THE EFFECT OF SURFACE ROUGHNESS UPON 25 ST ALUMINUM ALLOY SUBJECTED TO REPEATED TENSILE STRESSES ABOVE THE PROPORTIONAL LIMIT

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SUMMARY

Utilizing the Repeated Load Hydraulic Testing Machine at the Daniel Guggenheim Aeronautical Laboratory, California Institute of Technology, Pasadena, California, the author, in collaboration with Lt. Comdr. D. J. Hardy, U. S. Navy, investigated the effects of surface roughness upon the cyclic life of 25 ST aluminum alloy when subjected to repeated constant tensile stresses in the region above the proportional limit.

The stress impulses are of such low frequency as to allow consideration of single impulses. The rate of build-up of the impulse, and the duration of the impulsive load are such as to create an equivalent static load of substantially the same magnitude as the peak of the impulse loading.

It was found that surface roughness has some effect upon the cyclic life. In the lower stress regions, the greater the degree of surface roughness, the shorter the life appears to be. However, for the range of roughness investigated, 5μ to 200μ, the effect is not so pronounced as is usually found below the proportional limit.

Where the applied stresses reached far up into the plastic range the effect of surface roughness does not seem to follow quite as specific a pattern. Since the loading impulse featured a 0.33 second duration of maximum load, the effects of creep may well have taken over in shaping the life cycle curve with little regard for surface roughness.

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I. INTRODUCTION

This investigation seeks to carry further, in a definite direction, the immense task of completely determining the effects of repeated tensile stresses upon aluminum alloy material.

Certain structural members, more commonly found in the aircraft industry, are subjected to tensile stresses applied many times during the desired life of the part, but which members are commonly designed to operate within the proportional limit. If definite criteria could be built up such that the member can be designed very close to the proportional, or fatigue limit, as circumstances warrant, with the knowledge that a definite number of overstresses of given magnitude above the proportional limit can be accepted without failure or undue permanent deformation, then the savings in weight and cost are obvious. The first step, carried out here, is to investigate the region above the proportional limit for 25 ST aluminum alloy subjected to repeated tensile loads of constant magnitudes. The variable is surface roughness in the range of 5μ to 200μ. The extensions of this investigation must proceed in several directions. At least one other aluminum alloy should be studied before a general statement might be considered safe. Then the problem of varying the magnitudes of the stresses on a single specimen must be investigated. Finally, other types of stressing must be brought in, such as combined bending and tension.

The first of a series of steps toward amassing useful data on this subject was the design of a testing machine. This work was carried out by Lieut. Comdrs. Edward G. Bull and Robert L. Mastin, U.S. Navy, and reported on in their thesis "Repeated Loads Above the Proportional Limit on 24 ST Aluminum Alloy," C.I.T. 1947.

The work was carried forward by Captain Conrad N. Nelson, U.S. Air Force, as reported in his thesis, of the same title, C.I.T. 1948. The results showed that permanent deformation caused by overstressing could not be used for forecasting the life expectancy. It was also found by Nelson that "rest periods" during cyclic load application had an effect on the life cycle, as did initial stresses and magnitudes of overstresses applied. However, quantitative results could not be derived in the time available.

The results of the presently reported investigation showed that surface roughness was a factor in the life expectancy of similar specimens at the same loading, at least in the region just above the proportional limit.

Since even the slightest amount of bending coupled with the pure tension causes a pronounced drop in the cyclic life, these results are applicable only to members with freely hinged ends.

All work was carried out by the author, working with Lt. Comdr. D. J. Hardy, U.S. Navy, as partial fulfillment of the requirements for the Degree of Aeronautical Engineer, at the Daniel Guggenheim Aeronautical Laboratory, California Institute of Technology, Pasadena, California, during the Academic Year, 1948-1949.

II. EQUIPMENT

Test Specimens

The material used for all tests came from a 25 ST forging, with the following properties:

Yield Strength -- 39,400 - 41,250 p.s.i.

Tensile Strength -- 58,000 - 61,396 p.s.i.

% Elongation in 2 inches -- 16½ - 17

Chemical Composition %: (Remainder Aluminum)

Cu	Si	Mg	Fe	Mn	Zn	Cr
4.43	0.67	0.016	0.45	0.73	0.25	0.02

The test specimens were machined to the shape and dimensions shown in Fig. 1. Note here that upon the recommendations of previous investigators, (Ref. 1), the fillets were made 3/8" radius, instead of the 3/16" radius used in previous investigations. Machining and application of the surface roughness were carried out in the C.I.T. Machine Shop. Surface roughness was applied by circumferential grooving to give a constant mean diameter, but with ridges of 5μ, 50μ, 100μ, and 200μ.

A round tool, radius 3/64", was used on a Pratt & Whitney 13-inch lathe, Model B. The advance used for the grooving was as follows:

Roughness	Advance
5 μ	0.0012 in/rev
50 μ	0.0070 in/rev
100 μ	0.0100 in/rev
200 μ	0.0143 in/rev

The roughness was checked on a Profilometer built by Physicists

Research Company.

<u>Testing Machine</u>

This machine was designed and built in 1946-47 at C.I.T. by Lieut. Comdrs. Soli, Bull, and Mastin, and Lieut. Ditch, all of the U.S. Navy. (Ref. 1). It was subsequently modified by Mr. Chintakindi V. JogaRao and Captain Nelson, U.S. Air Force, to stiffen the test platform, (Ref. 2). Further modifications which will be indicated herein were made by the author and Lieut. Comdr. D. J. Hardy, U.S. Navy.

An aircraft hydraulic cylinder applies a tensile load through a universal joint and load coupon, (Fig. 2), located between its piston and the test specimen. The other end of the test specimen is secured through another universal joint to the frame of the test platform.

Figs. 3 and 4 are photographs of the testing machine.

The hydraulic cylinder is actuated periodically by a Vickers solenoid acting on a sleeve valve in the pressure line to the cylinder. The solenoid is operated through contact points opened and closed by a cam driven by a 1/20 HP universal-wound 110-volt a.c. electric motor. The same motor operates a mechanical counter which records the number of cycles of load application. Since there are two complete working curves cut on the single cam, the recorder, operating on the cam shaft, will record exactly half the actual stress cycles applied.

The hydraulic system starts at a reservoir with filler strainer, (Figs. 5, 6, 7, 8, 11), which supplies fluid to a positive displacement gear pump driven through a step-up reduction gear of 3.06 to 1 ratio by a 5 HP 220 volt a.c. electric motor, rated at 1140 RPM. An accumulator, strainer and pressure regulating valve are in the line. Pressures up to 1000 p.s.i. are claimed by the designers. However, no occasion to use more than 500 p.s.i. was experienced in the present investigation. A pressure-relief valve is installed and set for 1250 p.s.i. The effective piston diameter is 11.5 sq. in. Hence 500 p.s.i. will apply 5,750 pounds on the specimen. With cross-section area of 0.0707 sq. in. for the test section of the specimen, this corresponds to about 81,400 p.s.i.

A Bourdon hydraulic pressure gage is installed in the line just ahead of the solenoid-operated valve. A one-way valve prevents rapid drop of pressure from **injuring** the gage at the instant of load application. As will be discussed presently, this gage gives the coarse setting of load, but is not used for the accurate determination.

Everything from Military Specification Hydraulic Oil to third rate automobile crankcase oil was used in the system, with no failures attributable to the type of fluid.

The test platform is essentially a pair of 5" steel H-beams, six feet long, bolted together. Upon the beams are mounted heavy steel fittings to anchor the hydraulic cylinder and the fixed end of the test specimen. Obviously the length of the entire machine could have been halved by mounting the oil reservoir and accumulator adjacent to the main motor, rather than in line with the hydraulic pump. As indicated in Fig. 5 everything is mounted below the table top except the specimen, its fittings, the hydraulic cylinder, the electric controls, pressure gage, counter, and micro-switch cut-out system. All other hydraulic lines and fittings as well as the main motor and its shaft chain are below the table top. This is especially fortunate in keeping the constant oil leaks from damaging the electric system as well as giving a clean space for recording and changing specimens.

The universal joints at either end of the specimen carry counterweights to statically balance them. It was found necessary for the present investigators to install guides for these balances since there was a definite tendency for them to rotate the universal joints, resulting in jamming of the system in addition to inadvertently actuating the cut-out switch. These guides have been made very loose to allow movement of the weights both axially and a few degrees of rotation. This was necessary to allow movement of the universal joints when changing specimens. However, since the weight will seldom move from its position vertically above the specimen axis, the guides are primarily a safety feature.

Since reworking of the hydraulic system resulted in a set-up which would hold constant load to a remarkable accuracy, it was found possible to leave the test in operation for extended periods of time with assurance that the load would not drop off. Hence, the writers were able to carry out much more testing than would have been possible had their presence been constantly required as in previous work. This feature made necessary the installation of some sort of cutoff so that upon fracture of the specimen, the motors would stop, especially the counter-motor. Since the hydraulic piston is ordinarily operated with its free end about one inch outside the cylinder, and since it will be pulled completely back to the cylinder upon release of the load (i.e. fracture of the specimen), it was possible to use this return feature of the piston as the cut-out actuator. Fortunately, the piston has a collar raised about one-quarter inch from its circumference. Thus it was possible to install a micro-switch next to this collar, so that as the piston returned toward the cylinder after fracture of the specimen, the collar would strike the actuating arm of the micro-switch. The micro-switch was modified from a "normally closed" to a "normally open" type because there was none available of the type desired. This micro-switch was in the circuit with the solenoid motor and was led to a three-pole, double-throw relay. The relay was in turn connected to the counter circuit, the solenoid circuit, and the main motor cut-off switch. Thus when the micro-switch was actuated it opened the circuit which energized the relay solenoid, thereby dropping the solenoid plunger, opening all circuits to shut down all operation. The stopping of the counter motor at the time of fracture left a record which, of course, was the essence of the entire test. Fig. 12 diagrams this electrical circuit.

Load Measuring Coupon

It was mentioned above that the hydraulic pressure gage offered a coarse means of setting a definite load upon the specimen. However, the means of accurately obtaining readings of the actual load being applied, was through SR-4 resistance wire strain gages. Four of these gages are mounted at ninety degree spacing on a steel sleeve, called the "load coupon", (Fig. 2). This coupon is mounted between the hydraulic piston and the specimen. Knowing the cross-sectional area of the coupon and that of the test specimen, a correspondence can be set up between the strain of the coupon and the stress applied to the specimen. This is done by comparison as described below.

Load Measuring Equipment

After the strain gages are cemented onto the coupon and checked, the coupon is placed in any standard tension testing machine and the gage readings recorded by galvanometer, as known loads are applied. Thus, knowing the cross-sectional area of the coupon, readings on the galvanometer can be translated directly into load in pounds or into p.s.i. on the test specimen. During the calibration run it is possible to note that all gages are performing correctly and that their readings can be averaged by putting them in series and applying a factor of four. Thus all effects of bending are taken out. Table I and Fig. 15 detail this calibration.

However, since a galvanometer would be useless for measuring loads which revert to zero 52 times each minute, a comparison system is used during testing. A control board and amplifying system are provided. See electrical diagram, Fig. 13. Behind the control board a selector can connect to the recording system any one of four sets of resistances. These four resistances correspond to applying 1000, 2000, 3000, or 4000 pounds to the load coupon. The installation of these resistances can be made while calibrating the load coupon on a tension testing machine. With 1000 pounds applied load, the average strain gage reading is recorded. Then enough resistance is put into the selector system to give the same identical electrical reading. Similarly the resistances are set up for 2000, 3000, and 4000 pound loads. Thereafter, during testing, any reading of the load coupon strain gages can be compared to these standard values to determine its magnitude, in pounds of load.

The reason for this method is that while the test is in progress the most feasible system found for reading the strain gages was to use a Heiland Recording Oscilloscope which makes a photographic record of electric resistance against time, using an amplifier to get reasonable accuracy. Then, by running the known electrical impulses for 1000, 2000, 3000, and 4000 pound loads through the same circuit, the actual load line can be compared to them by direct measurement, and the actual load ascertained. Thus the comparison is unaffected by fluctuations in supply current or temperature of the amplifying circuit, since the strain gage readings and the comparison loads run through the same circuit.

An example of a typical oscillograph recording is shown in Fig. 8. From it the following information is obtained.

Duration of zero load	0.63 sec.
Duration of Maximum Load	0.33 sec.
Time - No Load to Full Load	0.14 sec.
Time - Full Load to No Load	0.025 sec.
Time for one complete cycle	1.125 sec.
Number of cycles per minute	52
Maximum Rate of Loading	41,700#/sec.
Maximum Rate of Unloading	184,000#/sec.

This information was used in preliminary analyses as will appear later herein. However, during actual testing only the magnitude of the load was required.

III. TEST PROCEDURE

In Fig. 9 a typical section of film is shown. The first three sections are the result of passing 1000, 2000, and 3000 pound equivalent electric loads across the screen. The last section, photographed immediately afterward, is the result of passing the actual electric load on the strain gages across the screen. To analyze this reading, draw base lines and measure with dividers the heights. In this case the calibration lines are:

1000 --- 0.32"

2000 --- 0.65"

3000 --- 0.96"

Thus it appears that 0.32" closely corresponds to 1000 pounds load. Accuracy to 0.01" is all that can be expected due to widths of recording lines and the development of the film. This maximum accuracy can be best achieved by using at least three calibration loads as was done here.

The height of the load line is 0.84". In actual testing, two or three of these loads would be photographed in succession. They would be found to be of identical height almost invariably.

Thus, by comparison, the load was:

$$\frac{0.84}{0.32} \times 1000 = 2625 \text{ pounds}$$

For specimen section of 0.0707 in^2 the tensile stress is then

$$\frac{2625}{.0707} = 37{,}130 \text{ p.s.i.}$$

A perfect test run would show periodic readings which, although both calibration and load lines varied with temperature, would, for each reading, calculate out to 37,130 p.s.i. tensile stress. Unfortunately the hydraulic system does not keep the load perfectly constant. Since it is impossible to photograph, develop, dry the film, and read it, without considerable time ensuing, errors in load may continue for long periods of time before correction. However, these load variations are not great as evidenced from a typical record sheet, Fig. 10.

