specification C273-61. Special plates were machined for this test and are shown in photograph #15. The specimen size was 2.0" x 1.0" and both 'L' direction and 'W' direction tests were run. A measurement of the head travel was used for the deflection because of the small angle (3^{o}) of the plates.

Shear stress was calculated by using the following formula:

$$S = \frac{P}{Lb}$$

Where S = Core shear stress (psi)
P = Load (pounds)
L = Core length (inches)
b = Core width (inches)

The shear modulus was calculated using:

$$G_c = \frac{P}{d}\,\frac{t}{A}$$

Where G_c = Core shear modulus (psi)
P = Load (pounds)
d = Deflection (inches)
t = Core thickness
A = Core area

The results of this test are listed in Table 2.

3. Flatwise Compressive Test

For this test the specimen size was 1.0" x .1". No modulus was measured but the specimens were tested in both core and sandwich form.

The results of this test are recorded in Table 3.

4. Edgewise Compressive Test

The procedures for this test were based on ASTM Specification C364-61. Special fixtures were made for this test and are shown in photograph #16. The sample size was 1" wide x 1" to 1 1/2" long; both 'L' direction and 'W' direction tests were run.

Failure always occurred in the ends of the sample by a crushing type failure. To overcome this the ends of two samples were filled with epoxy adhesive. These two samples failed at a higher load but still failed by a crushing of the ends. Only the load was measured; the strain in the facings was not measured.

Results of this test are in Table 4.

5. Peel Strength

The core to facing peel strength was measured as shown in photograph #17. This is not the usual test procedure followed but was necessary because of the very thin sandwich facings (.0016").

It was found that the peel strength was dependent on the adhesive thickness and increased with increasing adhesive thickness.

The type of failure also varied with the core to facing adhesive thickness. With the thinnest adhesive application the failure occurred primarily in the adhesive. With the thickest adhesive application the failure was nearly 100% core failure. This value with the thickest adhesive is probably optimum for this type core.

The results of this test are in Table 5.

The test results of the strength properties of honeycomb that are usually measured are plotted on graphs (figures 1 - 3) showing the properties of regular production honeycomb as reported in Hexcel TSB 120.

As can be seen from these graphs, the compressive strength and the 'L' direction shear strength of the .025" - 5052 - .0001" honeycomb are slightly lower than the minimum properties of the Hexcel 5052 production honeycomb. The 'W' direction shear strength is between the typical and the minimum value of the production honeycomb.

The shear modulus values calculated from the plate shear tests were smaller by a factor of approximately 7 to 10 times what is reported as minimum values of production honeycomb for the same density as shown below:

	Average Measured Value	Minimum Production Value
L Direction	1600 psi	17,000 psi
W Direction	1300 psi	8,500 psi

Because of the limited number of tests conducted, no satisfactory explanation can be given for these anomalous results.

B. Honeycomb Sandwich Panels

1. Specimens

The full scale specimens were nominally 80 inches long, 10 inches wide and one inch thick. The hexagonal honeycomb core was of 0.001 inch 5052 H39 aluminum foil with a cell size of 0.250 inches. The facing was of 7075T6 aluminum and was 0.016 inch thick. Facing and core were adhesive bonded.

The tenth scale specimens were nominally 8 inches long, one inch wide and 0.1 inch thick. The hexagonal honeycomb core was of 0.0001 inch 5052 H39 aluminum foil with a cell spacing of 0.025 inch. The facing was of 2024T81 aluminum and was 0.0016 inch thick. Facing and core were adhesive bonded.

After the first tests on the modeled panels, a thin coating was applied in order to increase their weight slightly. Data are reported for both cases.

One of the four one-tenth scale specimens (No. 27) was defective, having a sharp wrinkle in the facing. Excessive shearing motion at the wrinkle was evident in the first vibration tests. Thus, very few meaningful data from this specimen were obtained.

2. Density

Length and width of the full scale beams and length of the modeled beams were measured with a scale. Width of the modeled panels and

thicknesses of both sizes were measured with a micrometer. Three readings of width and thickness were taken and averaged. Panel weights were measured on beam balances. Weight of the full scale panels is accurate to at least $\pm$ 5 grams. Weight of the modeled panels is accurate to at least $\pm$ 5 milligrams. Density was computed and noted in units of pounds per square foot.

Table 6 gives the density data. It is seen that the modeled beams are somewhat less dense than the target value. The ratio of model to full scale density is .0815 before coating and .0921 after coating, as compared to a target value of 0.10. The anticipated density of the full scale material was 0.88 pounds per square foot. Comparing the coated material with this value, the ratio is.0 966 for an average error of 3.4%.

3. Stiffness

One beam of each size was clamped and loaded as a cantilever beam, photograph #18. Clamp engagement was 12.4 inches and 1.24 inches, respectively, for the two sizes. Dead weight loads were applied 4 inches and 0.4 inches, respectively from the free ends. End deflections were measured by a dial indicator in the case of the full scale beam and by a filar telescope in the case of the modeled beam. Data are given in Table 7 and plotted in Figures 4 and 5. The modeled beam was in the coated condition. Stiffness of the modeled beam after the application of two Type FAP-06 strain gage and connector was also measured. No significant change was noted, as indicated in Table 7 .

It can be seen that some hysteresis is present. Neglecting this and computing the slopes of the increasing load curves, we obtain stiffnesses of 7.87 lb. per inch and 0.773 lb. per inch for the full scale and modeled sizes, respectively. These give a model to full scale ratio of .0983 compared with the target value of 0.10, for an error of 1.7%.

4. Vibratory Response

The honeycomb beams were vibrated in free-free bending modes in order to determine their frequency response and damping characteristics. Beams were supported at two nodes for each mode of vibration, (photograph #19 and #20). Nodal supports were of soft sponge rubber on wooden bases. Each support was the width of the specimen and of about one-quarter inch extent in the longitudinal direction of the specimen. Acoustic excitation was provided by an Atlas Model PD-8 25 watt exciter driven by a Hewlitt-Packard Model 205A sine-wave oscillator. Frequency indication was by the oscillator control dial and by a Berkely Model 5500 digital counter. Specimen vibration was detected by a Lion Research Model PX321 capacitive proximity probe with Model 201 driver. Probe output was observed on a DuMont 324 cathode ray oscilloscope. Probe output for measurement of damping was amplified by a Quan-Tech Model 206 amplifier and fed to a Type M1650 galvanometer in a Honeywell Model 906C "Visicorder" recording oscillograph. Frequency response of the galvanometer is essentially flat up to 1650 Hz. Linearity of the capacitive probe is a

function of specimen distance. This distance was kept large enough so that no significant nonlinearity was encountered.

In performing a test, the oscillator frequency was gradually increased until a resonance was encountered. Nodal supports were adjusted to optimum location. Mode shapes were determined by palpation and by observation under stroboscopic light. Frequency was adjusted to obtain peak amplitude of the oscilloscope indication. The frequency was then observed. If a recording was to be made, the oscillograph was turned on at an appropriate paper speed and the acoustic excitation removed, thus permitting recording of the decay of vibration. In some cases of fundamental mode vibration, the beams were simply struck and the resulting vibration and decay recorded. The oscillograph records were measured and the number of cycles to half-amplitude, n, noted. Damping factor, g, was then computed as: $g = \frac{0.22}{n}$

Results of the vibratory response tests on honeycomb beams are given in Table 8. This table also includes frequencies of one beam with two Type FAP-06 strain gages and connectors cemented to the midlength and of one beam with a 4.3 gram accelerometer cemented to the midlength.

Looking first at the effect of the coating, it is seen that, as desired, the frequencies were lowered by the coating. The effect was greatest

at the lowest frequencies, averaging 4.7% reduction at the first mode and 3.1% at the fourth mode. Extremes of frequency reduction were 5.5% maximum and 2.8% minimum. Damping was consistently increased by the coating, the amount ranging from 25% to 57% and averaging 38%.

The target ratio of frequency of the modelled beams to that of the full scale beams was 10.0. Analysis of the data in Table 8 reveals that average values for each mode came very close to this target (using the coated condition). The first and third mode averages differ by 0.25%, the second mode by 0.37% and the fourth mode by 0.41%. Maximum deviation for the entire group taking the lowest full scale frequency against the highest model frequency for each mode, and vice versa, are 3.54%, 3.76%, 3.53% and 4.1%, respectively, for the first four modes.

Looking now at the damping data in Table 8, the most striking observation is that the modelled beams exhibit about ten times as much damping as the full scale beams. It is to be expected that structural damping should scale one-to-one. Crude qualitative consideration of the acoustic damping, however, offers a likely explanation. Damping force from the movement of air is proportional to velocity-squared and thus to frequency squared. This would give a factor of 100. Further, since the mass per unit surface area of the modelled beams is lower by

a factor of 10, the total acoustic force per unit mass could be up by a factor of 1000. This value is probably not achieved, since displacement amplitude is lower in the smaller beams. Nevertheless, it is appears that acoustic damping is greater in the smaller beams and possibly accounts for the larger overall damping. It should also be noted that acoustic damping is a function of amplitude. Since no effort to control amplitude was made, the actual degree of acoustic damping cannot be determined.

C. Corner Joint Assemblies

1. Specimens

The full scale corner joint assembly consisted essentially of a section of honeycomb panel 22 inches square with an "h" beam along one edge and a "Z" beam along an adjacent edge, the two beams being attached at the common corner by a riveted structure. The free flange of the "Z" beam (the closeout) contained ten holes for attachment to a drilled support. Attachment was made via bolts and nuts.

The one-tenth scale model corner joint assembly was identical to the full scale except for size. Honeycomb panel of the same construction as in the honeycomb beams was used. Miniature screws (0.6 mm watch screws) and nuts were used for closeout fastening.

2. Weight

Weights of the various parts and comparison of model to full scale are given in Table 9 .

Since the target ratio of weights is 0.001, it is seen that the model is 4.1% light. The panel structure alone is 4.9% light while the support and screws are 3.0% light.

3. Stiffness

Corner joint assemblies were loaded in cantilever bending while supported in two different manners, (photograph #21). In one case, the closeout was firmly clamped. A machinist vise was used for this on the tenth-scale model. In the case of the full scale assembly, a heavy aluminum clamping assembly was attached to the closeout and dogged down to a bed plate. As a second clamping condition, the closeout was screwed to the support beam using full scale or miniature screws, respectively. The support beam was then clamped. Load was applied through a sling 1.5 inches from the free end (full scale) or 0.15 inch from the free end (model). Load-deflection data are given in Table10 and plotted in Figures 6, 7, and 8.

Some hysteresis is apparent, particularly in the case of the modelled assembly. Also, because of a limitation in the number of suspending-type gram weights available, two runs were made on the modelled joint; the first to delineate the lower end of the curve and the second to delineate the upper end. The fallacy of this procedure became obvious in the case of the screwed joint test, wherein the addition of 500 grams in one step overloaded the specimen to failure. Failure was confined to the screws and nuts, appearing to be principally

stripping of the nuts. Evidence indicates that the 0.6 mm screws were slightly short and did not fully engage the miniature nuts. No other damage occurred.

In the case of the modelled structure, deflection was measured by a filar telescope with a resolution of only about 0.001 inch. This accounts for some irregularity in the curves for this case (Figure 7).

Taking the initial increasing-load portions of each curve (using the average in case of Figure 7)bending stiffnesses are computed as indicated below:

	Full Scale	One-Tenth Scale
Closeout Clamped	417 lb/in	29.2 lb/in
Closeout Screwed to Support	314 lb/in	18.0 lb/in

The target ratio of model to full scale stiffness should be 0.1. Actual ratios are 0.07 and 0.057, for errors of 30% and 43%, respectively. Thus, the model has much less stiffness than anticipated. Since the honeycomb structural stiffness scales within 2% (noted in a previous section), the deviation must lie in the other parts or in the assembly. Detailed examination of the scaled structure might reveal the cause. It should be noted that, in the case of the screwed attachment, part of the increased flexibility may have resulted from inadequate engagement of the screws.

4. Vibratory Response

The corner joint assemblies were surveyed for frequency response

and damping (of the lower modes) in both the clamped and the screwed-support conditions. Excitation and measurement instrumentation were the same as that described in Section B. In addition, mode shapes were observed by the movement of salt sprinkled on the specimens.

The principal frequencies observed were at the first and second bending modes and the first torsional mode. The first torsional nodal line was along the centerline of the scaled model but was displaced somewhat to one side in the full scale. A number of higher responses were measured, mostly on the full scale specimen. Most of these were of complex shape difficult to determine, although the "second torsional" was clear. This was like the first torsional except that two nodal lines extended approximately perpendicular to the supported edge. It might more appropriately be called a second lateral bending mode.

Results of these tests are given in Table 11. Comparing the frequency ratios with the target value of 0.1, it is seen that the one-tenth scale assembly ran relatively low in all cases. Frequencies were 7.8% to 12.1% low with the closeout clamped and 7% to 18.1% low with the closeout screwed to the support. This is consistent with the low stiffness measured from the load-deflection tests. As in those tests, part of this may be attributed to inadequate screws and nuts in the

case where the closeout was screwed to the support. This shows up predominantly in the first bending mode. Low stiffness at the joint would not be as influential on the higher modes, as borne out by the data.

As before, damping in the one-tenth scale model runs about an order of magnitude higher than in the full scale assembly.

D. Radial Beam

The radial beams are unit structures located in the longitudinal-radial plane of the service module and intended to support lateral shearing loads. At one-tenth scale each beam is approximately 16 inches long, 5.5 inches wide, and 0.2 inch maximum thickness, having eight webs of about 0.0018 inch thickness.

Weight and transverse stiffness of a one-tenth scale model beam were determined. Since a full-scale beam was not available, no comparison could be made.

Weight of the beam was 42 grams.

The beam was placed in a fixture which supported it in a vertical plane with the long dimension horizontal (photograph #22). Bottom support was provided by two rollers at the bottom edge of the beam, spaced 13.95 inches apart. Load was applied through a carriage placed at the center of the upper edge of the beam. The carriage contacted the upper edge through two rollers 3.7 inches apart.

Because of the geometry of the beam, the loading was inherently unstable. Maximum load was thus limited in order to prevent excessive

lateral movement at the center of the upper edge of the beam.

Deflection was measured by a 0.0001 inch division dial indicator in contact with the midpoint of the lower edge of the beam.

Test results are given in Table 12 and plotted in Figure 9.

Computation of stiffness from the line drawn through the data points yields a value of 8300 lb per inch for this loading condition. Visible and audible indications of buckling of the shear webs were noted during the loading.

V. SUMMARY

A. Honeycomb Core Material

Fabrication of honeycomb core material with .0001 inch 5052-H39 foil and .025 inch cell size has been successfully accomplished. This is possible due to the ability to produce .0001 inch 5052-H39 foil. The honeycomb material produced was approximately 10% low in density. Core shear in the "L" or ribbon direction, and compressive strengths were found to be slightly lower than the minimum for production core material. Core shear in the "W" direction was between minimum and typical values for production core material. Shear modulus was as much as ten times smaller than the minimum value for production honeycomb.

Core panels as large as 4 inches wide x 8 inches long x .10 inch thick have been attained in the configuration described above. Panel sizes 8 inches wide x 8 inches long can be fabricated for use in the construction of model panels.

Core material of the same cell size but of higher density (5.7#/ft.^{3}) has been fabricated using heavier foil (.0003 inch). Other cell configuration and density can be produced with the proper tooling.

B. Honeycomb Sandwich

Sandwich assemblies in both full scale and 1/10 scale were fabricated and tested for comparison. Alloy 2024-T81 .0016 inch thick was substituted for 7178-T6 facing material on the 1/10 scale specimens. A process for the heat treatment of 7178 alloy foil has never been developed, therefore a substitution was necessary. Tensile properties of 2024-T81 are 80% of 7178-T6. Alloy 7075-T6 was used for facings on the full scale specimens. This has tensile properties 84% of 7178-T6.

Core to face bonding was effected with a spray applied nylon phenolic epoxy adhesive. This application was practical and produced bonds of excellent strength.

The density of the 1/10 scale panels was low by approximately 18%. This is attributed to the low core density and adhesive weight. Panels were coated externally with additional adhesive which brought the panel density up to within 3.4% of desired weight.

One tenth scale panels exhibited bending stiffness 1.7% less than the scaled value of full scale panels.

Scaled values of free-free bending mode frequencies of the scaled and full size panels were very close, based on average values from the several specimens. The average

frequencies scaled within .5% for the first four modes. Taking extreme values from individual panels, the worst case represented a difference of 4.1%.

Damping factors were somewhat variable but averaged about an order of magnitude higher for the one-tenth scale panels than for the full scale panels. This is tentatively attributed to two factors. Acoustic damping and/or the energy absorbing characteristic of the face-to-core adhesive. The effect on damping caused by coating the specimens with additional adhesive seems to indicate the adhesive as being the major factor.

A series of vibratory response tests on flat aluminum bars has been initiated to determine the effect of a variety of adhesive applications on damping.

The presence of two strain gages at the midpoint of a one-tenth scale panel had negligible effect on the stiffness. These caused a reduction in resonant frequency of 1.5% to 3.7%, depending on mode of vibration.

The presence of a relatively heavy accelerometer at the midpoint of a scaled specimen had a major effect on frequency, causing a reduction of 6.9% to 27.6%, depending on the mode of vibration. No visual local effects of accelerometer presence could be detected.

C. Corner Joint Assembly

A corner joint assembly in both full scale and 1/10 scale were fabricated and tested for comparison.

Deviation in materials used for fabrication of the sandwich assemblies were also necessary in these assemblies.

Weight of the one-tenth scale specimen was 4.9% lighter than the full scale specimen.

The ratio of bending stiffness of model to full scale with the closeout clamped was .07 compared with the target value of .10. With the closeout attached to the support with screws, this ratio was .057. Thus the modeled structure is considerably less stiff than the scaled value of the full size structure. A close examination of the assembly indicates the need for additional application of adhesive in the sandwich-to-closeout joint. This would unquestionably improve the stiffness. Vibration frequencies of the model were about 8% to 12% lower than scaled full size values with the closeout clamped. The values were 7% to 18% low with the screwed attachment. Besides the insufficient joint bond described in the previous paragraph, screw engagement, being less than the desired value, allowed additional flexibility. Adequate miniature screws will be obtained to satisfy the modeling needs.

D. Radial Beams

Fabrication of a 1/10 scale radial beam by machine and chemical milling processes, similar to full scale beam processing, was highly successful.

The beam weighed 42 grams and had a stiffness, under loading conditions described, of 8300 pounds per inch.

No full scale radial beam was available for comparison.

VI. CONCLUSION

Overall fabrication accomplishments during phase I indicate that a 1/10 scale replica of the Apollo Service Module can be produced.

Test data indicates that similitude in such a model can also be attained. One questionable area which warrants further investigation lies in the damping properties exhibited by the test specimens.

A preliminary investigation into the affects of a number of adhesive applications on the damping of flat aluminum bars is currently underway. This test data is not complete for inclusion in this report but will be provided in the form of an addendum as soon as available.

TABLE 1

Core Shear Strength Flexure Test

Direction	Load (lbs)	Thickness (total)	Density (lbs/cu.ft)	Shear Strength (psi)
L	14.7	.106	2.1	70.4
L	15.5	.107	2.1	73.8
L	13.1	.100	2.2	66.5
L	13.6	.101	2.1	68.4
L	10.5	.098	1.9	54.5
L	10.7	.098	1.9	55.5

TABLE 2

Shear Strength Plate Shear Test

Direction	Load (lbs)	Area	Thickness	Density	Shear Strength	Modulus
L	125	2.0"	.096"	1.9	62.5	1728
L	118	2.0"	.096"	1.9	59.0	1470
W	120	2.0"	.096"	1.9	60.0	1290
W	112.5	2.0"	.096"	1.9	56.3	1320

TABLE 3

Flatwise Compressive Strength

Type	Core Density (lbs/cu.ft.)	Compressive Strength (psi)
Sandwich	2.1	82.5
Sandwich	2.1	75.0
Sandwich	2.2	85.0
Bare	1.9	71.5
Bare	1.9	75.0

TABLE 4

Edgewise Compressive Strength

Type	Direction	Size	Load (lbs)
Bare	L	1"xl 1/4"x.1"	37
Bare	L	1"xl 1/2"x.1"	36
Bare	L	1"x 1" x .1"	34
Stabilized	L	1"x 1" x .1"	42
Stabilized	W	1"x 1" x .1"	45

TABLE 5

Core to Facing Peel Strength

Width	Adhesive Thickness	Load
1.0"	.0014" to .0016"	1.2 lbs.
1.0"	.0016" to .0020"	1.7 lbs.
1.0"	.0020" to .0023"	2.5 lbs.

TABLE 6

DENSITY OF HONEYCOMB Panels

Beam	Scale	Length (in.)	Width (in.)	Thickness (in.)	Weight (grams)	Density (lb./ft.^{2})
1	Full	80.0	10.0	1.003	2265	0.897
2	Full	80.0	10.0	1.005	2291	0.908
3	Full	80.1	10.0	1.009	2279	0.904
					Average	0.903
27	0.1	8.04	1.027	.0987	1.930	0.0740
29	0.1	8.00	1.017	.0988	1.960	0.0764
31	0.1	8.04	1.013	.1000	1.780	0.0693
32	0.1	8.00	1.037	.1048	1.950	0.0746
					Average	0.0736
29(1)	0.1			.1009(2)	2.200	0.0840
31(1)	0.1			.0994(2)	2.050	0.0800
32(1)	0.1			.1040(2)	2.200	0.0855
					Average	0.0832
29(3)					2.265	
31(4)					6.430	

(1) After coating (No. 27 not coated).

