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LIGHTWEIGHT GEARBOX COMPOSITE CASE. PHASE II.

W. N. Holcomb

General Motors Corporation

Prepared for:

Naval Air Systems Command

12 January 1971

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PHASE II
LIGHTWEIGHT GEARBOX
COMPOSITE CASE

FINAL REPORT

(18 MARCH 1970 TO 18 JANUARY 1971)

12 January 1971

by

W. N. Holcomb

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Prepared under Contract N00019-70-C-0224

for

Naval Air Systems Command

Department of the Navy

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FORM 2250-1 (REV.11/60)

ALLISON
DIVISION
GENERAL MOTORS CORPORATION
INDIANAPOLIS 6, INDIANA

ENGINEERING DEPT. REPORT NO. 7008

PHASE II

LIGHTWEIGHT GEARBOX
COMPOSITE CASE

FINAL REPORT

Details of illustrations in this document may be better studied on microfiche

MODEL ______ PROJECT ______

CONTRACT N00019-70-C-0224 REFERENCE ______

PREPARED BY W. N. Holcomb DATE 12 January 1971

APPROVED CBWright 1/12/71

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I. INTRODUCTION

This report is submitted as the Phase II summary report in compliance with Contract N00019-70-C-0224. The work accomplished by Detroit Diesel Allison Division in the machining, static testing and dynamometer testing of the boron-glass-epoxy composite reduction gear front case is described herein. Since this report and the December, 1970, monthly report are due on nearly the same date, the December report has been included in this final report. The program period covered by this report is from 18 March 1970 through 18 January 1971.

The Phase I program which included the design and molding of three composite material cases for the Naval Air Systems Command was completed under Contract N00019-68-C-0514. Phase II proposed the machining and assembling the best of the three molded cases and comparative deflection and structural and compatability testing with the magnesium case. This report covers the Phase II effort.

II. SUMMARY AND RECOMMENDATIONS

The machining, static testing and dynamometer testing of a boron-glass-epoxy composite material gearcase has been successfully completed under Contract No. N00019-70-C-0224.

The housing involved in this program was satisfactorily machined to design tolerances. The required use and somewhat short life of diamond cutting tools plus other machining problems such as tapping, makes machining to the extent done on this program quite expensive. The tooling and processing approach utilized were tailored to the requirements of this one part and could not be economically applied to production. However, this material could be satisfactorily handled in production by incorporating the following recommendations:

- The amount of finish stock left on the molded part should be minimized wherever possible.
- Steel inserts should be molded in place for all tapped holes. These could be solid or tapped, depending upon location tolerances.
- Machined areas requiring surface finishes better than 80 RMS should be minimized.
- A definitive study should be initiated to obtain a wider range of machinability data on this material.

A general view of the finished case is given in Figure 1.

A static deflection test was performed to simulate the maximum case loading. A 12,000 lb. load was applied to the main drive gear bearing support by means of a hydraulic ram. Deflections in the case were measured by dial indicators. This test set-up and loading method duplicated a previous test on the magnesium case so that a direct comparison could be made. The composite case showed 16% less deflection at the front face and 11% less deflection at the support than the magnesium case. The composite case also showed less radial translation of the main bore than the magnesium case; 26% less at the support and 14% less at the front face.

The original design study (Program BC15) predicted a deflection (slope) reduction at the front face of up to 50%. The shortcomings of this program in element size, number of elements and nodes, computer precision and in-plane stiffness resulted in its inability to properly model the composite case. A new finite element program, BC83, greatly improves upon these shortcomings. This program will accept an idealization consisting of 200 nodes and 300 elements and run in double precision. The new program will also model external loads in the plane of the structure as long as the structure does not approach a beam configuration.

The following recommendations apply if increased stiffness is desired:

1. Eliminate the twelve (12) small external ribs and make the thickness transition from the front face to the side wall more gradual.

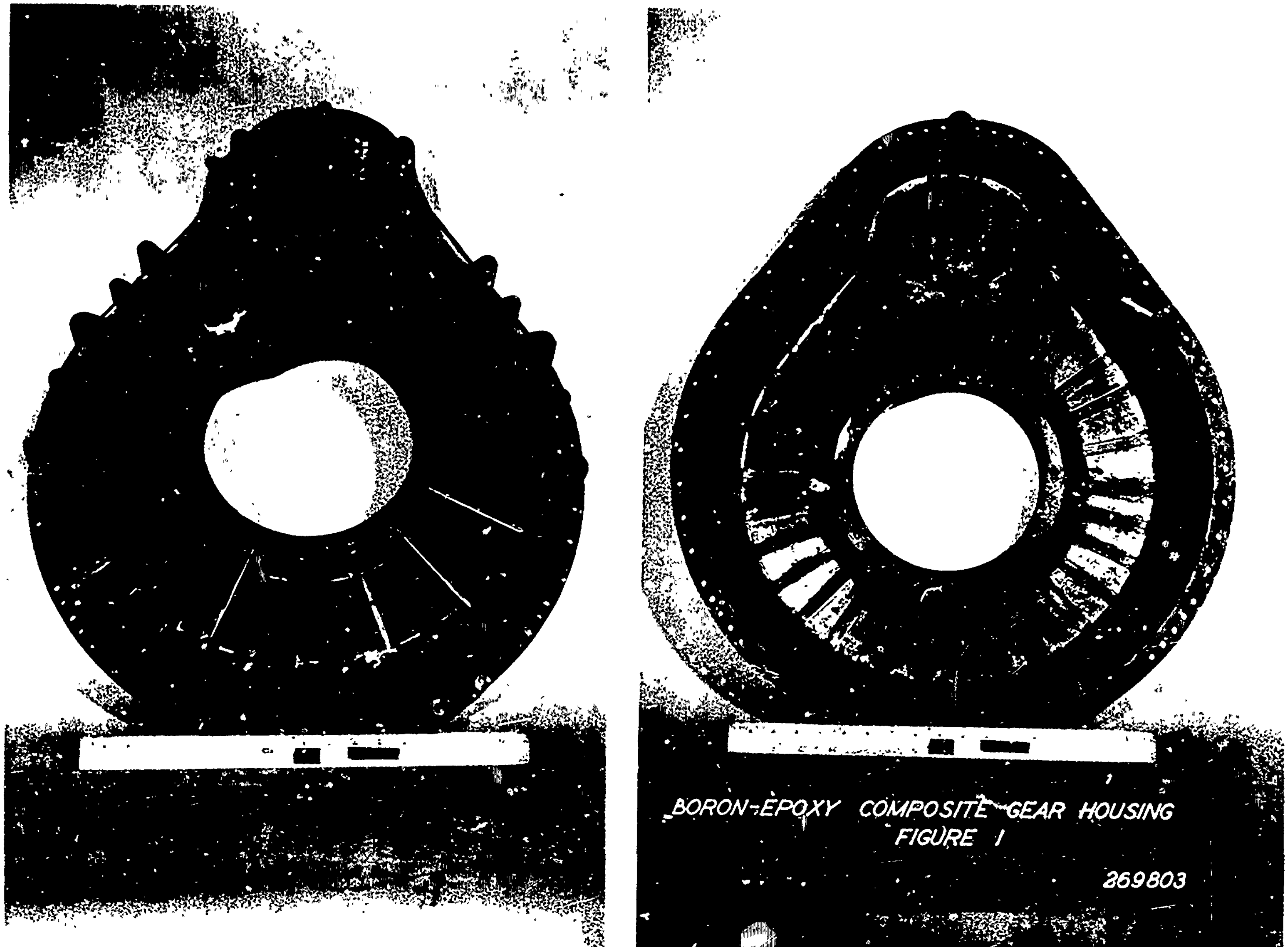
2. Increase the effective bending modulus of the main ribs.

3. Increase the thickness of all webs.

A ten-hour dynamometer test was completed at 5500 horsepower and 150,000 in. lbs. propeller shaft moment. The composite case completed this testing satisfactorily. Photographs were taken of the main drive gear bearing, propeller shaft bearing, pinion gear teeth, main drive gear teeth, sun gear teeth, planet gear teeth, and pinion drive shaft spline prior to the test.

Photographs of these same components after the testing indicated that their condition was unchanged. This fact leads to the conclusion that the stiffness of the composite material front gear case was adequate in reacting the imposed loads to maintain proper gear and bearing alignment.

The expenditure curve for this program is shown on page 43. This curve reflects expenditures through the end of December 1971. The program master schedule is given on page 44.



BORON-EPOXY COMPOSITE GEAR HOUSING
FIGURE 1

269803

3 b

III. COMPOSITE CASE MACHINING

Three housings were molded under Phase I of the composite case program. Of the two housings retained at Allison, one "good" housing was to be finish machined and the other was to be used for development and tool tryout. This tryout piece preceded the "good" housing on all new machining operations. No machining was attempted on the "good" housing until satisfactory results were obtained on the practice piece.

The general properties of boron filaments plus a small study made on drilling and tapping prior to the release of the actual housings indicated a need for special cutting tools. The drilling and tapping study on flat samples showed extreme wear on solid carbide drills. At this point, diamond tooling was indicated, but it was thought desirable to try other tool materials early in the machining development.

Tool inserts initially tried and discarded were G. E. Carboloy - C883, Dupont Baxtron (tungsten carbide), and Kennametal C06 aluminum oxide. All three materials showed extremely high wear. Resultant surface finishes were poor (over 250 RMS) and the taper generated by tool wear was out of design tolerances. Thus, these tool materials were discarded in favor of diamond tooling. Figures 3 and 4 present photographs of the non-diamond tools after use. It should be noted that, because of the machine used, the ceramic insert could not be evaluated at optimum cutting speeds. Thus, further evaluation is needed for this material.

<u>Single Point Diamond Tools</u> - Initial attempts at finishing the main center bore and pinion bore (see figure 2) were made with single point diamond tools. The housing was set up on a 66 in. vertical lathe for these trials. Enough wear on the diamonds was encountered to make design tolerances of ± .0005 over a 1.40 bore length impossible to hold. (See figure 5 for photographs of a diamond tool after use). This, again, is not a total evaluation of diamond tool life. The machine used limited the cutting speed to 550 surface feet per minute (SFM) whereas diamond tools are most efficient at speeds of 1000 SFM or above. Tool configuration was also not ideal. As indicated in further development, a high negative rake tool provides better tool life. The tools used on the lathe were zero degree rake.

Additional single point tool usage involved machining a mounting face adjacent to the pinion bore and a clearance cut near the main flange face (see figure 2). Because these areas were contoured, the housing was set up on a 2-dimensional vertical spindle contour mill with templates. A single point diamond tool with 20° negative rake was set at .70 radius in a fly cutter. The necessary machining was done with one tool and very little diamond wear was experienced. The machined surfaces met design tolerances with flatness held within .001 total indicator reading on the pinion bore mounting face. The best surface finish obtained was 150 RMS and this was deemed acceptable by Engineering Design. The improved cutting efficiency on this operation over those on the lathe was attributed to the high negative rake and a slightly increased cutting speed (660 SFM).

Machining parameters used for both boring and flycutting are presented in Tables I and II.

Diamond Grinding - A 5 in. OD - 180 grit diamond cup wheel was used to finish machine the main flange face, center bores and pinion bore (see Figure 2). This wheel was mounted on an Ex-Cell-O tool post grinder set up on the ram of a vertical lathe. Machining was done at the parameters specified in Table I.

Wheel wear was quite evident, with approximately 1/4 in. of diamond length used to remove .080 average stock from the main flange face and to finish the bores. Design tolerances were easily held by using several .002 deep finish cuts. This effectively removed the taper generated by tool wear from the heavy roughing passes.

Surface finish, as with single point tools, did present a problem. The best finish obtained by grinding was 20-80 RMS. The high roughness resulted when bundles of boron filaments near the surface were cut. Because of the boron hardness and the relatively soft epoxy backup, the filaments tended to fracture rather than cut cleanly. In figure 6, the roughness of the machined boron filaments is evident next to the fine finish on the base epoxy.

Some specific areas required a 32 RMS or better finish for "O" ring seals. These areas were machined to allow for a .002 per side coating of a Dow Epoxy Novalac 438-Nadic Methyl Anhydride cured resin system. This resulted in a 20-30 RMS finish on the pure epoxy after regrind, and was acceptable from a design standpoint. Figure 8 is a Proficorder peak-to-valley trace of both a boron-epoxy and pure epoxy surface.

Drilling and Tapping - Diamond drills were planned for use from the start of the housing machining program. These were diamond core drills purchased from Starlite Industries, Rosemont, Pennsylvania, and consist of steel tubing with diamonds applied to one end (see figure 7). Along with the core drills, a Starlite combination collet-water swivel was obtained to use in conjunction with the hollow drills. This device was to apply pressurized coolant (40 PSIG) thru the ID of the drill for cooling and chip flushing purposes.

During the tool tryout phase on the "practice" housing, the water swivel was tried and discarded. It did supply a good coolant stream through the drill, but inaccuracies in the collet section created too much drill runout for practical use (.003-.010 total indicator reading). The final set-up used for the "good" housing was a standard chuck with an external coolant supply.

Initial development drilling showed most of the core drills cutting oversize and out of design tolerance limits. This was due in part to some drill runout in the chuck, but primarily to most of the drills being on the high size limit specified (see Table III). Where possible, the drills were reworked by dressing with a diamond grind wheel. After rework, as shown in Table III, the drills did cut within design tolerances.

The machining parameters used for drilling are shown in Table II. On the "practice" housing, power feed was tried with the water swivel and with external coolant. Results were poor in that the coolant would not flush entirely around the drill. This caused excessive heat buildup in the drill and some burning of the epoxy matrix. The epoxy would load the drill and reduce cutting efficiency. Also the plug created by the core drill would remain in the drill and presented quite a problem in removal. The final method used of raising the spindle to clear and cool the drill after every .010-.020 cut depth proved quite satisfactory. No drill loading was experienced and the plug came out easily. The plug left in a blind hole was easily removed by using an undersize carbide spade drill or end mill. These tools dulled rapidly, but the unsupported plug delaminated easily and was simple to remove.

The diamond plated twist drills shown in Figure 7 were not tried in this program because of lack of time. If data on these drills are obtained in the future, this report will be amended to include the results.

Tapping was done by hand using standard 4-flute high speed steel taps. All the threaded holes were modified class 2B fits and GH-5 lead and bottom taps were used. A tool life of (2) holes per lead tap and (4) holes per bottom tap was obtained. For the one housing involved in this development program it was considered acceptable to obtain this short life on inexpensive taps rather than to try nitrided taps. It is doubtful that nitrided taps would have given much improvement in tool life. This short tap life is the basis for the recommendation of molded in place inserts.

Miscellaneous Operations - Some required machining operations were not specifically tooled with respect to the composite material machining characteristics. These operations were handled in the following manner:

Chamfering - Chamfering on the large center bore was done accurately with a diamond grind wheel. In areas where the diamond wheel would not work, chamfers were done by hand with silicone carbide grind sleeves.

Countersinking - 90° countersinking on threaded holes was done with miniature aluminum oxide grind wheels. Because of wear on these wheels, the resulting countersink was more like a rounded break.

Small Hole Boring - Jig boring was required on small, close tolerance holes such as dowel holes. This was done by using an undersize diamond core drill mounted in a boring head. This worked quite satisfactorily and tolerances of ± .00025 were held.

Burring - All burring was done by hand using an air grinder with silicone carbide grind sleeves.

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TABLE I

Boring and Grinding

Machine: Bullard Model 75 - 66" Cut Master VTL
Allison Tag #81892

Grinding Adapter: Ex-Cell-O Corporation Tool Post Grinder

Tooling: Boring - Single Point Diamond - .040 & 3/32 Radius - 0 to 7° Positive Rake (See Note)

Grinding - Norton 5" OD - 180 Grit Diamond Cup Wheel

Machining Parameters:

Boring:	Table Speed	- 125 RPM
	Feed	- .0013 - .002
	Cut Depth	- .004-.006 Rough
	Coolant	- Water Soluble Oil (See Note)
Grinding:	Table Speed	- 47 RPM
	Quill Speed	- 2600 RPM
	Feed	- .0078 Rough & Finish
	Cut Depth	- .006 Rough .001-.002 Finish
	Coolant	- Water Soluble Oil

Note: The single point diamond tooling used and the machining parameters established are useable but not optimum. See the [illegible] section of this report for further explanation.