IV. DISCUSSION

Although the work herein is conducted in the region of stresses which lies above the proportional limit, the metal may still be considered an elastic body and as such it is necessary to examine the testing sequence applied with a view toward determining the effects of vibrations which may be excited.

Looking first at the breakdown of load vs time as portrayed by oscillograph recordings, Fig. 8, it can be seen that the frequency of load application is 52 cycles per minute, or 0.867 cycles per second.

To compute the natural frequency of vibration of the test section of the test specimen, in the longitudinal mode, it can be assumed that the test section acts as though clamped at the ends. Referring to Den Hartog's text, Ref. 3, in his Appendix II the natural frequencies are found from the formula:

$$f = \frac{1}{2}\sqrt{\frac{E}{m' l^2}} \qquad \text{where} \quad \left\{ \begin{array}{rl} f = & \text{fundamental natural frequency, cycles/second} \\ m' = & \text{mass/unit vol.} \\ & 0.101/386 \ \#\text{sec}^2/\text{in}^4 \\ l = & \text{length, 2 in.} \\ E = & \text{Mod. of Elast.} \\ & 10{,}300{,}000 \text{ p.s.i.} \end{array} \right.$$

$$f = \frac{1}{2}\sqrt{\frac{10{,}300{,}000\ (386)}{(0.101)(2)^2}} = 49{,}500 \text{ cycles/sec.}$$

The natural period is then $T = \frac{1}{f} = 2.02 \times 10^{-5}$ sec. Higher modes will, of course, give smaller periods.

Now, since there is an unloaded time of 0.63 seconds between impulses, the system can complete $0.63/2.02 \times 10^{-5} = 31,200$ natural periods before the next impulse begins. Thus all vibration will be damped out between cycles and the system can be considered as subjected to isolated impulses, with no effect due to the periodicity of the loading.

The effects on simple elastic systems of various forms of impulsive loads have been organized in a paper by Dr. J. M. Frankland, Ref. 4. In order to apply his conclusions to an elastic system, several conditions must be fulfilled, thereby allowing the system to be treated as having one degree of freedom. These conditions are:

(a) The duration of impact must be sufficiently long so that there are no complications due to stress waves and other phenomena foreign to the system of one degree of freedom. Dr. Frankland suggests that the impulse should last at least a tenth of the fundamental natural period of the system. Obviously, this condition is fulfilled by the system under consideration herein.

(b) The impact load should be distributed fairly uniformly over the structure. Since the load is transmitted to the test section of the specimen through the homogeneity of the material, a uniform load distribution is closely approximated here.

(c) The fundamental mode of the structure is uncoupled with higher modes. In the case under observation, the fundamental mode in question is longitudinal and may be considered uncoupled with higher modes.

For such idealized systems, Dr. Frankland offers approximate, as well as exact solutions for the equivalent static load impressed upon the system. For a type of impulse which is of uniform magnitude and is long in comparison to the natural period, as in this case, the important parameter is the rate of build-up of the impulse. Using the following nomenclature, the relation below applies:

e.s.l. = equivalent static load

n = dynamic load factor = $\dfrac{\text{e.s.l.}}{\text{impulse peak load}}$

p = circular natural frequency

t_o = time required for build-up of impulse

Formula: $n = 1 + \dfrac{2}{pt_o} \sin \dfrac{pt_o}{2}$

In the system under consideration, $pt_o = 43{,}500$ and thus the second factor closely approaches zero. Hence, the equivalent static load may be taken as identical to the peak of the impulse loading.

For a graphical method of obtaining the equivalent static load due to an impulse of any form, a paper by Dr. G. E. Hudson, Ref. 5, is recommended.

From the foregoing analysis it can be concluded that the system is actually subjected to the stresses set up by the loads as indicated by reading the strain gage loads as previously described under "Test

Procedure". However, in any future attempt to compare impulsive loading tests made by different types of impulses, the equivalent static loading must be carefully computed, since that factor is, after all, the determining factor for the actual stresses induced. For impulses of duration close to the natural period of the system, the equivalent static load may approach twice the impressed load. Furthermore, it is obvious that the frequency of applying the impulses must be investigated for approaching resonance.

One other parameter must be mentioned when dealing with the region above the proportional limit. "Creep" is a definite function of time. In the type of loading applied here the full magnitude of applied force endures for an appreciable period. When this time is added for the relatively large numbers of cycles applied in these tests the deformation operates through Poisson's ratio to reduce the cross-sectional area progressively. Thus, for constant loading the induced stresses progressively increase. This factor has not been considered herein. However, it is again worthy of mention that the duration of load application, as well as rapidity of build-up and release of load, must be weighed when attempting to correlate these results with those obtained for identical magnitudes of loadings but with different types of impulses.

It might be worthy of comment that in the foregoing computations it was not necessary to strive for great accuracy since it was obvious from the start that the natural frequency might be considerably different and yet not alter the dynamic load factor from unity.

V. RESULTS

Tables III through LVI indicate the degree of accuracy achieved in attempting to hold a constant load during a test. It is believed that all tests were held to sufficient tolerances to justify plotting all results on the appropriate curves. However, many other tests were started but not completed for a variety of reasons.

Many of the tests involve prolonged periods between readings for checking the load. In cases where the reading following the interval showed that the load was as desired, it appeared reasonable to assume that constant loading had prevailed.

It was thought that there might be some weakness exemplified by a fillet break, and hence the type of break was noted in every case, and was specified on the plot. However, fillet breaks did not change appreciably the scatter of the test result points.

Figs. 16, 17, 18, and 19 are the plots of all test results for the roughness factors investigated. Fig. 20 shows the effect of roughness by comparison of the replotted curves.

Although scatter made it difficult to be too specific regarding the position of a curve, it is felt that a definite tendency toward decreasing the life of a specimen is revealed as roughness increases. Although this result appears, from Fig. 20, to be quite general, the author feels safe only in applying this statement to the relatively low stress region, just above the proportional limit.

Note that there is too much scatter in the higher stress regions to permit making a definite determination of where the curves lie. The 200 μ Roughness tests gave a very good grouping of points in a region which overlapped the curves of less roughness, as can be seen from Fig. 19. This apparent reversal of expected results is another factor which influences the author to refrain from drawing conclusions concerning the effect of roughness in the higher stress regions.

VI. CONCLUSIONS

For 25 ST aluminum alloy, surface roughness affects the life expectancy of a member subjected to repeated tensile stresses of magnitudes greater than the proportional limit. In the range of 5μ to 200μ, each step of increasing roughness results in a decrease in cyclic life over a range of loadings embracing the proportional limit and on up to about 42,500 p.s.i. For higher stresses, the effect is in general the same, but specific statements cannot be made without further testing.

The degree of shortening in cyclic life appears rather small when compared with similar tests below the proportional limit. With the scatter as broad as it appears herein it might be concluded that the expense incident to reducing roughness during finishing operations on structural parts is possibly not justified if carried out for the purpose of increasing life expectancy above the proportional limit for this type of loading.

VII. RECOMMENDATIONS

It is not expected that other aluminum alloy structural materials will exhibit different results than the 25 ST tested here. However, in the interests of completeness, this work might well be extended to cover those materials in common use in industry.

Since few aircraft members are subjected to such constant loadings as were applied during these tests, the effects of varying stresses should be investigated. Captain Nelson, in Ref. 2, did a slight amount of work in this direction. However, some definite statistical pattern is mandatory.

In regard to the equipment, several points are worthy of mention. The rate of load application, 52 cycles per minute, could be increased many fold without increasing the dynamic load factor. Moreover, it would appear that if the loading was compared on a basis of equivalent static loading, the results should be irrespective of the manner of achieving this load. That is, if the load of a definite magnitude is applied directly, or if half that load is applied in such a manner as to give a dynamic load factor of two, the stresses incurred should be identical. Hence, redesign of the machine to increase greatly the cycles per minute seems justifiable. However, as pointed out in the "Discussion" section herein, the duration of load would thereby be reduced so that a greater cyclic life might be expected.

Captain Nelson, Ref. 2, investigated the use of an Oscilloscope with retentive screen for load measurement, and found that a rate of loading of at least 600 cycles per minute would be the minimum that could be so measured, and that even then, the mean load, rather than peak load would come out. As an alternative method of reducing the excessive workload of the present system it is suggested that precision type pressure-control valves might allow load control without any resort to strain gages.

At the very least, the present system could be vastly improved by eliminating the storage batteries. Checking, watering, charging, and moving of these batteries occupies more time than is justified.

REFERENCES

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TABLE 1

Calibration of Strain Gages

(Connected in Series)

Reading	Load (lbs.)	Millivolts
1	100	.310
2	200	.615
3	300	.930
4	400	1.22
5	500	1.55
6	600	1.85
7	700	2.15
8	800	2.49
9	900	2.78
10	1000	3.10
11	1100	3.41
12	1200	3.73
13	1300	4.03
14	1400	4.35
15	1500	4.68
16	1600	4.98
17	1700	5.29
18	1800	5.62
19	1900	5.93
20	2000	6.21
21	2100	6.56
22	2200	6.83

TABLE I (Cont'd)

Reading	Load (lbs.)	Millivolts
23	2300	7.19
24	2400	7.50
25	2500	7.82
26	2600	8.13
27	2700	8.44
28	2800	8.76
29	2900	9.09
30	3000	9.36

TABLE II

Static Tensile Test

25 ST 5μ Surface Roughness

Throop Hall--Materials Testing Lab.

Specimen Diameter 0.3" Area: 0.0707 sq. in.

Load lbs.	#79 Gage Rdg.	#79 Strain Rdg.	#80 Gage Rdg.	#80 Strain in/in	Average Strain in/in	Stress p.s.i.
0	0	0	0	0	0	0
300	2.0	3.05×10^{-4}	2.5	3.905×10^{-4}	3.477×10^{-4}	4243
600	5.3	8.082	4.5	7.029	7.555	8486
900	8.0	12.2	7.6	11.871	12.035	12729
1200	10.4	15.86	10.6	16.557	16.208	16972
1500	13.0	19.825	13.3	20.775	20.300	21215
1800	15.8	24.095	16.3	25.460	24.777	25460
2100	19.6	29.89	20.0	31.240	30.565	29701
2400	28.2	43.00	28.3	44.205	43.602	33945
2560	42.0	64.05	42.0	65.604	64.827	36209
2700	45.2	68.93	46.0	71.852	70.391	38189
2800	75.0	114.37	78.0	121.84	118.11	39604
2930	85.2	129.93	86.6	138.39	134.16	41442
3000	92.3	140.76	95.0	148.39	144.58	42430
3100	102.9	156.92	98.5	153.86	155.39	43847

#79 --- 1.525×10^{-4} in/in/division

#80 --- 1.562×10^{-4} in/in/division

TABLE III

Test 1

Approx. Gage Setting 200 p.s.i.

Reading	Cycles	Load (lbs.)
1	30	2285
2	100	2357
3	1500	2143
4	2000	2571
5	2400	2500
6	2600	2571
7	2800	2500
8	3000	2571
9	4000	2340
10	4500	2270
11	5000	2360
12	5500	2285
13	6900	2350
14	8240	2410
15	10000	2571
16	70000	2350
17	74200	2515
18	76200	2570
19	262204	Failure

Roughness - 5μ

Break - Fillet

Ave. Load - 2350 lbs.

Stress - 33,000 p.s.i.

TABLE IV

Test 2 — Approx. Gage Setting 220 p.s.i.

Reading	Cycles	Load (lbs)
1	100	2670
2	1000	2610
3	2520	2880
4	3000	2720
5	71050	2620
6	71200	2760
7	75000	2760
8	83600	No reading
9	105000	Failure

Roughness - 5μ

Break - Fillet

Ave. Load - 2700 lbs.

Stress - 38,200 p.s.i.

TABLE V

Test 3 — Approx. Gage Setting 240 p.s.i.

Reading	Cycles	Load (lbs.)
1	40	2960
2	3000	2950
3	8000	2970
4	12400	2750
5	16900	2850
6	56008	Failure

Roughness - 5μ

Break - Fillet

Ave. Load - 2900 lbs.

Stress - 41,000 p.s.i.

TABLE VI

Test 4 — Approx. Gage Setting 260 p.s.i.

Reading	Cycles	Load (lbs.)
1	40	3110
2	2500	3160
3	7600	2960
4	11760	No reading
5	13800	3170
6	20850	3100
7	21720	3020
8	22972	Failure

Roughness - 5μ

Break - Normal

Ave. Load - 3100 lbs.

Stress - 43,800 p.s.i.

TABLE VII

Test 24 — Approx. Gage Setting 260 p.s.i.

Reading	Cycles	Load (lbs.)
1	3000	3240
2	7900	3280
3	7920	3250
4	13100	3250
5	13200	3250
6	18000	3330
7	18060	3270
8	18374	Failure

Roughness - 5μ

Break - Fillet

Ave. Load - 3250 lbs.

Stress - 46,000 p.s.i.

TABLE VIII

Test 26 Approx. Gage Setting 270 p.s.i.

Reading	Cycles	Load (lbs.)
1	3000	3480
2	7680	3500
3	7700	3470
4	9900	3500
5	9920	3500
6	21002	Failure

Roughness - 5μ

Break - Fillet

Ave. Load - 3485 lbs.

Stress - 49,300 p.s.i.

TABLE IX

Test 5 Approx. Gage Setting 280 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	3240
2	7780	No reading
3	7812	3420
4	9220	3390
5	11600	No reading
6	11660	3360
7	17660	3380
8	Machine broke down	

Roughness - 5μ

TABLE X

Test 6 — Approx. Gage Setting 280 p.s.i.