(2) Variation in beam thickness apparently too great to allow determination of coating thickness by this method.

(3) With two strain gages.

(4) With accelerometer.

TABLE 7

LOAD-DEFLECTION DATA - HONEYCOMB Panels

Full Scale Beam No. 1		One-Tenth Scale Beam No.		
Load (lb.)	Deflection (in.)	Load (grams)	Deflection (in.)	
			Coated	2 Strain Gages Added
0	0			
0.5	.065	0	0	0
1.0	.128	10	.029	.027
1.5	.190	20	.057	.056
2.0	.252	30	.086	.085
2.5	.316	40	.115	.115
3.0	.383	50	.144	.144
3.5	.447	70	.201	.203
4.0	.512	100	.290	.290
3.5	.449	0	.001	.002
3.0	.391			
2.5	.328			
2.0	.266			
1.5	.201			
1.0	.138			
0.5	.073			
0	.009			

TABLE 8

VIBRATORY RESPONSE - HONEYCOMB Panels

No.	Scale	Frequency (Hz) Mode 1	Mode 2	Mode 3	Mode 4	Damping Factor, g Mode 1	Mode 2
1	Full	39.5	109	212	342	.0018	.0012
2	Full	39.6	108	212	343	.0022	.0015
3	Full	39.8	109	212	343	.0015	.0018
27(1)	0.1	327	980	2130	2980	———	———
29	0.1	403	1091	2114	3379	.020	.008
31	0.1	411	1141	2208	3540	.018	.007
32	0.1	428	1164	2260	3648	.014	.008
29(2)	0.1	387	1049	2055	3290	.029	.010
31(2)	0.1	388	1080	2125	3430	.026	.011
32(2)	0.1	409	1120	2195	3520	.017	.011
29(3)	0.1	379	1035	1980	3240		
31(4)	0.1	281	975	1625	3200		

(1) Defective Specimen. Data not used in analysis.
(2) After coating.
(3) With two strain gages at midpoint.
(4) With accelerometer at midpoint.

TABLE 9

WEIGHTS OF CORNER JOINTS

Item	Full Scale	One-Tenth Scale	Ratio Model/Full Scale
Panel Structure	2577 grams	2.450 grams	.000951
Support and Screws	1835 grams	1.780 grams	.000970
Total	4412 grams	4.230 grams	.000959

TABLE 10

LOAD-DEFLECTION DATA - CORNER JOINT ASSEMBLIES

A. Closeout Firmly Clamped

Full Scale		One-Tenth Scale			
		Run #1		Run #2	
Load(lb)	Deflection(in)	Load(grams)	Deflection(in)	Load(grams)	Deflection(in)
0	0	0	0	0	0
10	.024	50	.004	500	.036
20	.048	100	.008	600	.045
30	.072	200	.015	650	.048
40	.096	400	.029	700	.052
50	.120	900	.069	800	.059
40	.099	400	.037	850	.063
30	.075	200	.022	900	.067
20	.052	100	.012	850	.065
10	.027	50	.007	800	.062
0	.002	0	.003	700	.055
				500	.043
				0	.004

B. Closeout Screwed to Support

Full Scale		One-Tenth Scale		
Load(lb)	Deflection(in)	Load(grams)	Deflection(in)	
0	0	0	0	
10	.030	50	.005	
20	.061	100	.011	
30	.093	200	.024	
40	.126	400	.049	
50	.159	900	.135	(Screws and nuts failed)
40	.131			
30	.100			
20	.068			
10	.036			
0	.004			

TABLE 11

VIBRATORY RESPONSE - CORNER JOINT ASSEMBLIES

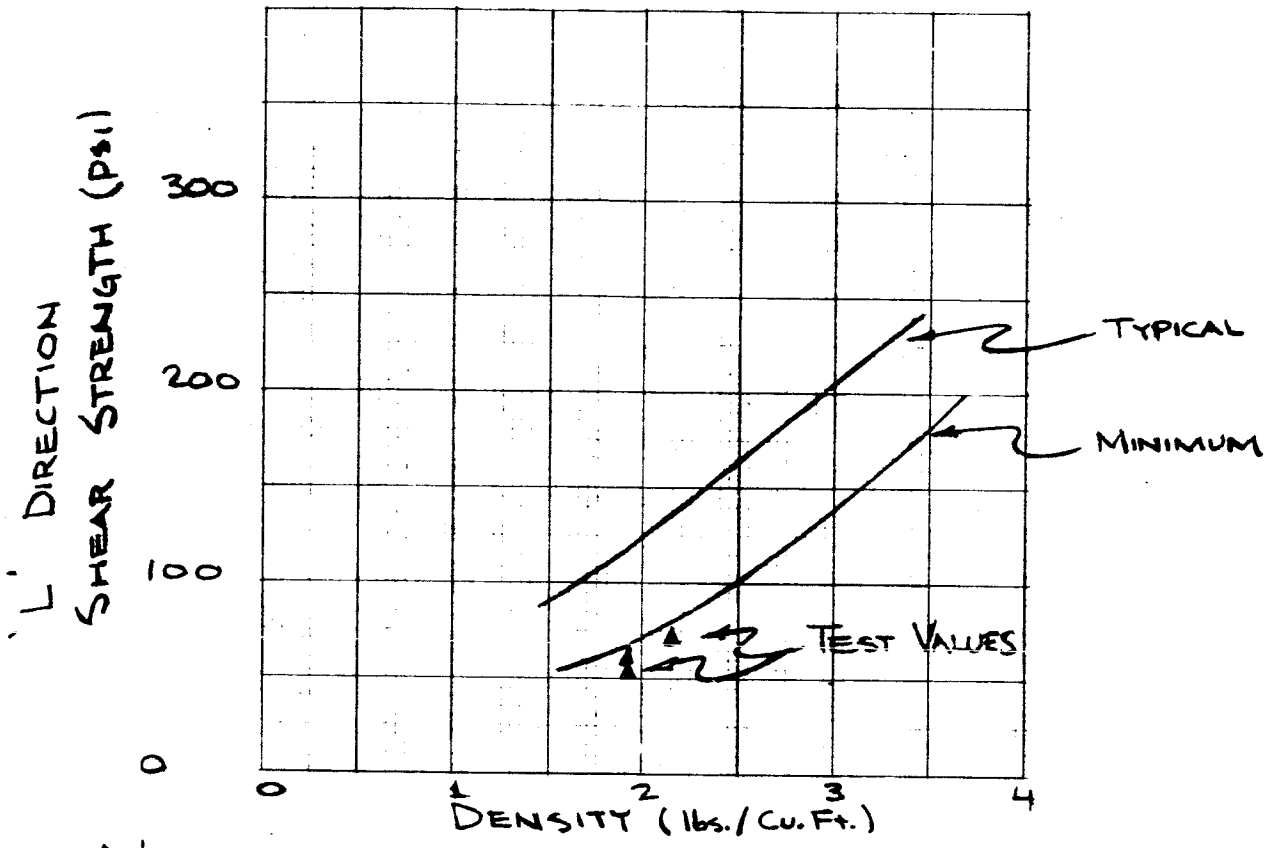
Mode	Frequency(Hz)		Damping(g)		Frequency Ratio
	Full Scale	One-Tenth Scale	Full Scale	One-Tenth Scale	Model/Full Scale
A. Closeout Clamped					
First Bending	54.2	500	0.0044	0.04	9.22
Second Bending	355	3280	0.0055	0.055	9.24
First Torsion	136	1195	0.0024	0.044	8.79
Second "Torsion"	478				
(Other weak modes)	158	7620	0.0116		
	1420				
	1930				
	4000				
	4300				
	7700				
B. Screwed to Support					
First Bending	47.8	391	0.011	0.059	8.19
Second Bending	329	3060	0.0176	0.044	9.30
First Torsion	128	1135	0.0037	0.0314	8.86
Second "Torsion"	473		0.0157		
(Other weak modes)	1400	7800			
	1930				
	4000				
	4300				
	7700				

TABLE 12

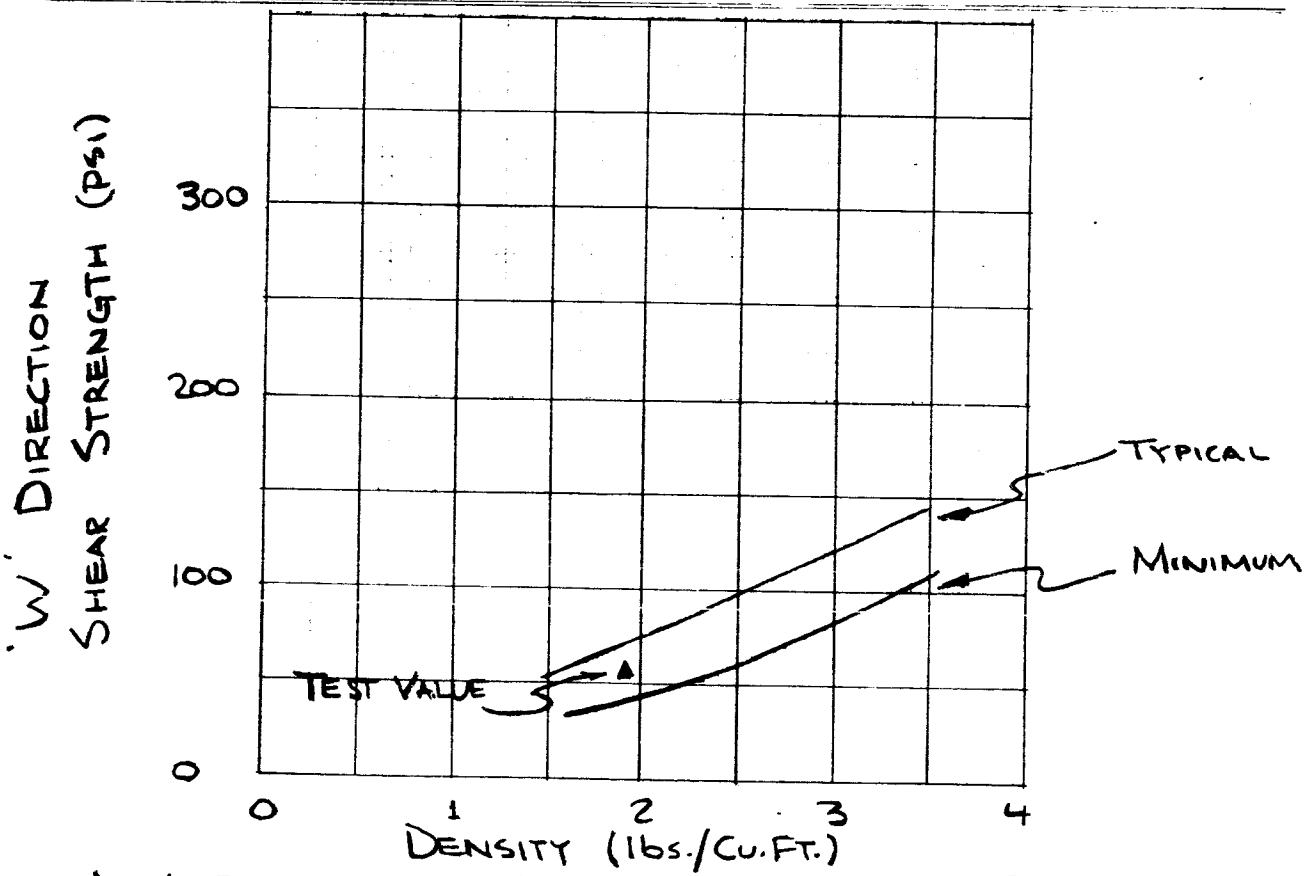
LOAD-DEFLECTION DATA - RADIAL BEAM

Load (lb)*	Deflection (in)
0	0
5	.00055
10	.00121
15	.00180
20	.00241
24	.00294
0	0