TABLE II

Flycutting, Drilling and Tapping

Flycutting

Machine: Cincinnati Hydro-Tel - Single Spindle Vertical Mill
U. S. Ordnance Tag No. 93775

Tooling: Single Point Diamond - .040 Tip Radius -
20° Negative Rake - .70 set radius

Machining Parameters:

Spindle Speed	-	1800 RPM
Feed	-	.001
Cut Depth	-	.030 Rough .002 Finish
Coolant	-	Water Soluble Oil

Drilling and Tapping

Machine: SIP - Hydroptic 7A Jig Bore
Allison Tag No. 210825

Tooling: Starlite Diamond Core Drills
Standard Beasly 4 Flute H.S. Steel Taps

Machining Parameters:

Drilling -	Spindle Speed	-	1600 to 2000 RPM
	Feed	-	Hand
	Cut Depth	-	.010-.020 Between Drill Clearing Stroke
	Coolant	-	Water Soluble Oil

Tapping - By hand with no cutting fluid

TABLE III

Diamond Core Drills

Requested Drill Dia.	Actual Drill Dia.	Diameter Hole Cut	Hole Diameter After Rework
.250	.252	.252/.254	No Rework Required
.161	.1595	.160	No Rework Required
.271/.276	.2705	.274	No Rework Required
.497/.498	.5085	.508/.513	Not Reworkable
.3746/.3751	.381	.382/.385	Not Reworkable
.331/.336	.3355	.3385/.3390	.3365
.279/.288	.287	.290	.285
.452/.457	.461	.463/.464	.4565
.396/.401	.395	.399	No Rework Required

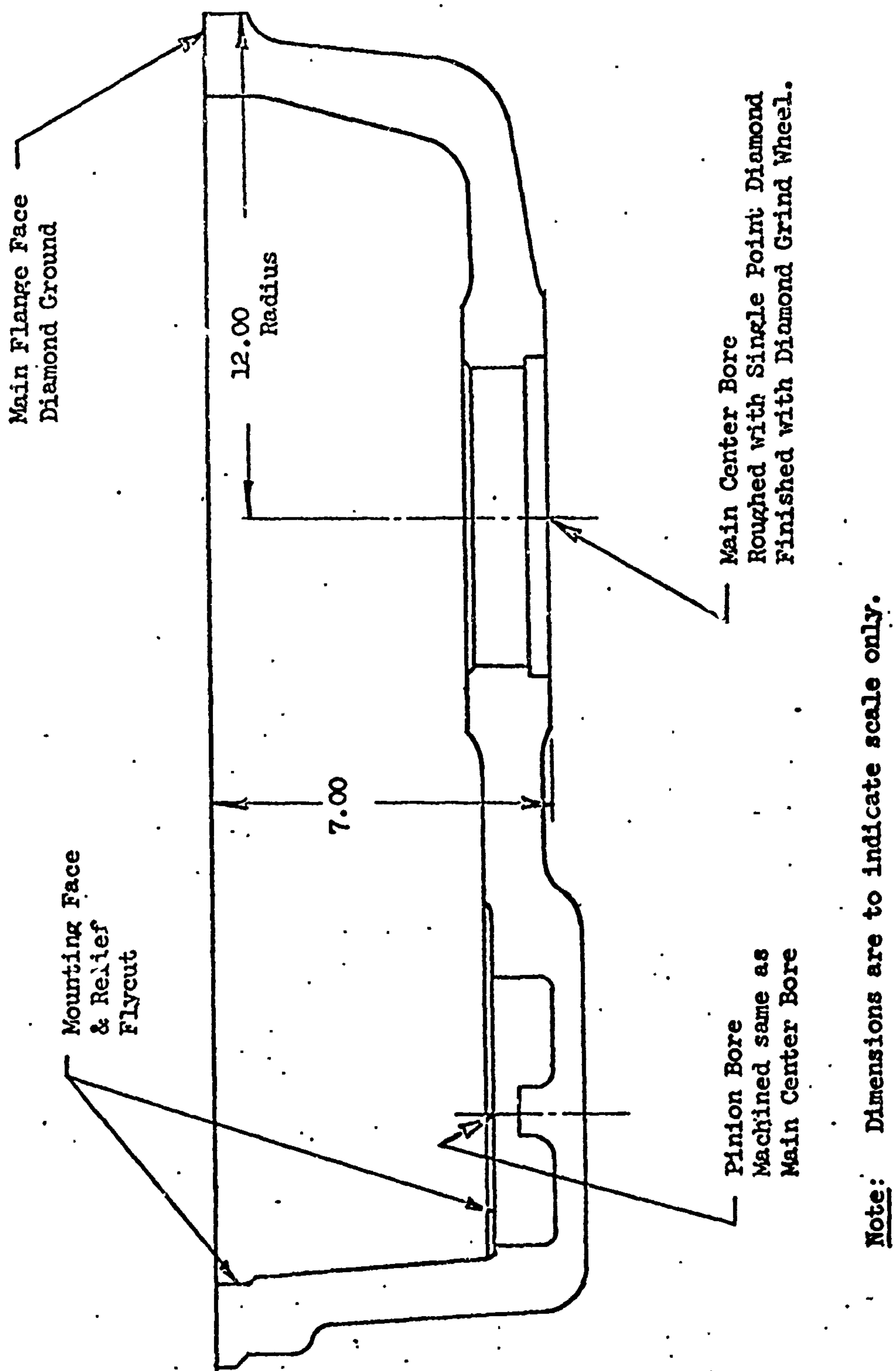
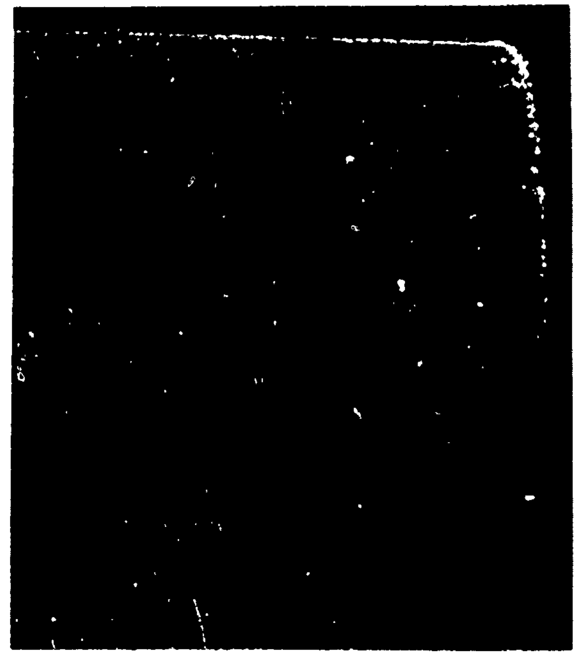


Figure 2
Front Reduction Gear Housing

VIEW A
25X Magnification



VIEW B
25X Magnification

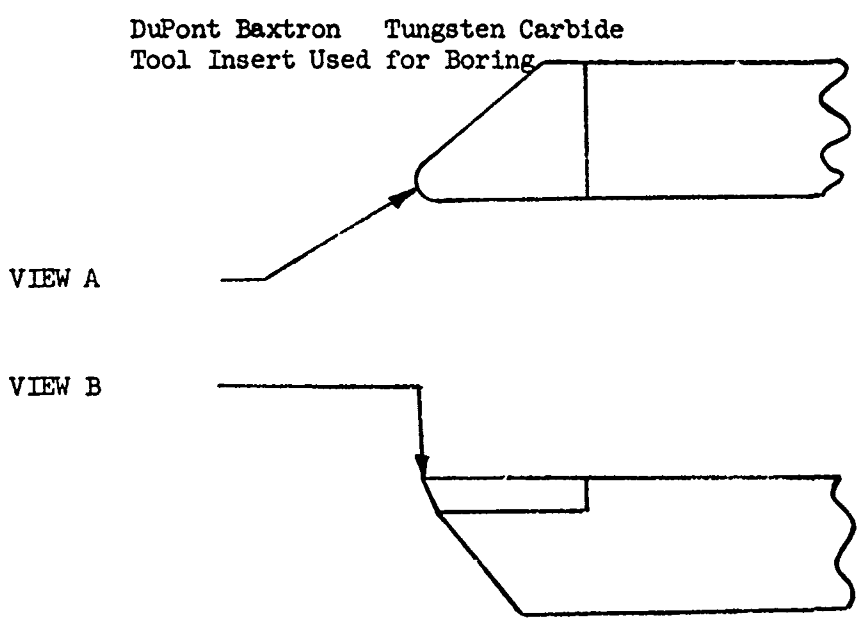
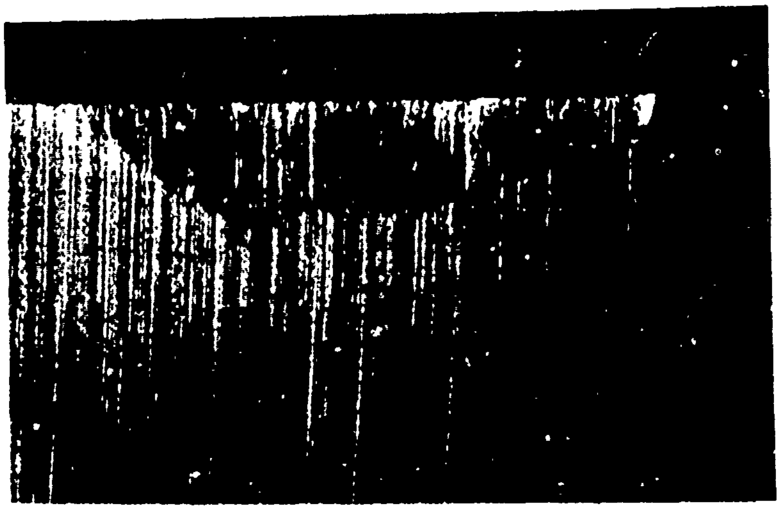
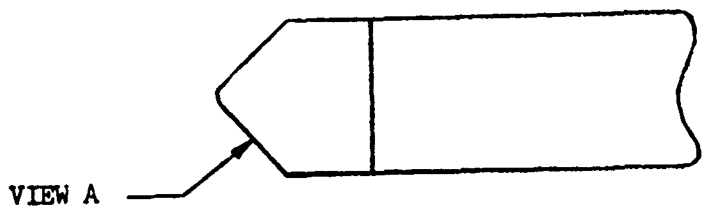


FIGURE 3



VIEW A - 0883 Carboloy
25X Magnification



VIEW A - Ceramic Insert
25X Magnification

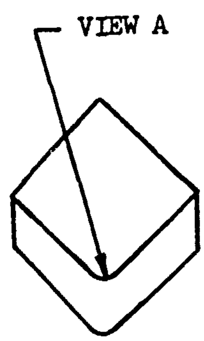


FIGURE 4

VIEW A
25X Magnification

VIEW B
25X Magnification

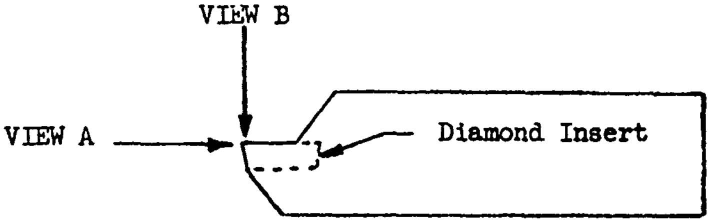
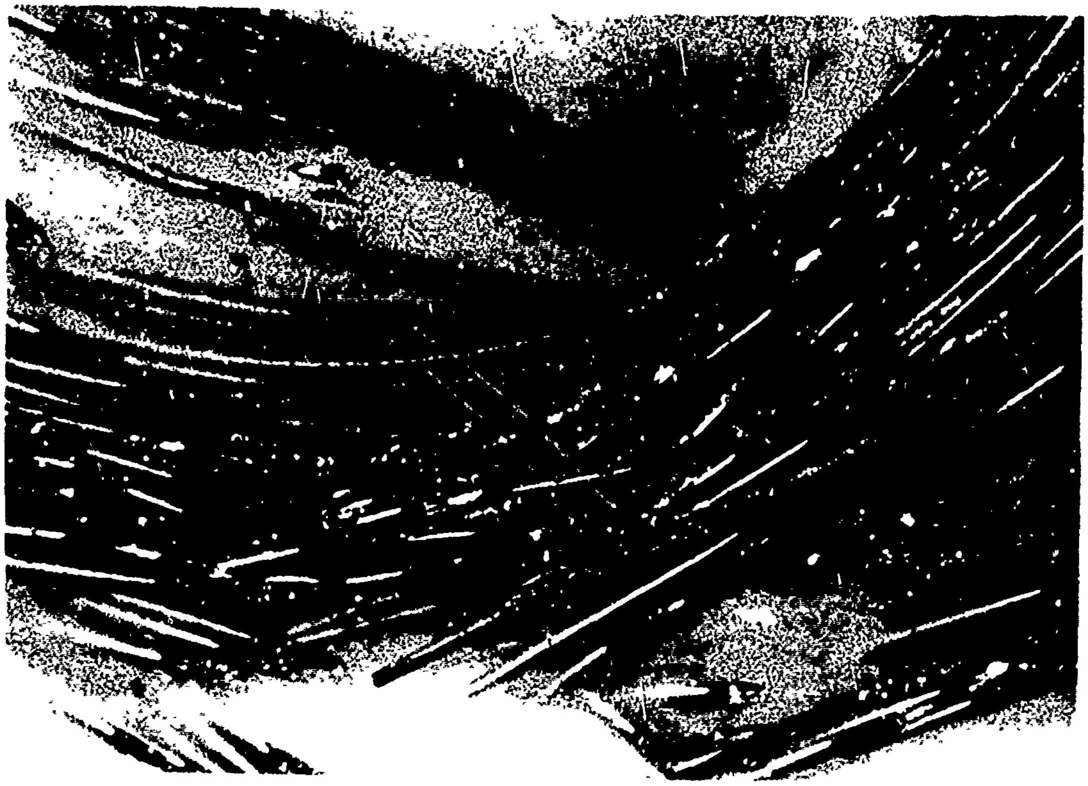