Reading	Cycles	Load (lbs.)
1	20	3260
2	1060	3470
3	1120	3560
4	1620	3520
5	1650	3440
6	Machine broke down at 4000 cycles	

Roughness - 5 μ

TABLE XI

Test 22 — Approx. Gage Setting 280 p.s.i.

Reading	Cycles	Load (lbs.)
1	30	3260
2	1930	3560
3	5420	3500
4	9820	3620
5	13020	3560
6	16280	3580
7	18870	Failure

Roughness - 5 μ

Break - Fillet

Ave. Load - 3360 lbs.

Stress - 47,500 p.s.i.

TABLE XII

Test 13 Approx. Gage Setting 280 p.s.i.

Reading	Cycles	Load (lbs.)
1	20	3460
2	7540	3390
3	7560	3570
4	9400	3570
5	9440	3500
6	13460	3570
7	13490	3570
8	14564	Failure

Roughness - 5μ

Break - Normal

Ave. Load - 3520 lbs.

Stress - 49,780 p.s.i.

TABLE XIII

Test 25 Approx. Gage Setting 290 p.s.i.

Reading	Cycles	Load (lbs.)
1	80	No reading
2	600	3550
3	3420	3590
4	7500	3580
5	10340	No reading
6	10540	No reading
7	11240	3510
8	11260	3590
9	13700	Failure

Roughness - 5μ

Break - Normal

Ave. Load - 3560 lbs.

Stress - 50,300 p.s.i.

TABLE XIV

Test 7 | Approx. Gage Setting 300 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	3440
2	3280	3540
3	3320	3540
4	3400	3540
5	10750	3660
6	10800	3730
7	12750	No reading
8	12800	No reading
9	12980	No reading
10	13430	3670
11	19326	Failure

Roughness - 5μ

Break - Normal

Ave. Load - 3590 lbs.

Stress - 50,700 p.s.i.

TABLE XV

Test 23 Approx. Gage Setting 300 p.s.i.

Reading	Cycles	Load (lbs.)
1	3000	3640
2	7300	3830
3	9580	3450
4	9600	3610
5	13832	Failure

Roughness - 5μ

Break - Normal

Ave. Load - 3630 lbs.

Stress - 51,300 p.s.i.

TABLE XVI

Test 9

Approx. Gage Setting 320 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	No reading
2	50	3630
3	6740	3680
4	6760	No reading
5	7000	3870
6	7060	3870
7	9316	Failure

Roughness - 5 μ

Break - Normal

Ave. Load - 3760 lbs.

Stress - 53,100 p.s.i.

TABLE XVII

Test 10

Approx. Gage Setting 340 p.s.i.

Reading	Cycles	Load (lbs.)
1	30	3920
2	1080	4015
3	2080	4120
4	3240	Failure

Roughness - 5 μ

Break - Normal

Ave. Load - 4020 lbs.

Stress - 56,800 p.s.i.

TABLE XVIII

Test 11

Approx. Gage Setting 360 p.s.i.

Reading	Cycles	Load (lbs.)
1	50	4450
2	100	4510
3	400	4390
4	550	Failure

Roughness - 5 μ

Break - Normal

Ave. Load - 4450 lbs.

Stress - 63,000 p.s.i.

TABLE XIX

Test 8

Approx. Gage Setting 220 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	2670
2	4650	2620
3	4800	2730
4	5400	2800
5	9320	2690
6	13400	2710
7	17200	2680
8	77380	Failure

Roughness - 50 μ

Break - Fillet

Ave. Load - 2700 lbs.

Stress - 38,200 p.s.i.

TABLE XX

Test 28 | Approx. Gage Setting 230 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	No reading
2	4260	No reading
3	4500	No reading
4	6420	2700
5	6440	2870
6	8072	2930
7	8080	2880
8	9460	2890
9	16180	2970
10	16200	2730
11	21860	3000
12	21800	2950
13	23660	2870
14	48892	Failure

Roughness - 50 μ

Break - Fillet

Ave. Load - 2880 lbs.

Stress - 40,700 p.s.i.

TABLE XXI

Test 12 Approx. Gage Setting 240 p.s.i.

Reading	Cycles	Load (lbs.)
1	5	3000
2	30	3000
3	660	2780
4	680	3050
5	1600	3090
6	1630	3170
7	7000	3000
8	11200	3050
9	16830	2950
10	36840	Failure

Roughness - 50μ

Break - Normal

Ave. Load - 3000 lbs.

Stress - 42,400 p.s.i.

TABLE XXII

Test 27 Approx. Gage Setting 250 p.s.i.

Reading	Cycles	Load (lbs.)
1	2980	3120
2	7540	3160
3	7560	3280
4	9560	3150
5	9600	3090
6	10650	3140
7	10670	3170
8	16200	3010
9	23740	Failure

Roughness - 50μ

Break - Fillet

Ave. Load - 3140 lbs.

Stress - 44,400 p.s.i.

TABLE XXIII

Test 15 Approx. Gage Setting 260 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	3300
2	175	3470
3	5010	3350
4	11100	3460
5	14600	3250
6	17400	3290
7	20534	Failure

Roughness - 50 μ

Break - Fillet

Ave. Load - 3350 lbs.

Stress - 47,400 p.s.i.

TABLE XXIV

Test 29 Approx. Gage Setting 270 p.s.i.

Reading	Cycles	Load (lbs.)
1	3810	3570
2	3830	3570
3	7280	3440
4	7300	3470
5	9710	3450
6	9730	3390
7	19300	3290
8	19310	3290
9	20970	3490
10	21000	3410
11	27370	Failure

Roughness - 50μ

Break - Normal

Ave. Load - 3430 lbs.

Stress - 48,500 p.s.i.

TABLE XXV

Test 14 — Approx. Gage Setting 280 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	3410
2	40	3410
3	3760	3390
4	4000	3560
5	8840	3540
6	12860	3430
7	20280	3570
8	24612	Failure

Roughness - 50μ

Break - Normal

Ave. Load - 3470 lbs.

Stress - 49,000 p.s.i.

TABLE XXVI

Test 16 — Approx. Gage Setting 300 p.s.i.

Reading	Cycles	Load (lbs.)
1	520	3630
2	3280	3520
3	8320	3650
4	11832	Failure

Roughness - 50 μ

Break - Fillet

Ave. Load - 3600 lbs.

Stress - 50,900 psi

TABLE XXVII

Test 19 Approx. Gage Setting 320 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	3860
2	200	3860
3	1020	3860
4	2497	Failure

Roughness - 50μ

Break - Normal

Ave. Load - 3860 lbs.

Stress - 54,600 p.s.i.

TABLE XXVIII

Test 18 Approx. Gage Setting 340 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	4170
2	100	4170
3	850	4170
4	880	4170
5	1900	4080
6	2218	Failure

Roughness - 50μ

Break - Normal

Ave. Load - 4150 lbs.

Stress - 58,600 p.s.i.

TABLE XXIX

Test 17

Approx. Gage Setting 360 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	4160
2	18	Failure

Roughness - 50 μ

Break - Normal

Ave. Load - 4160 lbs.

Stress - 58,800 p.s.i.

TABLE XXX

Test 38 — Approx. Gage Setting 210 p.s.i.

Reading	Cycles	Load (lbs.)
1	44	2630
2	72	2570
3	550	2600
4	570	2550
5	2980	2550
6	3000	2580
7	3650	2550
8	3660	2550
9	7250	2640
10	7260	2550
11	11200	2440
12	14600	2520
13	21450	2610
14	56830	2520
15	56840	2630
16	60125	2670
17	64300	2550
18	64310	2620
19	68080	2600
20	68100	2600
21	73890	2660
22	73900	2660
23	81090	2500
24	81100	2500
25	91378	Failure

Roughness - 100 μ — Ave. Load - 2580 lbs.

Break - Normal — Stress - 36,500 p.s.i.

TABLE XXXI

Test 35

Approx. Gage Setting 230 p.s.i.

Reading	Cycles	Load (lbs.)
1	868	2780
2	3140	2810
3	4320	2790
4	8100	2760
5	11200	2670
6	14100	2760
7	18500	2850
8	22650	2710
9	23945	Failure

Roughness - 100μ

Break - Fillet

Ave. Load - 2760 lbs.

Stress - 39,000 p.s.i.

TABLE XXXII

Test 34 Approx. Gage Setting 250 p.s.i.

Reading	Cycles	Load (lbs.)
1	1670	3020
2	3380	3000
3	8190	3000
4	10260	3000
5	10800	3070
6	10820	3090
7	14710	3070
8	18640	3090
9	18680	3150
10	20100	3180
11	23060	3070
12	23080	3130
13	25100	3180
14	27492	Failure

Roughness - 100 μ

Break - Fillet

Ave. Load - 3080 lbs.

Stress - 43,600 p.s.i.

TABLE XXXIII

Test 30 Approx. Gage Setting 260 p.s.i.

Reading	Cycles	Load (lbs.)
1	1780	3305
2	4590	3370
3	4600	3300
4	7930	3320
5	11360	3270
6	13200	3350
7	16800	3290
8	22338	Failure

Roughness - 100μ

Break - Fillet

Ave. Load - 3300 lbs.

Stress - 46,700 p.s.i.

TABLE XXXIV

Test 32 Approx. Gage Setting 280 p.s.i.

Reading	Cycles	Load (lbs.)
1	1720	3420
2	4510	3610
3	4530	3530
4	5570	3440
5	5590	3310
6	11430	3440
7	11450	3370
8	12754	Failure

Roughness - 100μ

Break - Normal

Ave. Load - 3420 lbs.

Stress - 48,300 p.s.i.

TABLE XXXV

Test 31 Approx. Gage Setting 300 p.s.i.

Reading	Cycles	Load (lbs.)
1	80	3330
2	110	3350
3	3850	3540
4	3880	3720
5	8570	3460
6	8600	3640
7	8680	Failure

Roughness - 100μ

Break - Fillet

Ave. Load - 3505 lbs.

Stress - 49,500 p.s.i.

TABLE XXXVI

Test 33 Approx. Gage Setting 310 p.s.i.

Reading	Cycles	Load (lbs.)
1	80	3660
2	100	3600
3	3906	3630
4	4000	No reading
5	4924	Failure

Roughness - 100μ

Break - Fillet

Ave. Load - 3630 lbs.

Stress - 51,300 p.s.i.

TABLE XXXVII

Test 20

Approx. Gage Setting 320 p.s.i.

Reading	Cycles	Load (lbs.)
1	66	3720
2	500	3890
3	7700	3950
4	8310	Failure

Roughness - 100μ

Break - Fillet

Ave. Load - 3850 lbs.

Stress - 54,500 p.s.i.

TABLE XXXVIII

Test 37

Approx. Gage Setting 320 p.s.i.

Reading	Cycles	Load (lbs.)
1	30	3680
2	50	3680
3	1620	3940
4	2640	4060
5	2916	Failure

Roughness - 100μ

Break - Normal

Ave. Load - 3840 lbs.

Stress - 54,300 p.s.i.

TABLE XXXIX

Test 21 Approx. Gage Setting 340 p.s.i.

No readings. Machine broke down after 20 cycles.

Roughness - 100μ

TABLE XL

Test 36 Approx. Gage Setting 340 p.s.i.

Reading	Cycles	Load (lbs.)
1	30	3820
2	140	3880
3	200	3940
4	540	4100
5	560	4120
6	724	Failure

Roughness - 100μ

Break - Normal

Ave. Load - 3970 lbs.

Stress - 56,100 p.s.i.

TABLE XLI

Test 51 — Approx. Gage Setting 210 p.s.i.

Reading	Cycles	Load (lbs.)
1	20	2520
2	1390	2600
3	3790	2540
4	5060	2600
5	5260	2720
6	8300	2650
7	11700	2500
8	17400	2520
9	24250	2630
10	29600	2520
11	34850	2500
12	42096	Failure

Roughness - 200μ

Break - Fillet

Ave. Load - 2580 lbs.

Stress - 36,500 p.s.i.

TABLE XLII

Test 45 Approx. Gage Setting 220 p.s.i.

Reading	Cycles	Load (lbs.)
1	70	2460
2	700	2570
3	3400	2660
4	6750	2670
5	10610	2690
6	12030	2670
7	15790	2720
8	18250	2700
9	22800	2720
10	30140	2720
11	37662	Failure

Roughness - 200 μ

Break - Fillet

Ave. Load - 2660 lbs.

Stress - 37,600 p.s.i.

TABLE XLIII

Test 48 Approx. Gage Setting 240 p.s.i.

Reading	Cycles	Load (lbs.)
1	40	2820
2	1180	2910
3	3220	2930
4	6420	2880
5	8310	2880
6	11450	2910
7	14160	2930
8	17700	3010
9	20200	2980
10	23430	2920
11	26432	Failure

Roughness - 200 μ

Break - Fillet

Ave. Load - 2920 lbs.

Stress - 41,300 p.s.i.

TABLE XLIV

Test 54 Approx. Gage Setting 250 p.s.i.

Reading	Cycles	Load (lbs.)
1	230	2950
2	5216	3000
3	7220	2950
4	7530	2990
5	12480	3010
6	16270	3030
7	19450	3080
8	22840	3070
9	25972	Failure

Roughness - 200μ

Break - Normal

Ave. Load - 3010 lbs.

Stress - 42,600 p.s.i.

TABLE XLV

Test 42 Approx. Gage Setting 260 p.s.i.

Reading	Cycles	Load (lbs.)
1	40	3400
2	1160	3470
3	3180	3100
4	4170	3200
5	5220	3160
6	7350	3220
7	8160	3290
8	9250	3230
9	10100	3230
10	11374	Failure

Roughness - 200 μ

Break - Fillet

Ave. Load - 3255 lbs.

Stress - 46,000 p.s.i.

TABLE XLVI

Test 41 Approx. Gage Setting 280 p.s.i.