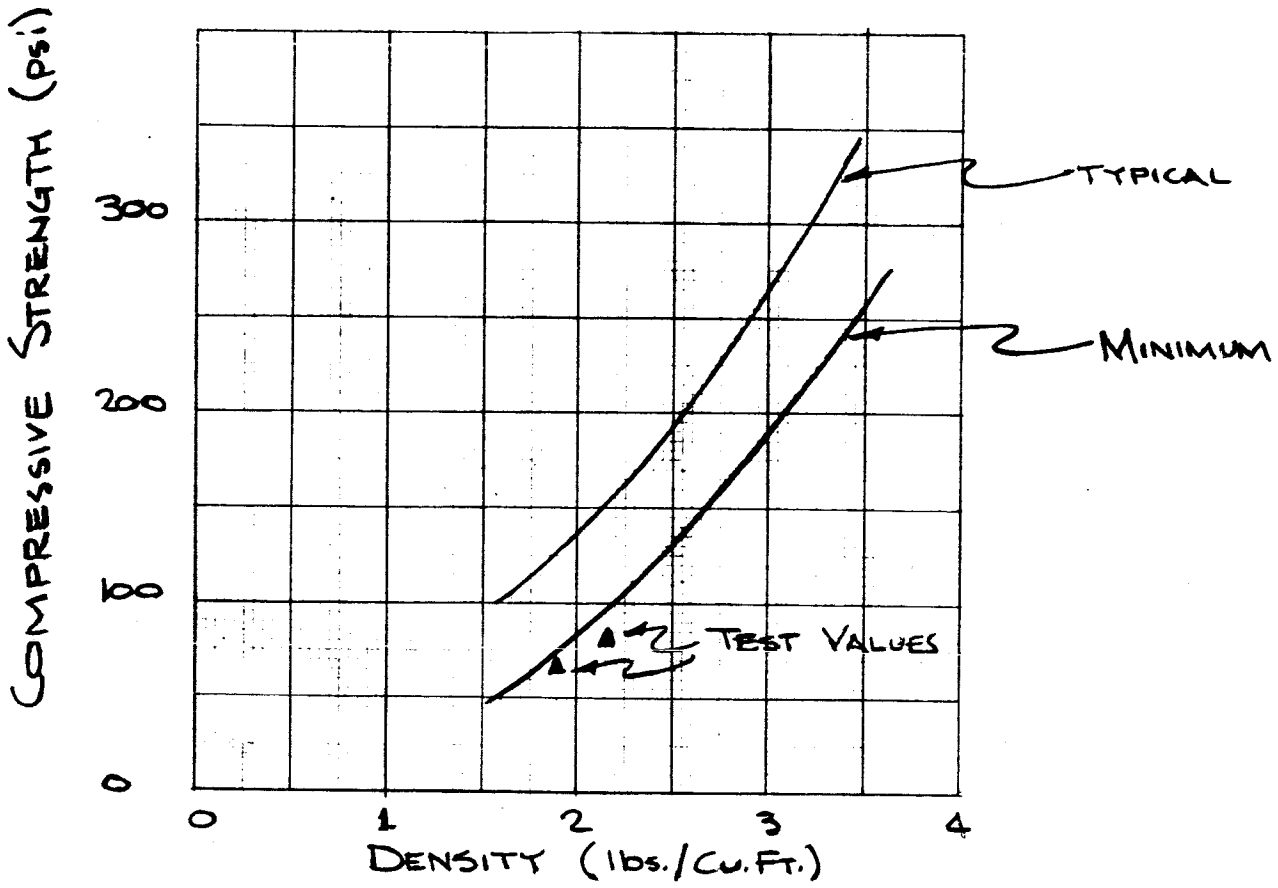
* In addition to tare weight of 2.01 lb



'L' DIRECTION SHEAR STRENGTH VS. DENSITY
5052 HONEYCOMB
FIGURE 1



'W' DIRECTION SHEAR STRENGTH VS. DENSITY
5052 HONEYCOMB
FIGURE 2



COMPRESSIVE STRENGTH VS. DENSITY
5052 HONEYCOMB

FIGURE 3

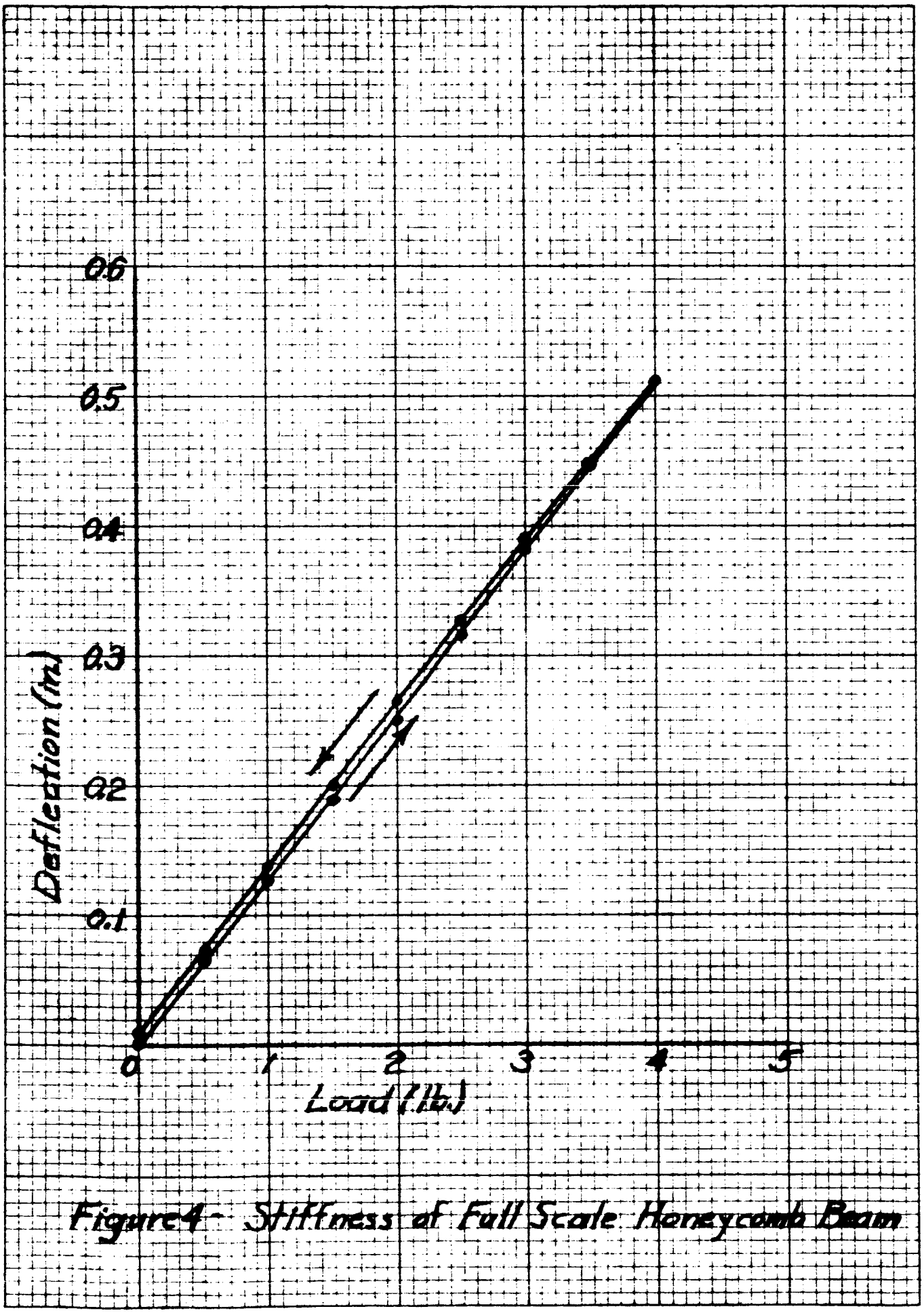


Figure 4 - Stiffness of Full Scale Honeycomb Beam

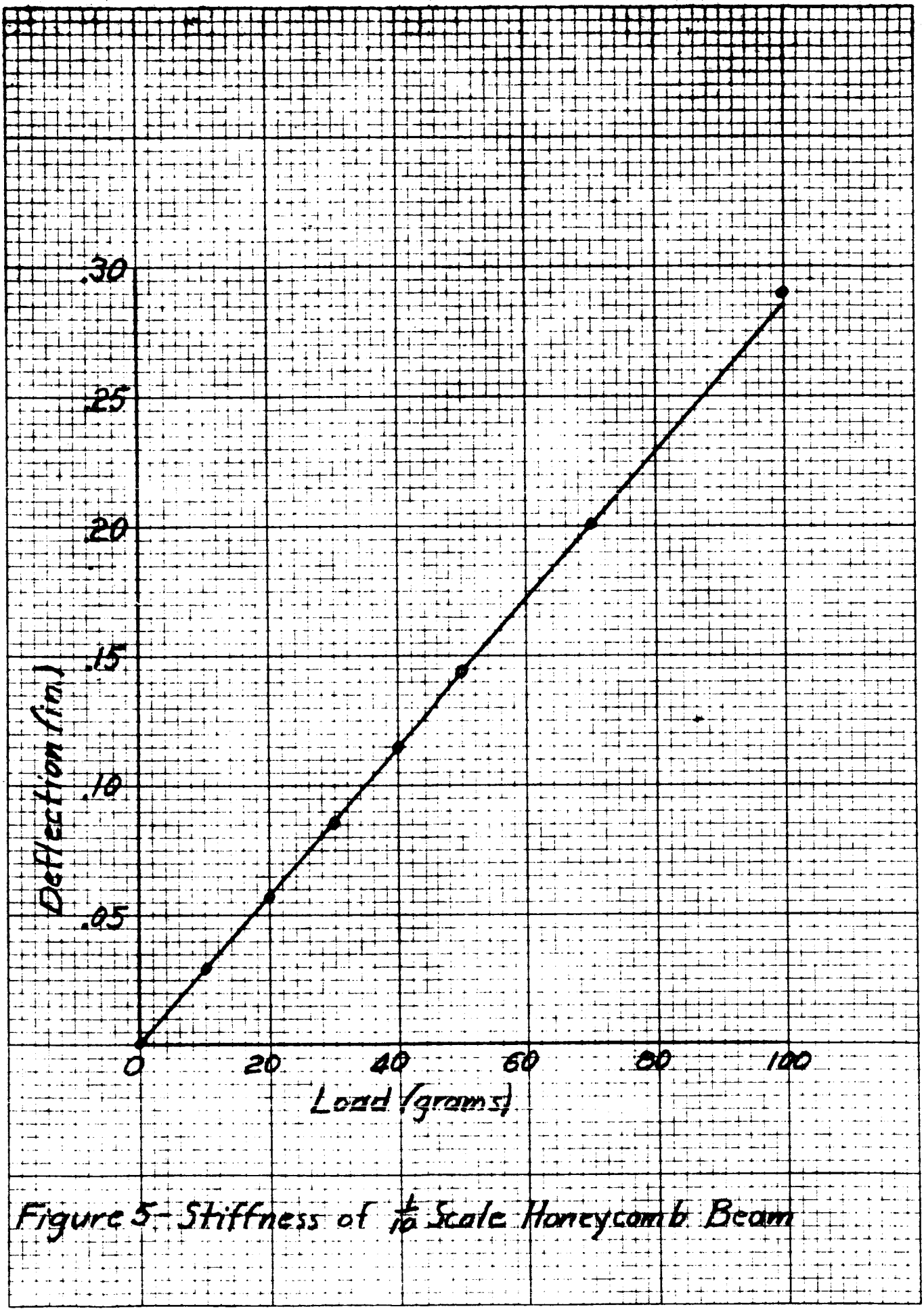


Figure 5 - Stiffness of $\frac{1}{10}$ Scale Honeycomb Beam

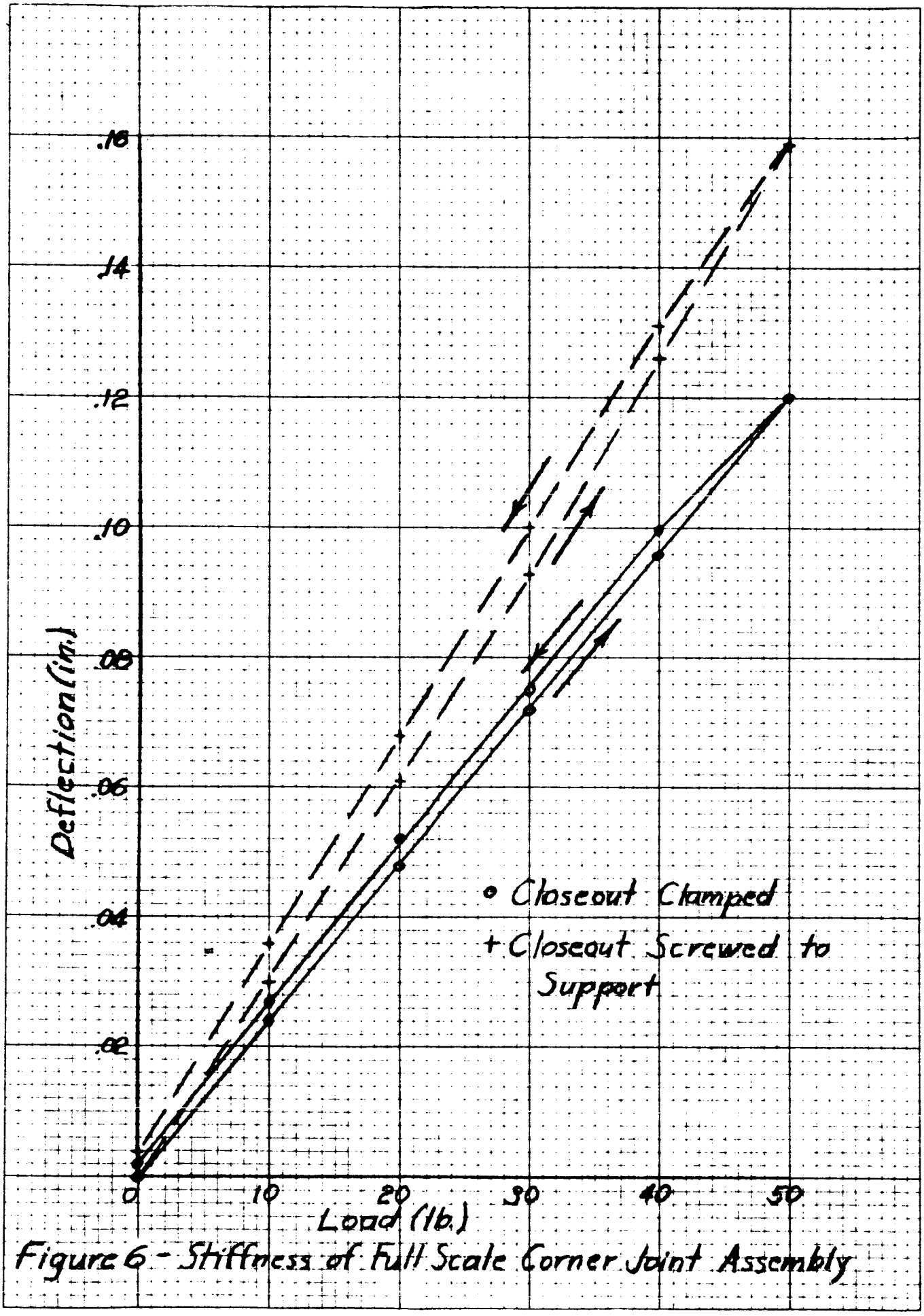


Figure 6 - Stiffness of Full Scale Corner Joint Assembly

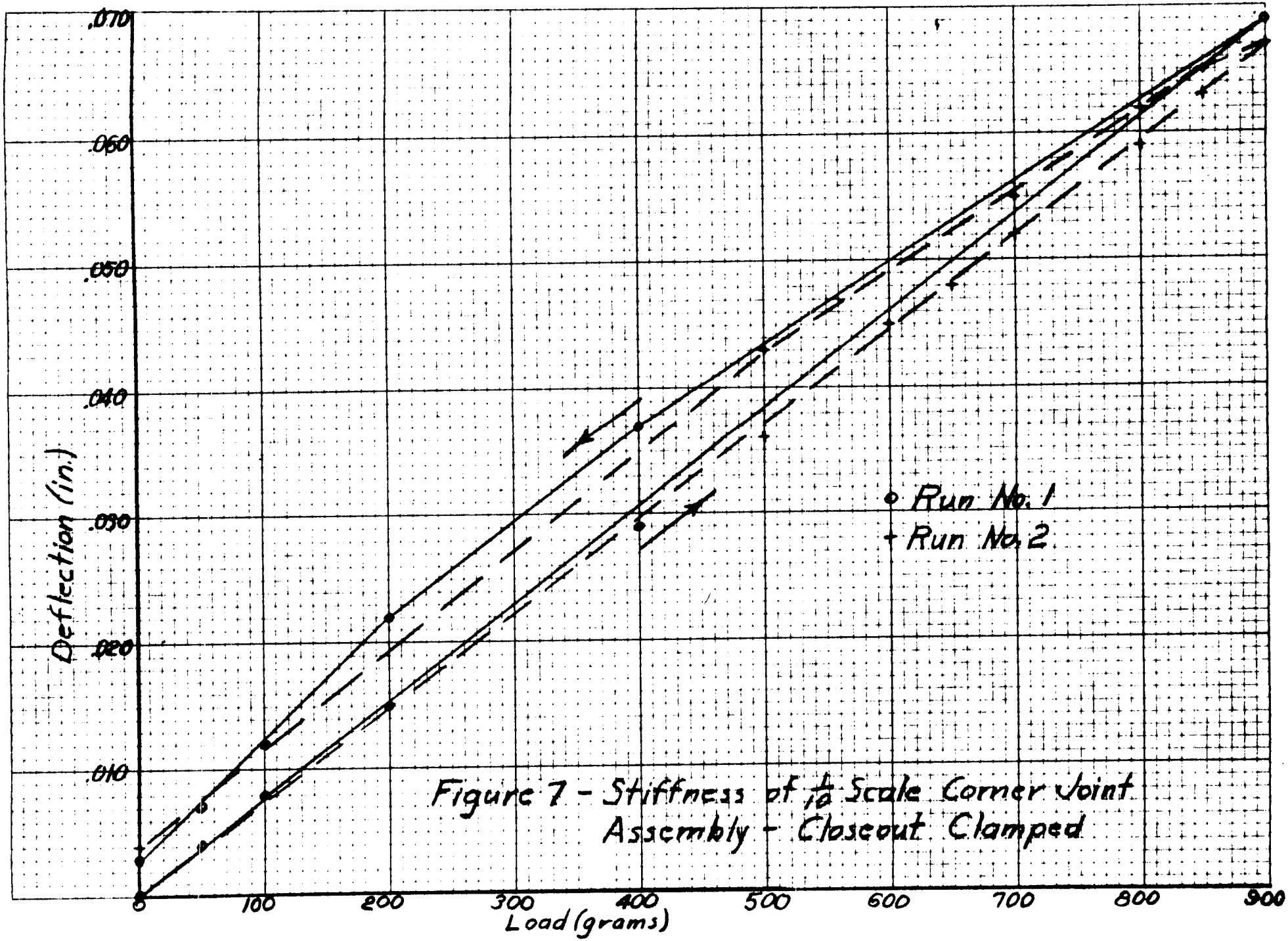


Figure 7 - Stiffness of ¼ Scale Corner Joint Assembly - Closeout Clamped

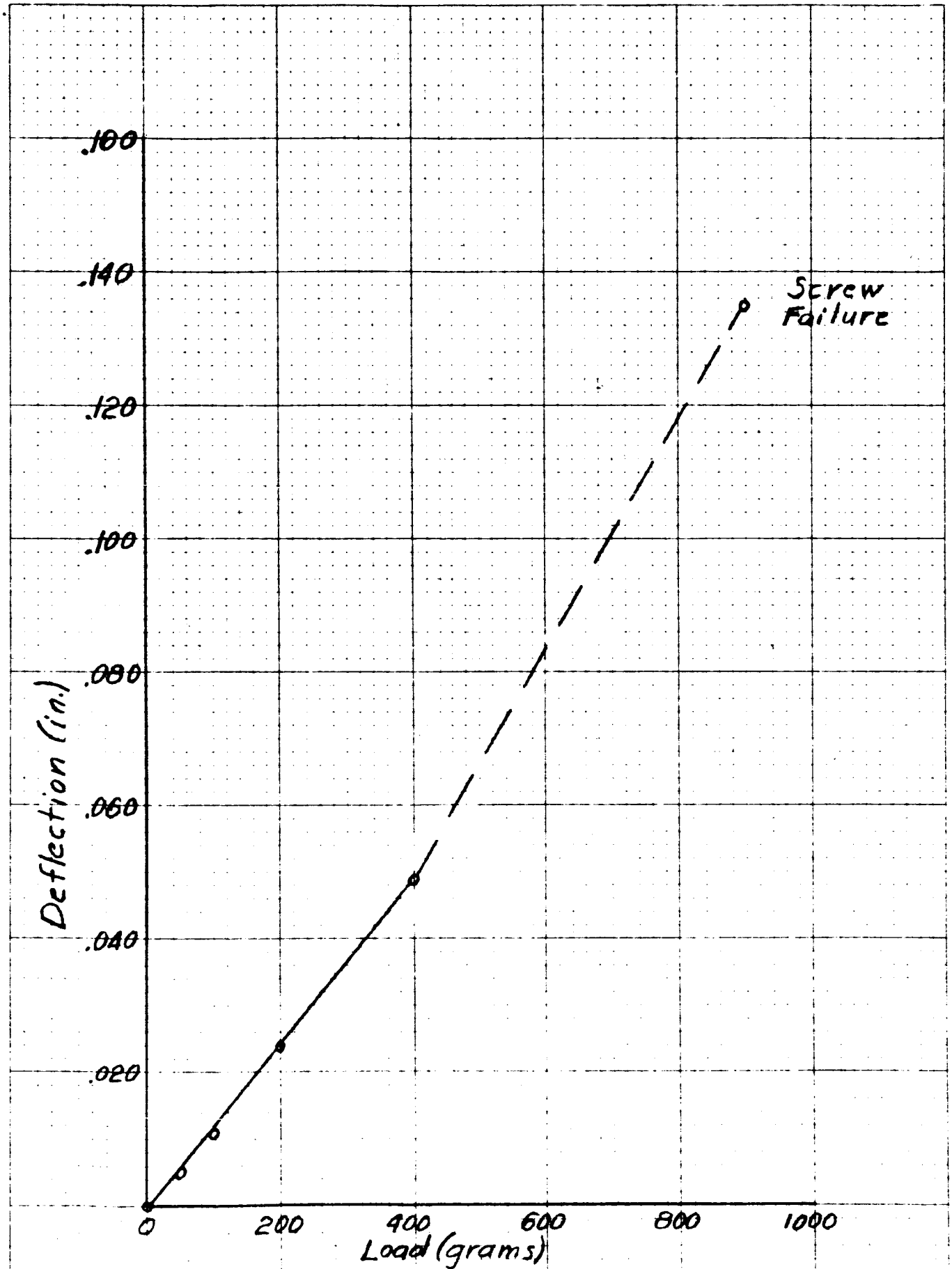


Figure 8- Stiffness of $\frac{1}{10}$ Scale Corner Joint Assembly - Closeout Screwed to Support