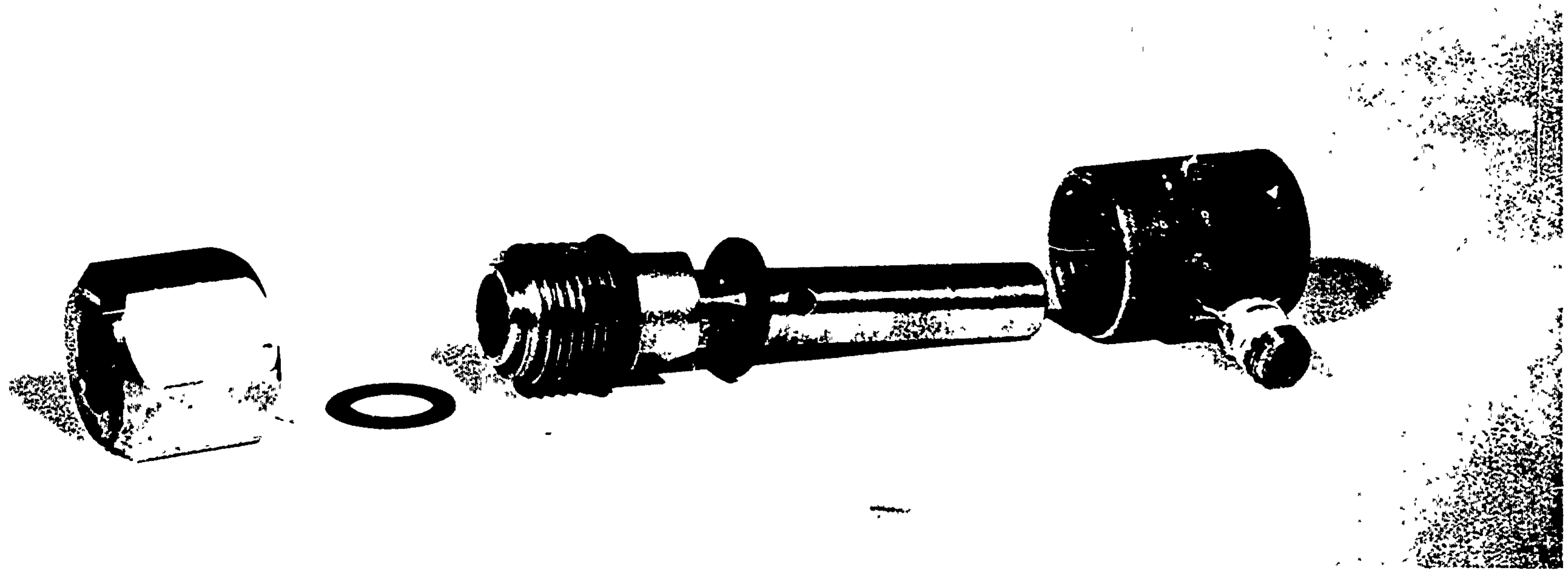
FIGURE 5
3/32 Rad. Diamond Boring Tool



25X Magnification

FIGURE 6

Boron-Epoxy Surface as Finished With A
Diamond Grind Wheel



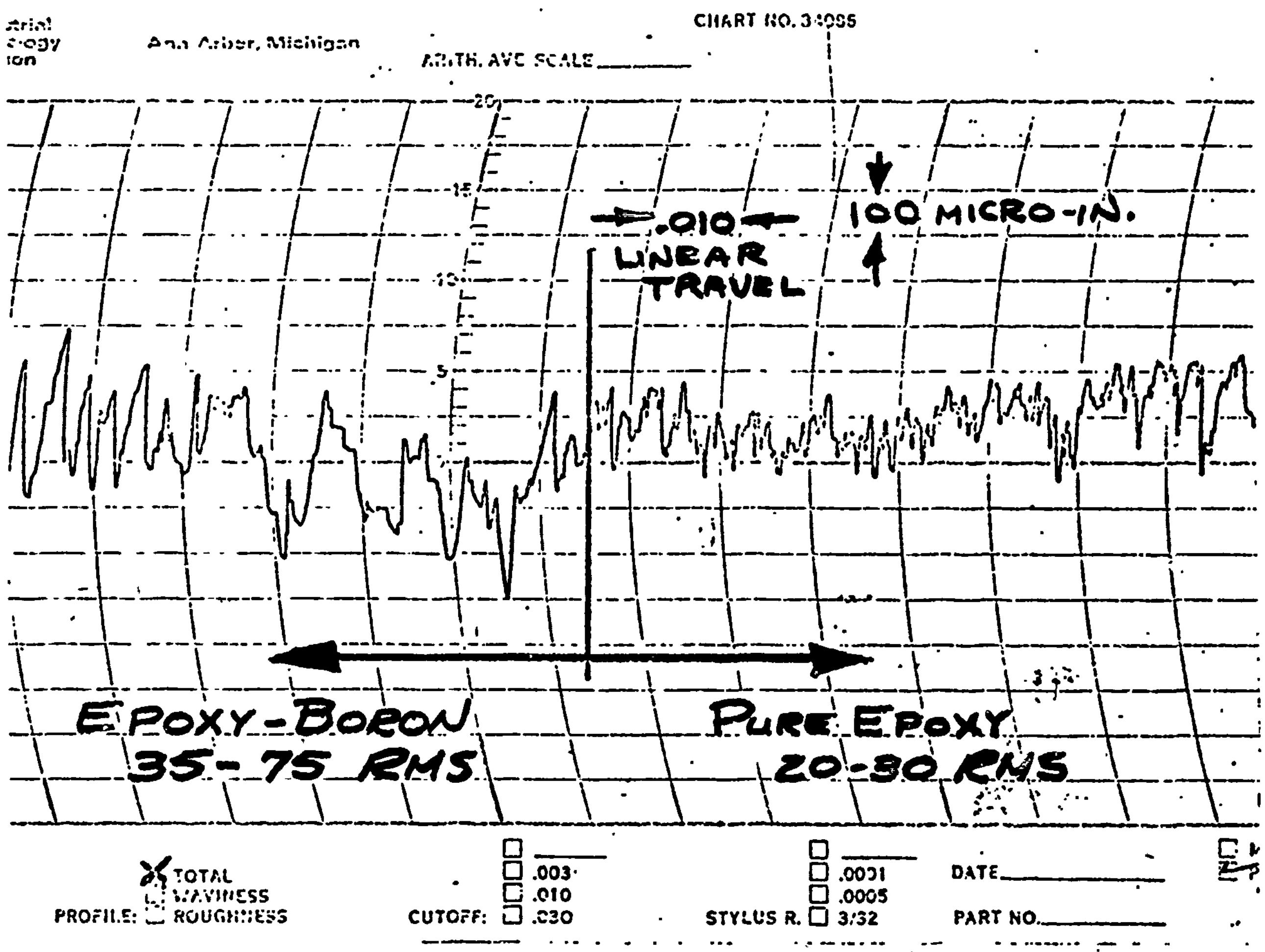


Figure 8

Proficorder Trace of Pure Epoxy and Boron-Epoxy Surface Finish

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IV. STATIC DEFLECTION TEST

The most severe load taken by the case is the 12,000 lb. torque component reaction from the first stage gearing. This load is applied at the main drive gear bearing support and produces a moment which creates deflections in the front case face. Development testing of the magnesium case indicated that the deflection of the front face significantly affects the power train gear and bearing alignment. A static deflection test was performed to simulate the maximum case loading.

Test Set-Up

The case was mounted to a steel plate which was supported by ram stands. A hydraulic ram supplied the force from outside the case by means of a fulcrum bar loading into a steel ring. The load was transferred from the ring through the bearing rollers to the bearing support. Figure 9 shows a sketch of the test set-up. A photograph of the loading method is shown in Figure 10.

Instrumentation

Dial indicators were installed to obtain front face deflection at the center bore, front face deflection at the pinion bearing pad, main drive gear bearing support deflections and other deflections on the outside of the case. The locations of the indicators are shown in Figure 11. A photograph of the instrumentation is presented in Figure 12.

Test Results

The test results on boron-epoxy case are given along with data previously obtained on the magnesium case in Figure 13. These values have been corrected for any deflection of the steel mounting plate. The composite case showed less slope change than the magnesium case; 16% less at the front face and 11% less at the support. The composite case also showed less radial translation of the main bore than the magnesium case; 26% less at the support and 14% less at the front face.

Engineering Evaluation of Test Results

In general, the composite gear case was designed for a specific deflection, i.e. slope, across the center hub of the front face. The following conclusions are based upon this deflection.

1. The composite case is 1.16:1 times as stiff as the magnesium case instead of the 2:1 as predicted.

2. The original design prediction for the composite case, based on computer program BC15, was 33% below the measured test data.

3. New calculations, using an improved finite element program, BC83, gives a slope prediction that is 7.8% above the test data.

Confidence in the original design calculations, using program BC15, was based on an analysis of the T56-A-18 magnesium case for which existing deflection test data were available. The analysis predicted the front case face average deflections for the magnesium case within 10% of those measured during testing. The shortcomings of this 1st generation finite element program in element size, number of elements and nodes, computer precision and in-plane stiffness has contributed to its inability to properly model the composite case.

Due to computer storage limitations, the idealization of the case had to be limited to 66 node points and the computer was limited to single precision. This limitation greatly restricted the analysis such that all beams and plate thicknesses could not be properly modeled. The BC15 program was also limited to external normal or bending moment loads and thus would not model external loads in the plane of the structure. This restriction means that only the moment due to the main drive gear bearing load could be applied to the case and not the bearing load itself.

A new finite element program, BC83, greatly improves upon these shortcomings. This program will accept an idealization consisting of 200 nodes and 300 elements and run in double precision. The new program will also model external loads in the plane of the structure as long as the structure does not approach a beam configuration.

An 1130 computer plot of the front projection of the finite element idealization of the composite case for program BC83 is given in Figure 14.

A plot of the various front face slopes, computed and measured, is given in Figure 15. The plot shows that the BC83 program results agree with the test data as close as could be expected. All material properties, tolerances and all test variables would have to be known for better correlation.

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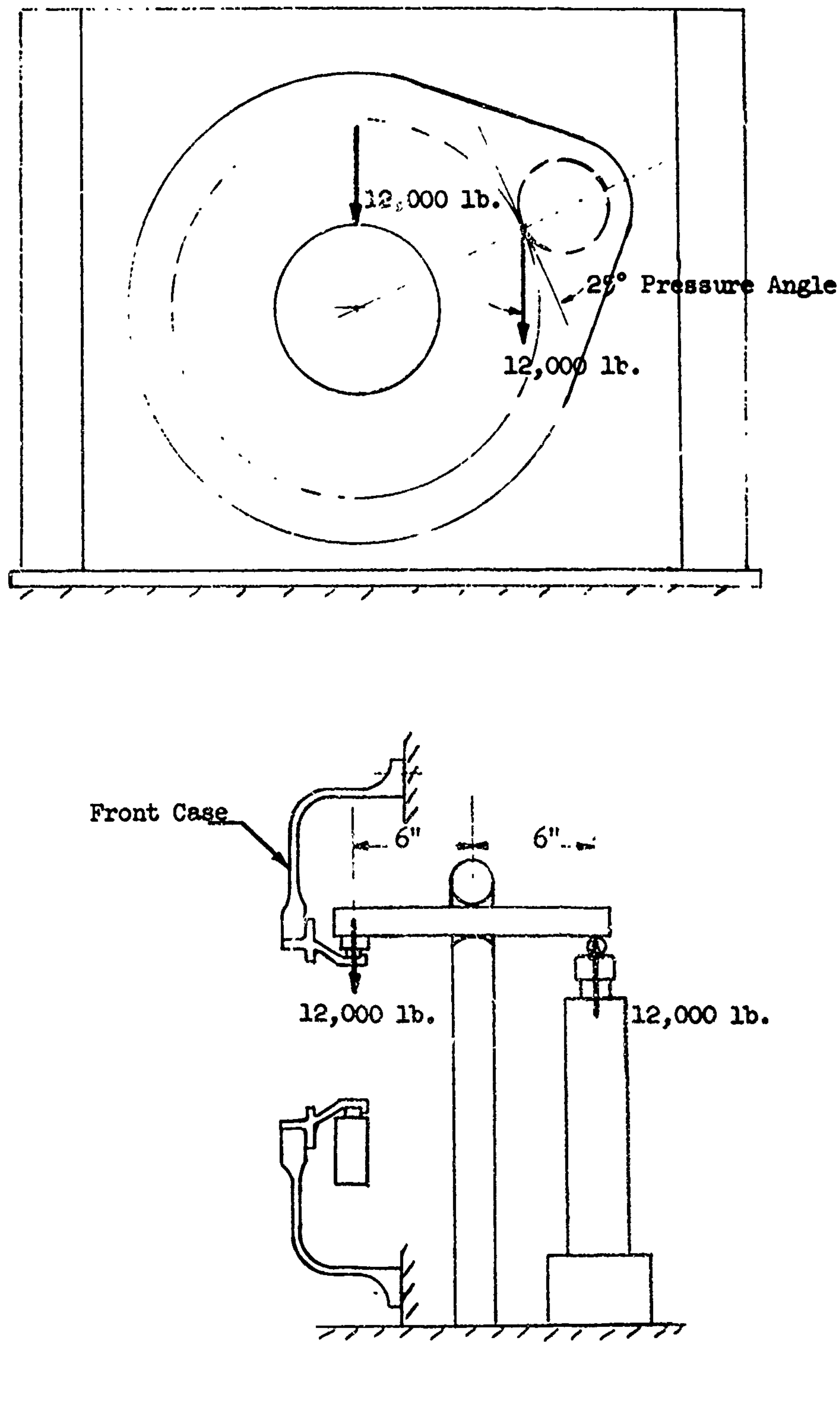