No readings. Machine broke down

Roughness - 200μ

TABLE XLVII

Test 49 Approx. Gage Setting 280 p.s.i.

Reading	Cycles	Load (lbs.)
1	40	3540
2	2520	3420
3	4090	3440
4	6940	3440
5	8210	3420
6	10810	3530
7	10830	3460
8	12550	3450
9	14476	Failure

Roughness - 200μ

Break - Fi llet

Ave. Load - 3460 lbs.

Stress - 49,000 p.s.i.

TABLE XLVIII

Test 44 — Approx. Gage Setting 290 p.s.i.

Reading	Cycles	Load (lbs.)
1	40	3630
2	2010	3630
3	4360	3650
4	6820	3590
5	8150	3630
6	9058	Failure

Roughness - 200μ

Break - Normal

Ave. Load - 3625 lbs.

Stress - 51,200 p.s.i.

TABLE XLIX

Test 39 — Approx. Gage Setting 300 p.s.i.

Reading	Cycles	Load (lbs.)
1	40	3312
2	500	No reading
3	Machine failed	

Roughness - 200μ

TABLE L

Test 46 | Approx. Gage Setting 300 p.s.i.

Reading	Cycles	Load (lbs.)
1	46	3580
2	7680	No reading
3	2640	No reading
4	3690	No reading
5	Electrical failure	

Roughness - 200μ

TABLE LI

Test 43 Approx. Gage Setting 310 p.s.i.

Reading	Cycles	Load (lbs.)
1	10	3500
2	36	3660
3	50	3690
4	720	3770
5	730	3710
6	1230	3790
7	1250	3670
8	1770	3730
9	1800	3730
10	2680	3640
11	2700	3690
12	3150	3810
13	4670	3810
14	5390	3690
15	6280	3690
16	7850	Failure

Roughness - 200 μ

Break - Normal

Ave. Load - 3705 lbs.

Stress - 52,400 p.s.i.

TABLE LII

Test 40 Approx. Gage Setting 310 p.s.i.

Reading	Cycles	Load (lbs.)
1	30	3610
2	40	3640
3	810	3710
4	820	3740
5	1120	3740
6	1140	3710
7	2380	3640
8	2400	3700
9	3150	3710
10	3170	3710
11	3710	3660
12	3720	3800
13	5000	3620
14	5010	3570
15	6440	3590
16	6940	Failure

Roughness - 200 μ

Break - Fillet

Ave. Load - 3680 lbs.

Stress - 52,000 p.s.i.

TABLE LIII

Test 47 Approx. Gage Setting 320 p.s.i.

Reading	Cycles	Load (lbs.)
1	88	3550
2	100	3670
3	640	3790
4	650	3820
5	2420	3770
6	2450	3820
7	4140	3810
8	5270	3810
9	7556	Failure

Roughness - 200 μ

Break - Fillet

Ave. Load - 3755 lbs.

Stress - 53,100 p.s.i.

TABLE LIV

Test 52 Approx. Gage Setting 320 p.s.i.

Reading	Cycles	Load (lbs.)
1	30	3650
2	390	3610
3	800	3720
4	820	3550
5	1260	3570
6	1280	3540
7	2050	3740
8	2870	3740
9	3200	3830
10	3420	3830
11	4960	3850
12	6210	3730
13	7234	Failure

Roughness - 200 μ

Break - Fillet

Ave. Load - 3710 lbs.

Stress - 52,500 p.s.i.

TABLE LV

Test 53 Approx. Gage Setting 330 p.s.i.

Reading	Cycles	Load (lbs.)
1	24	3830
2	40	3870
3	540	3830
4	780	3850
5	1016	Failure

Roughness - 200μ

Break - Normal

Ave. Load - 3845 lbs.

Stress - 54,300 p.s.i.

TABLE LV1

Test 50 Approx. Gage Setting 340 p.s.i.

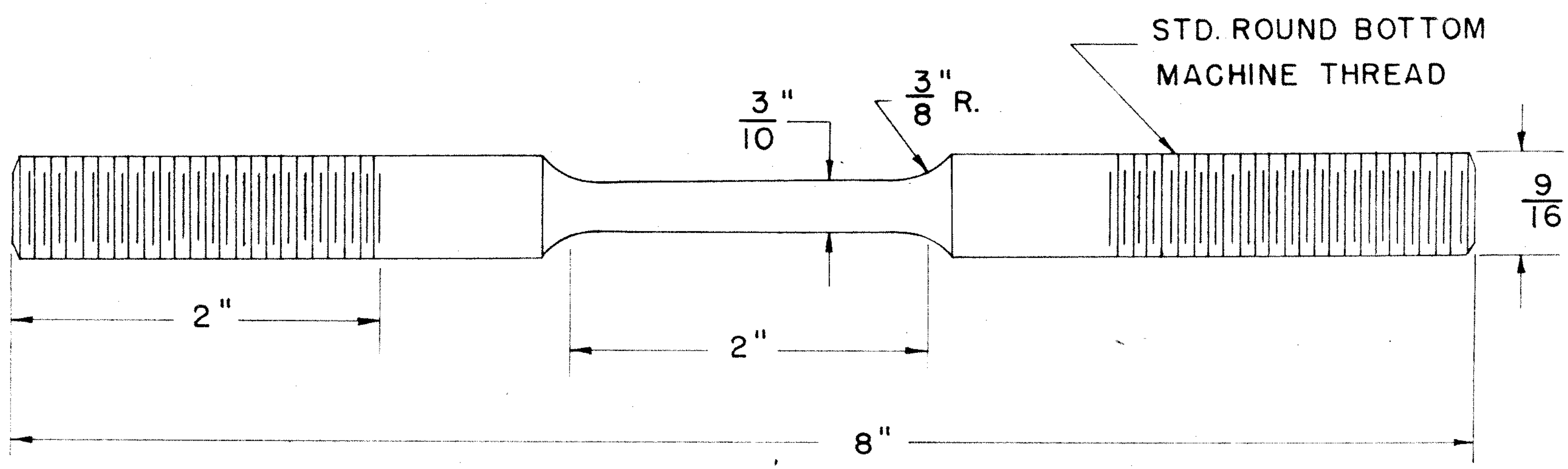
Reading	Cycles	Load (lbs.)
1	20	3660
2	75	3725
3	150	3800
4	300	Failure

Roughness - 200μ

Break - Normal

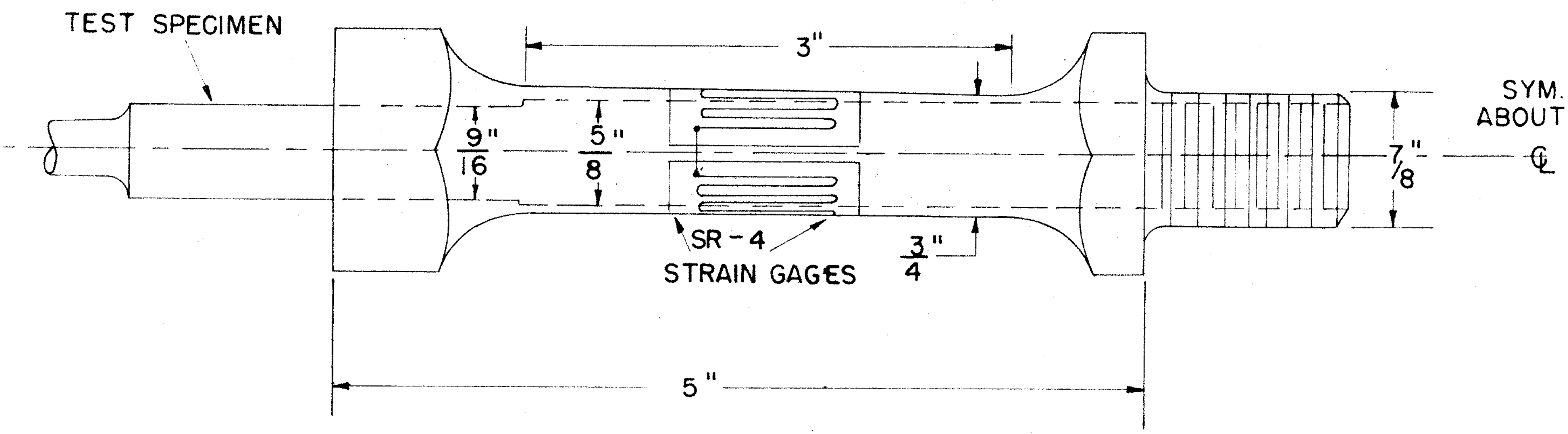
Ave. Load - 3730 lbs.

Stress - 52,750 p.s.i.