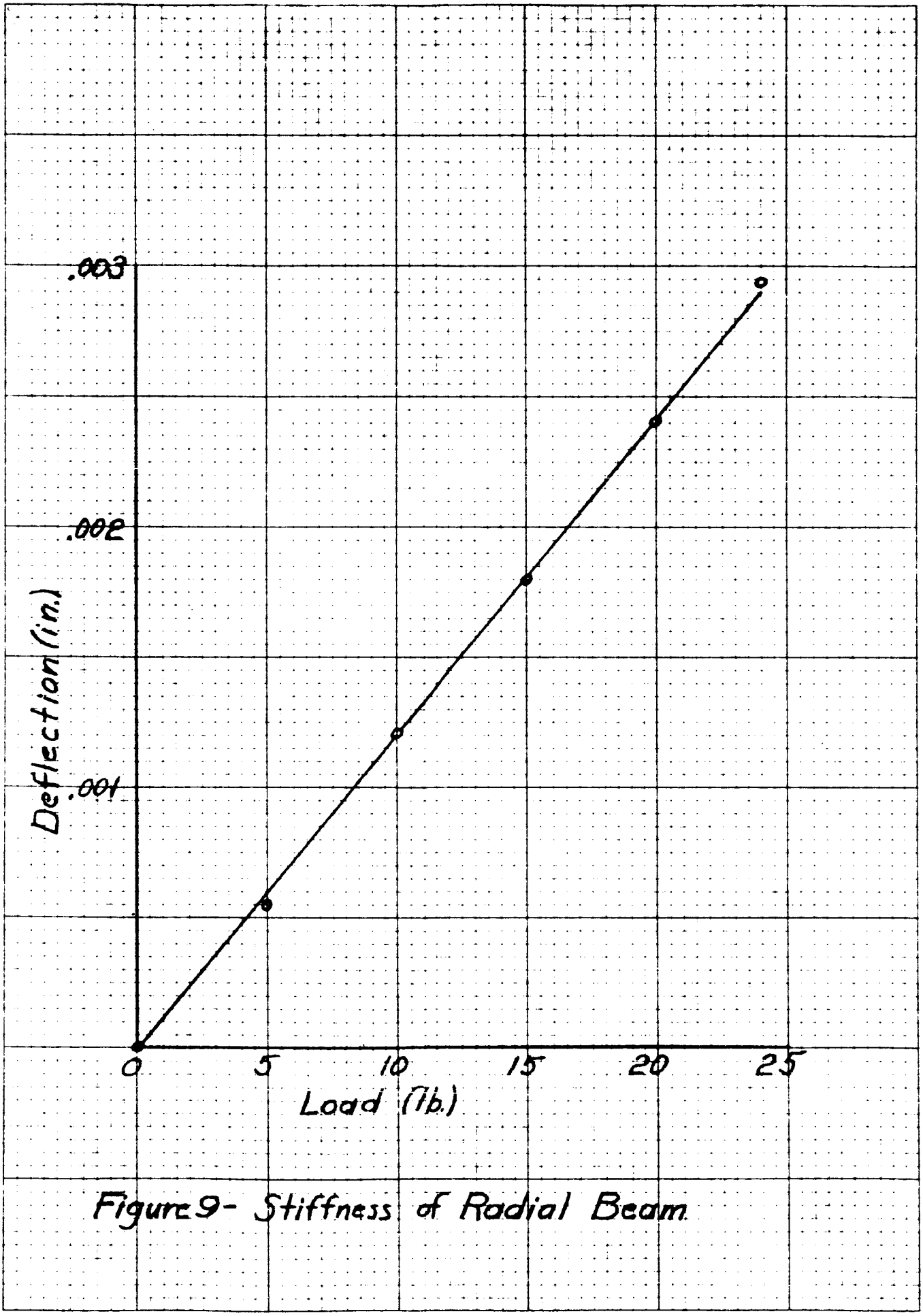
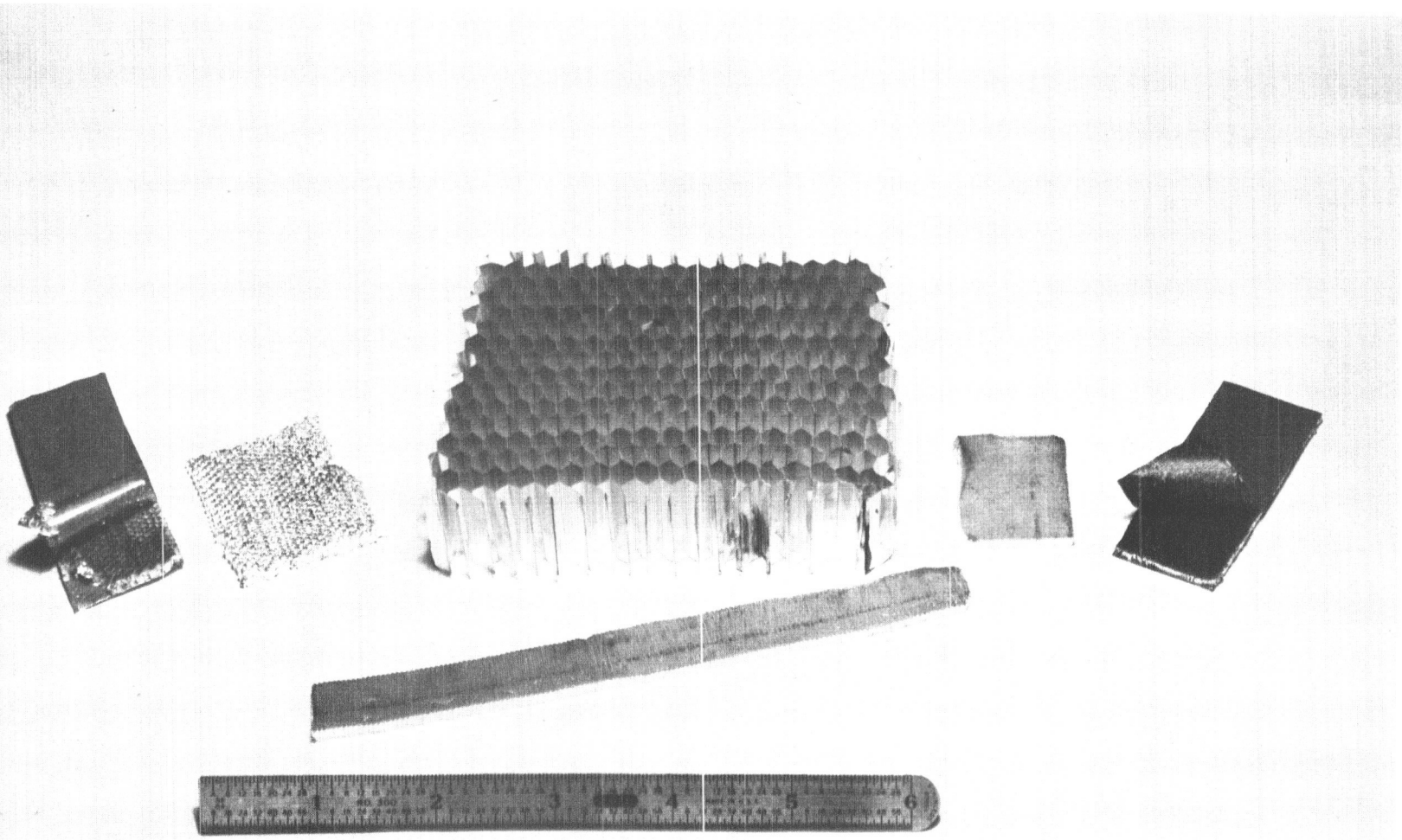
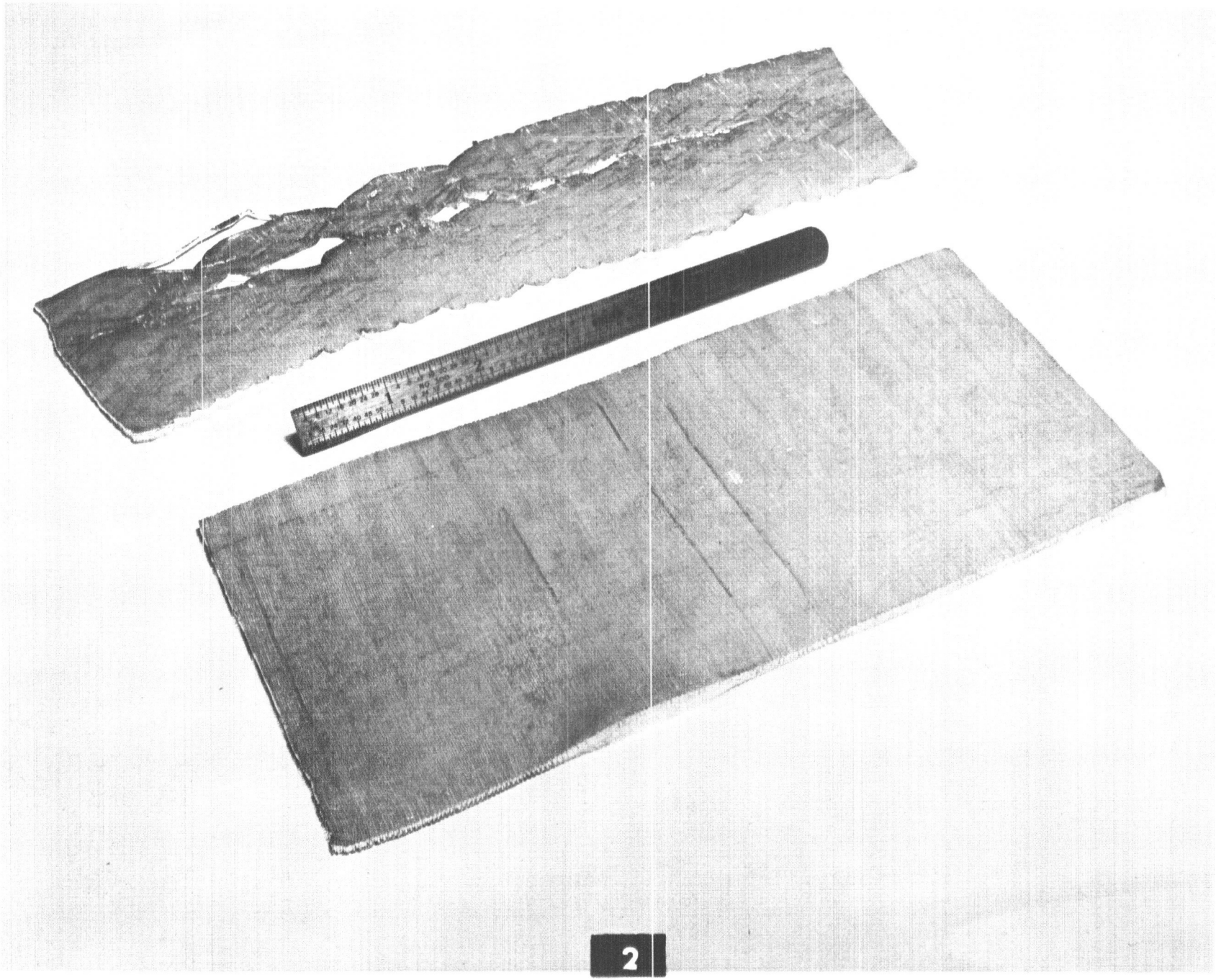
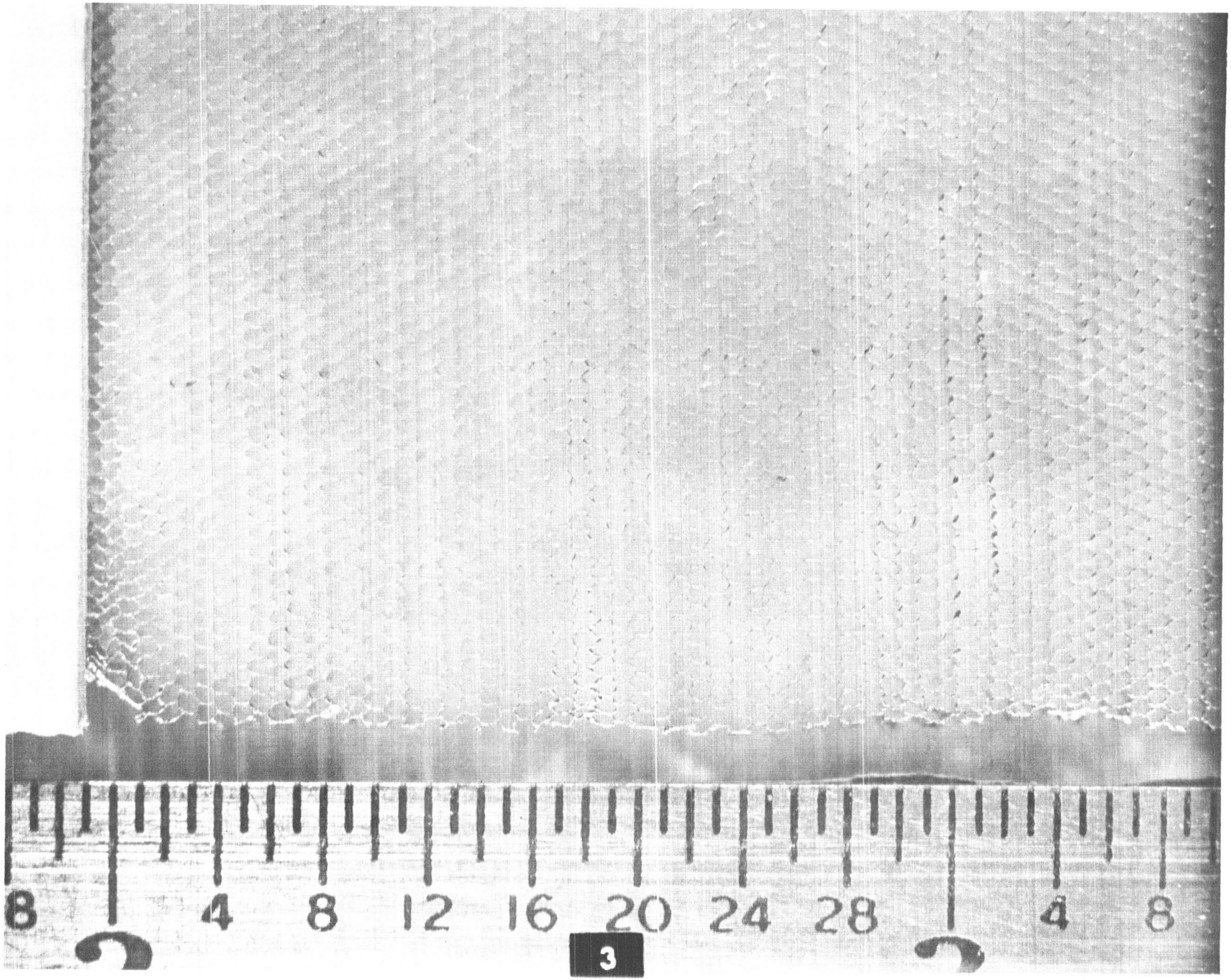