Figure 9 - Static Deflection Test Set-Up

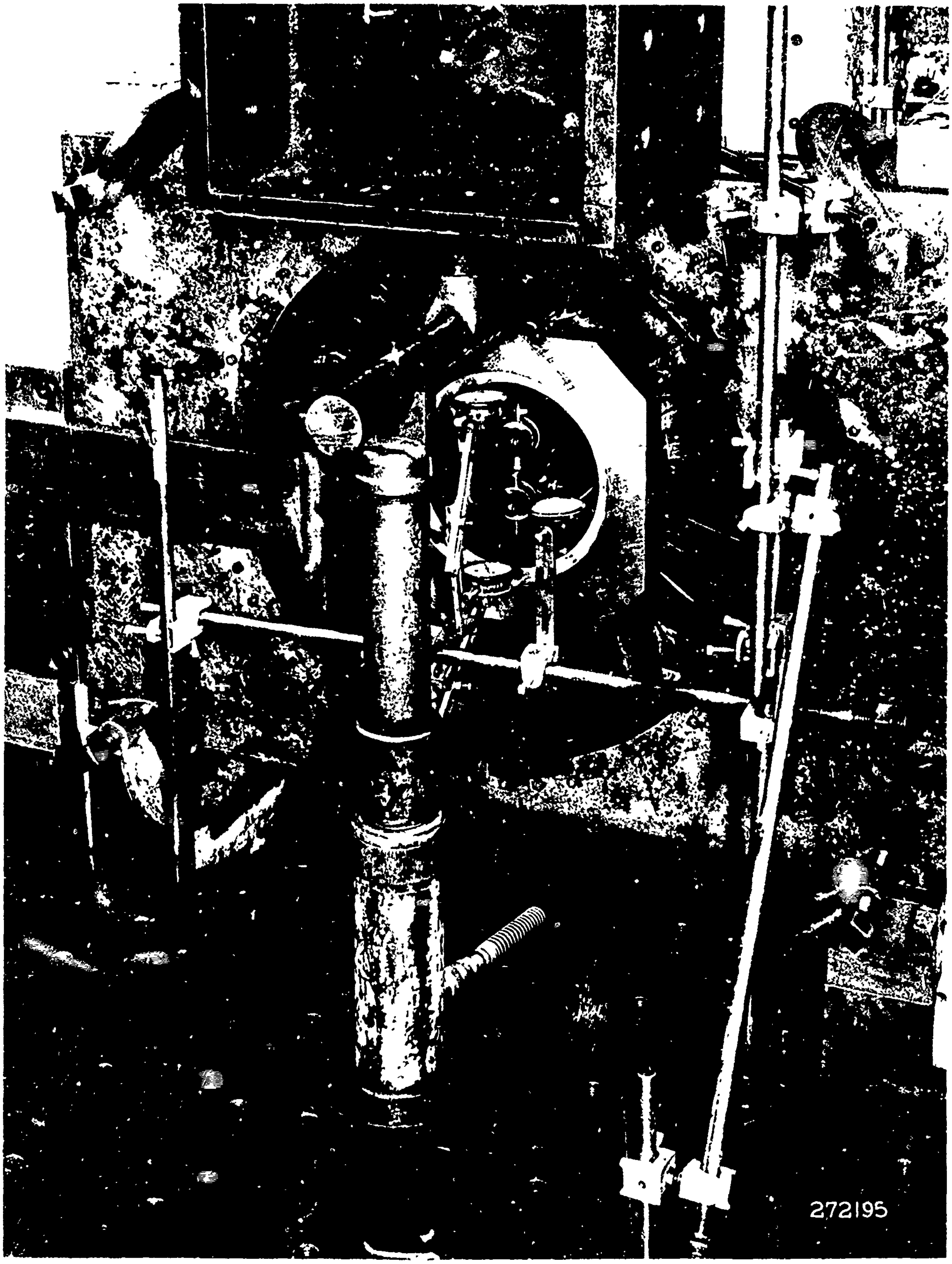


FIGURE 10 STATIC DEFLECTION TEST LOADING

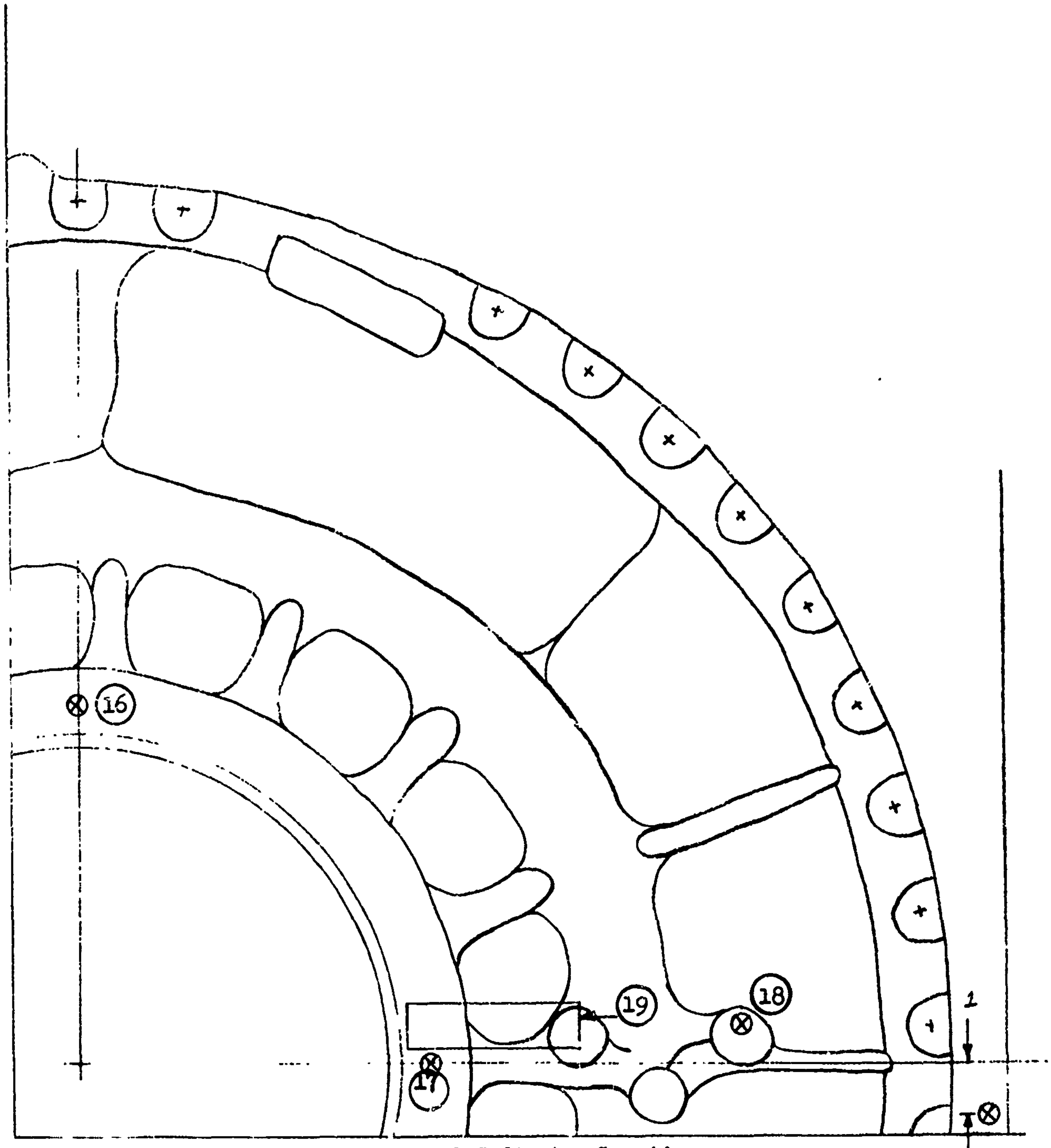


Figure 11 - Dial Indicator Locations

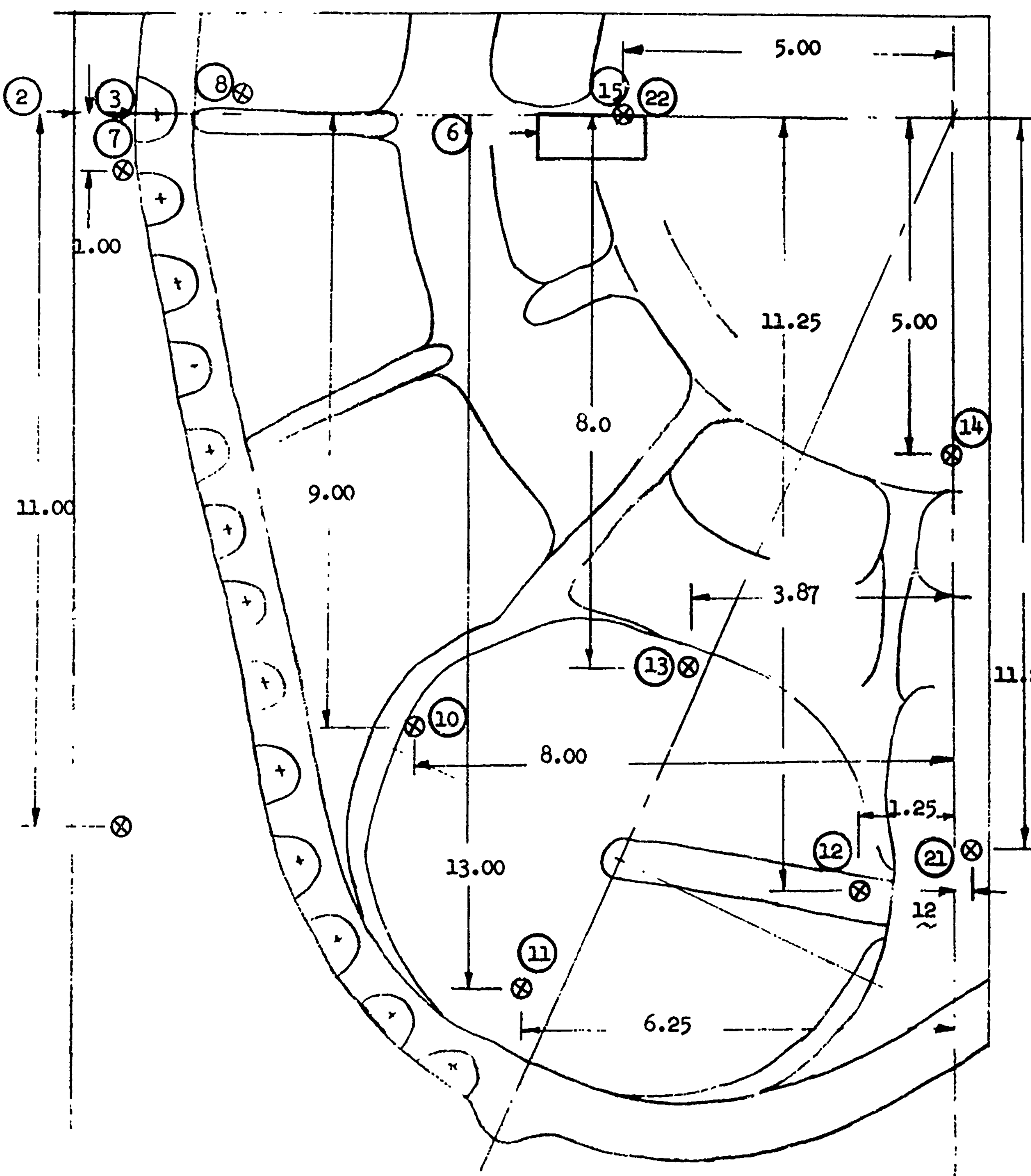


Figure 11 - Dial Indicator Locations

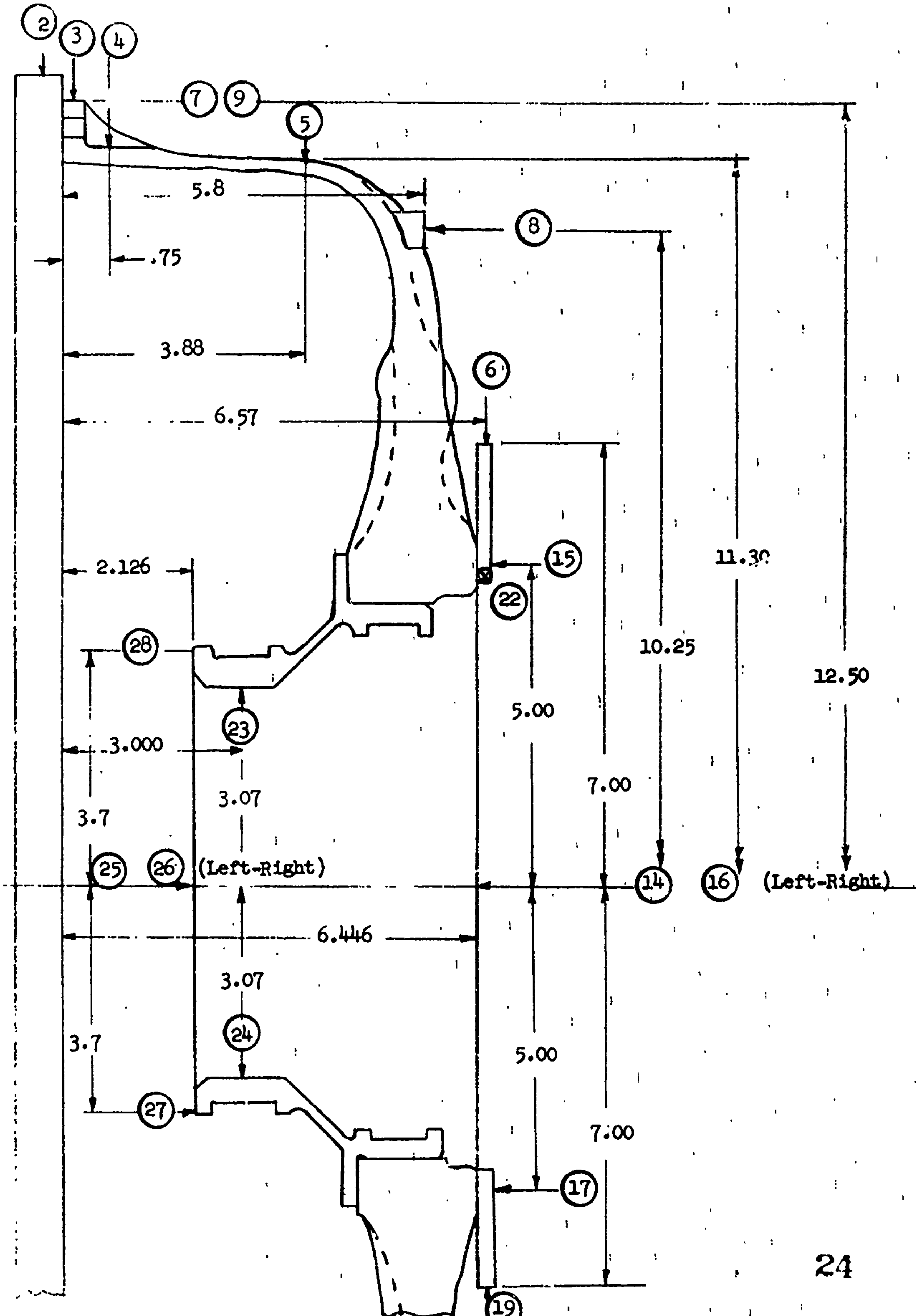


Figure 11 - Dial Indicator Locations

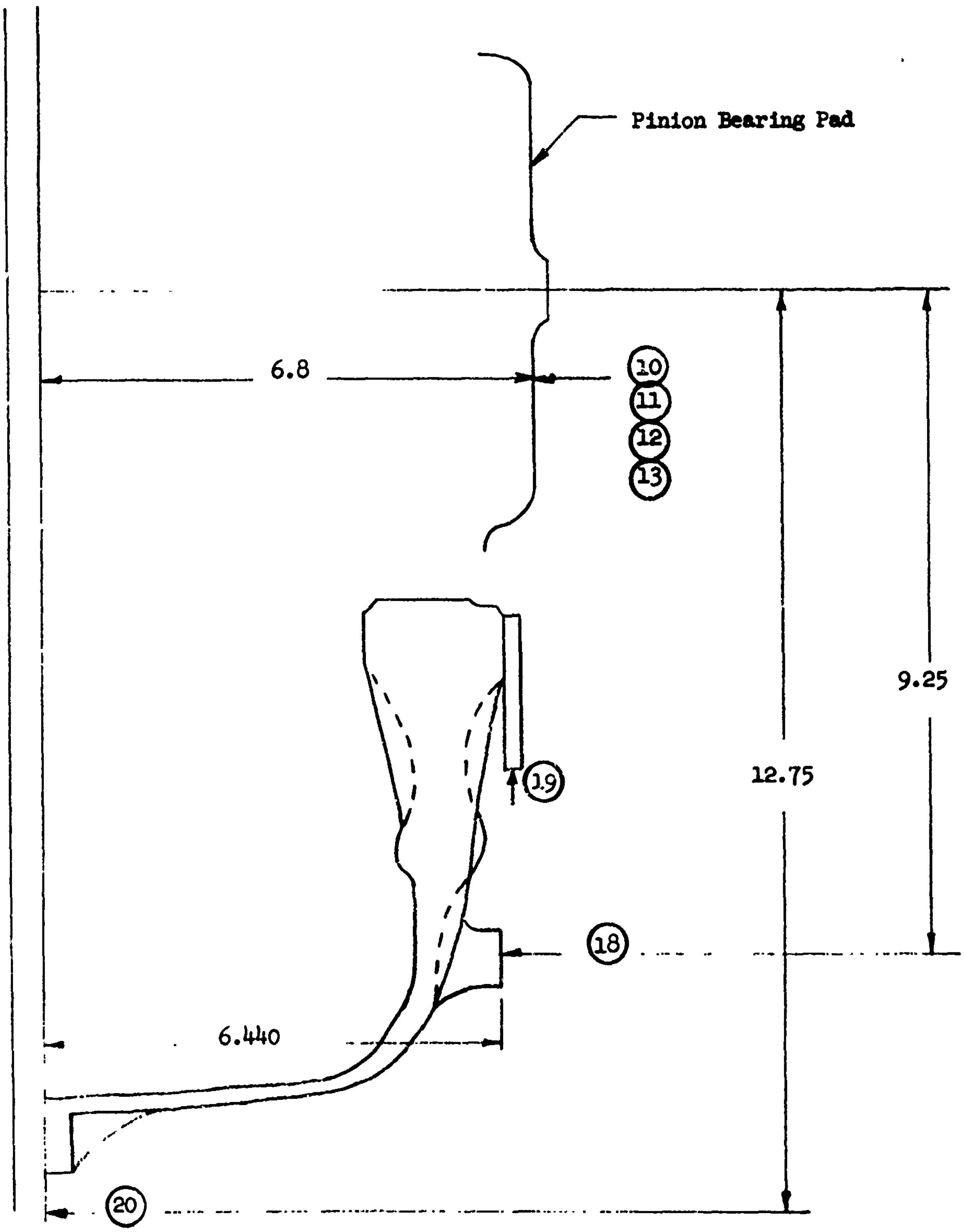


Figure 11 - Dial Indicator Locations

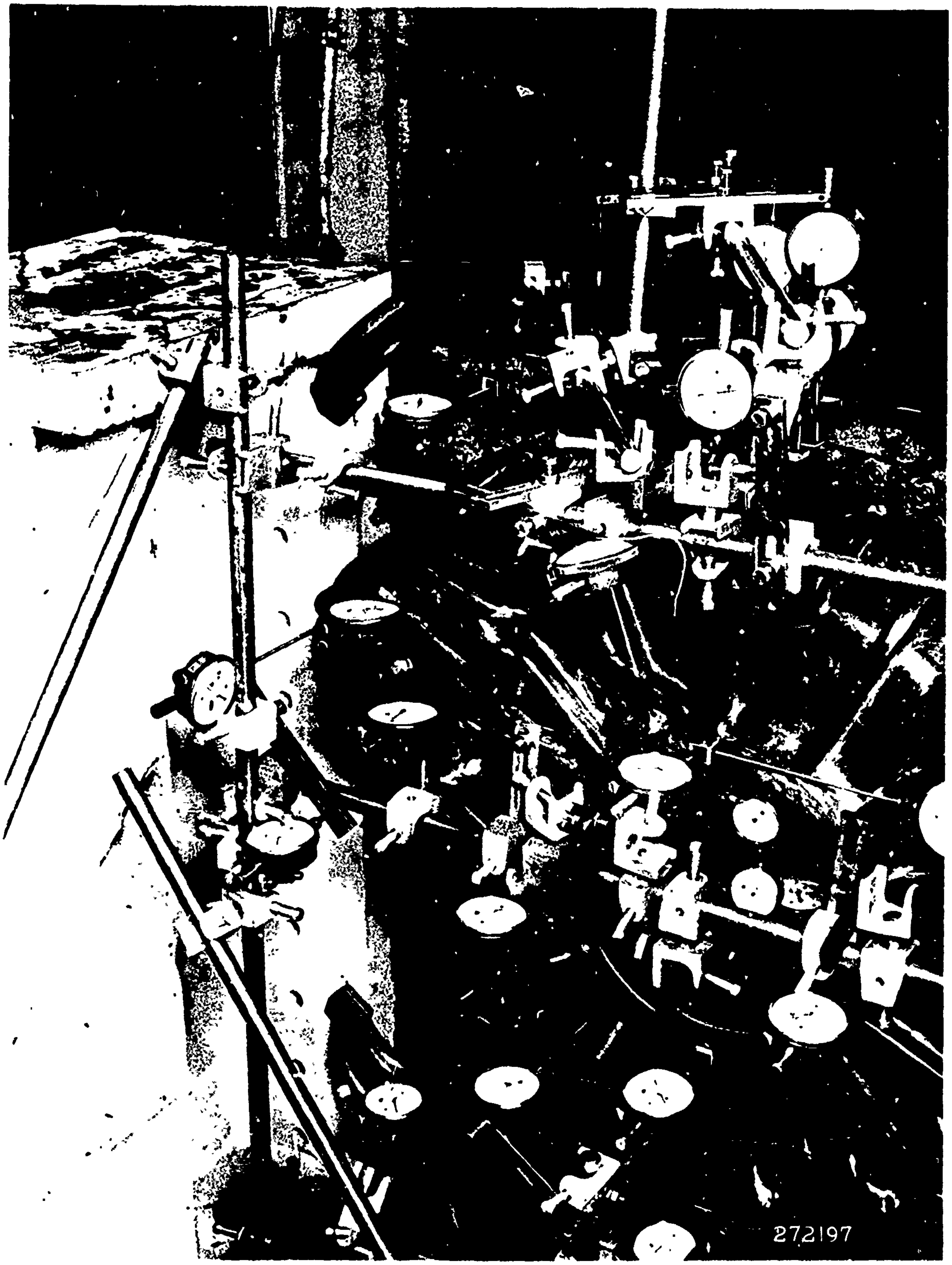
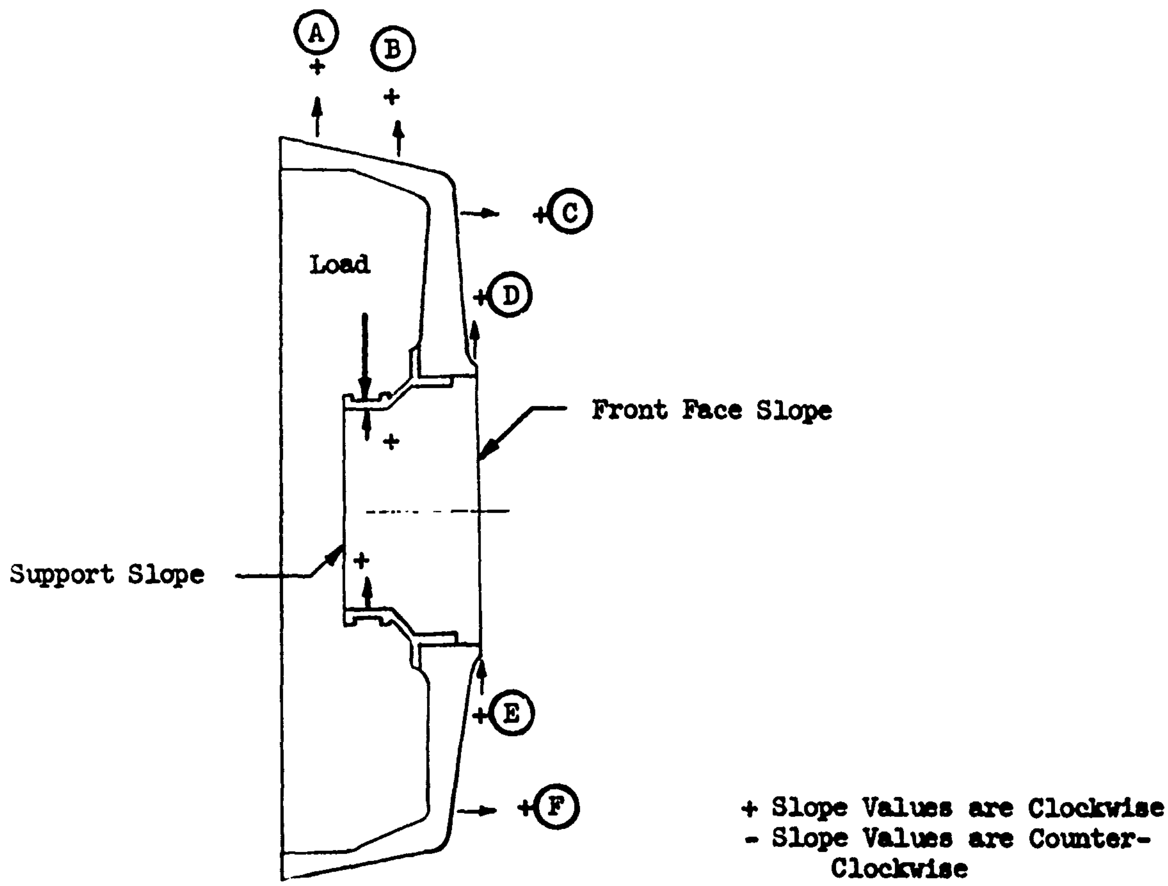


FIGURE 12 STATIC DEFLECTION TEST INSTRUMENTATION

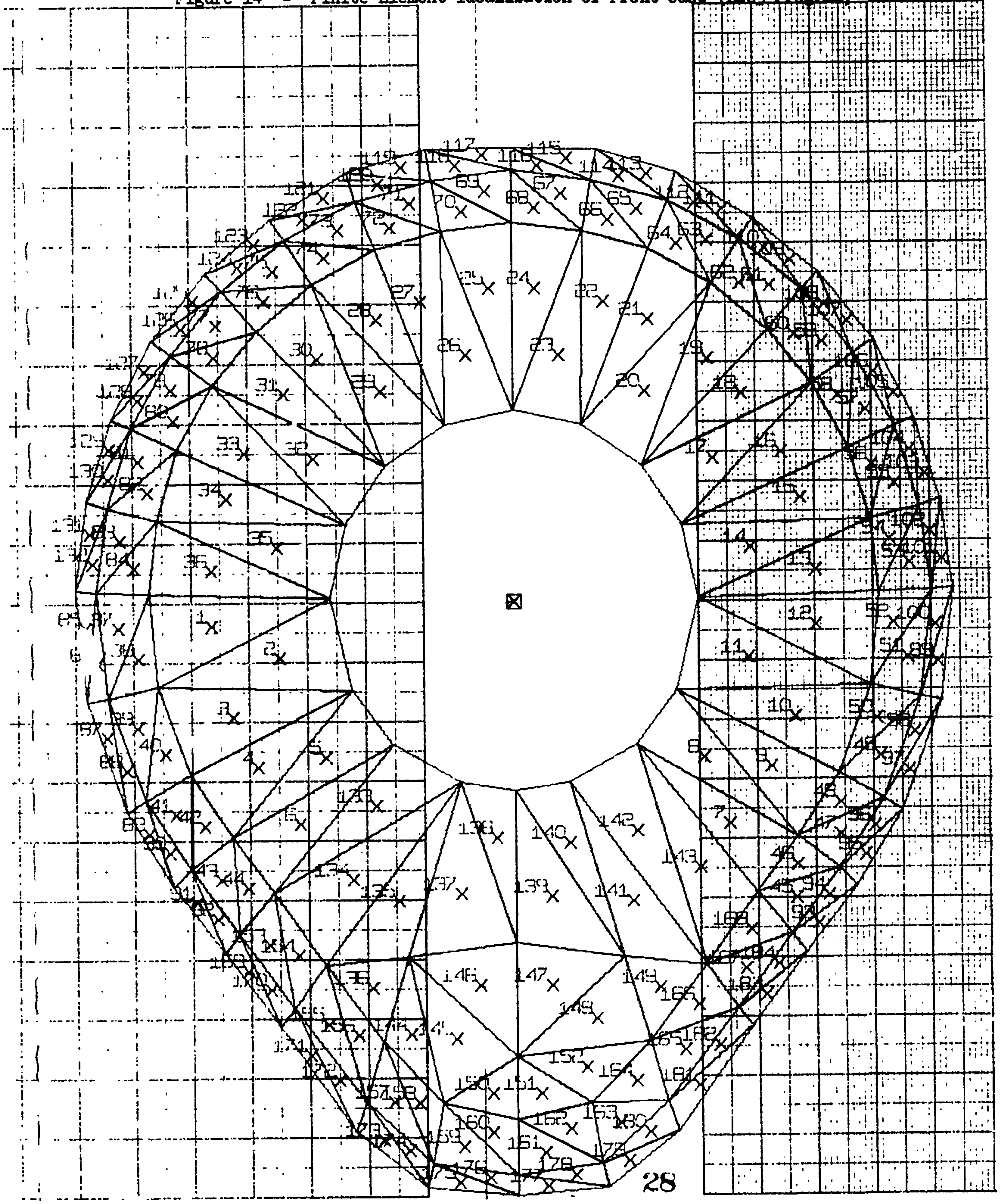


Deflection Item	Magnesium Case	Boron-Epoxy Case
Front Face Slope (in/in)	-.00212	-.00179
Support Slope (in/in)	-.00221	-.00197
Pinion Face Slope (in/in)*	-.00003	-.00042
Support Radial Def. (Top)	-.0196	-.0145
Support Radial Def. (Bottom)	-.0116	-.0085
Point A	-.0009	+.0005
Point B	+.0008	-.0051
Point C	-.0055	+.0028
Point D	-.0088	-.0076
Point E	-.0078	-.0058
Point F	+.0064	+.0092

* This slope measured by indicators #10 & 12 (Figure 11)

Figure 13 - Static Deflection Test Results

Figure 14 - Finite Element Idealization of Front Case (BC83 Program)



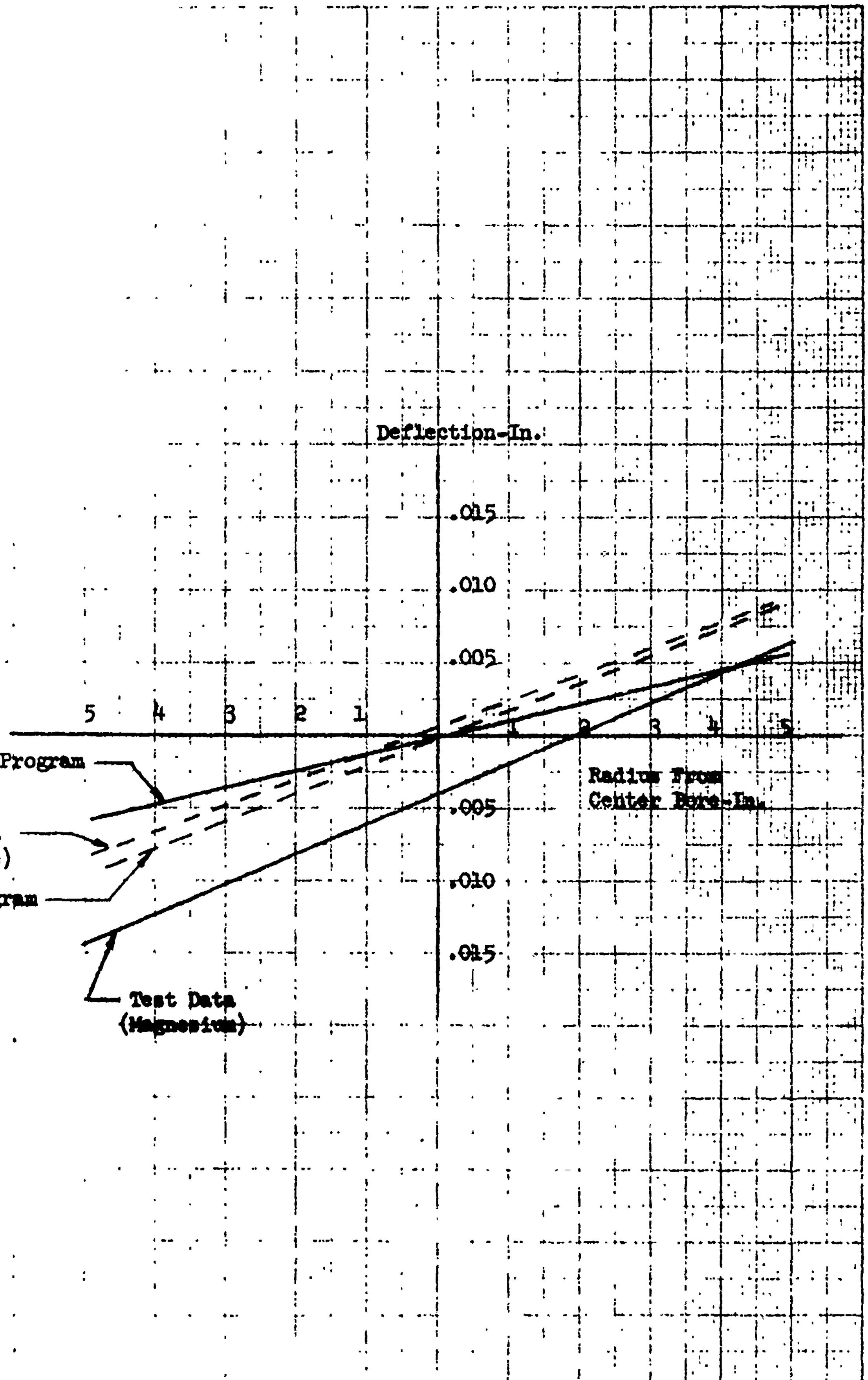


Figure 15 - Comparison of Slope Deflections

V. DYNAMOMETER TESTING

The composite material front housing was tested on the back-to-back dynamometer as part of Reduction Gear S/N 507, Buildup 18. A moment shaft rig was used in lieu of a propeller to couple the test unit to the back-to-back rig.

The test rig consisted of two separate heavy duty gear boxes and interconnecting shafting so arranged that output rotation of the test gearbox propeller shaft was redirected to the test gearbox input shaft with a speed increase ratio exactly matching the test gearbox reduction ratio. A torque applier device was build into the high speed gear box which permitted introducing torsional wind-up into the back-to-back torque loop. The test rig was motored to 13,820 RPM at the torquemeter shaft by two 500 horsepower dynamometers. Propeller shaft torque was transmitted to the low speed gear box by means of a heavy duty constant-velocity universal joint assembly. Propeller shaft moment was applied by hydraulic rams acting parallel to the propeller shaft centerline and extending from the gear box mount to arms attached to the moment rig housing. The net forward thrust of the rams was transmitted to the propeller shaft and taken to the test gear box rear housing through the propeller thrust bearing.