TEST SPECIMEN

FIG. 1



LOAD MEASURING COUPON

FIG. 2

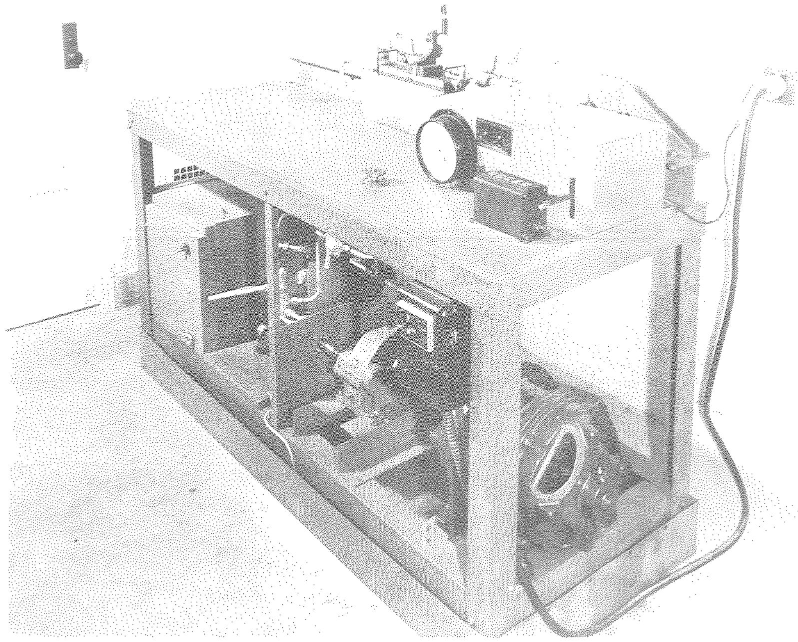


Fig. 3

General View of Machine

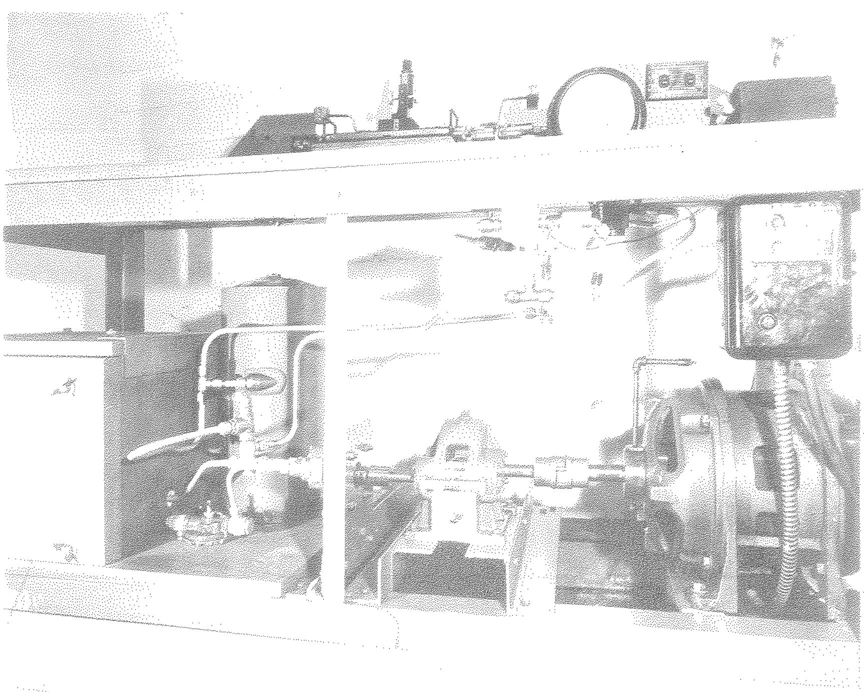


Fig. 4

Hydraulic Section

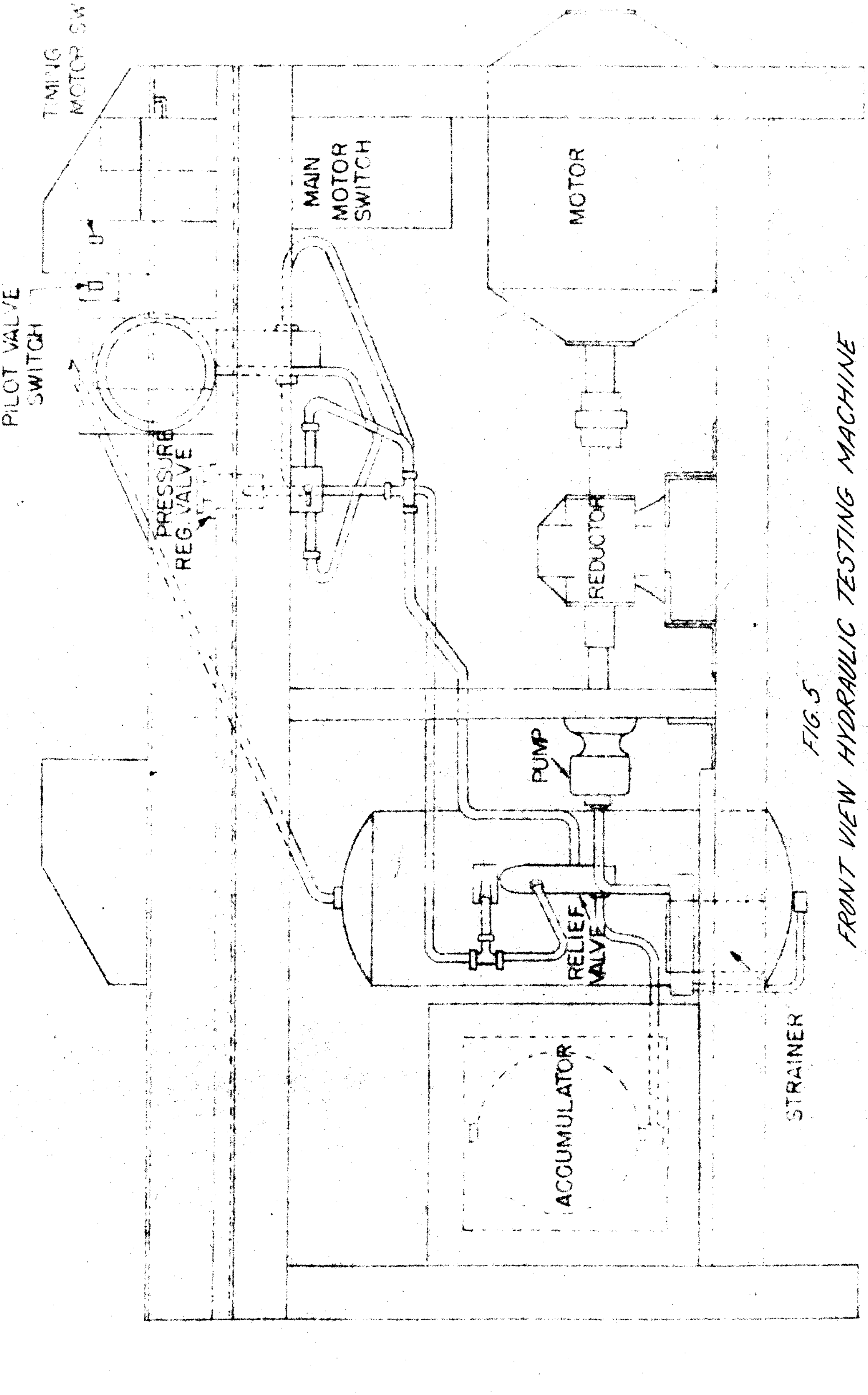


FIG. 5
FRONT VIEW HYDRAULIC TESTING MACHINE

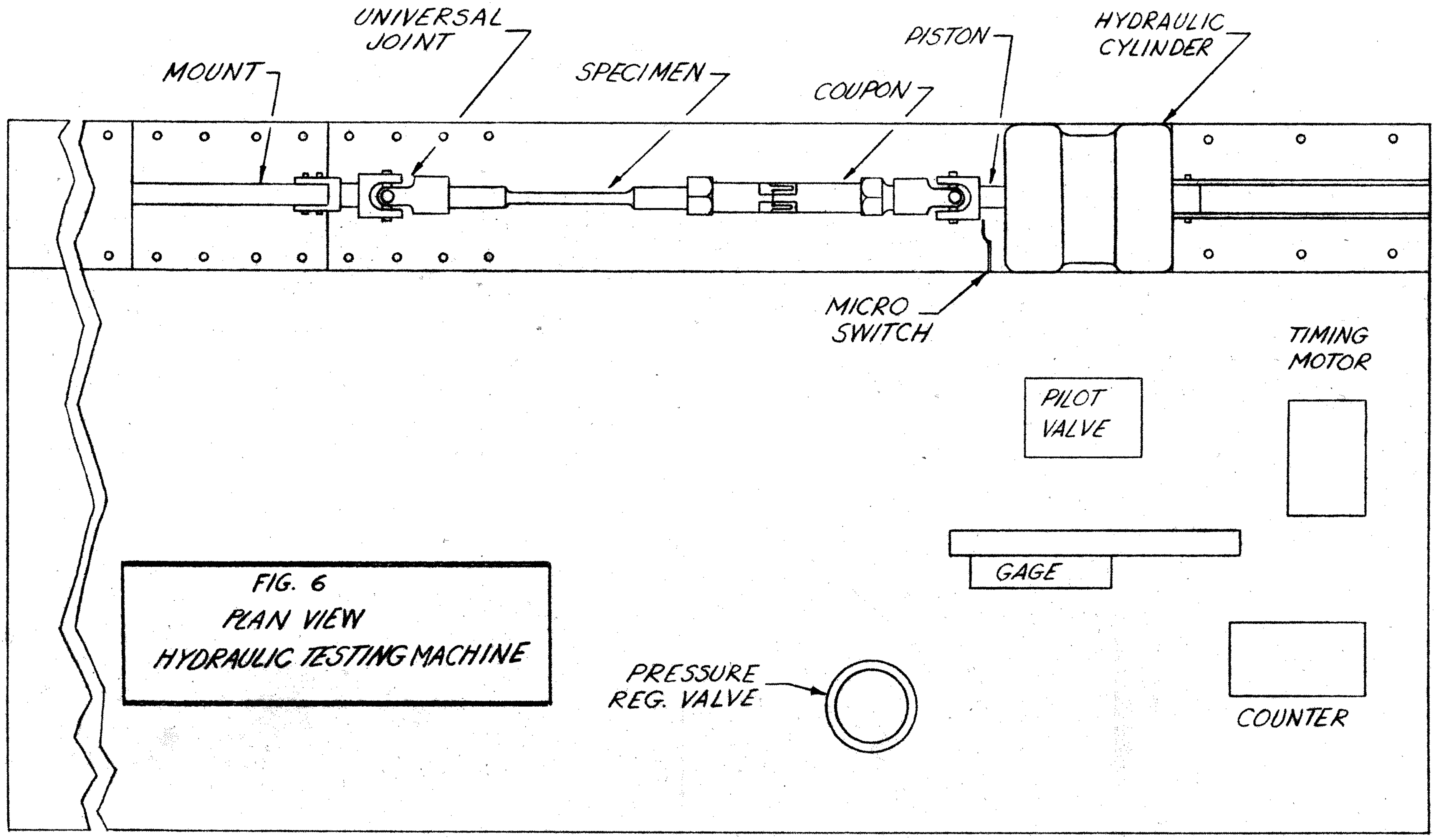


FIG. 6
PLAN VIEW
HYDRAULIC TESTING MACHINE

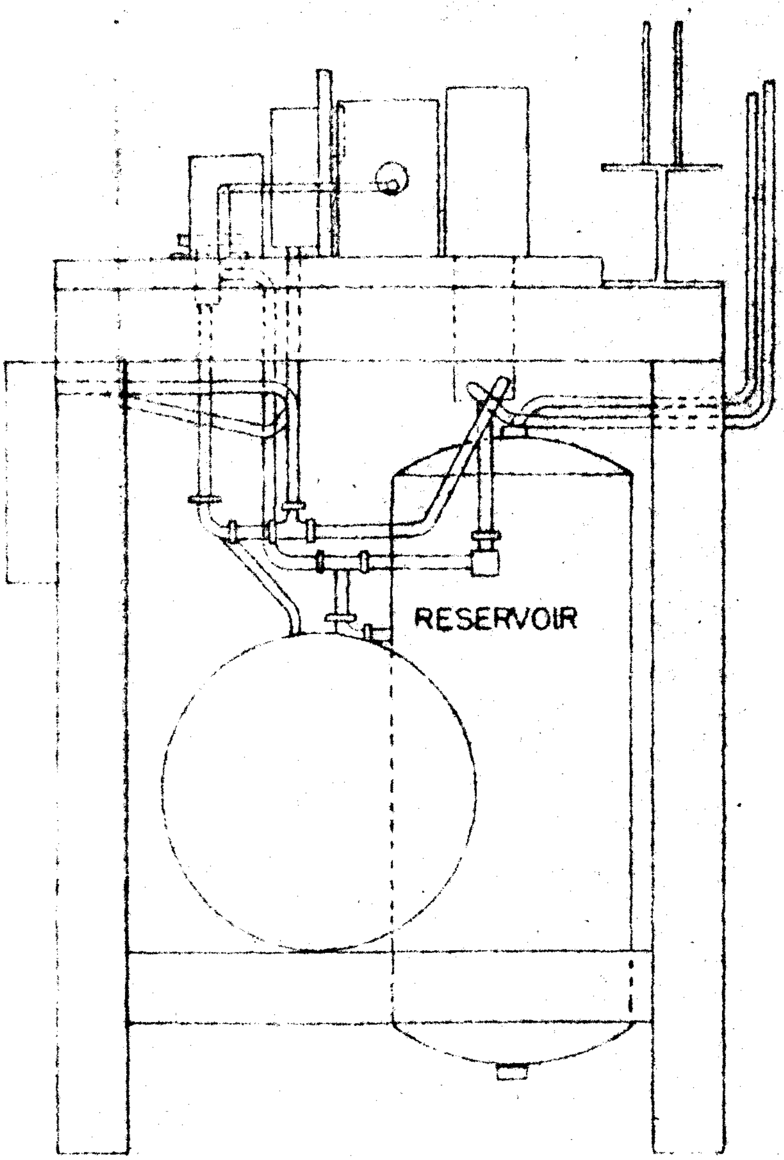


FIG. 7
END VIEW HYDRAULIC TESTING MACHINE

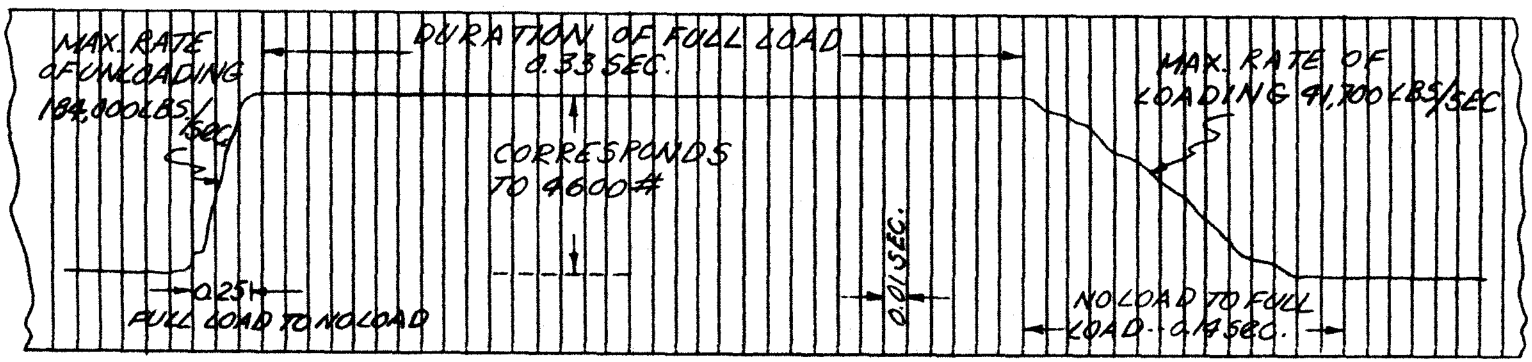


FIG. 8

STUDY OF LOAD APPLICATION

(FROM OSCILLOGRAPH PHOTOGRAPH)

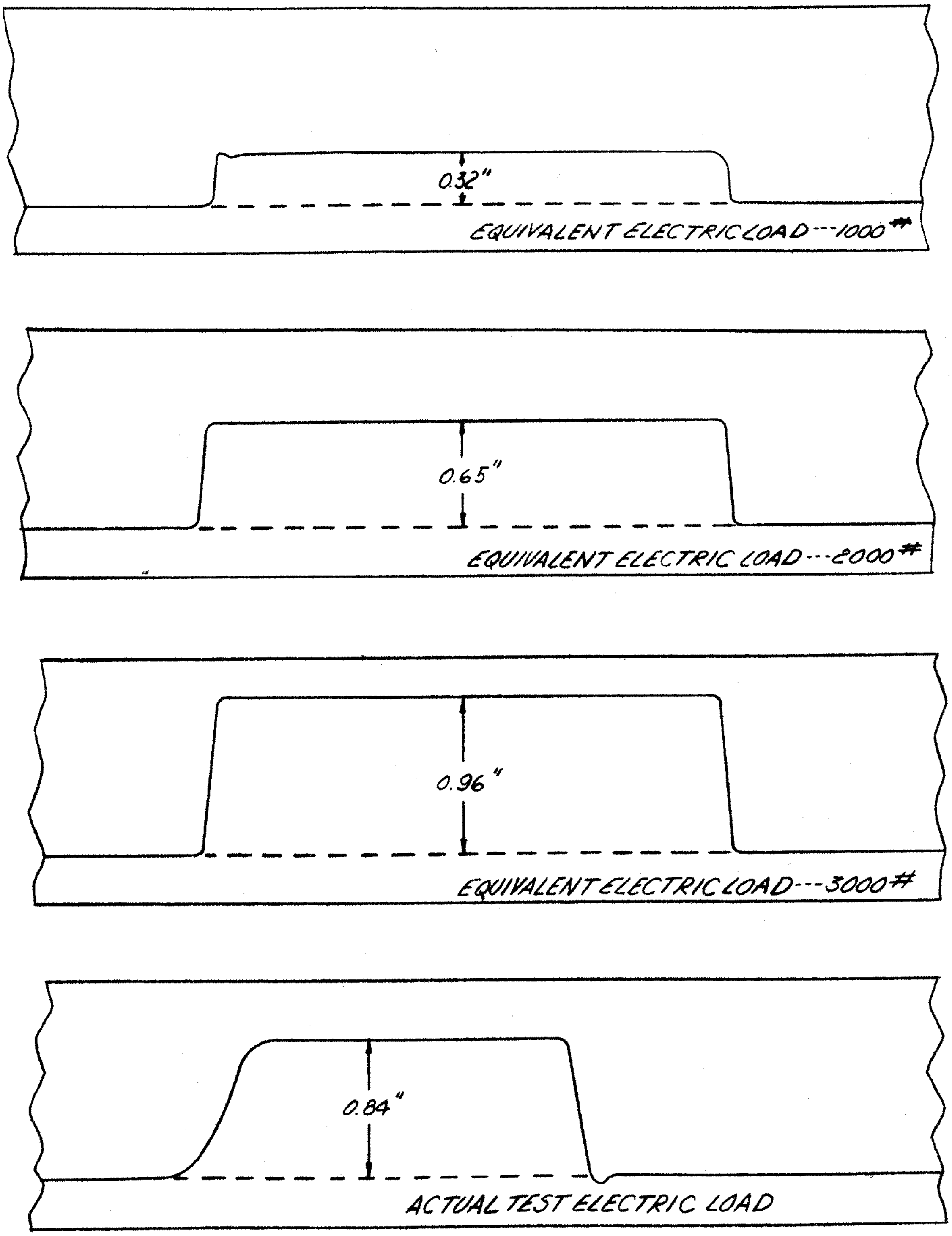


FIG. 9

TYPICAL TEST RESULT AS FILMED

Sample Test
Date_________

G.A.L.C.I.T.