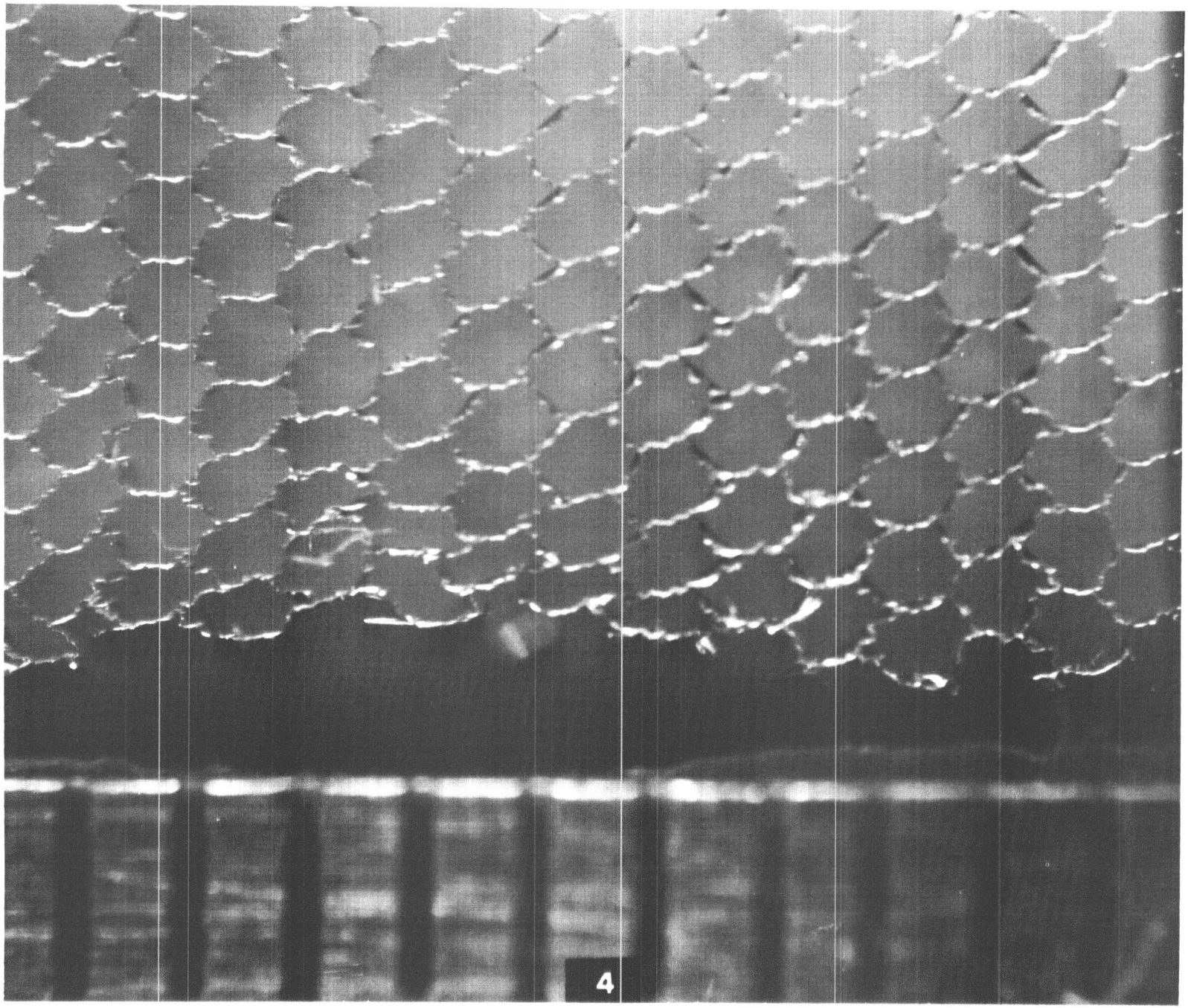
Figure 9- Stiffness of Radial Beam

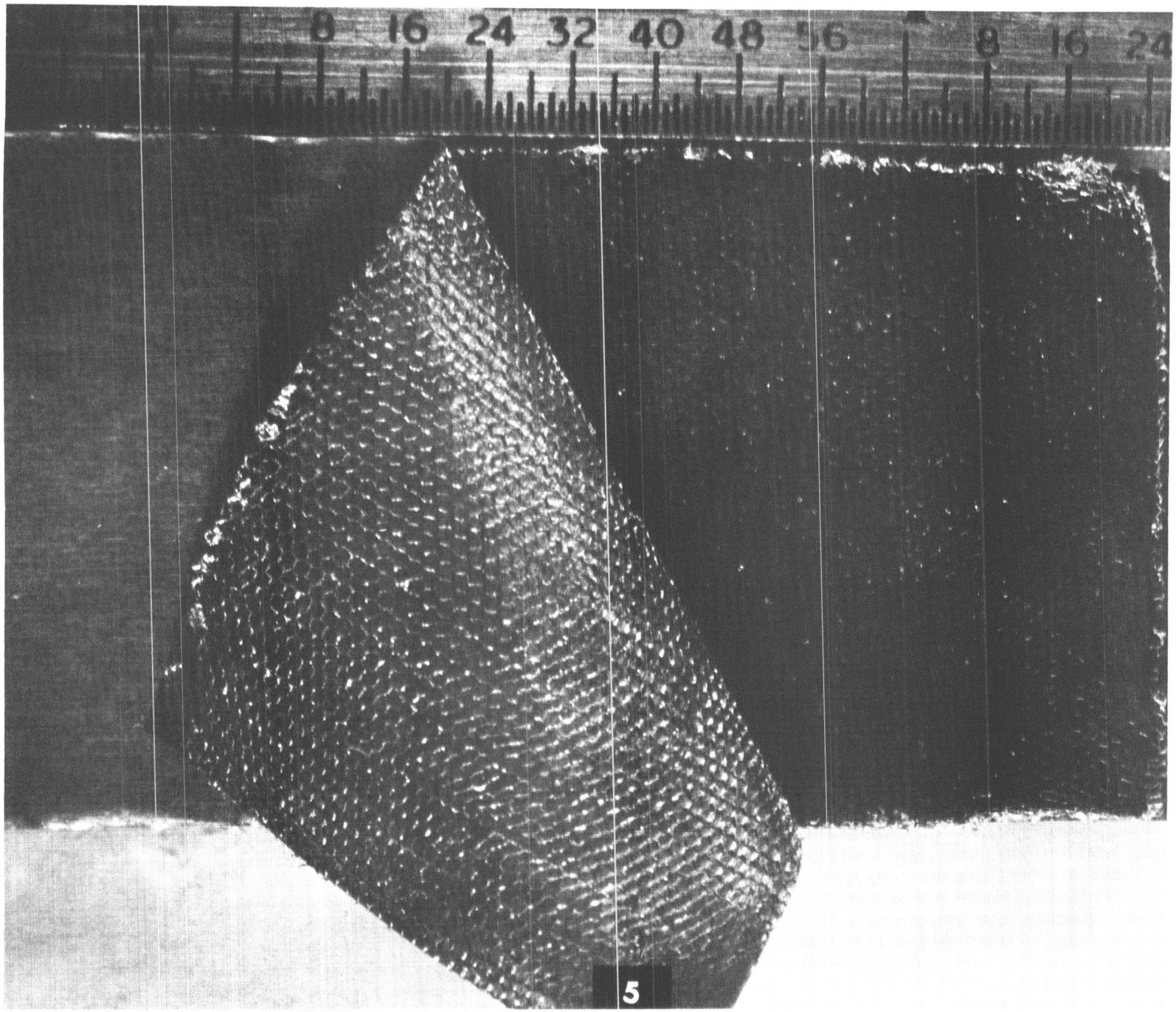


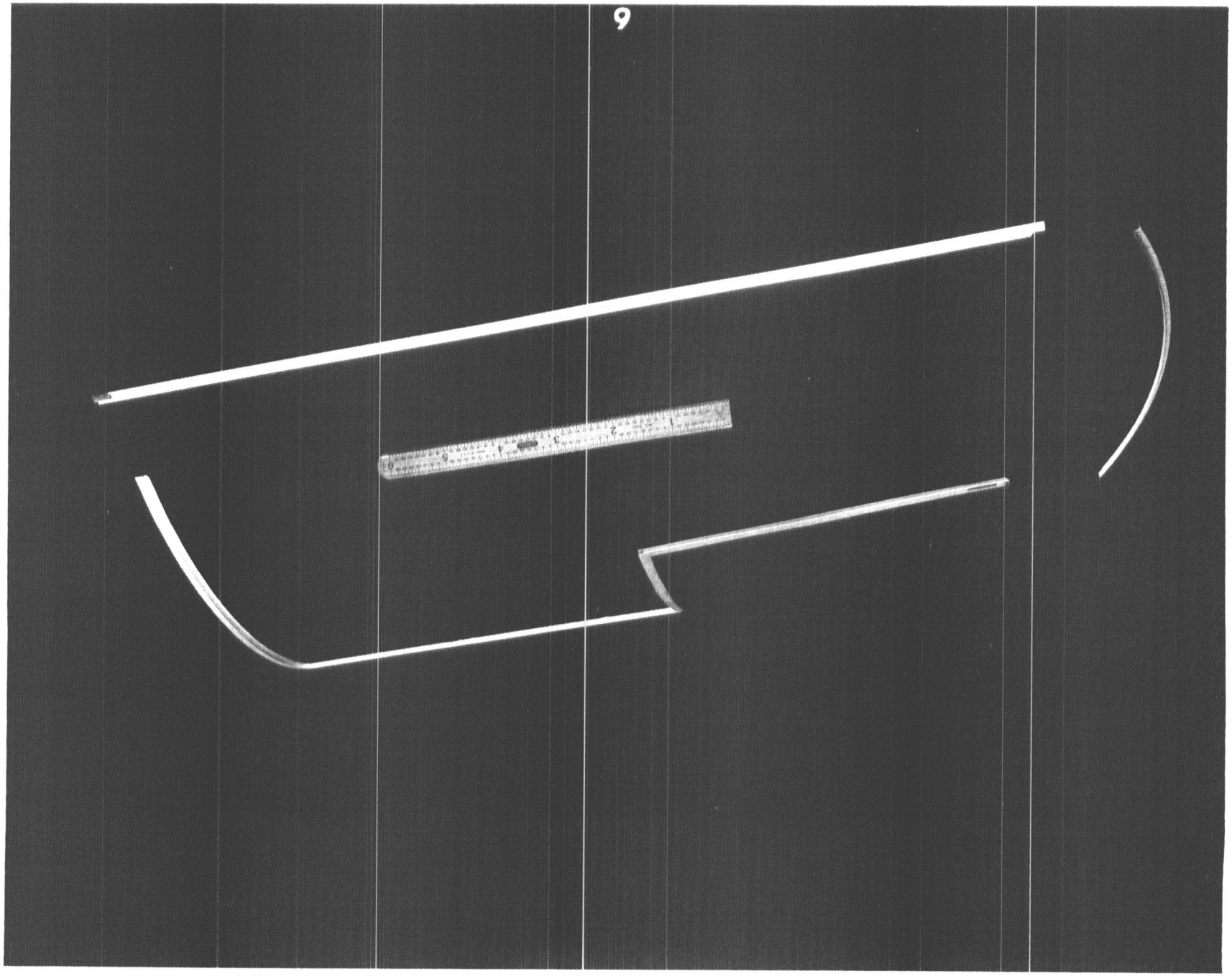


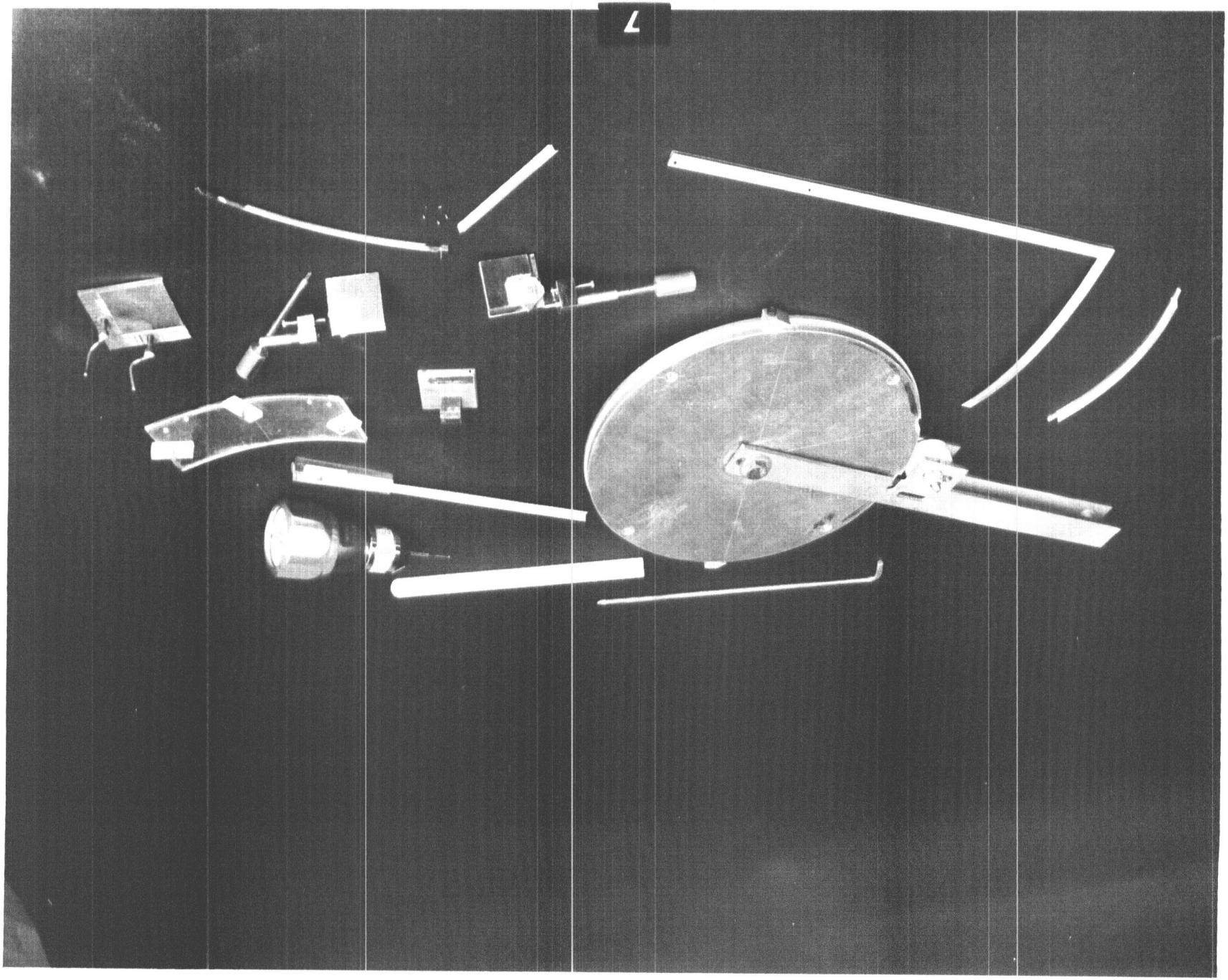
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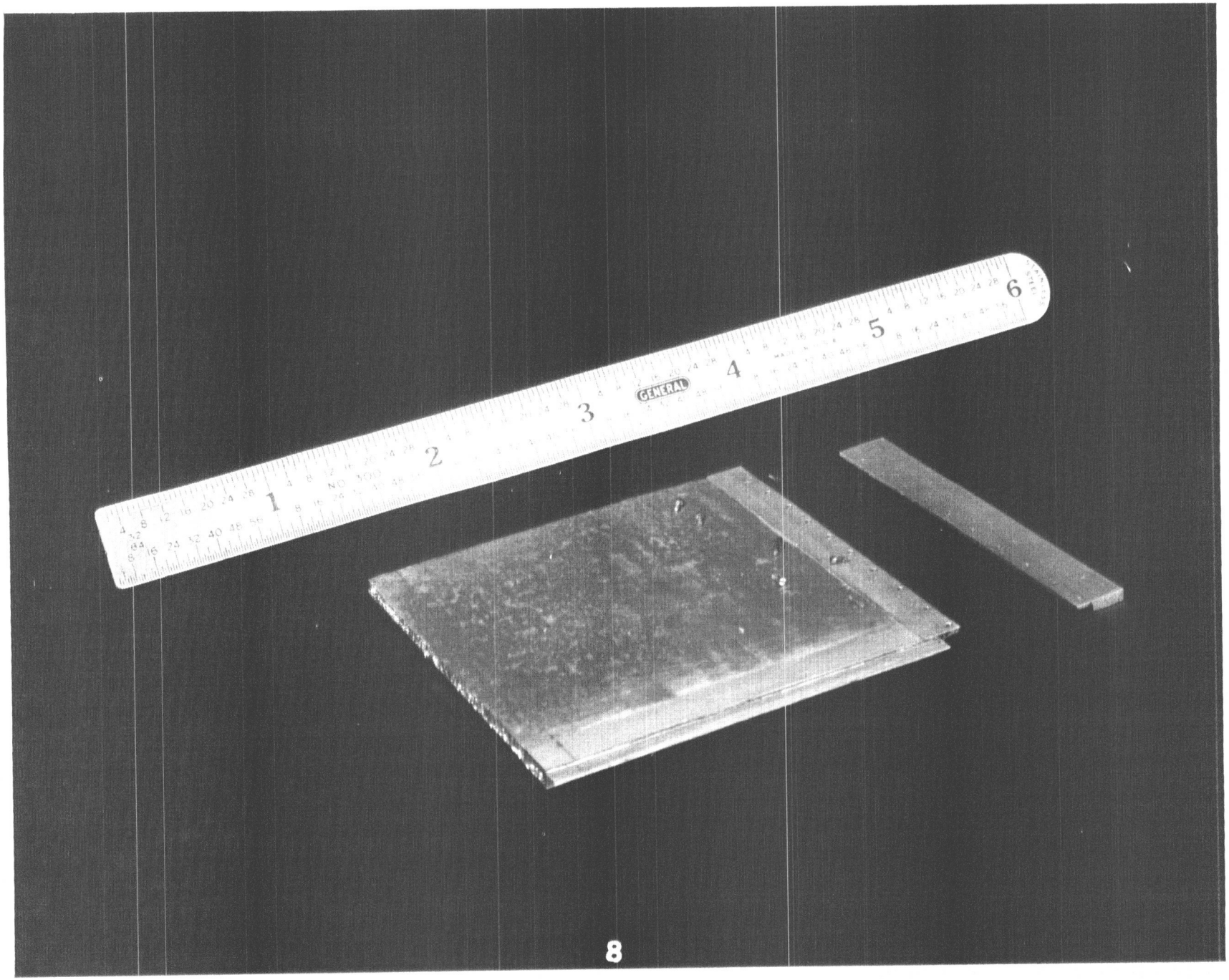

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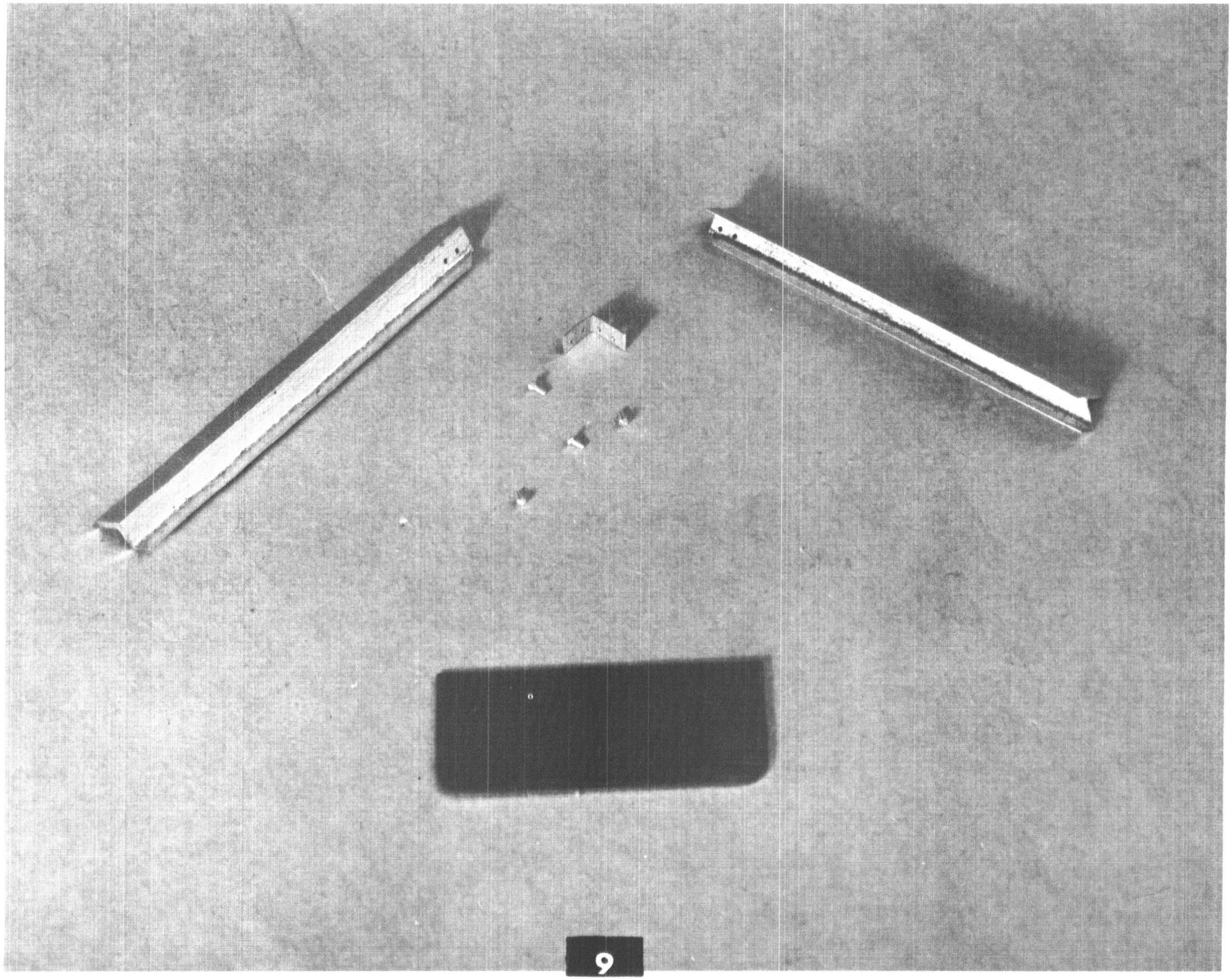

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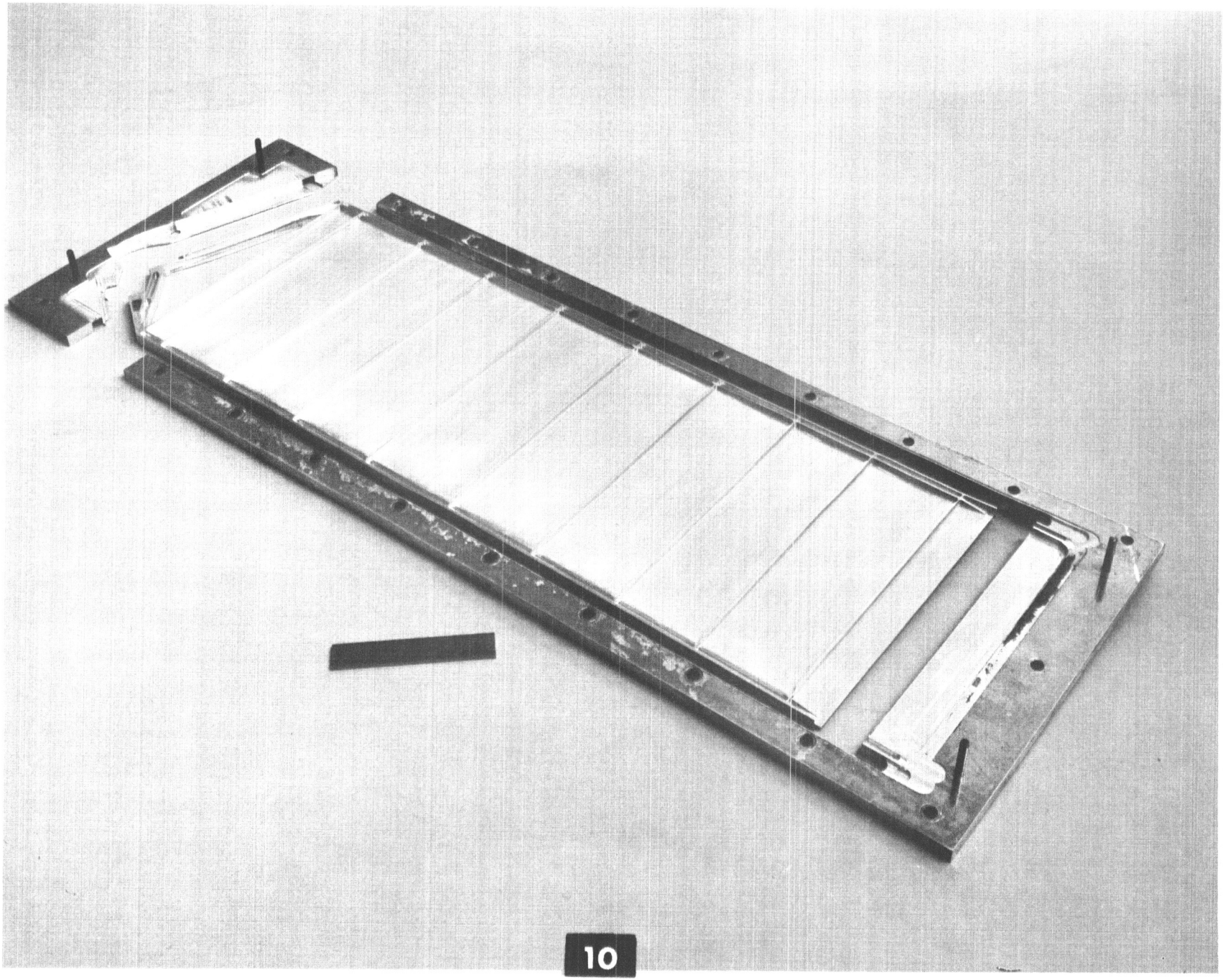

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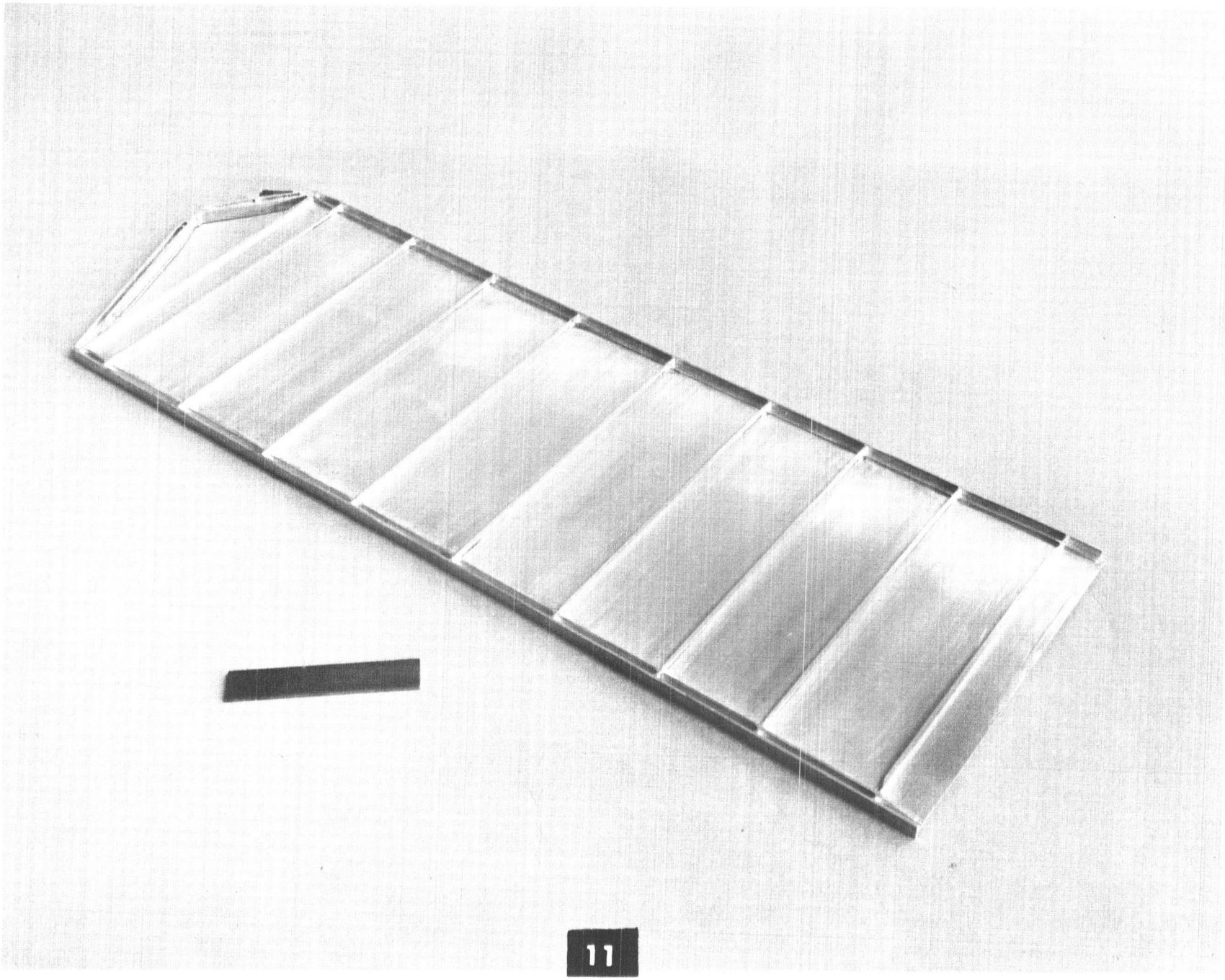
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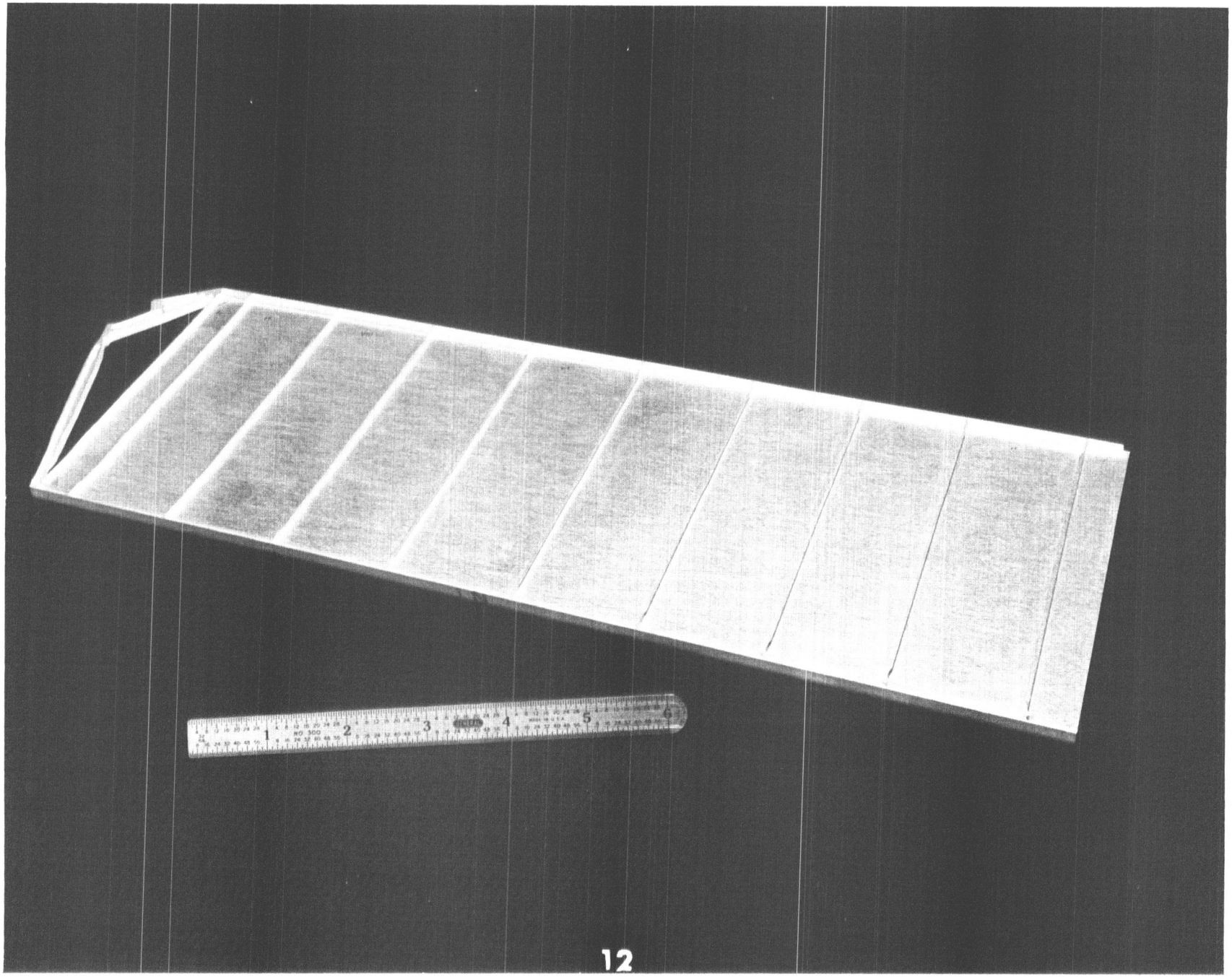
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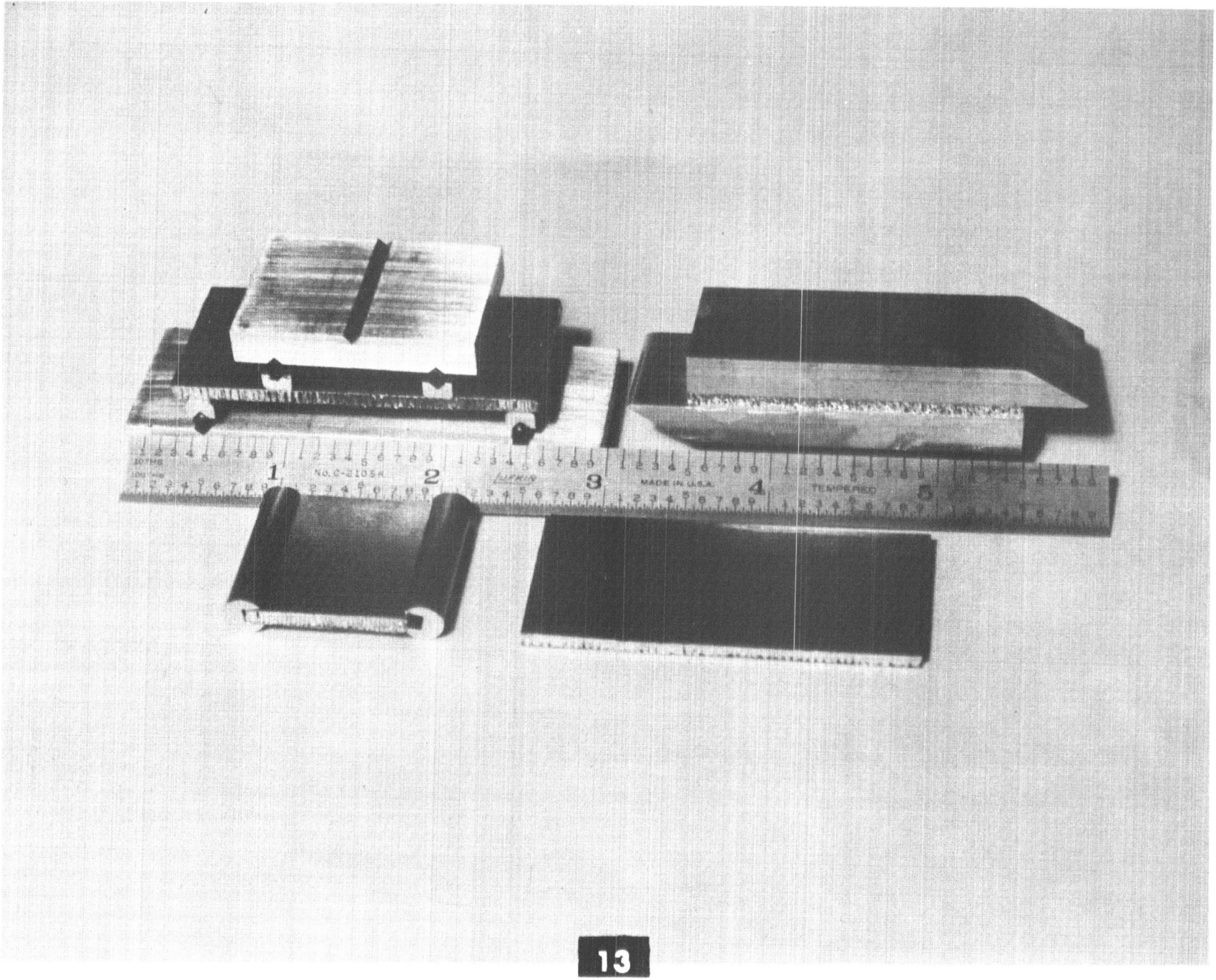