A general view of the test set-up showing heavy duty gear boxes, universal joint assembly, moment applying equipment and the test gear box is given by figure 16.

Lubrication System

MIL-L-23699 lubricant was used for this test. MIL-L-23699 lubricant was chosen because it had been used for the majority of running during development of the model 501-M22 gear box. The test stand oil system consisted of a large supply tank holding approximately 40 gallons of lubricant, a supply pump with remotely controlled bypass valve for inlet pressure control, filters and steam and water heat exchangers with an automatic mixing valve for temperature control. The reduction gear scavenge pumps returned scavenge oil to the tank. Total oil flow to the gear box was measured by a 1.25 inch Potter flowmeter. A 0.5 inch Potter flowmeter was used to measure oil flow rate in the line which supplied the planetary system.

During the test, oil inlet temperature was controlled so as not to exceed 150°F oil outlet temperature. This was done to avoid excessive radial forces at the splitlines due to differential case expansion. Typical inlet temperature was 88°F, and typical outlet temperature was 144°F.

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Test Schedule

The loads applied during 10 hours of loaded running were 5500 propeller shaft horsepower (5580 HP as measured by the torquemeter), 150,000 in. pounds horizontal moment applied by hydraulic rams, 50,000 in. pounds moment vertically down applied by weight of the moment rig and universal joint assembly, and 10,000 pounds forward thrust due to hydraulic ram loads. All running at the above loads was counted toward accumulating the desired 10 hours, regardless of the length of the period run. The direction of the horizontal moment was alternated each half hour of endurance from clockwise to counter-clockwise, beginning with clockwise as viewed from above.

Running Time Summary

The following running was accumulated on GB 507, BU18, during 2 phases of testing.

	Hours at Powers Below 5575 T.M. HP	Hours at Powers 5575 & Above, T.M. HP
TM Calibration & 10-Hour Endurance	6:13	10:00
Vibration Survey	1:35	0:00
Sub-Total	7:48	10:00
Total	17:48	

Reduction Gear 507 had accumulated a total of 828.7 hours of development testing prior to the composite case test.