Structures Laboratory

Gage

Roughness

Reading No.	Electric Load	Cycles	Height Inches	Actual Load lbs.	Tensile Load p.s.i.
1	1000	4	0.27		
2	2000	4	0.55		
3	3000	4	0.81		
4	4000	4	1.08		
		Ave.	0.27		
5		4	0.70	2590	36600
6		4	0.70	2590	36600
7			0.70	2590	36600
8	2000	4000	0.60		
9	3000	4000	0.91		
10	4000	4000	1.19		
		Ave.	0.30		
11		4000	0.78	2600	36800
12		4000	0.78	2600	36800
13		4000	0.78	2600	36800
14	1000	8000	0.20		
15	2000	8000	0.40		
16	3000	8000	0.61		
		Ave.	0.20		
17		8000	0.52	2590	36600
18		8000	0.52	2590	36600
19		8000	0.51	2550	36000

Reading No.	Electric Load	Cycles		Height Inches	Actual Load lbs.	Tensile Load p.s.i.
61	2000	76000		0.44		
62	3000	76000		0.67		
63	4000	76000		0.80		
			Ave.	0.22		
64		76000		0.57	2590	36600
65		76000		0.57	2590	36600
66		76000		0.57	2590	36600
Failure		77380		Fillet Break		

Fig. 10

Typical Data Sheet

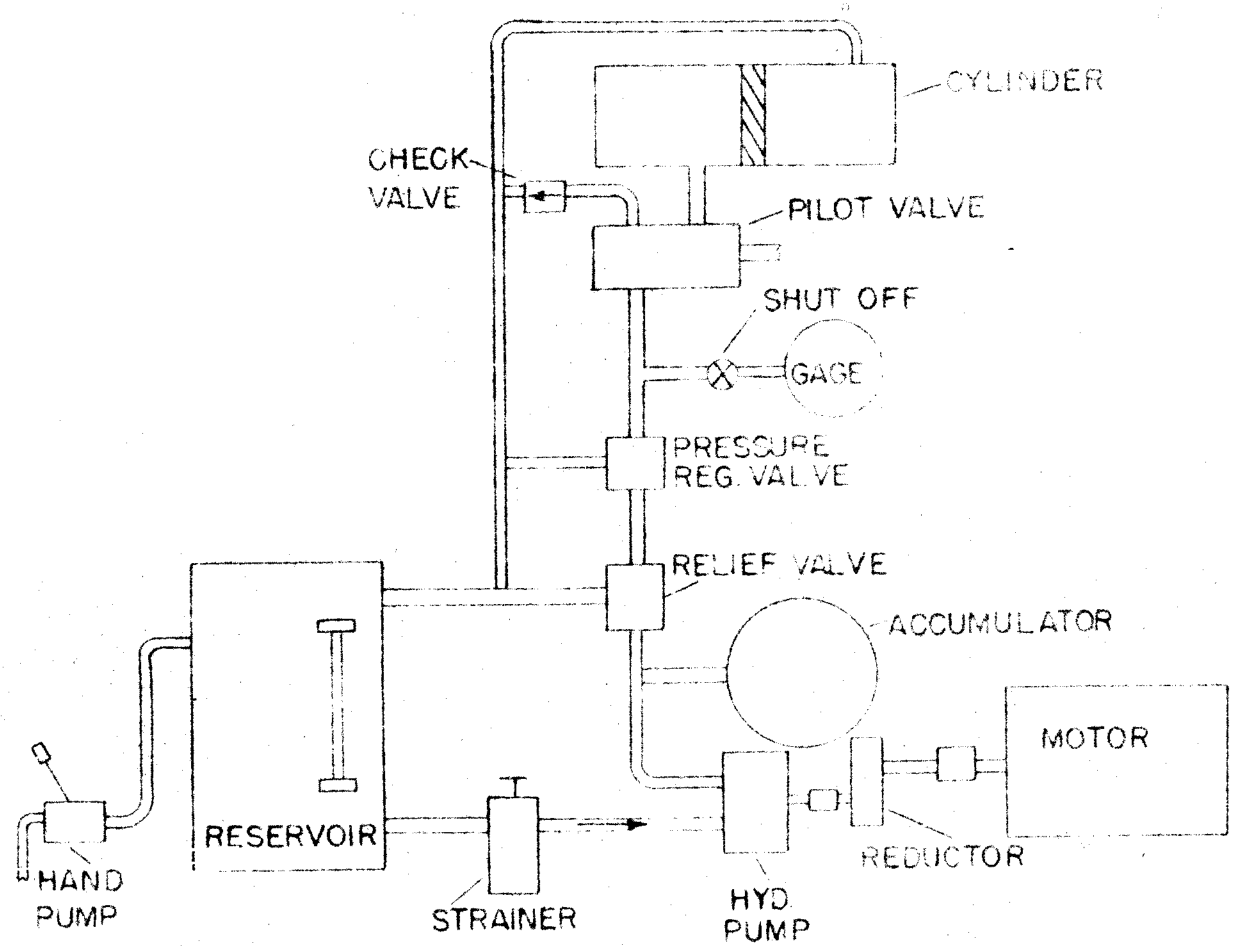


FIG. 11
SCHEMATIC DRAWING OF HYDRAULIC SYSTEM

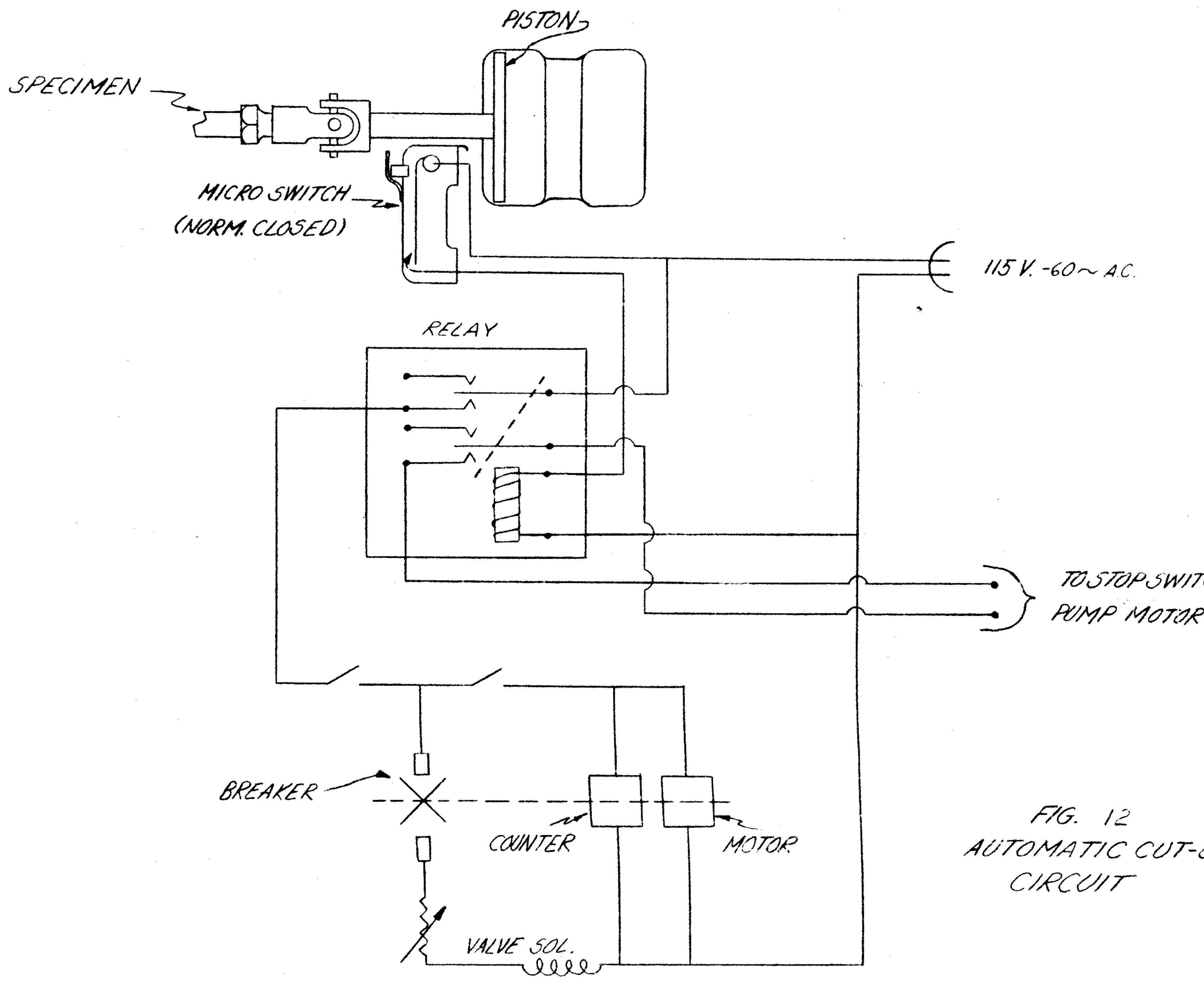
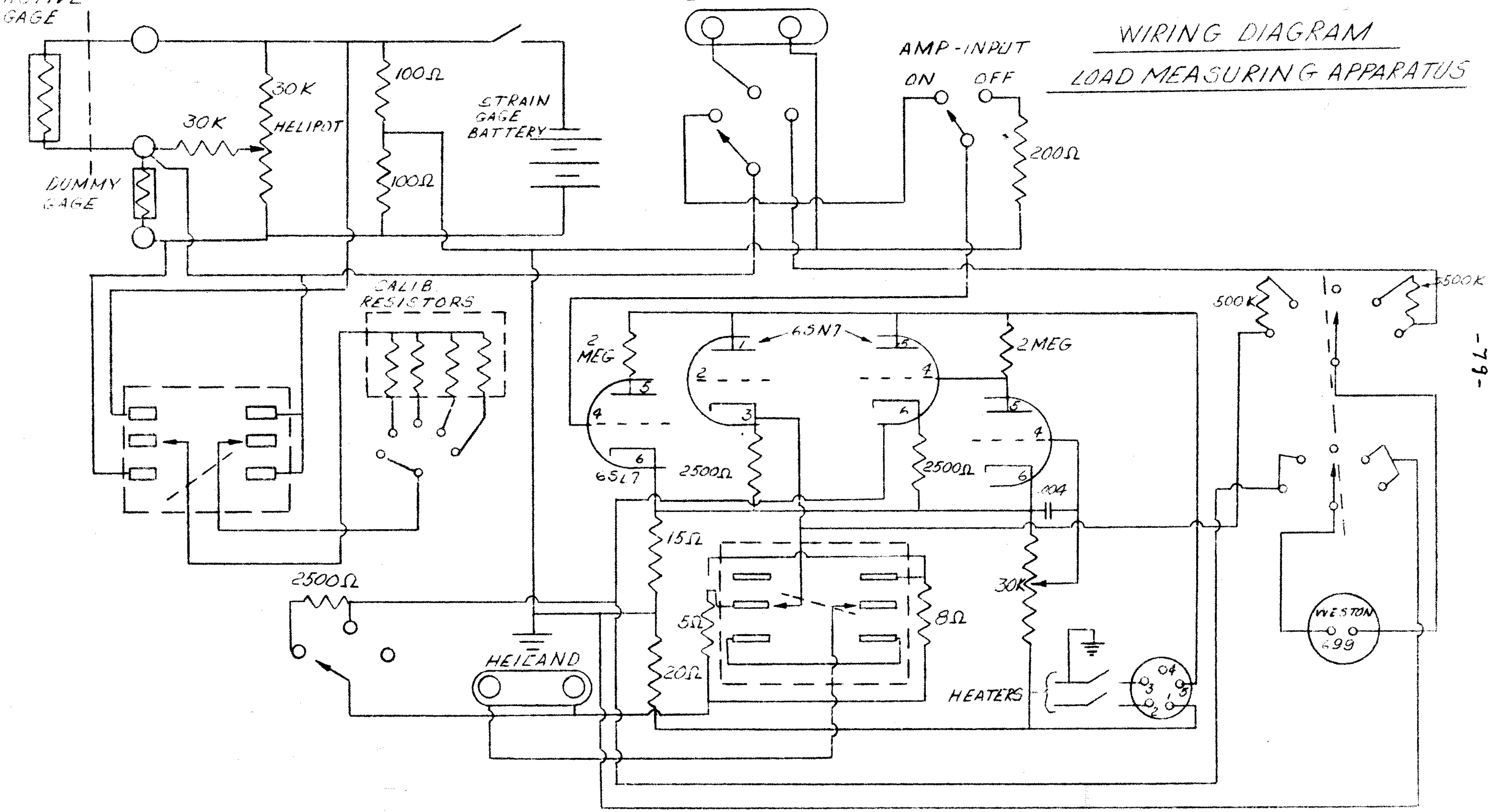