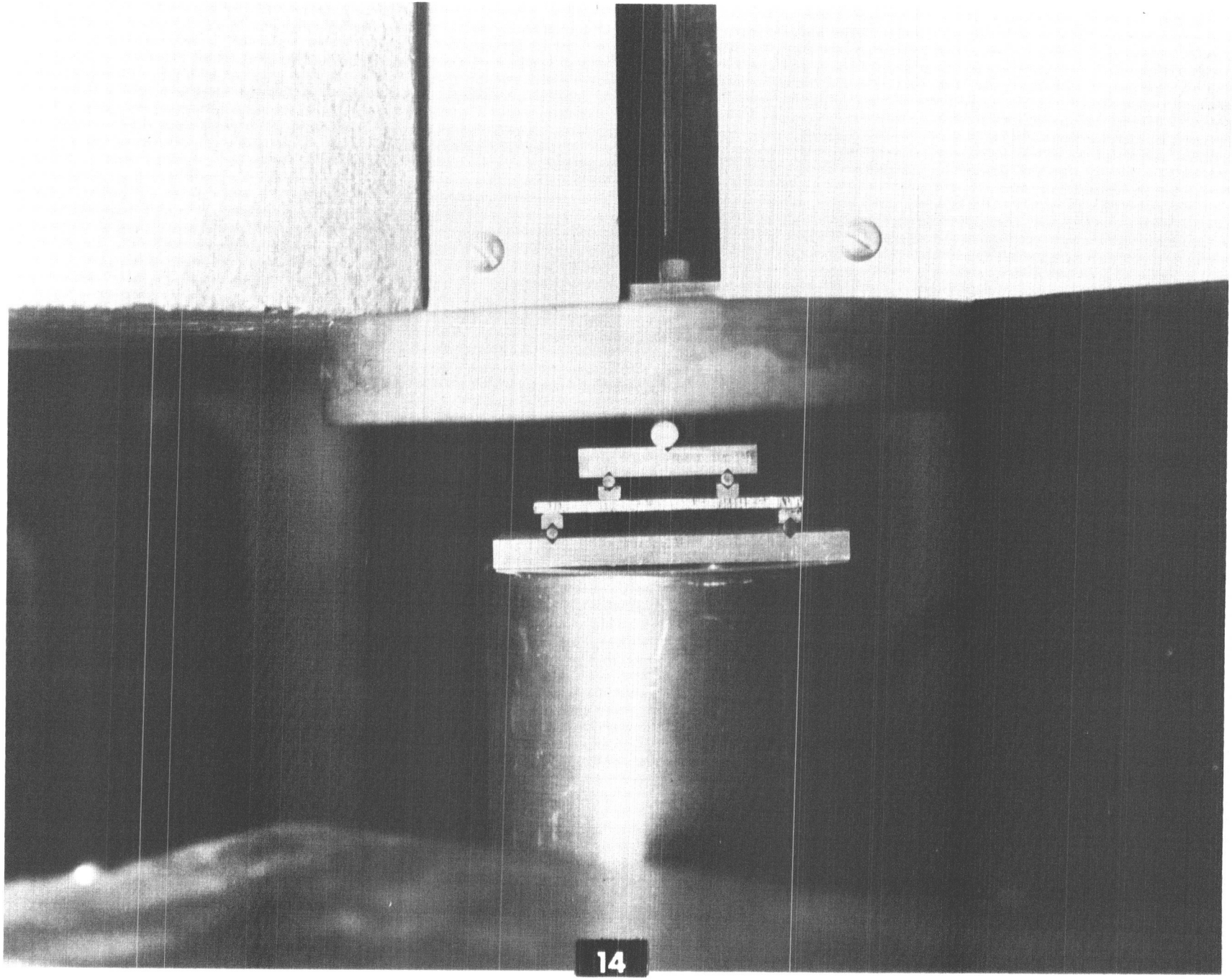
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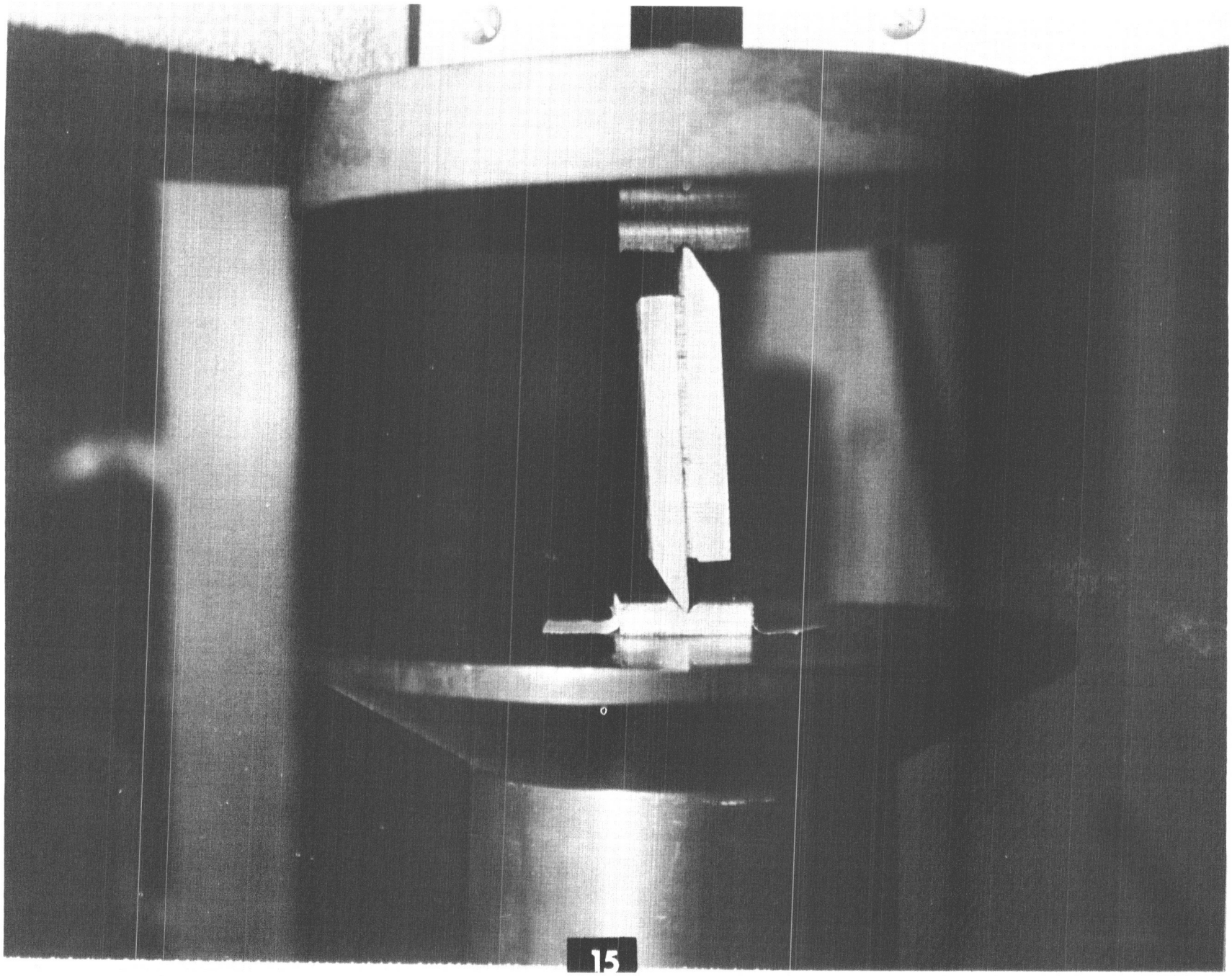
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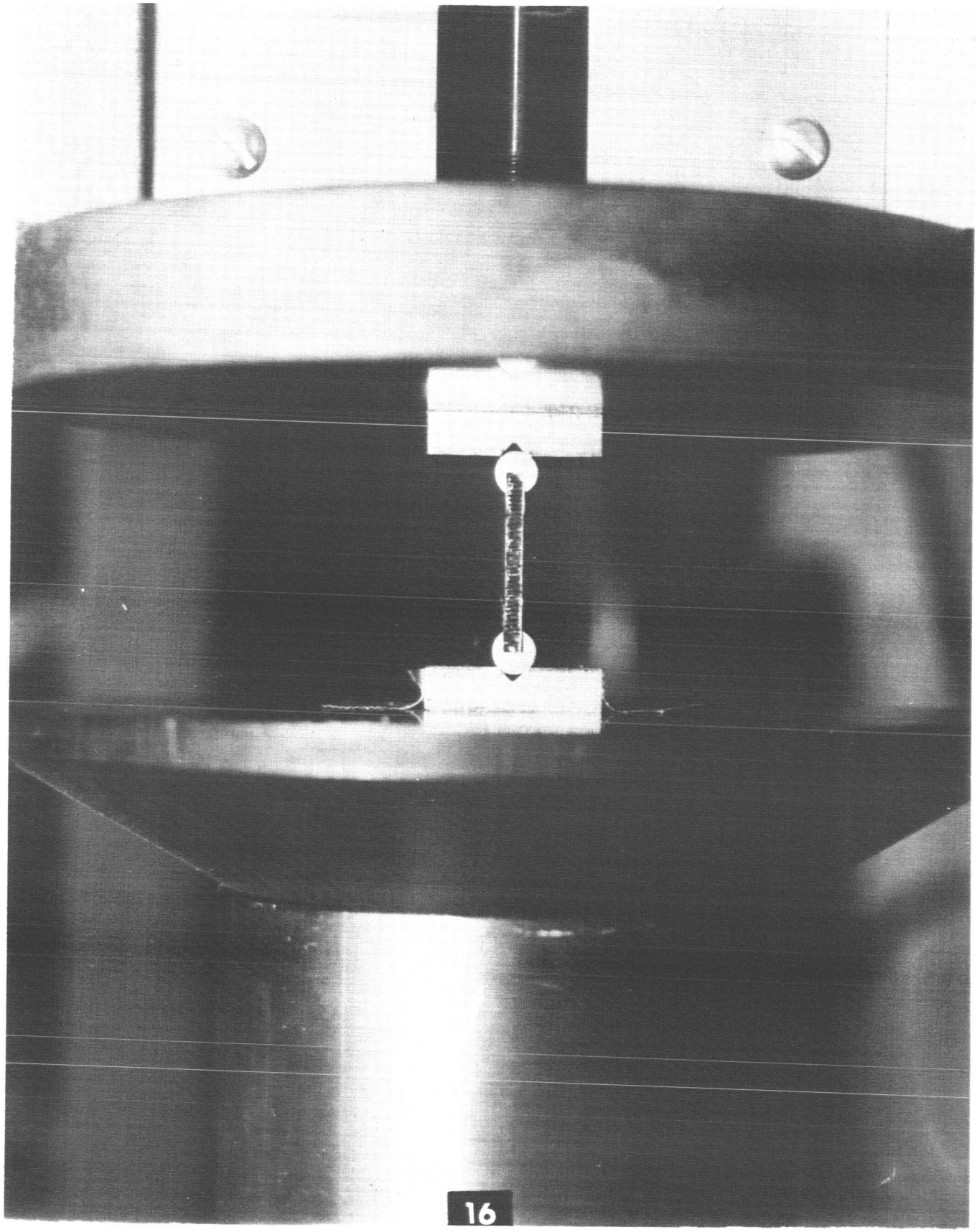

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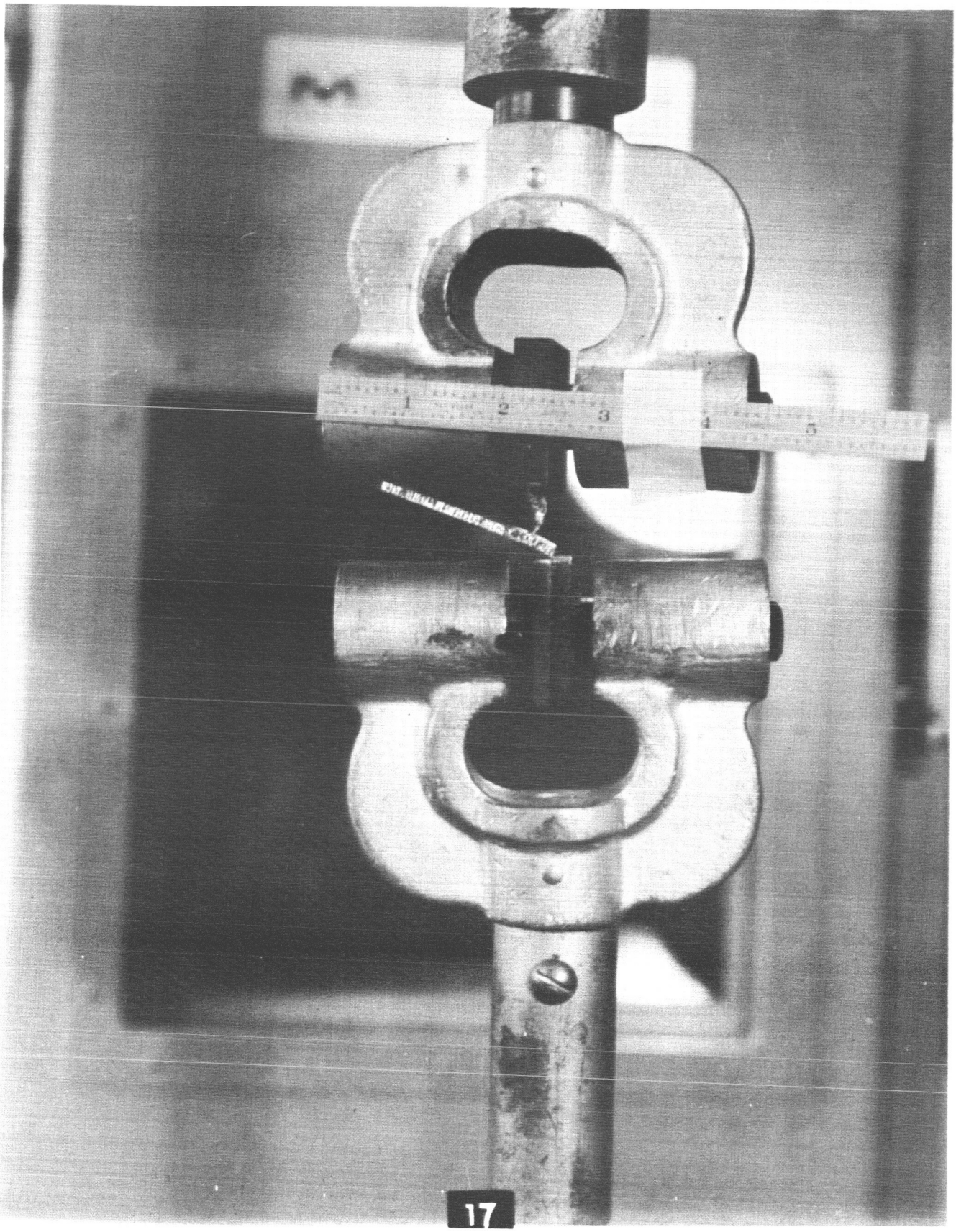

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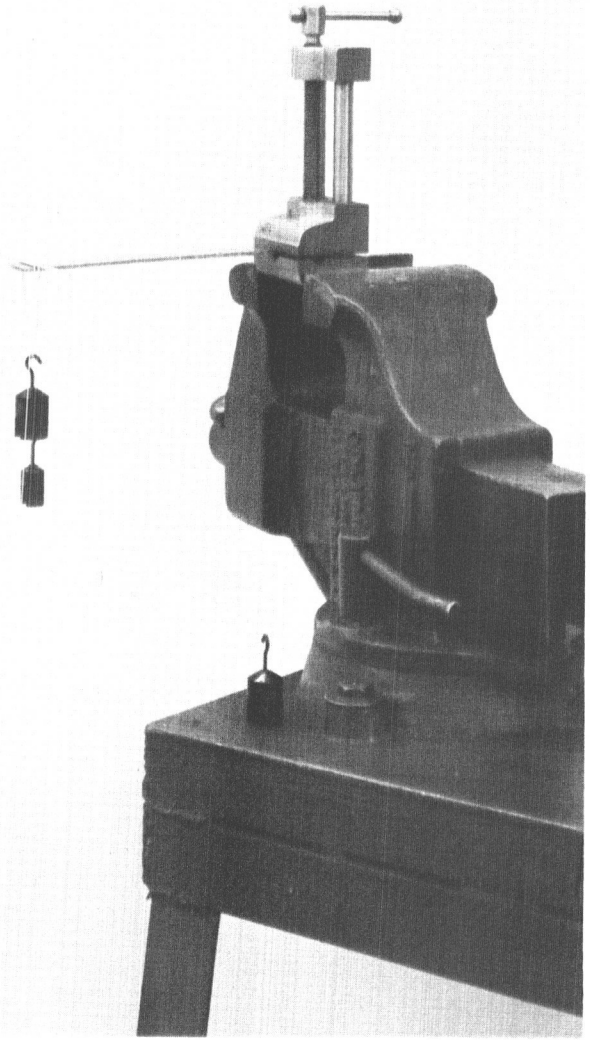
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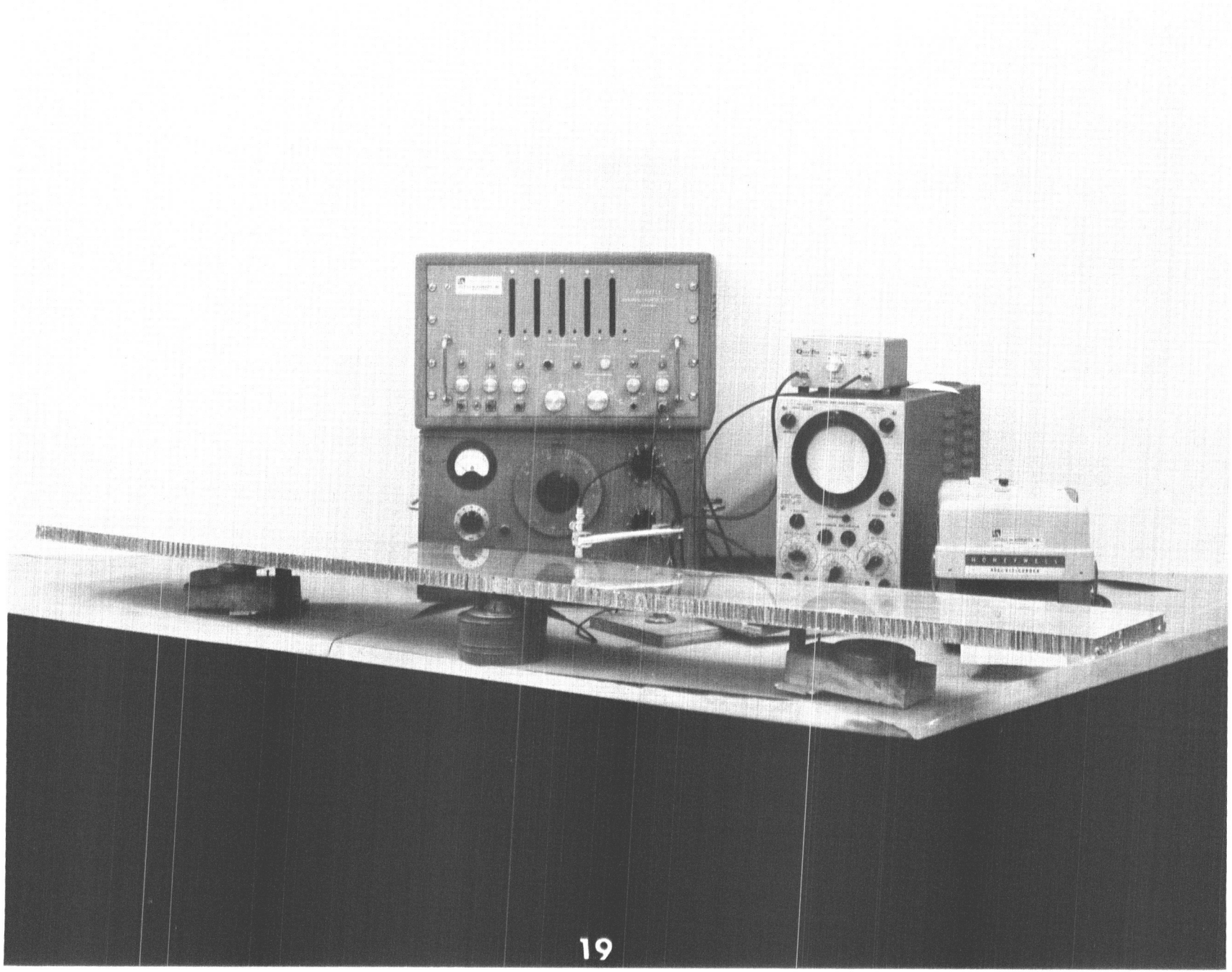