Parts Condition

The main drive gear is supported by the front housing through the main drive gear bearing and main drive gear bearing support. The main drive gear bearing and the main drive gear teeth, therefore, depend upon the stiffness of the front housing for proper alignment. Prior to build-up of the gear box photographs were taken of these components. Photographs were again taken after completion of the dynamometer test. The photos of the main drive gear inner raceway and rollers before and after testing is shown in Figures 17 and 18, respectively. The main drive gear bearing outer raceway before and after testing is shown in Figures 19 and 20, respectively. Comparison of these photographs show that the condition of the races and rollers was unchanged from the pre-test condition. This fact leads to the conclusion that the stiffness of the composite material front housing was adequate in reacting main drive gear and moment loads to meet the requirements of the 501-M22 gear box. Ten hours operation at 5500 HP and 150,000 in. lb. moment would have resulted in spalling or other distress of these raceways, had the case been as flexible as the original design magnesium front case.

Proper alignment of the main drive gear teeth is indicated by Figure 25 (before test) and Figure 26 (after test).

Other main power train components which were photographed before and after the test are as follows:

Propeller Bearing Outer Raceway and Rollers
Pinion Gear Teeth
Sun Gear Teeth
Planet Gear Teeth
Pinion Drive Shaft Spline

These photos are grouped in pairs (before and after condition) and are given in Figures 21 through 24 and Figures 27 through 32.

The integrity of the composite case was verified before and after testing by radiographic inspection.

Visual inspection indicated no yielding of studs or threaded bushings in the composite material.

The molded-in oil tubes performed satisfactorily. No oil leakage was encountered with the composite case.

Two defects were noted in the case after testing. Minor cracks appeared at the junction of the internal stiffening ribs to the case shell at the main splitline. A prism of filler material was ejected from the case at the front seal mounting surface as revealed by Figure 33. These defects do not seriously reflect upon the suitability of the composite material for future development work.

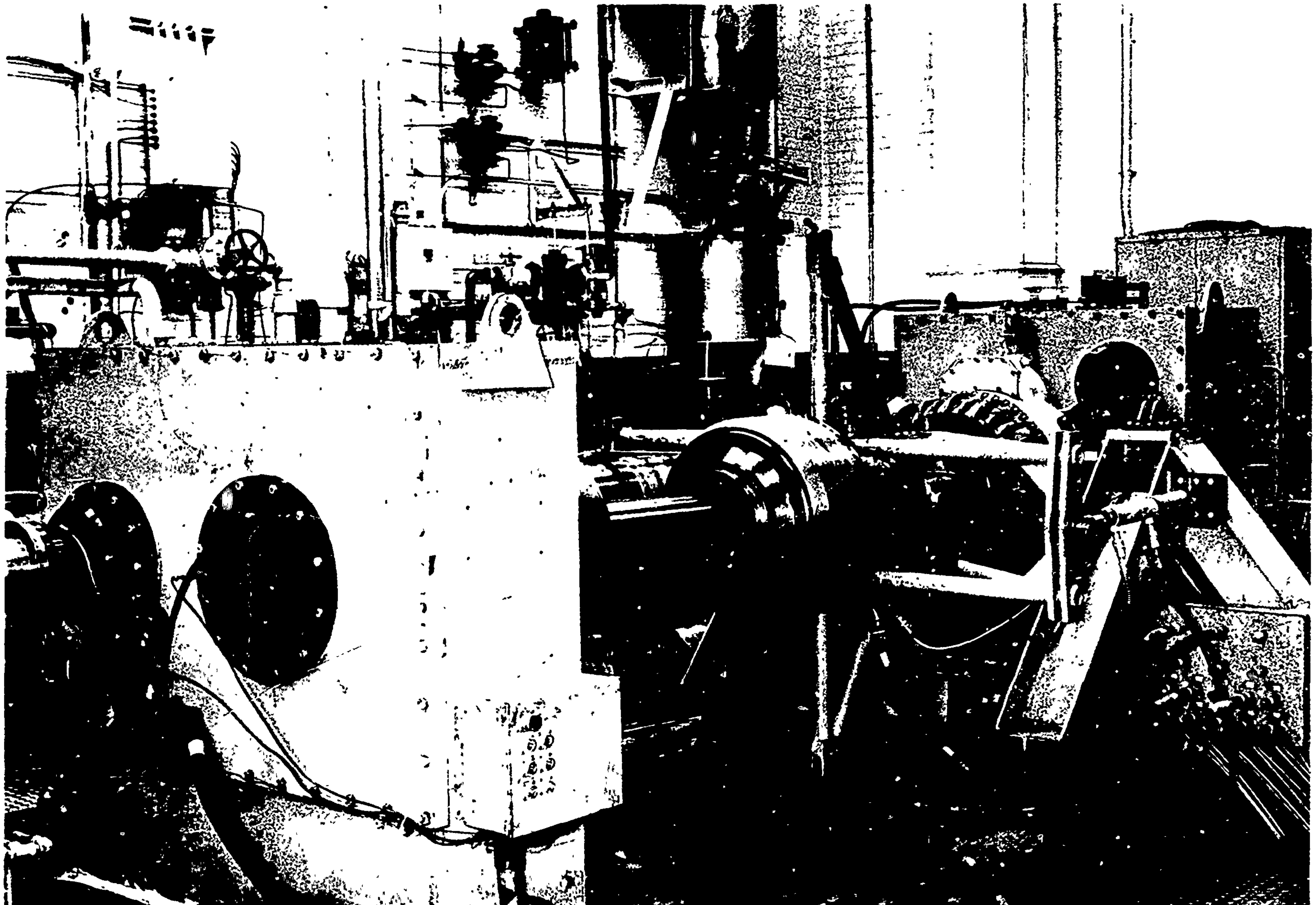


FIGURE 16 COMPOSITE CASE TEST UNIT INSTALLED ON BACK-TO-BACK DYNAMOMETER

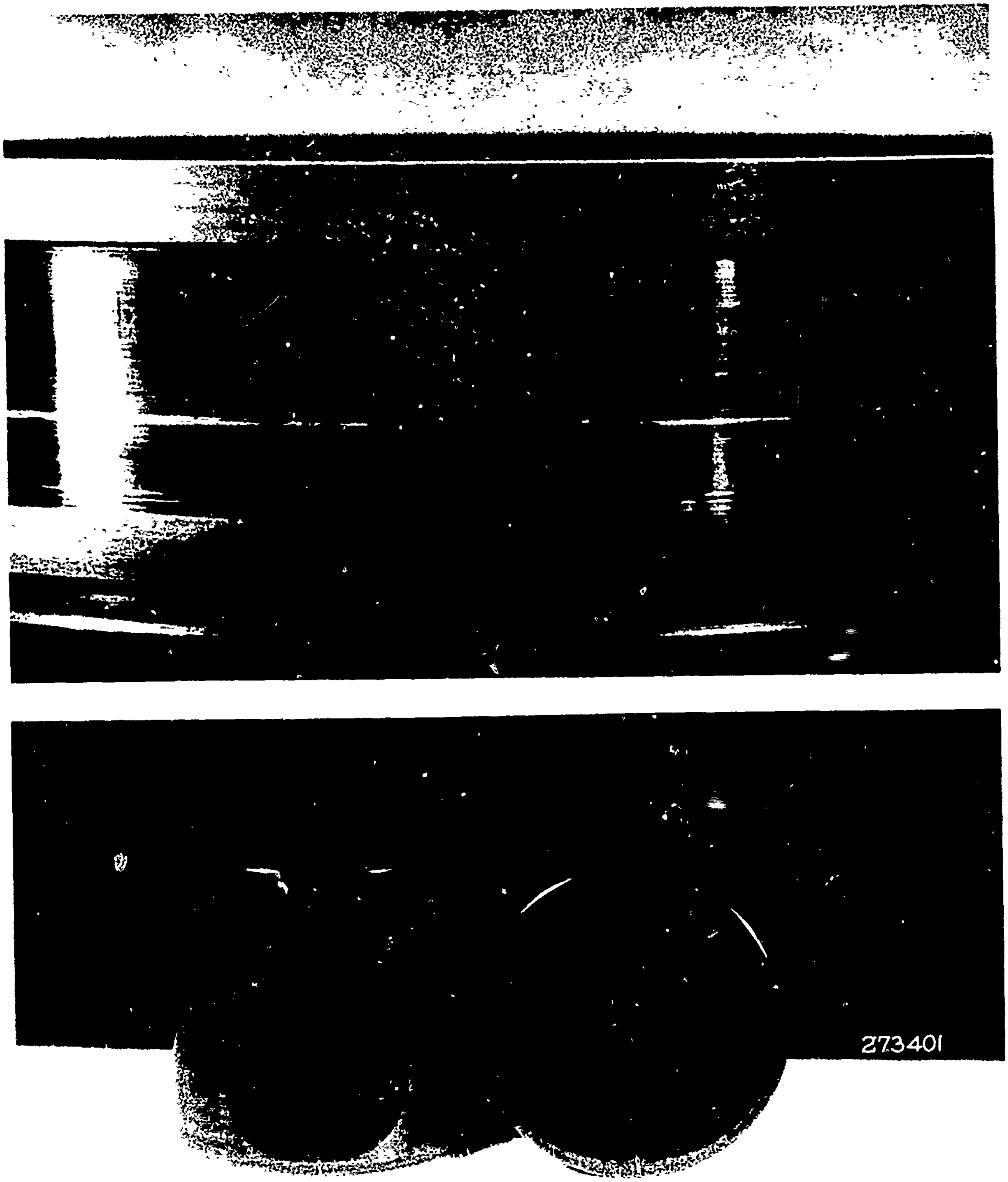


FIGURE 17 MAIN DRIVE GEAR BEARING SUPPORT - INNER RACEWAY AND ROLLERS
(BEFORE TEST)

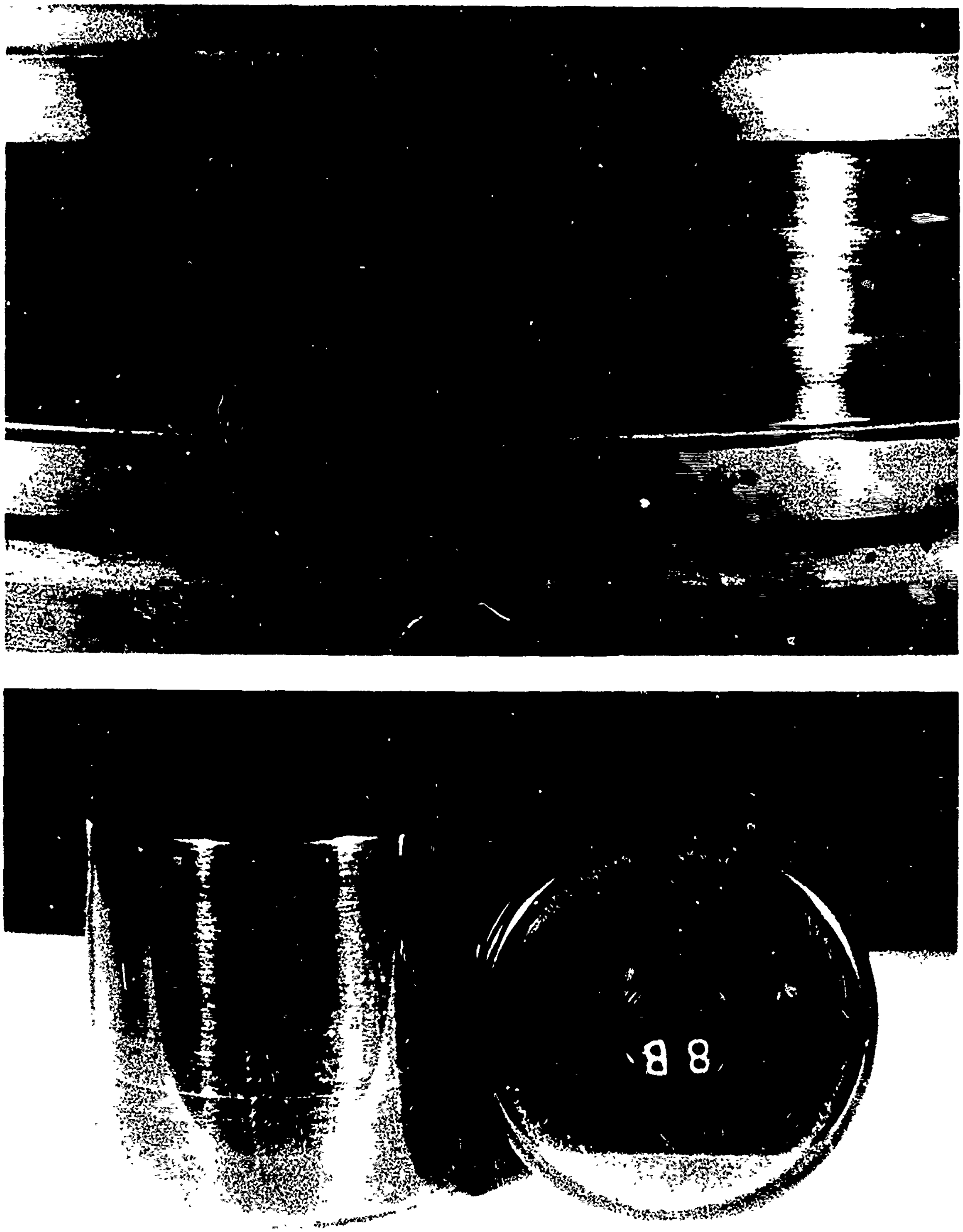


FIGURE 18 MAIN DRIVE GEAR BEARING SUPPORT - INNER RACEWAY AND ROLLERS
(AFTER TEST)

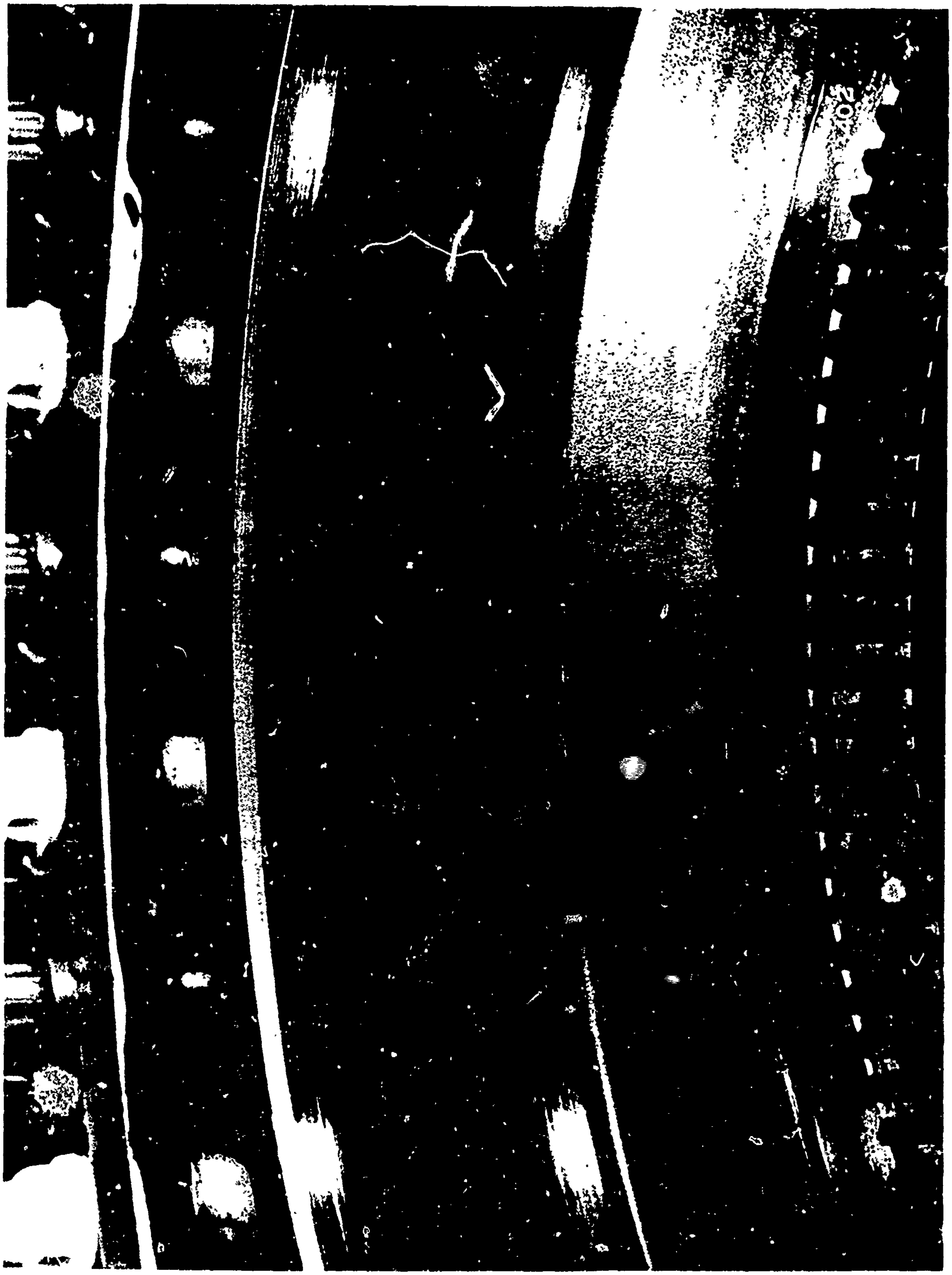


FIGURE 19 MAIN DRIVE GEAR BEARING-OUTER RACEWAY (BEFORE TEST)