FIG. 12
AUTOMATIC CUT-OFF
CIRCUIT

FIG. 13

WIRING DIAGRAM

LOAD MEASURING APPARATUS



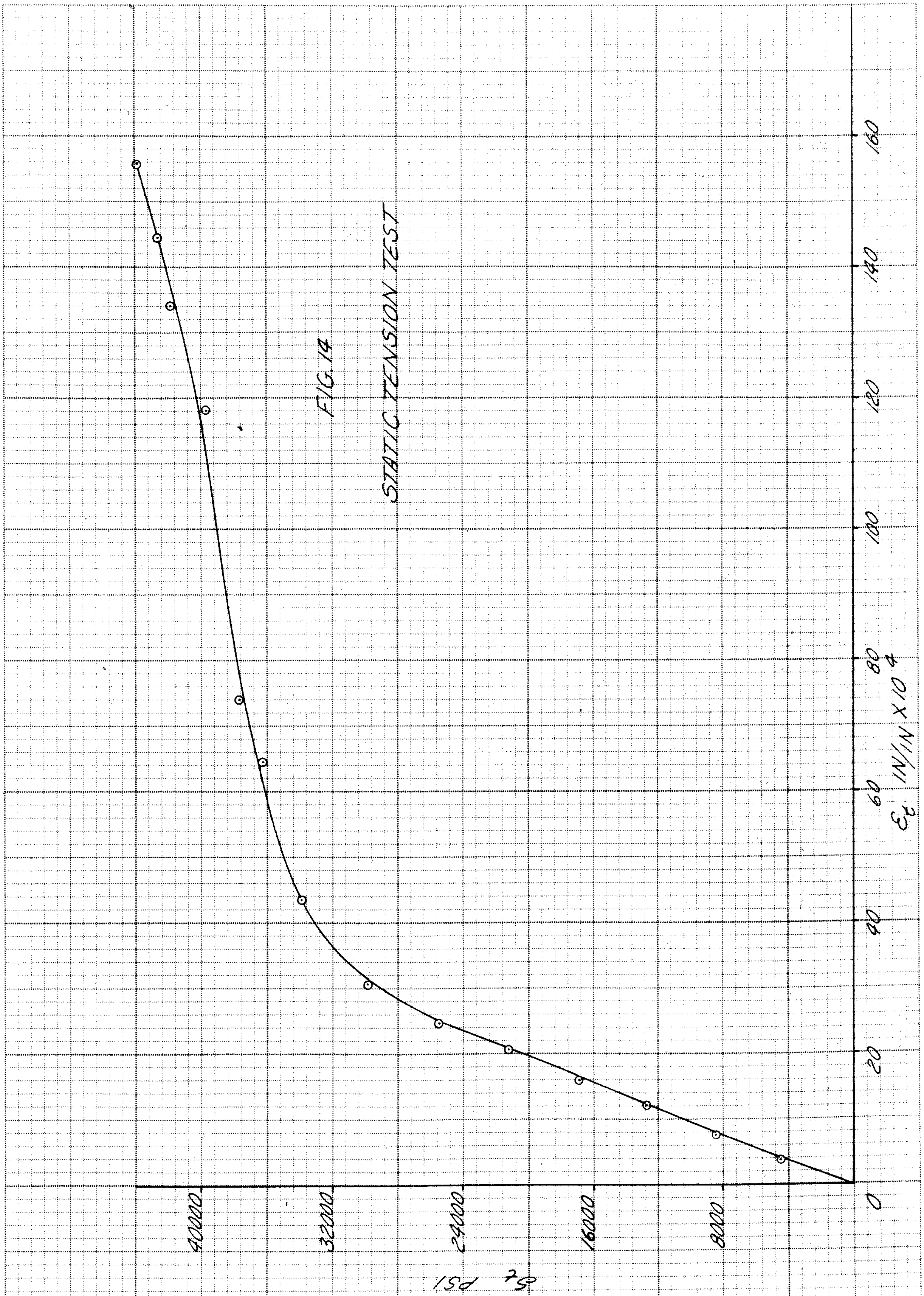


FIG. 14

STATIC TENSION TEST

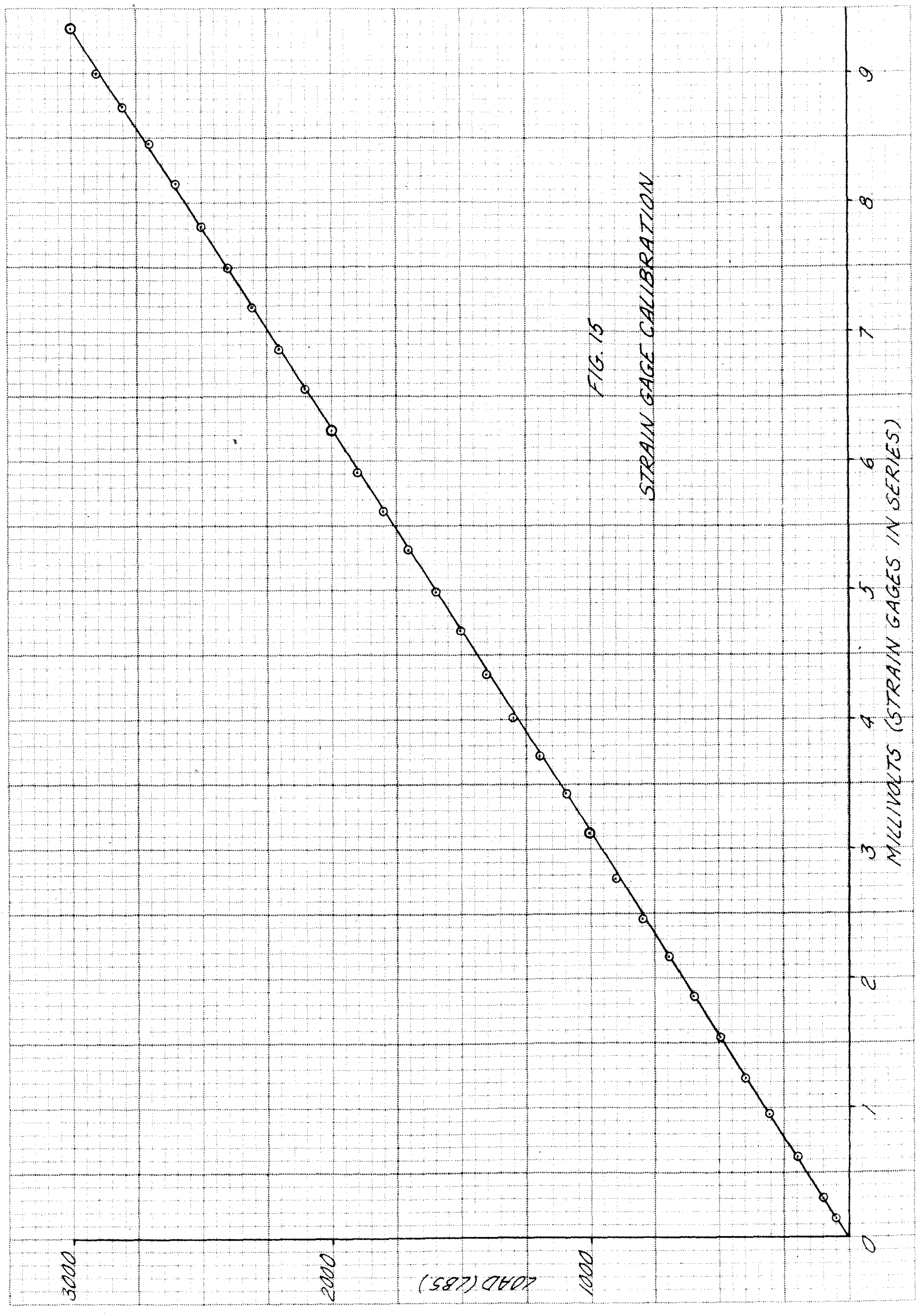


FIG. 15

STRAIN GAGE CALIBRATION

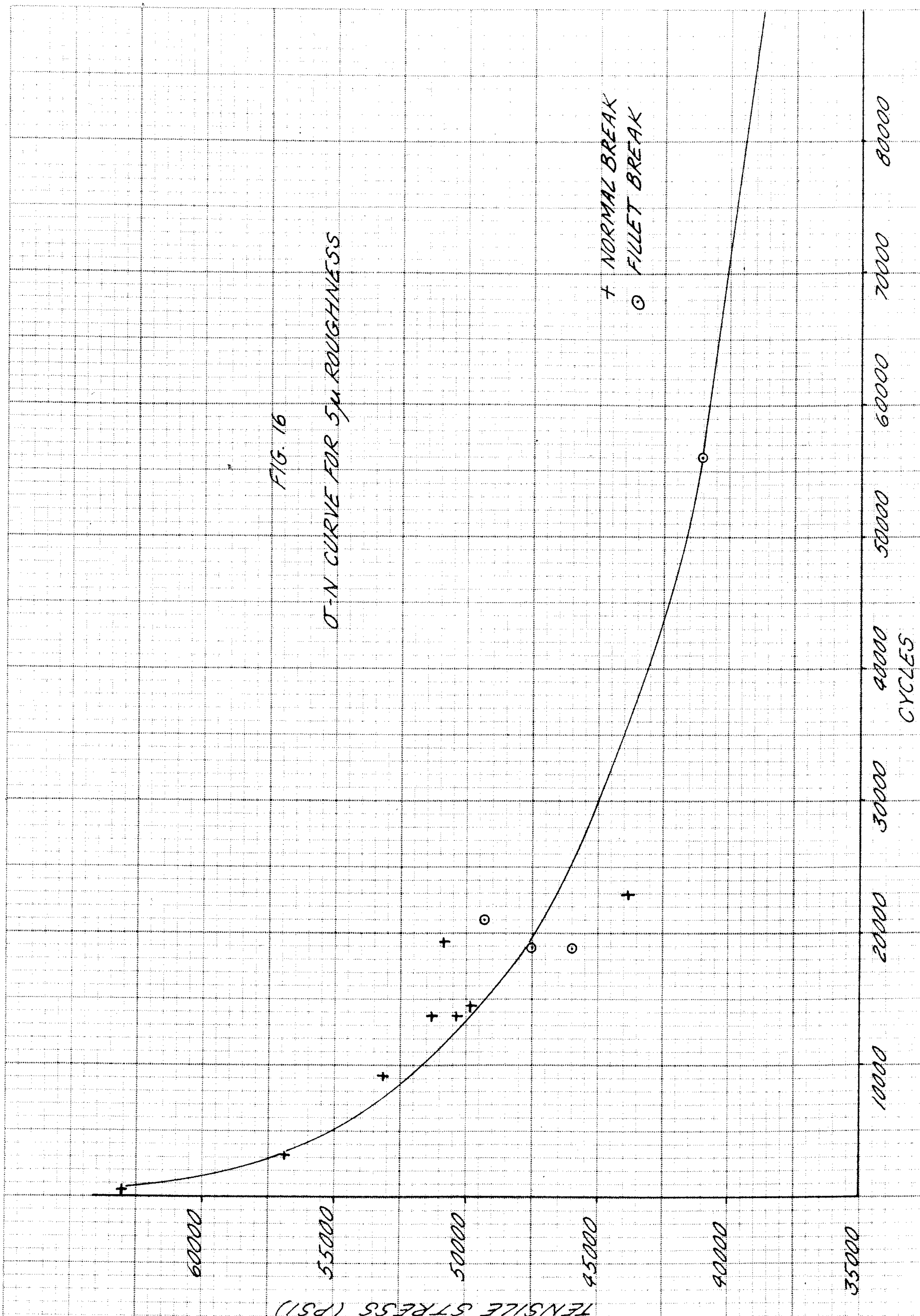


FIG. 16
σ-N CURVE FOR 5μ ROUGHNESS
+ NORMAL BREAK
⊙ FILLET BREAK
TENSILE STRESS (PSI)
60000
55000
50000
45000
40000
35000
CYCLES
10000
20000
30000
40000
50000
60000
70000
80000