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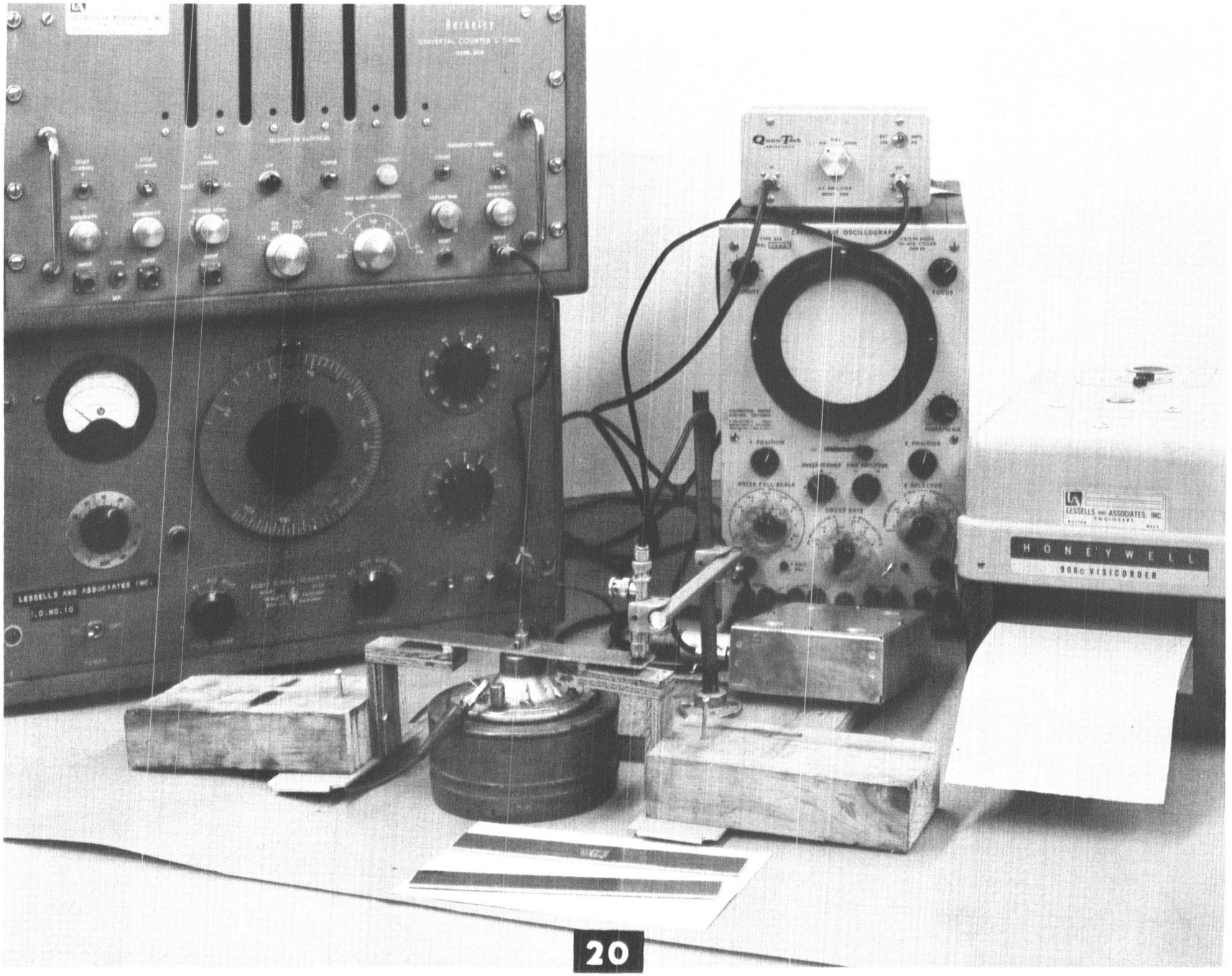


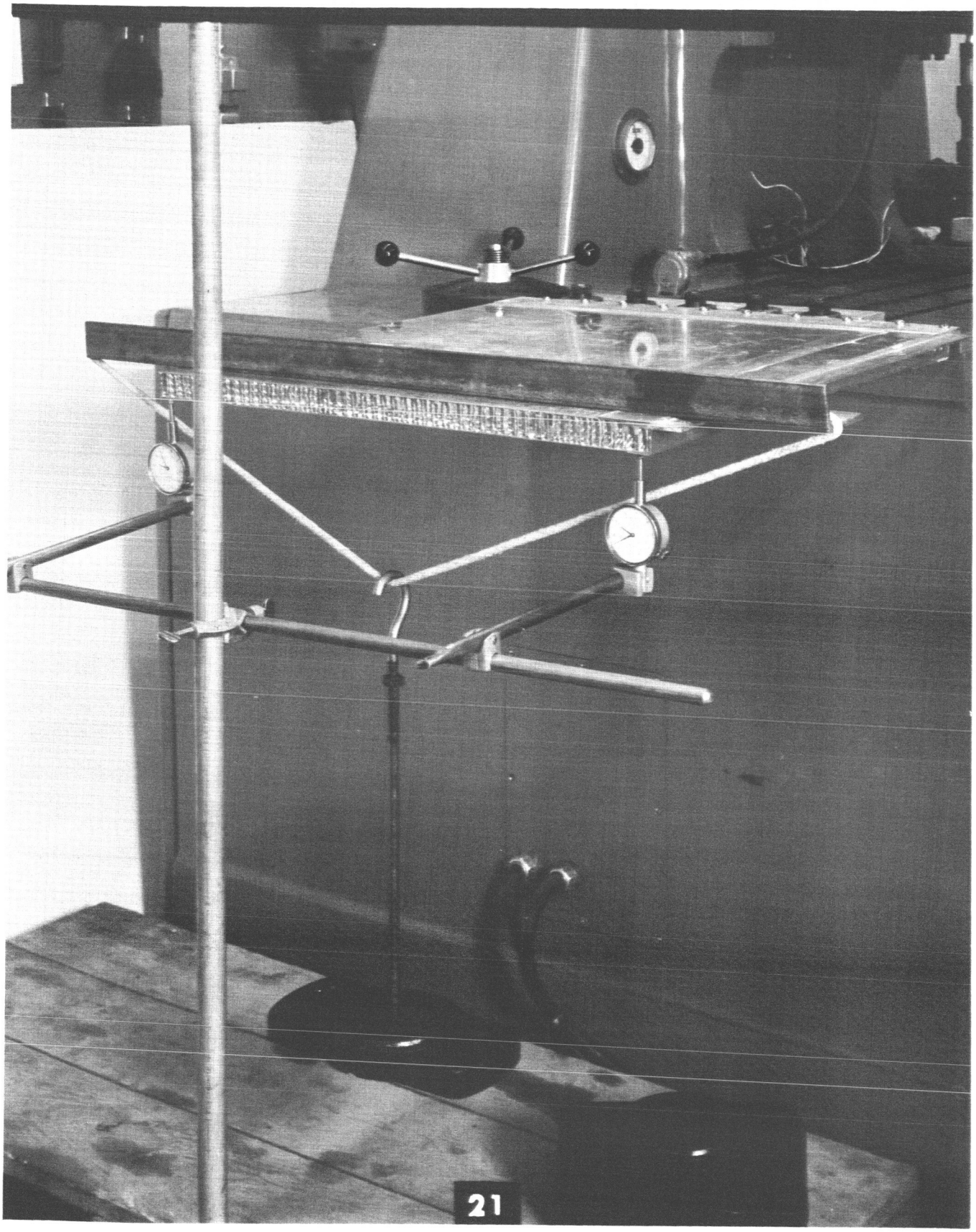

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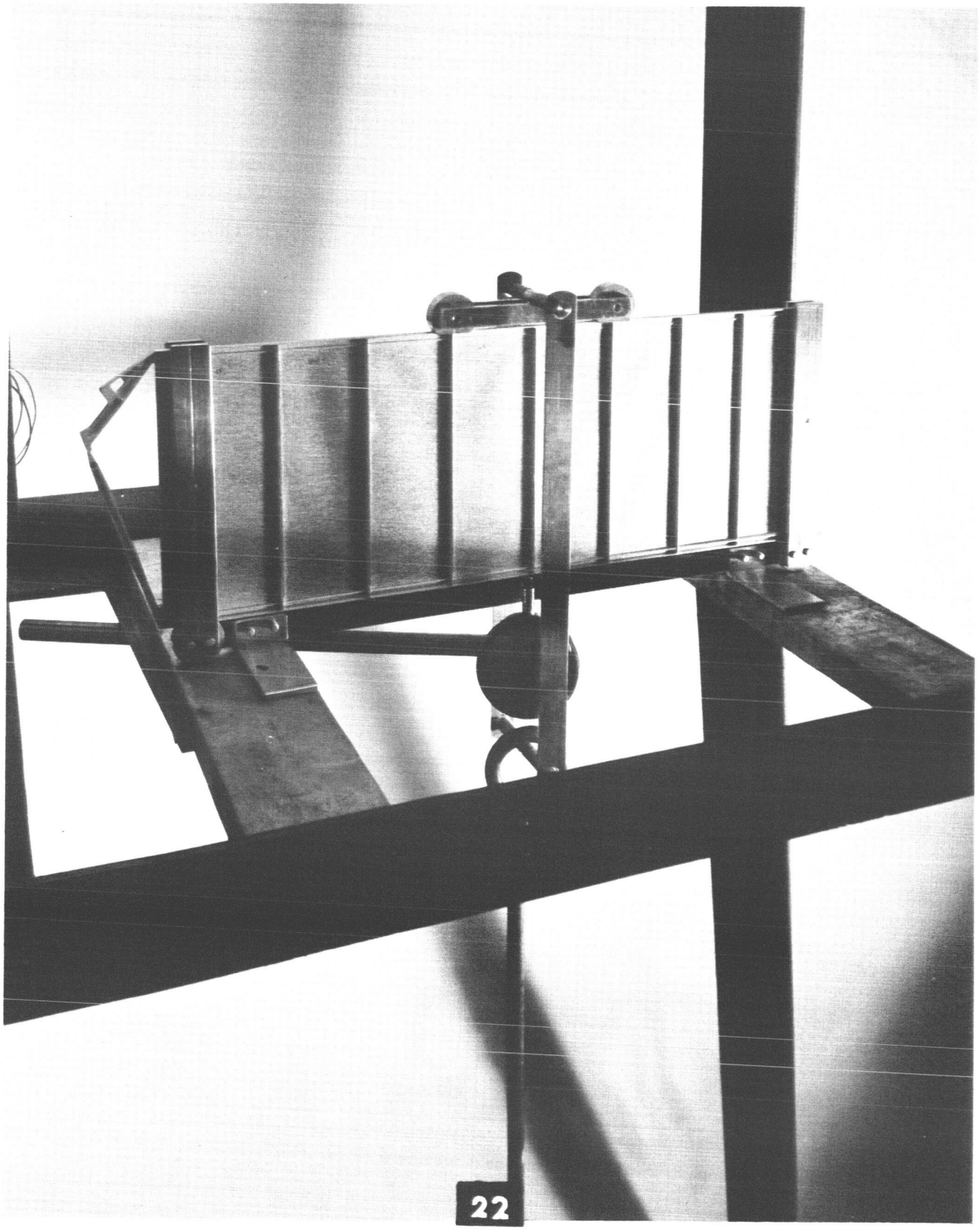







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FEASIBILITY STUDY OF CONSTRUCTING A 1/10 SCALE APOLLO SERVICE MODULE SIMILITUDE MODEL

NASA CONTRACT NAS 9-4567

ADDENDUM

Damping Properties of Aluminum Beams with Various Coatings And Retest of Honeycomb Specimens

MARCH 1966

Submitted to:

NATIONAL AERONAUTICS & SPACE ADMINISTRATION
Manned Spacecraft Center
Houston, Texas

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I. INTRODUCTION

Among the results noted in earlier tests of the original report were the high structural damping values of the one-tenth scale honeycomb beams, as compared with full-scale honeycomb beams. The scaled structures exhibited about an order of magnitude higher damping, though damping should remain unchanged since the materials of construction are the same. Increased effect of acoustic damping in the smaller beams was suggested as a possible factor. Subsequent considerations also suggested the possible influence of stress level and of bonding adhesive characteristics as being of possible influence on the damping characteristics.

As a result of these various considerations, it was decided to conduct tests on various sized specimens at various stress amplitudes and on specimens coated with various adhesives. The specimens would be excited in the fundamental free - free bending mode, with frequency and damping factors observed.

In addition, the original one-tenth and full scale honeycomb beams and corner specimens were retested. A slightly different technique was used in order to minimize acoustic coupling between the one-tenth scale specimens and the acoustic exciter upon removal of excitation, thus reducing the influence of the exciter in supplying extraneous damping. The original full scale honeycomb specimens were retested substituting a mechanical shaker for the acoustic driver in order to obtain higher stress amplitudes.

II. SIZE - EFFECT AND STRESS EFFECT TESTS

Six solid specimens of 6061-T6 Aluminum Alloy were prepared. Configurations are indicated in Table I.

Specimens were excited as described in page 32 of the original report with the following exception in procedure: Instead of placing the acoustic exciter adjacent to the specimen, it was placed about 18 inches below. The acoustic energy was transmitted to the specimen surface through an 18 inch long tube. When it was desired to remove the excitation, the upper end of the tube was suddenly displaced laterally away from the specimen, thus removing essentially all coupling between exciter and specimen.

The use of this procedure had little influence on the damping observed from the heavier specimens, but indicated significantly lower damping in the specimens of lesser area density (lbs. per sq. in. of surface).

The method of excitation allowed the achievement of relatively high stress levels in the high-frequency specimens, but was not capable of creating high stresses in the low-frequency specimens. Specimen No. S2 was particularly difficult to excite due to the low powered driving system.

The six sizes of specimen were first tested at approximately the maximum amplitude attainable. Damping was measured at two decay levels; the first from full amplitude to one-half amplitude; the second from half amplitude to one-quarter amplitude. Results of these tests are given in Table II.

In addition to the above, Specimens No. S1 and S3 were compared through the fourth successive reduction of one-half amplitude, starting at 10,000 psi stress amplitude through to one-sixteenth of original amplitude. Results are given in Table III.

Examination of Tables II and III reveals that damping increases as the stress amplitude increases. This is not to say that the stress, per se, is the cause, since acoustic damping and possibly other effects are related to stress within a given specimen. The magnitude of reduction, from Table III, is about 40% in going from maximum stress amplitude of 10,000 psi to one-sixteenth of that value.

The effects of size and frequency are shown in Figure 1, where the damping factor is plotted against area density. Damping is seen to be lower as specimen area density is increased. Damping is also seen to be lower in the higher frequency specimens. The 40 cps specimens were tested at extremely low stress amplitudes and it is expected as observed in the previous paragraph, raising the stress level would result in increased damping. This would present a curve moved further about the 400 cps curve.

Values for honeycomb beams are also included in Figure 1, the values for one-tenth scale being taken from current test results.

III. INFLUENCE OF VARIOUS COATINGS.

Twelve specimens similar to Specimen S3 (table I) were prepared. These most closely compare dimensionally with the one-tenth scale honeycomb samples described on page 30 of the main report. Weight, frequency and damping of each specimen were measured. Each specimen was then coated with one of the various coatings as given in Table IV. The adhesive applied to specimens 1 thru 4 was spray applied in layers of approximately .5 mil at a time and oven cured. The glass cloth used on specimen 5 was spray impregnated and then dead weighted on the specimen during oven cure. The adhesive on specimen 6 was also dead weighted and oven cured. The adhesives used on specimens 7 thru 12 were hand combed and troweled then oven cured.

The spray application proved most easily accomplished and uniformity of thickness was reasonably controlled. Uniformity in thickness of the hand troweled coatings was least controllable. Weight, frequency and damping were measured again.

The initial level of stress amplitude was set at approximately 10,000 psi prior to removal of excitation. Damping was determined for decay periods from full to one-half amplitude and from one-half to one-quarter amplitude. Test results are given in Table V.

III. INFLUENCE OF VARIOUS COATINGS (continued)

It is noted first from Table V that the coatings caused a very slight drop in resonant frequency. This drop is of the same order as the error in ability to excite each bar exactly at resonance. The consistency in direction indicates, however, that the drop is real and occurs as would be expected.

It is also observed that all but one coating caused an increase in damping in both the first and second half-amplitude decay periods. Specimen 7 shows no change in frequency or damping after coating. This data is questionable. Figure 2 shows the original and final damping for the first and second decay period for each specimen. It also plots the same data in terms of change in damping.

IV. SUPPLEMENTARY TESTS ON ONE-TENTH AND FULL SCALE HONEYCOMB BEAMS

Subsequent to the change in procedure wherein the acoustic coupling between specimen and exciter was reduced, as described earlier, damping measurements were made on the same three one-tenth scale honeycomb sandwich specimens (externally coated with Hexcelite 830 adhesive) numbered 29, 31, and 32 as listed in Table VIII of the original report.

The specimens were excited to a maximum center deflection range of .060 inches. This is equivalent to a stress amplitude of approximately 10,000 psi. Damping factors are listed in Table VI and plotted against stress in Figure 3.

Attempts were made to record damping of the one-tenth scale specimens at low initial amplitudes for comparison with data obtained on the full scale specimens. These were unsuccessful due to the presence of a relatively high electronic background "noise" level compared to the output levels being measured.

The full scale honeycomb sandwich specimens were once again tested with the same acoustic exciter used to drive the one-tenth scale specimens. Maximum attainable center amplitude range was .021 inches for a relative stress level of 350 psi, too low for comparison with one-tenth scale specimens. This data is listed in Table VII.

Recognizing the need for testing at comparable stress amplitude, the three full scale sandwich specimens were then excited using a higher powered driving system, an electromagnetic shaker. A center deflection range of .375 inches was obtained. This is equivalent to a stress level of 6250 psi. Damping factors are listed in Table VIII and plotted against stress in Figure 3.

V. SUPPLEMENTARY TESTS ON ONE-TENTH AND FULL SCALE CORNER SPECIMENS

Acknowledging the adverse affects on the damping characteristics caused by test techniques initially used, it was decided to re-run some of the tests on the corner specimens using techniques as employed in the test of honeycomb sandwich specimens described in the previous section.

The full scale corner specimen was tested as a cantilever with the closeout firmly clamped in the manner described in the original report, but driven with the higher powered electromagnetic shaker. An amplitude range of .23 inches peak to peak was measured at the free end. Results are listed in Table IX.

The one-tenth scale corner specimen was also tested as a cantilever with the closeout firmly clamped in a manner similar to that described in the original report. Excitation was accomplished, however, with the acoustic driver through an 18 inch long tube. The excitation was removed by displacing the tube when damping data was recorded.

The recording indicated a gradual decay rather than a clean departure from full amplitude. This is attributed to the inability to quickly remove the exciting mechanism, allowing some input to the specimen during the decoupling. For this reason the values of g, for full to 1/2 stress in Table X, are not considered valid.