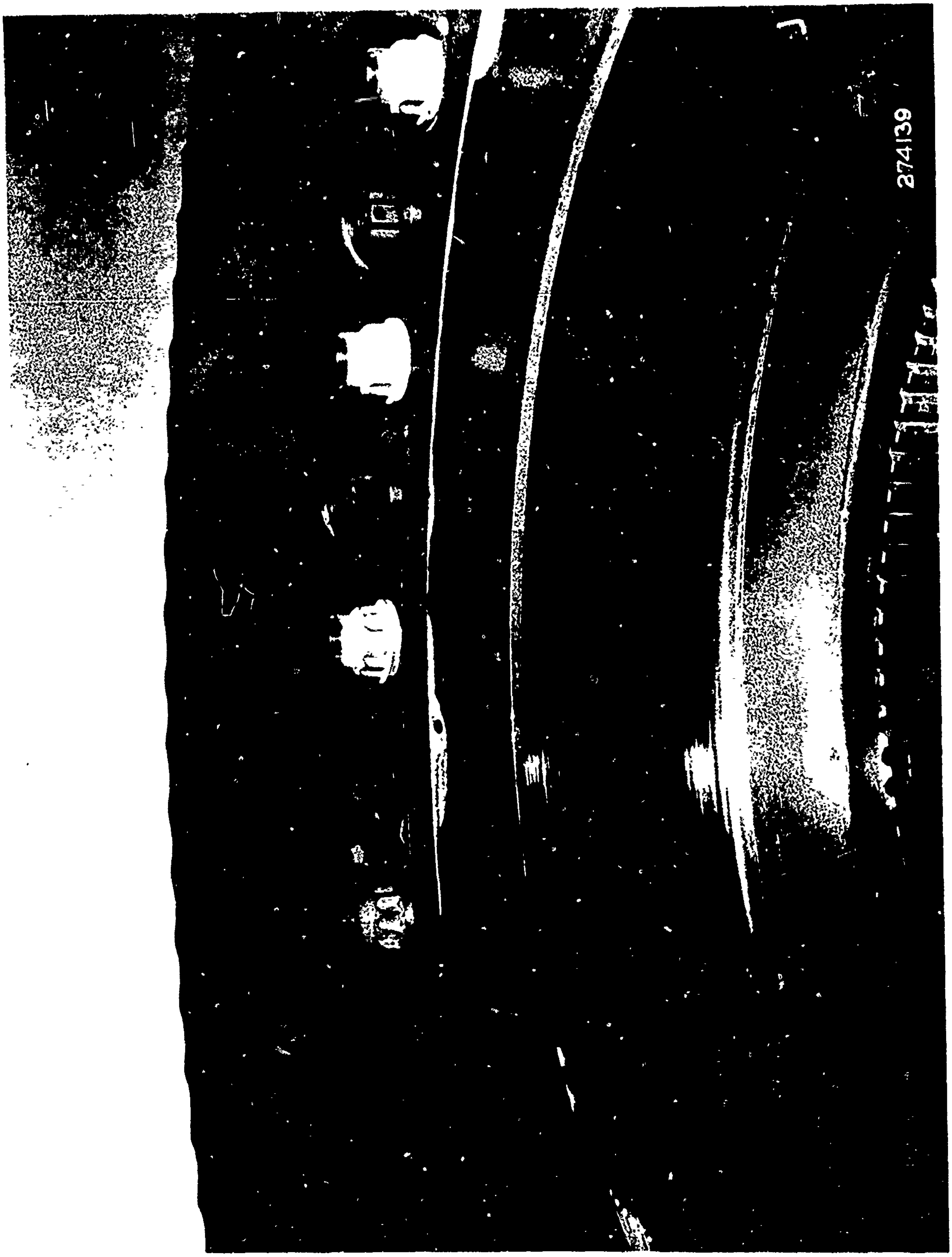


FIGURE 20 MAIN DRIVE GEAR BEARING-OUTER RACEWAY (AFTER TEST)

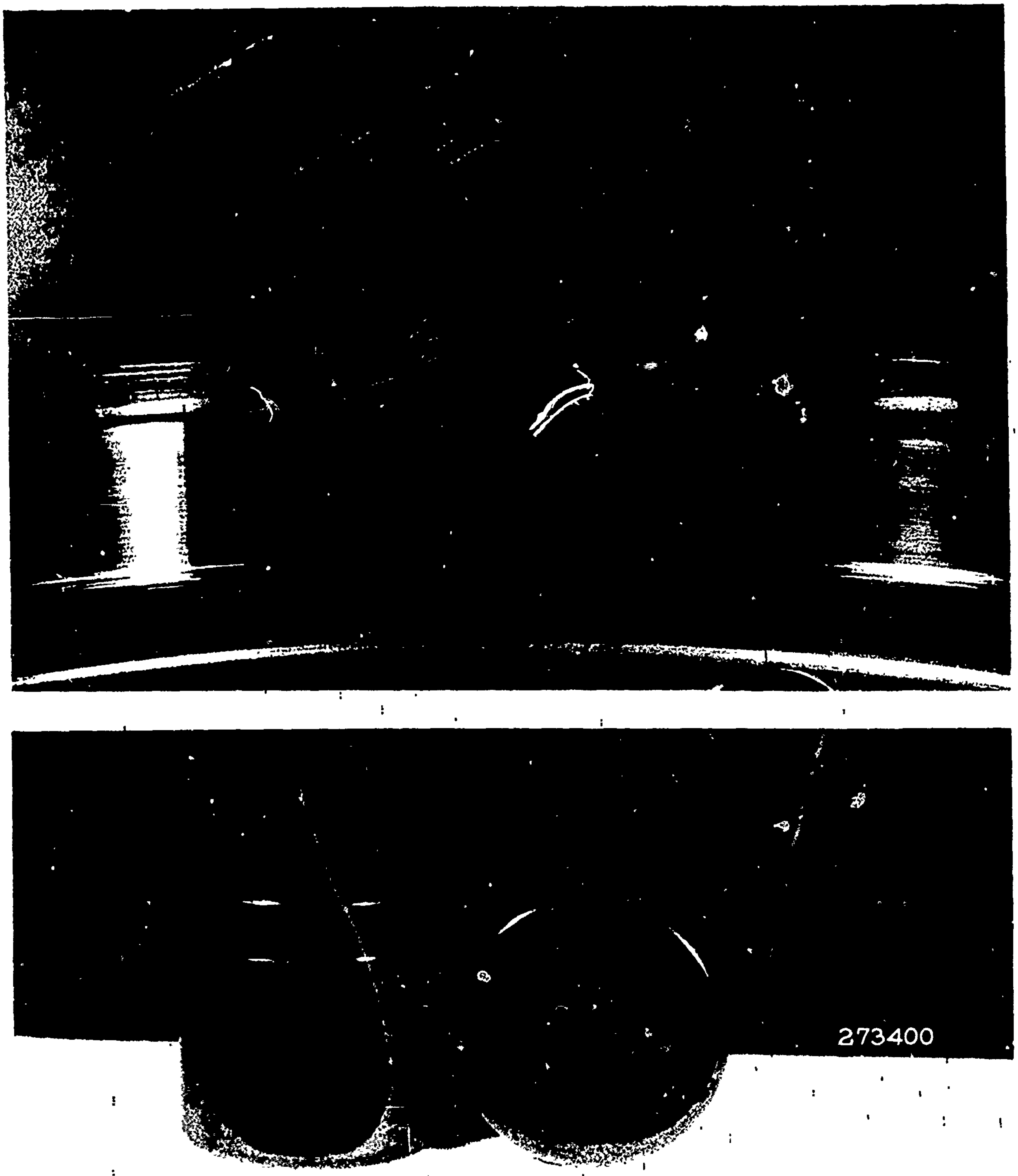


FIGURE 21 MAIN DRIVE GEAR SUPPORT-PROPELLER BEARING OUTER RACEWAY AND ROLLERS (BEFORE TEST)

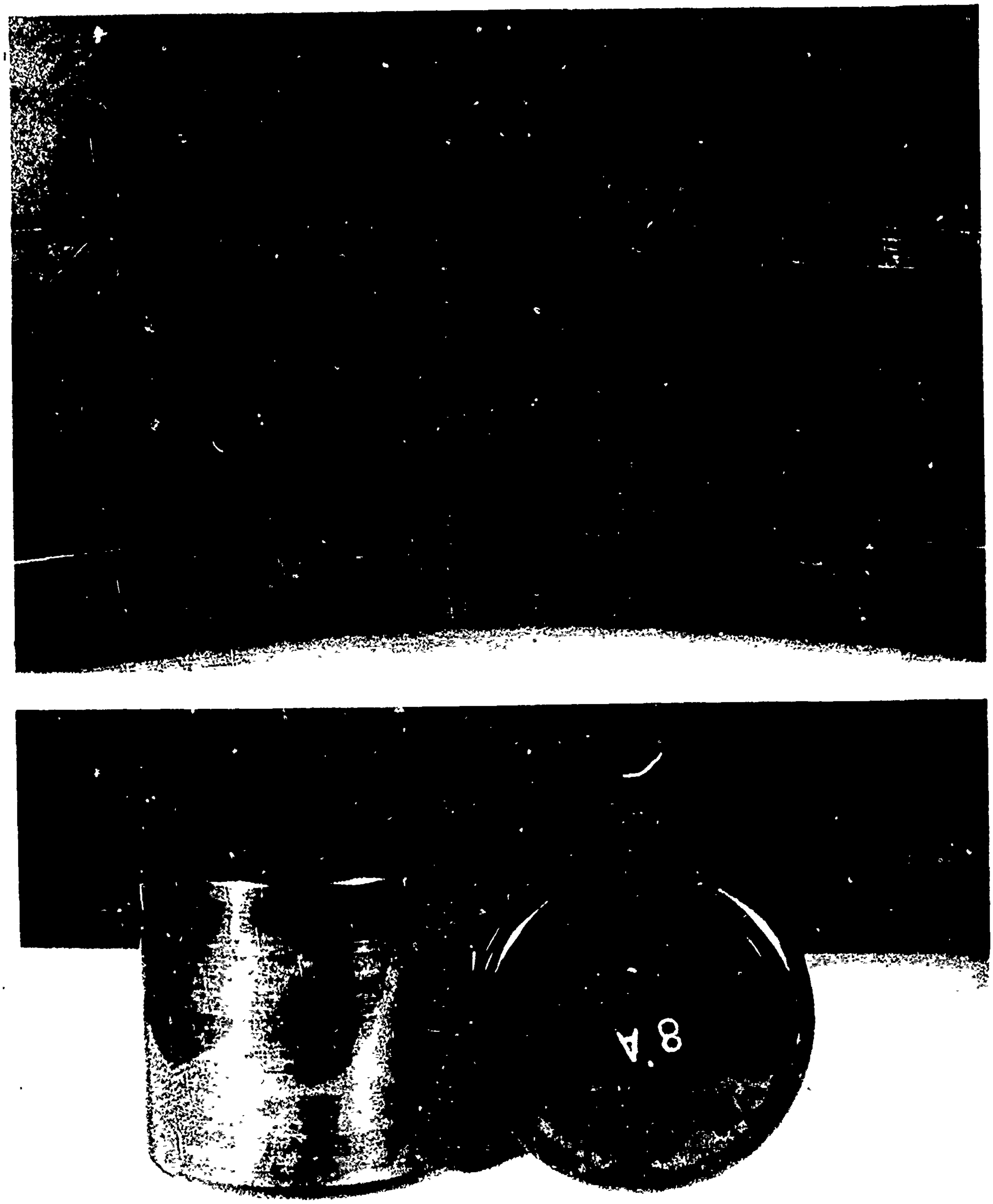


FIGURE 22 MAIN DRIVE GEAR SUPPORT-PROPELLER BEARING OUTER RACEWAY AND ROLLERS (AFTER TEST)

FIGURE ..? PINION GEAR TEETH-LOAD SIDE (BEFORE TEST)

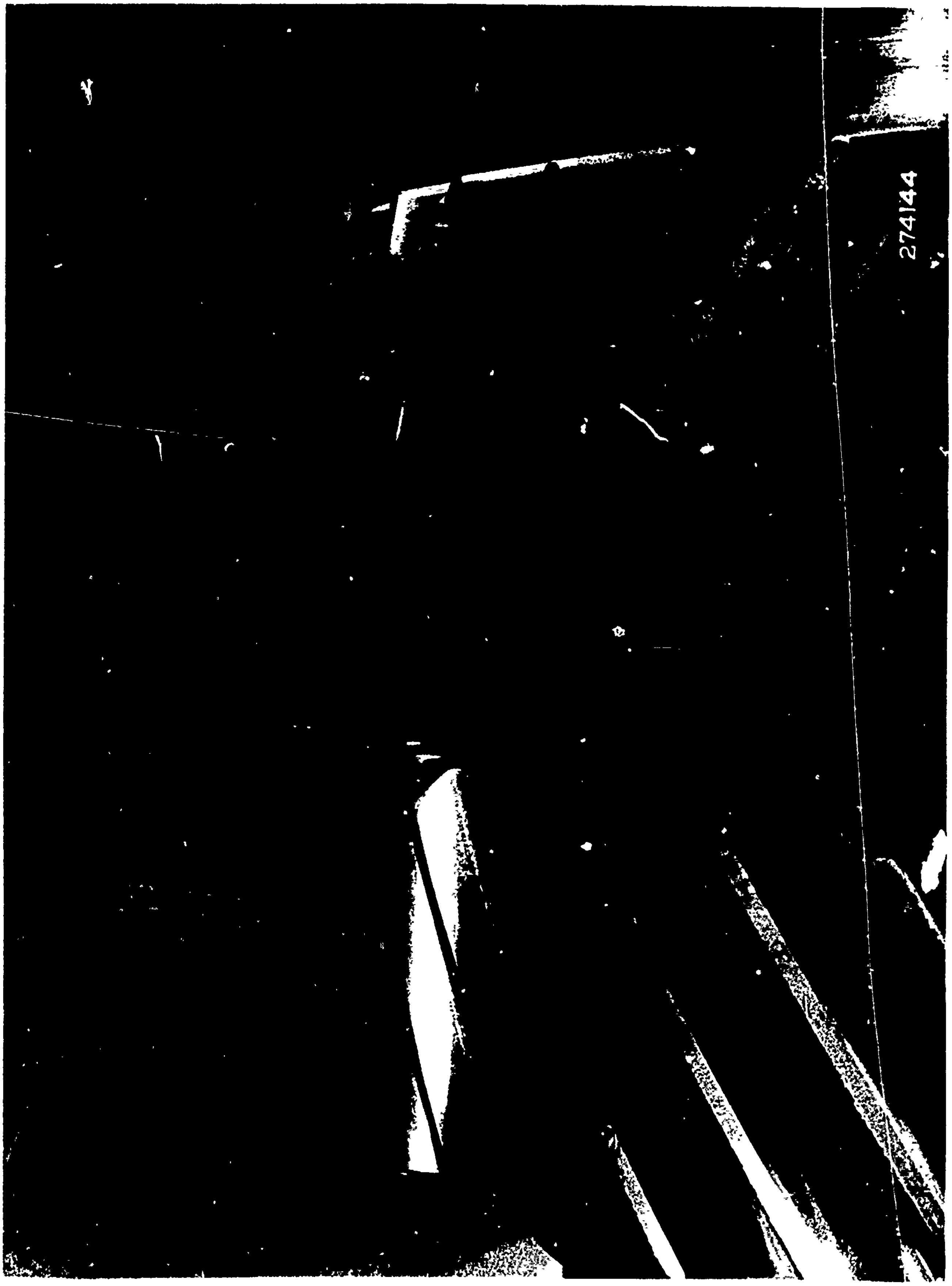


FIGURE 24 PINION GEAR TEETH-LOAD SIDE (AFTER TEST)