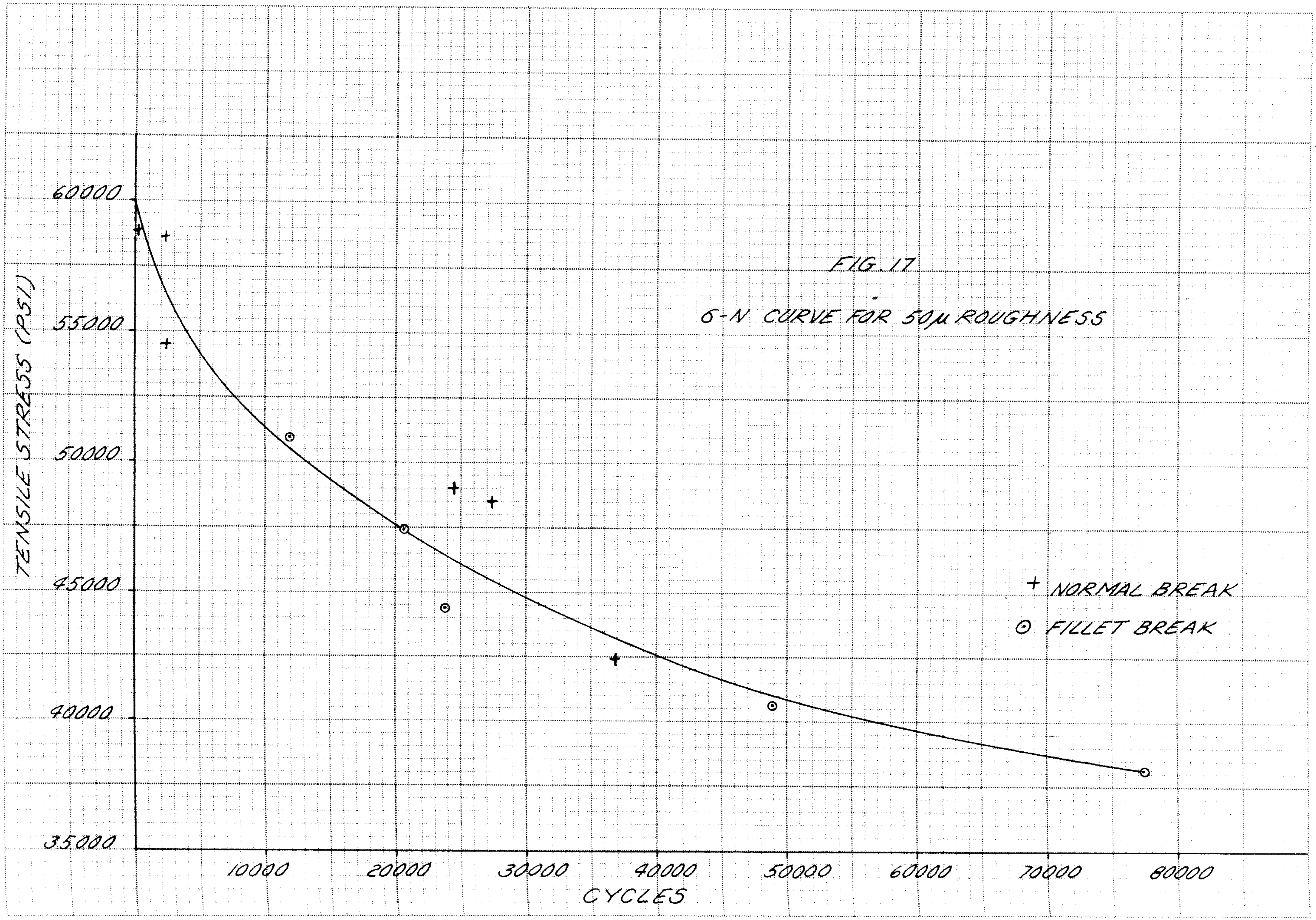


FIG. 17

σ-N CURVE FOR 50μ ROUGHNESS

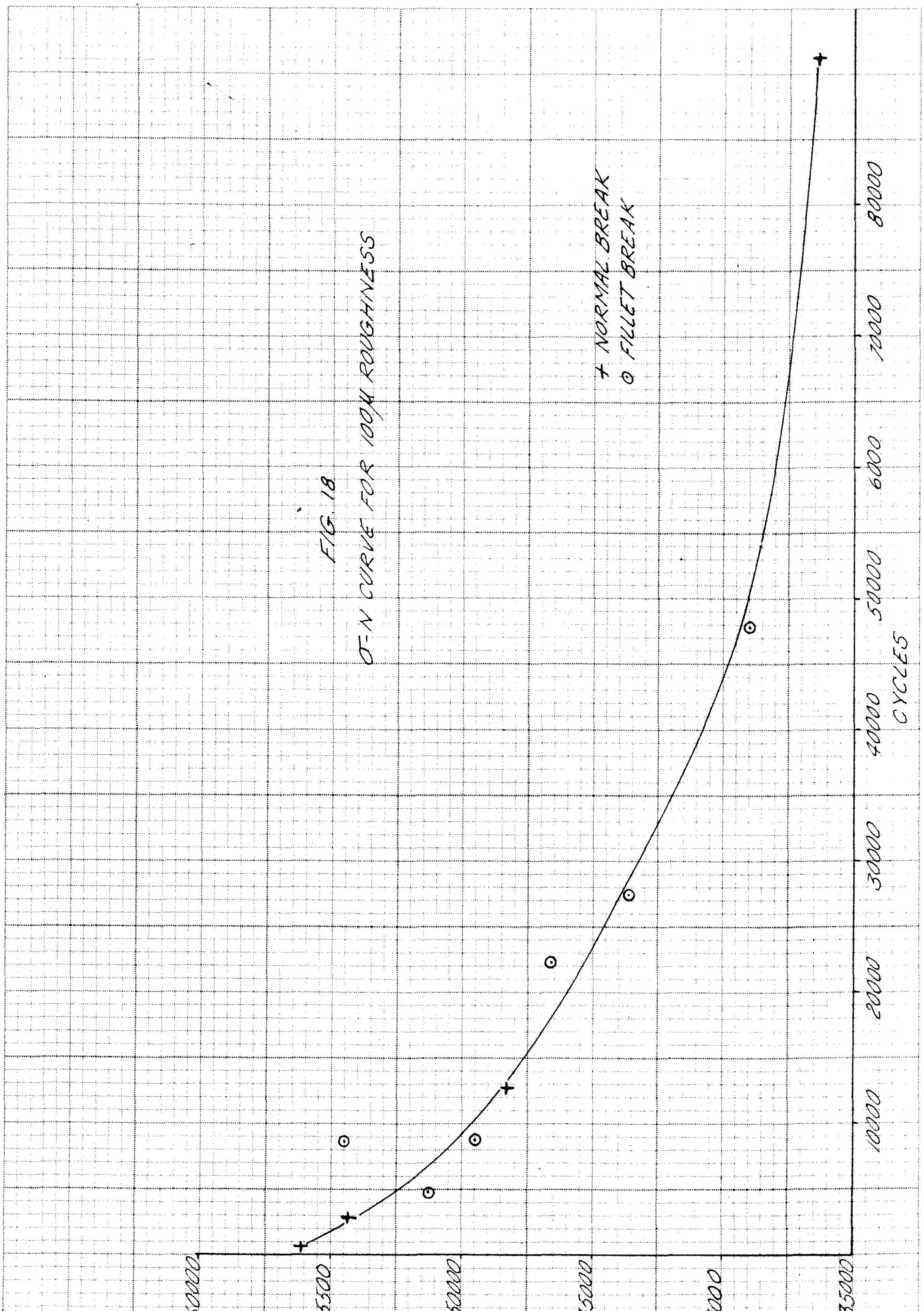


FIG. 18

σ-N CURVE FOR 100μ ROUGHNESS

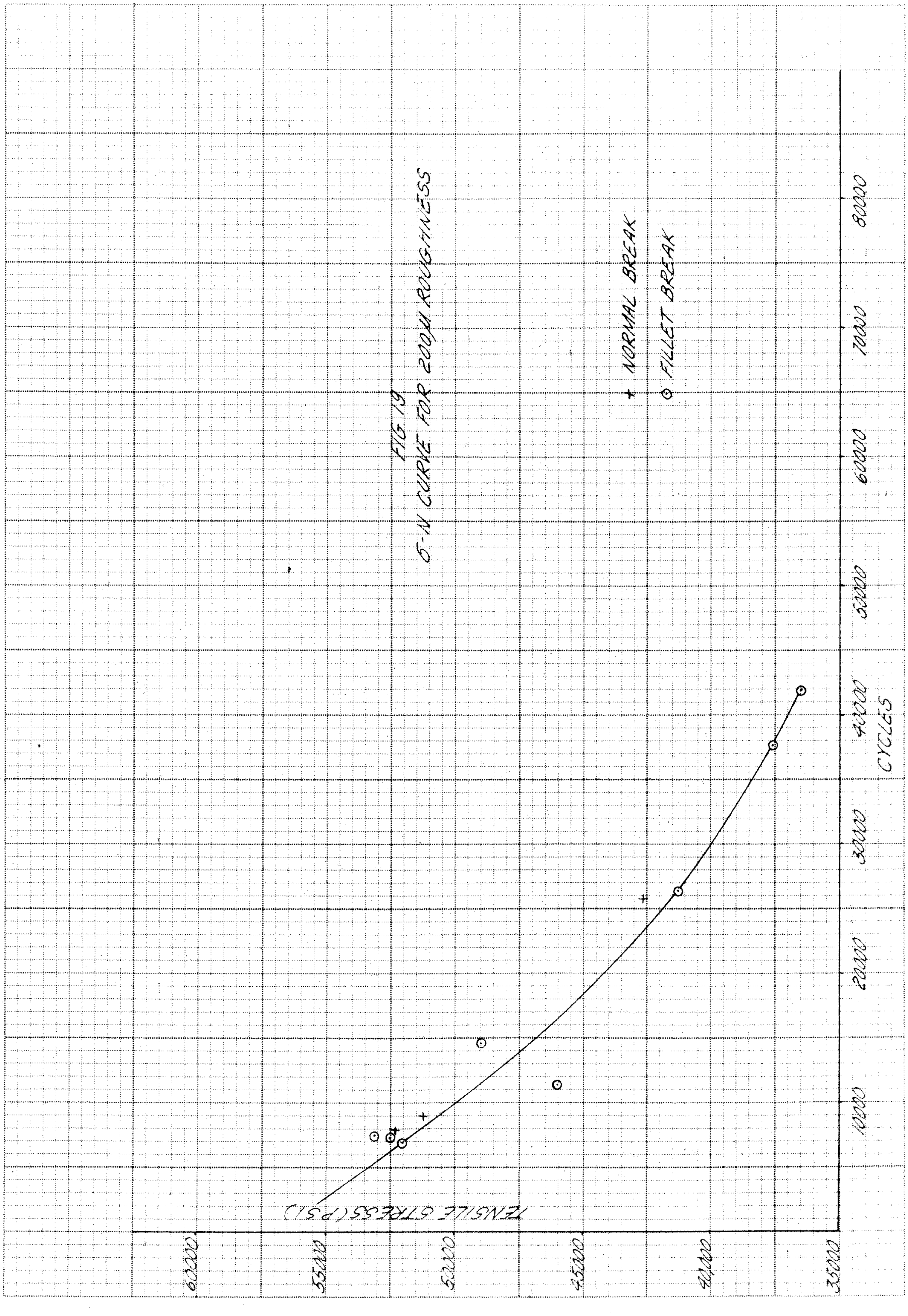


FIG 19
S-N CURVE FOR 200μ ROUGHNESS
+ NORMAL BREAK
⊙ FILLET BREAK
CYCLES
10000
20000
30000
40000
50000
60000
70000
80000
TENSILE STRESS (PSI)
60000
55000
50000
45000
40000
35000

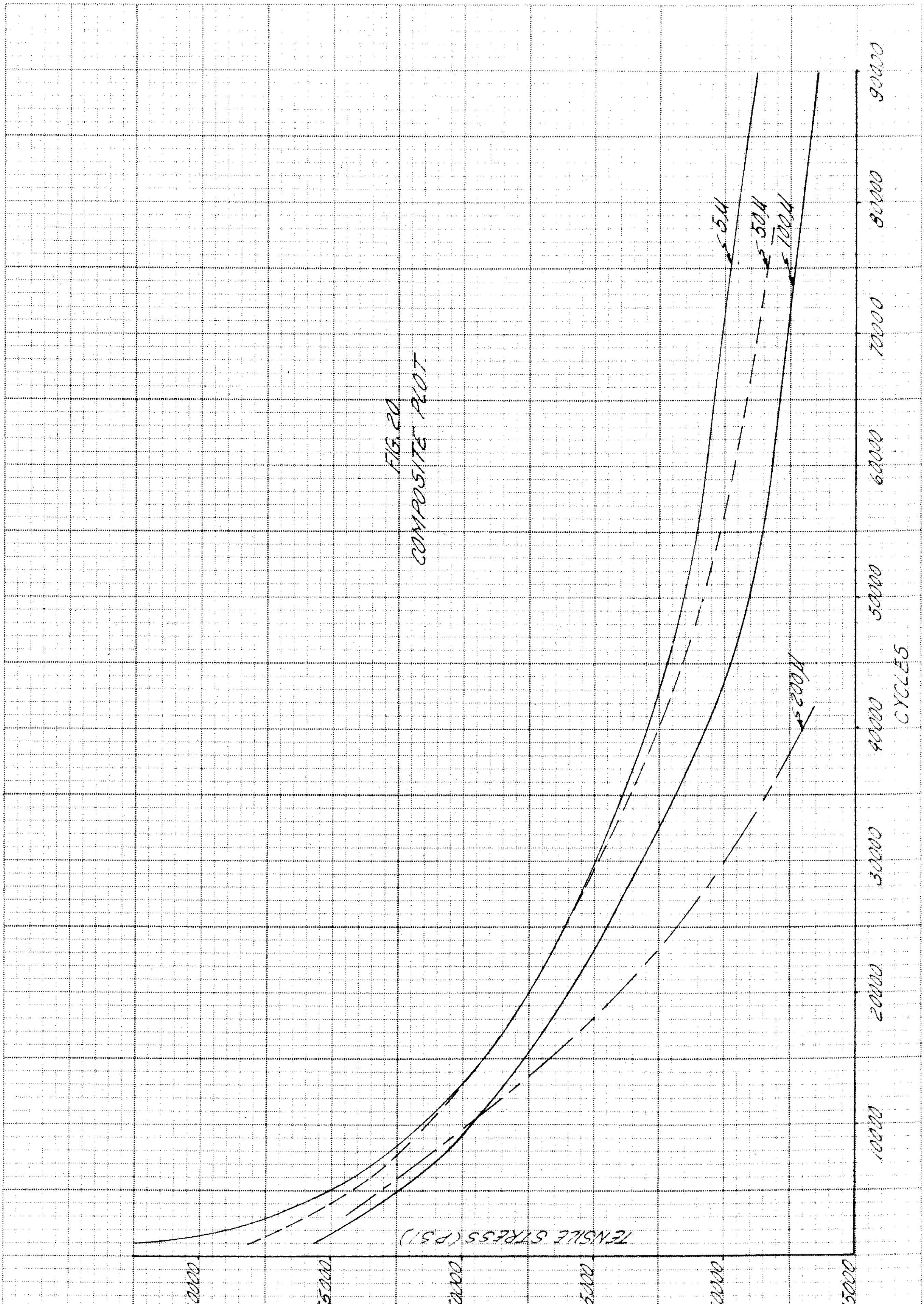


FIG. 20
COMPOSITE PLOT