A notable occurrence was the progressive inability of the specimen to reach previously attained frequencies, which suggests a fracture in the specimen. This is reflected in the results obtained, indicating higher damping than would be experienced were the specimen not fractured. Where and when the fracture occurred is unknown, the complete loading history not having been recorded. Original tests on this specimen were made at a natural frequency of 500 c.p.s. Current frequencies of 488, 472, and 418 c.p.s. were obtained, the latter at a stress amplitude slightly higher than that at which damping of the full scale specimen was measured. This data is listed in Table X.

SUMMARY

Data obtained from tests on size - effects specimens indicate a substantial influence of stress level (or amplitude) on damping. This led to the subsequent retesting of the full and one-tenth scale honeycomb specimens at more comparable stress amplitudes. A detailed study of the reasons for the frequency and density effects is beyond the scope of the present project and will, therefore, not be attempted.

Coating the specimens with adhesive causes a slight reduction in resonant frequency and an increase in damping which varies, dependant upon the type adhesive and the thickness applied.

The data obtained implies that the Chemlok 304 adhesive imparts less damping to the specimen than the coatings used on other specimens. However, these tests are not considered sufficiently adequate from which to draw a conclusion on the damping properties of these adhesives. Also, the non-uniformity of adhesive thickness application on Specimens 7 thru 12 may create local regions of increased mass which would influence the damping, producing lower values.

Re-test of the one-tenth scale honeycomb sandwich specimens using improved test procedures, reduced considerably the effect of external influences on the specimens while measuring damping. Also, as realized from the size - effects tests, matching stress levels is important in determining the damping.

Damping ratios, model-to-full scale, were found to be 1.5/1 at 2500 psi stress level, 1/1 at 6000 psi stress level and .76/1 (by extrapolation of the full scale data) at 10,000 psi stress level. The reasons for this variation are not obvious. Suggested causes are:

1. External influences
    a. Aerodynamic damping
    b. Specimen support system
    c. Inaccuracies in setting test conditions
    d. Inaccuracies in taking data

2. Structural influences

   a. Effect of additional adhesive coating on face skins

   b. Variation in the behavior of the face to core adhesive (full scale adhesive is on a cloth carrier).

   c. Behavior of core material (Model shear modulus is low)

   d. Difference between face material used in model versus full scale

Results obtained in re-test of the corner specimens indicates a reduction in damping, model-to-full scale, from 10/1 in the original tests to 6.5/1. This improvement is attributed to the removal of the external damping contributed by the presence of the acoustic driver. Acknowledging that the specimen is suffering, damping values would be further improved in a non-defective assembly.

CONCLUSION

Cause for the high damping values originally obtained in the test of the one-tenth scale honeycomb sandwich specimens has been determined. Primarily, the earlier test methods and conditions introduced deleterious effects to the behaviour of the one-tenth scale specimens. Corrections in procedure, incorporated in subsequent tests, produced a reduction in the ratio of model to full scale damping from more than 10/1 to less than 2/1.

Elimination of the questionable damping properties, coupled with the satisfactory accomplishments of Phase I, leaves little doubt that adequate similitude in a one-tenth scale Apollo Service Module can be accomplished.

TABLE I

SIZE-EFFECT SPECIMENS

Specimen No.	Length (In)	Width (In)	Thickness (In)	Weight (Grams)	Design Frequency (cps)
S1	13.8	1.75	0.383	407.20	400
S2	43.8	5.50	0.383	4074.50	40
S3	8.0	1.00	0.124	42.75	400
S4	25.0	3.25	0.124	426.04	40
S5	5.5	0.70	0.063	10.74	400
S6	17.0	2.25	0.063	102.05	40

TABLE II

TEST RESULTS - VARIOUS BEAM SIZES

Specimen	Frequency (cps)	Maximum Stress (Approximate) (psi)	Damping Factor (g)	
			Full to $\frac{1}{2}$	$\frac{1}{2}$ to $\frac{1}{4}$
S1	417	10,000	.00013	.00013
S2	41.3	300	.00096	.00092
S3	398	10,000	.00038	.00029
S4	40.7	600	.00188	.00171
S5	434	17,000	.00076	.00069
S6	45.7	700	.00322	.00320

TABLE III

DAMPING OF SPECIMENS NO. 1 AND NO. 3

	Damping Factor (g)			
Specimen	Full to 1/2 Stress	1/2 to 1/4	1/4 to 1/8	1/8 to 1/16
S1	.000132	.000132	.000109	.00008
S3	.000384	.000292	.000273	.000222

TABLE IV

SPECIMEN COATINGS

Specimen	Weight Gain (grams)[1]		Density (Estimated) per mil
1	-0.0128[2]	0.5 mil nominal Hexcelite 830	.0000326#/in.^{2}
2	0.0911	1.1 mil nominal Hexcelite 830	"
3	0.2219	1.8 mil nominal Hexcelite 830	"
4	0.3444	2.5 mil nominal Hexcelite 830	"
5	0.1562	1.4 mil nominal Hexcelite 830 Impregnated 1.0 mil glass cloth	.0000295#/in.^{2}
6	0.2367	2.2 mil nominal Scotchweld AF 42 unsupported film	.0000408#/in.^{2}
7	0.0325	.9 mil nominal Chemlok 304	.0000433#/in.^{2}
8	0.3744	2.8 mil nominal Chemlok 304	"
9	0.6146	4.5 mil nominal Chemlok 394	"
10	0.2642	2.0 mil nominal Epibond 1210 and 9615 hardener (50-50 by wt).	.0000346#/in.^{2}
11	0.3496	2.7 mil nominal Epibond 1210 and 9615 hardener (50-50 by wt.)	"
12	0.5133	4.0 mil nominal Epibond 1210 and 9615 hardener (50-50 by wt.)	"

(1) Bare weight ranged from 44.0574 to 44.5706 grams

(2) Weight loss in Specimen 1 may be from cleaning process. Since all specimens were chemically cleaned at the same time, it is assumed a similar loss was experienced throughout.

TABLE V

TEST RESULTS - COATED SPECIMENS

Specimen	Frequency (cps) (1) B	Frequency (cps) (1) A	Damping (g) Full to $\frac{1}{2}$ B	Damping (g) Full to $\frac{1}{2}$ A	Damping (g) $\frac{1}{2}$ to $\frac{1}{4}$ B	Damping (g) $\frac{1}{2}$ to $\frac{1}{4}$ A
1	406	405	.00032	.00037	.00031	.00034
2	406	404	.00034	.00037	.00029	.00033
3	403	402	.00033	.00042	.00029	.00038
4	405	403	.00034	.00048	.00032	.00043
5	405	402	.00032	.00039	.00030	.00033
6	406	403	.00032	.00038	.00031	.00033
7	403	403	.00032	.00032	.00028	.00028
8	403	402	.00031	.00041	.00028	.00037
9	403	401	.00033	.00043	.00030	.00048
10	406	404	.00032	.00040	.00030	.00037
11	406	403	.00032	.00042	.00028	.00039
12	403	401	.00032	.00047	.00029	.00043

(1) B = Before Coating
A = After Coating

TABLE VI

DAMPING OF ONE-TENTH SCALE HONEYCOMB SANDWICH

Specimen	Frequency (cps)	Initial (1) Amplitude (inches) Peak to Peak	Damping Factor (g) Full to 1/2 Stress	1/2 to 1/4	1/4 to 1/8	1/8 to 1/16
29	382	.060	.0079	.0073	.0063	.0045
31	388	.060	.0071	.0065	.0061	.0054
32	411	.060	.0067	.0060	.0056	.0039

(1) Approximate max stress - 10,000 psi.

TABLE VII

DAMPING OF FULL SCALE HONEYCOMB SANDWICH WITH ACOUSTIC EXCITATION

Specimen	Frequency (cps)	Initial Amplitude (inches) Peak to Peak (1)	Damping Factor (g) Full to 1/2 Stress	1/2 to 1/4	1/4 to 1/8	1/8 to 1/16
1	---	.015	.0023	.0022	--	--
2	---	.017	.0029	.0026	.0024	.0023
3	---	.021	.0018	.0016	.0014	.0013

(1) Approximate max stress - 350 psi.

TABLE VIII

DAMPING OF FULL SCALE HONEYCOMB SANDWICH WITH SHAKER EXCITATION

Specimen	Frequency (cps)	Initial Amplitude (inches) Peak to Peak (1)	Damping Factor (g) Full to 1/2 Stress	1/2 to 1/4	1/4 to 1/8	1/8 to 1/16
1	39.5	.375	.0071	.0048	.0031	.0031
2	39.5	.375	.0073	.0048	.0034	.0023
3	39.6	.375	.0065	.0044	.0031	.0020

(1) Approximate max stress - 6250 psi.

TABLE IX

DAMPING OF FULL SCALE HONEYCOMB CORNER SPECIMEN

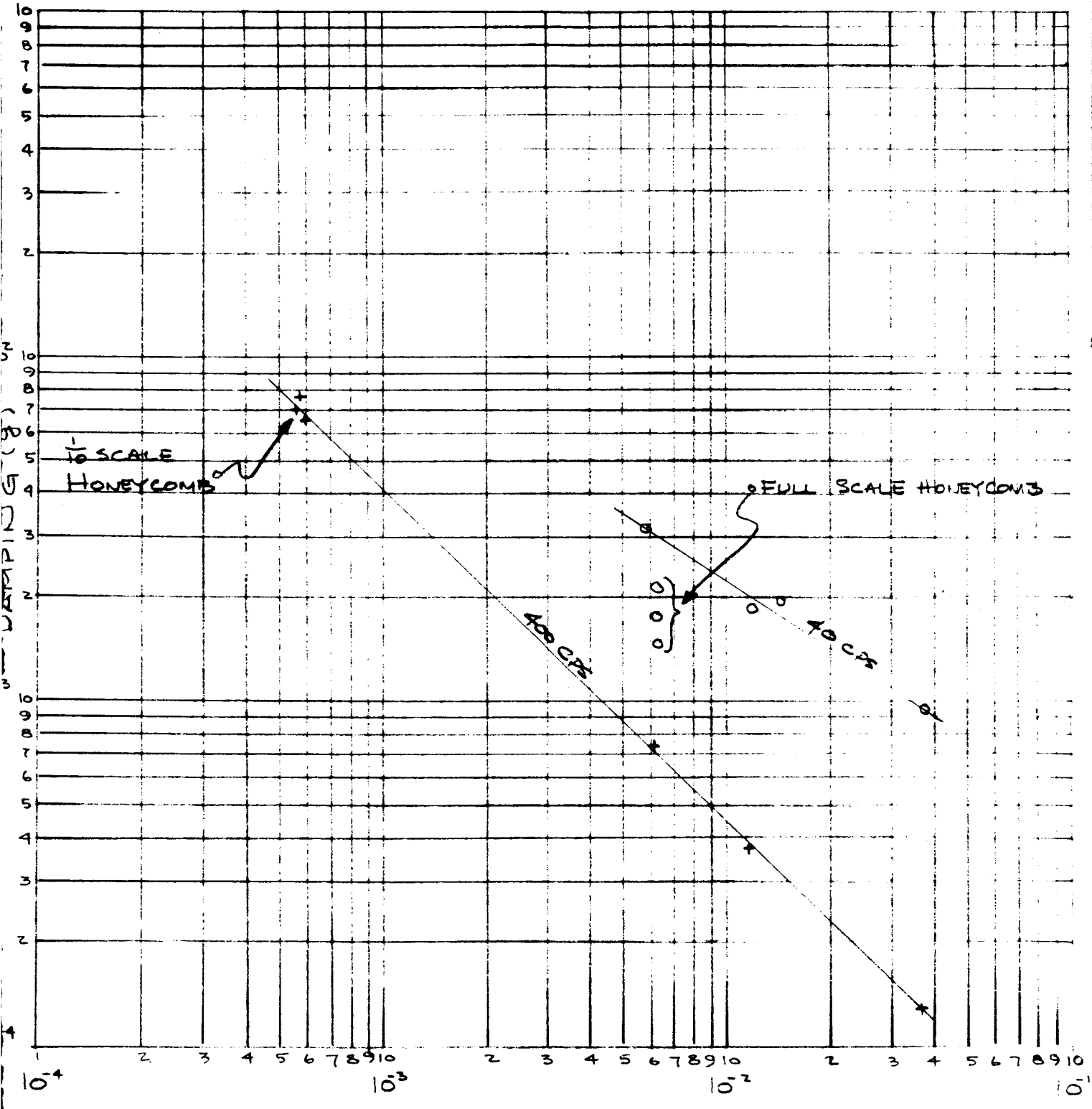
Frequency (cps)	Initial Amplitude (inches) Peak to Peak	Damping Factor (g)			
		Full to 1/2 Stress	1/2 to 1/4	1/4 to 1/8	1/8 to 1/16
54	.23	.011	.0074	.006	.005

TABLE X

DAMPING OF ONE-TENTH SCALE HONEYCOMB CORNER SPECIMEN

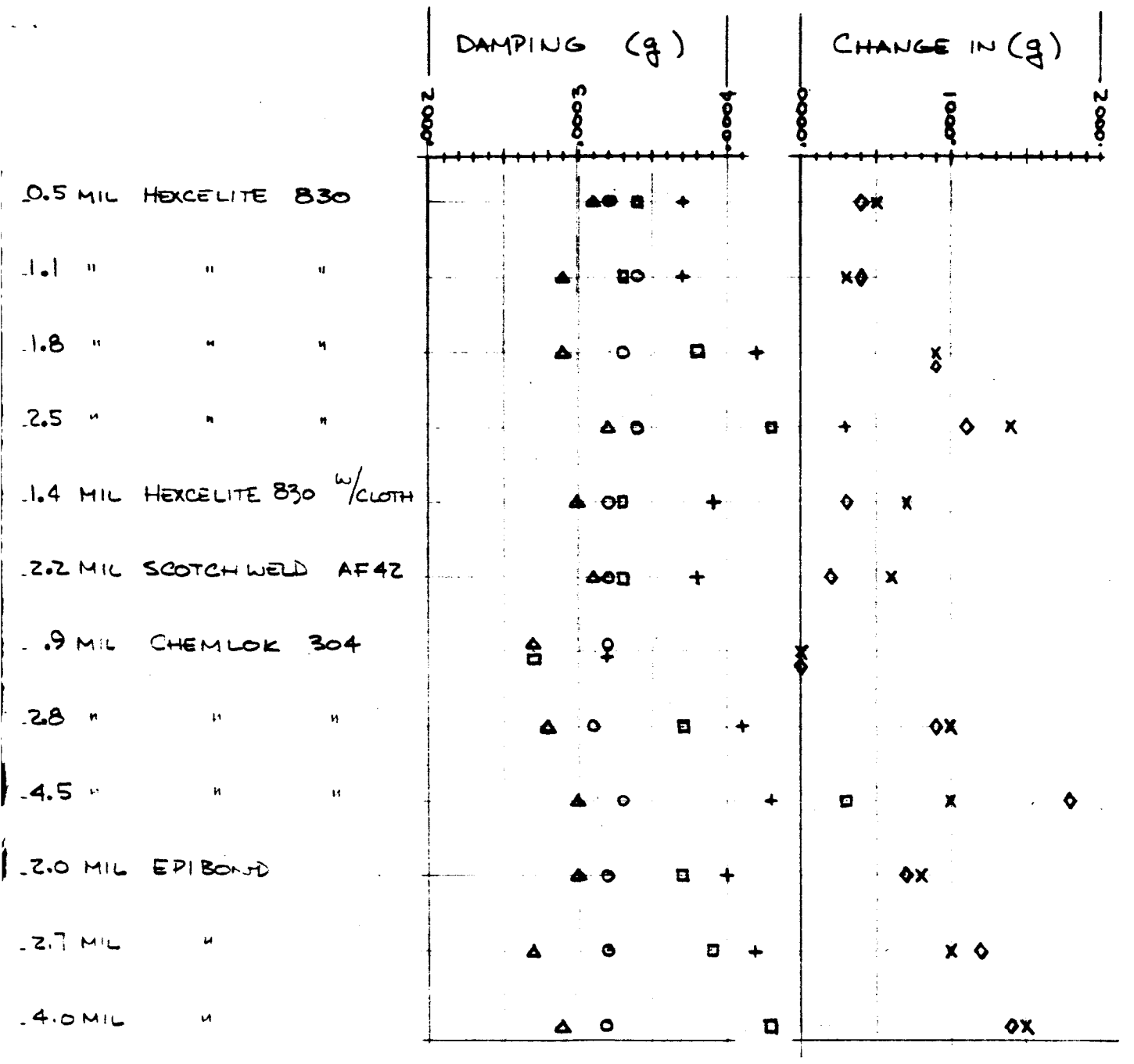
Frequency (cps)	Initial Amplitude (inches) Peak to Peak	Damping Factor (g)			
		Full to 1/2 Stress	1/2 to 1/4	1/4 to 1/8	1/8 to 1/16
418	.025	.015(1)	.048	.044	--
488	.050	.024(1)	.059	.055	.044

(1) These values not considered valid. Specimen subject to partial driving force.



AREA DENSITY (lb./in^2)

DAMPING VS. SPECIMEN DENSITY FIG. #1

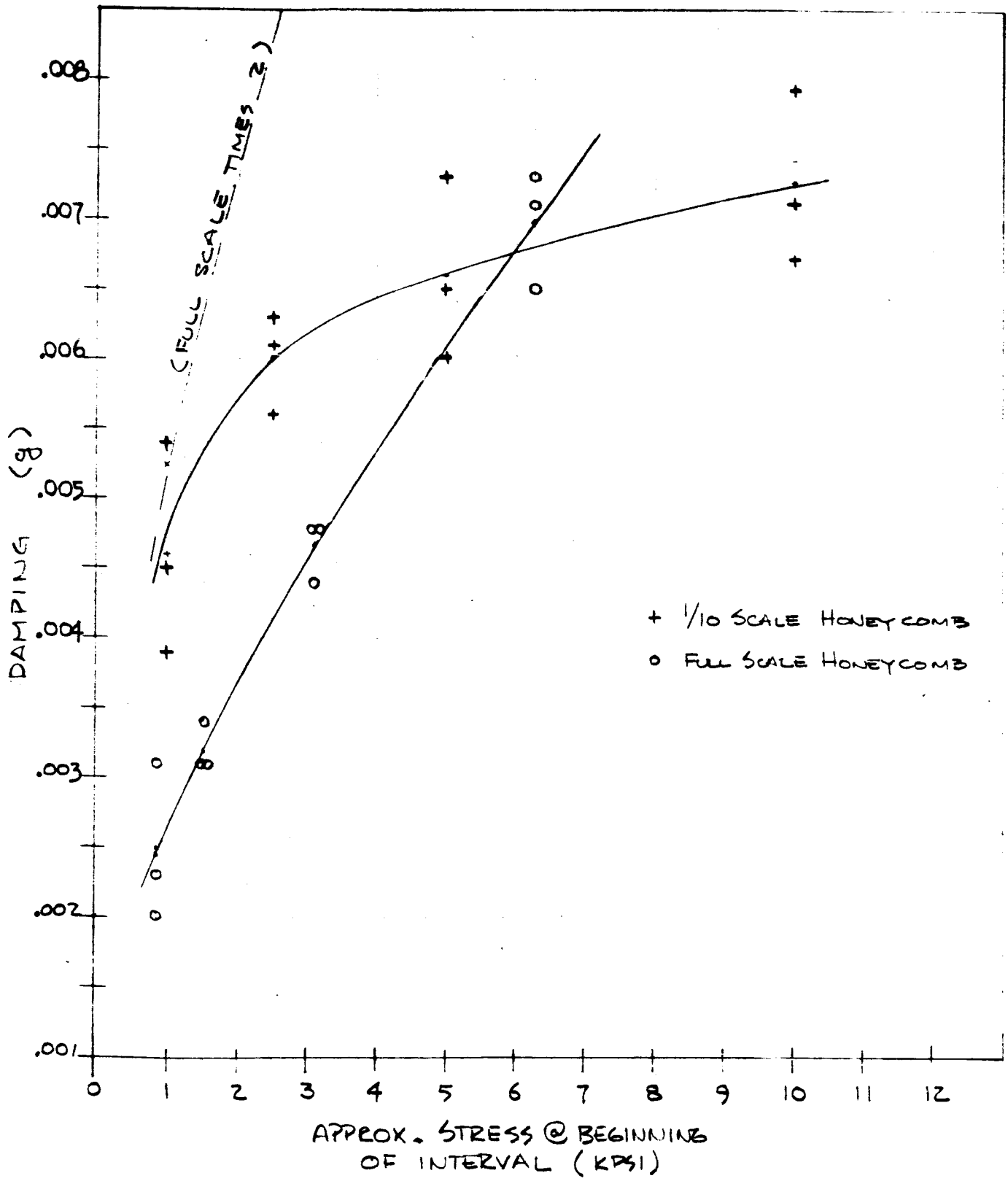


g
- ○ BEFORE COATING (FULL TO ½ AMP.)
- \+ AFTER COATING (FULL TO ½ AMP.)
- △ BEFORE COATING (½ TO ¼ AMP.)
- □ AFTER COATING (½ TO ¼ AMP.)

Δg
- × FULL TO ½ AMP
- ◇ ½ TO ¼ AMP

DAMPING OF COATED SPECIMENS

FIG# 2



DAMPING VS. STRESS LEVEL

FIG.*3