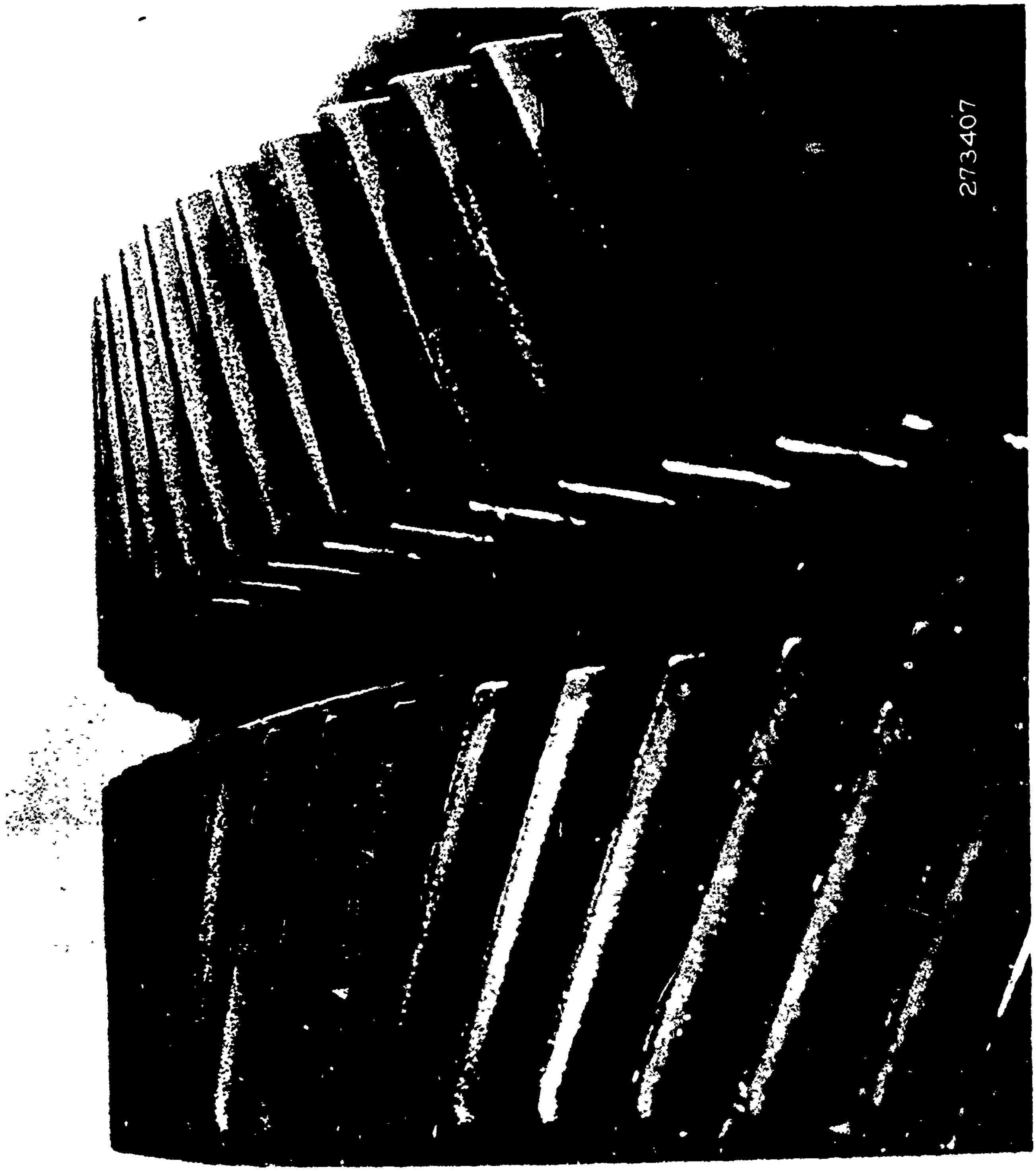


FIGURE 25 MAIN DRIVE GEAR TEETH-LOAD SIDE (BEFORE TEST)

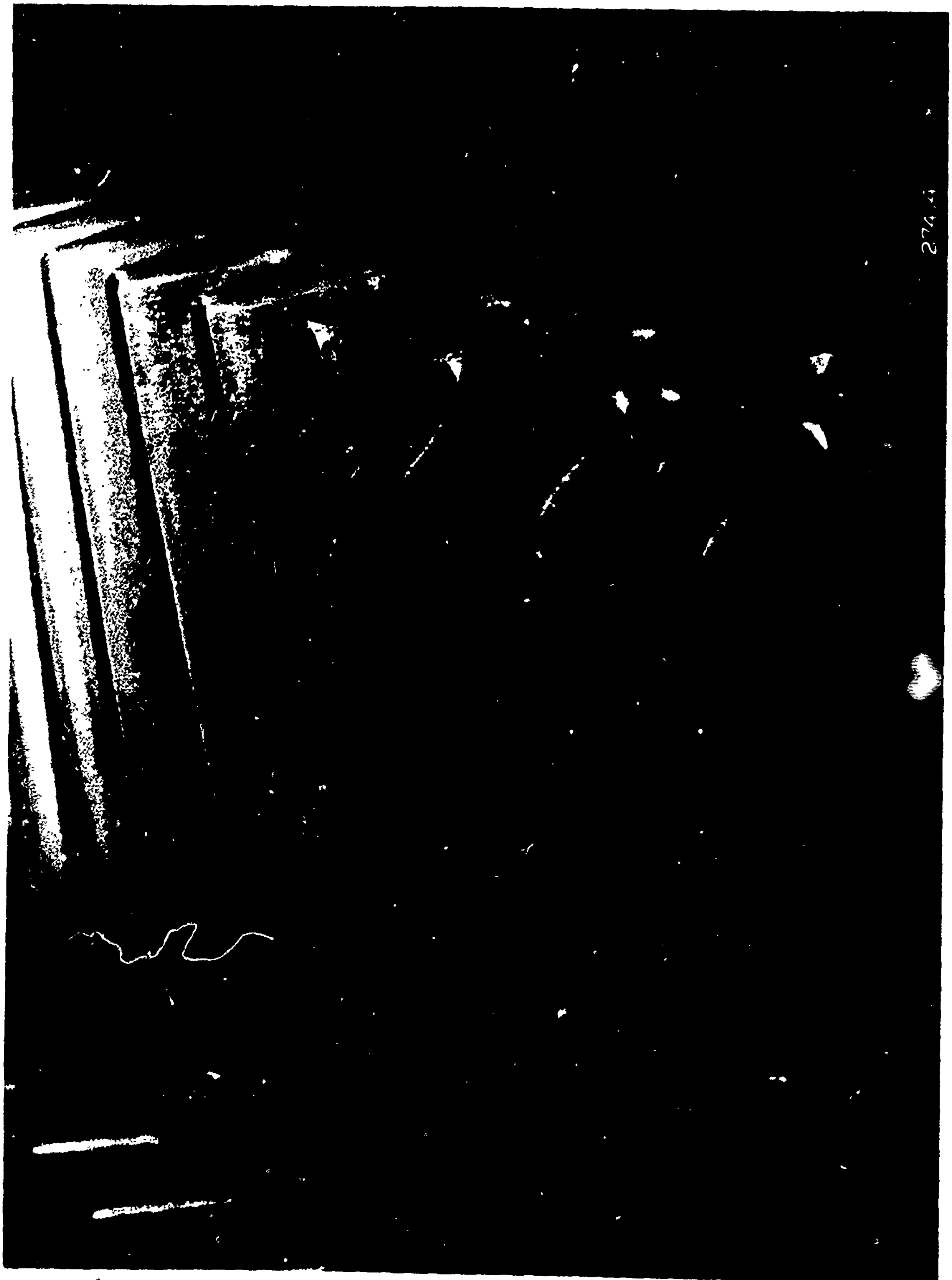


FIGURE 26 MAIN DRIVE GEAR TEETH-LOAD SIDE (AFTER TEST)

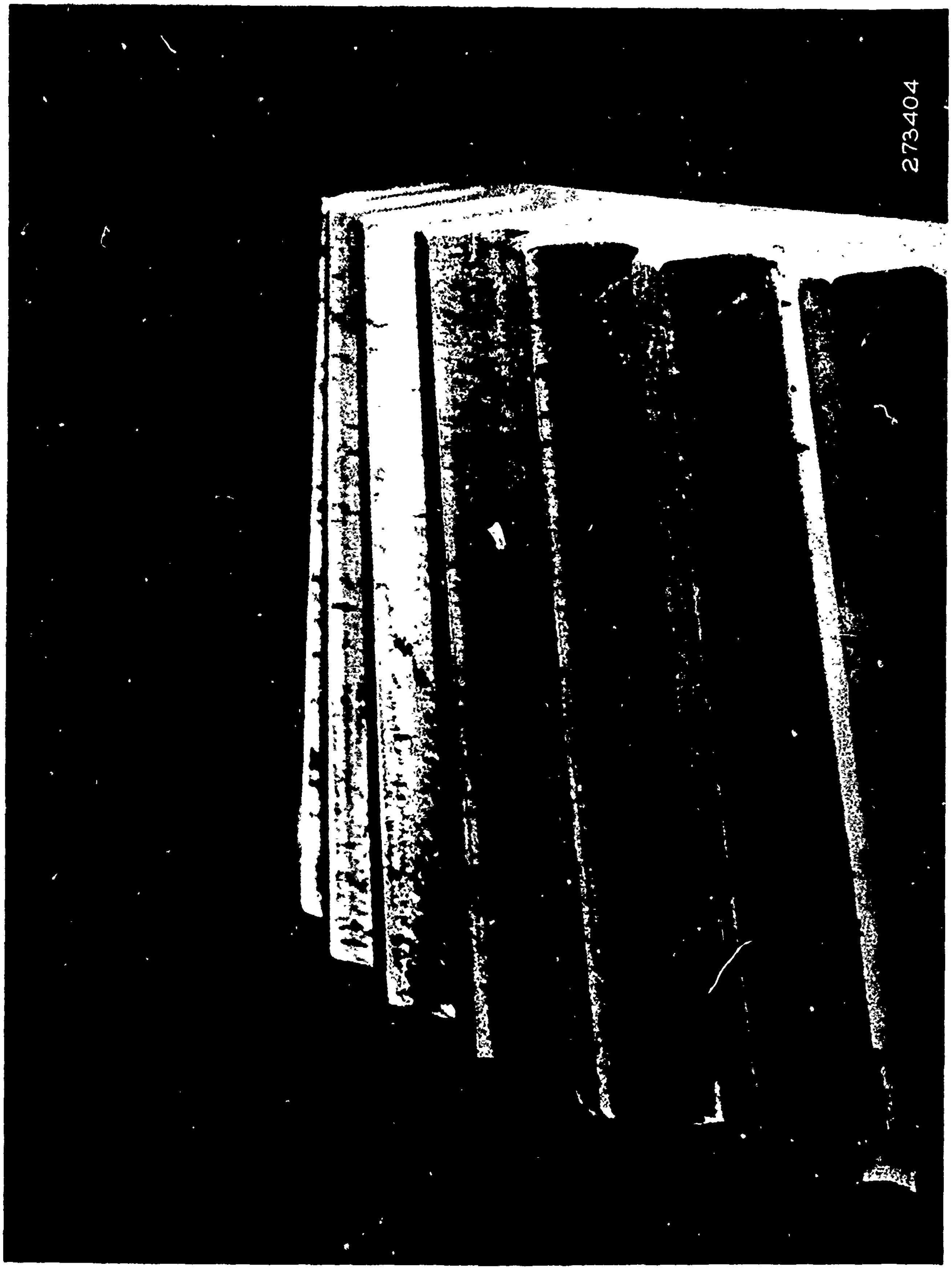


FIGURE 27 SUN GEAR TEETH-LOAD SIDE (BEFORE TEST)

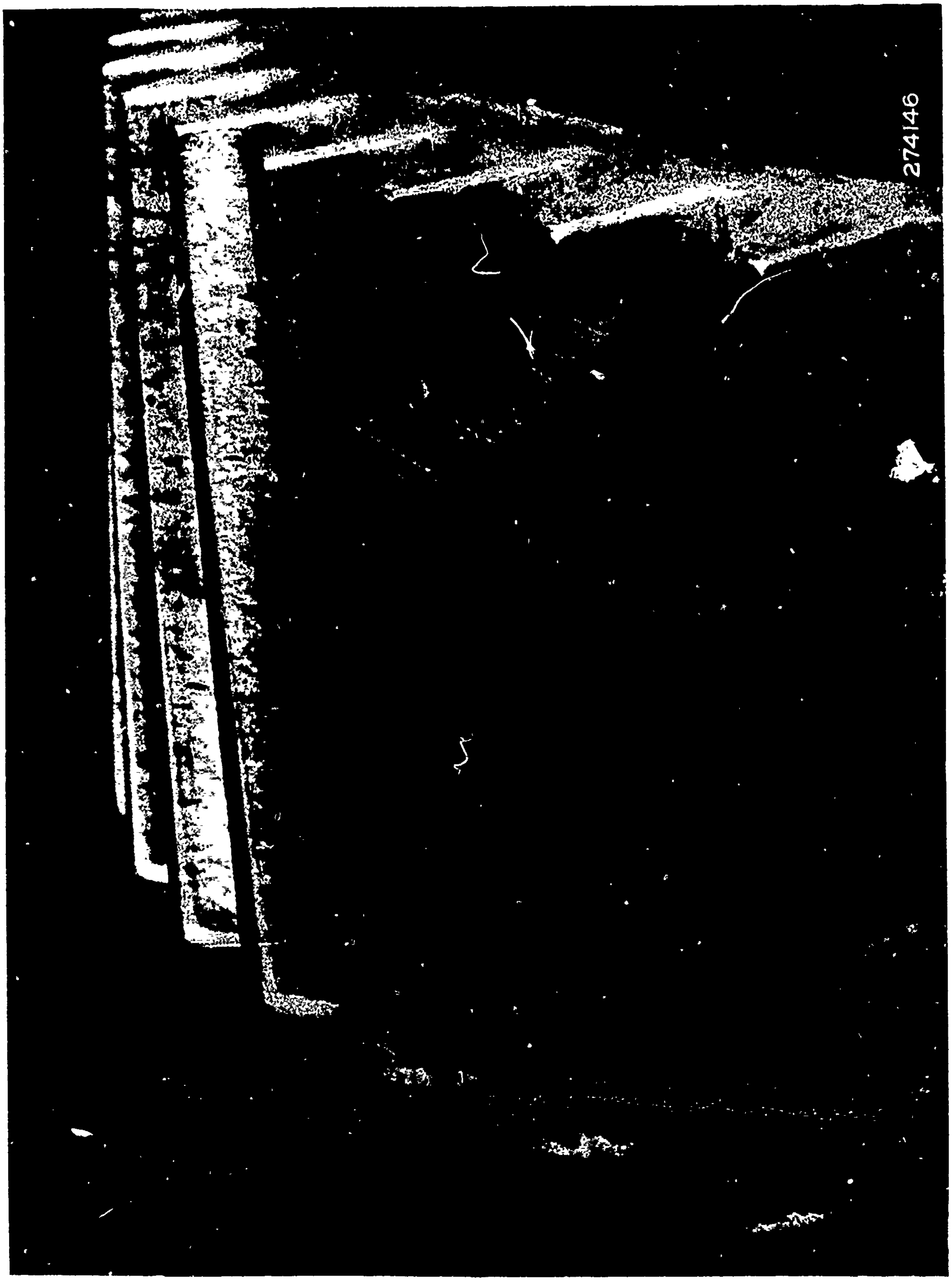


FIGURE 28 SUN GEAR TEETH-LOAD SIDE (AFTER TEST)

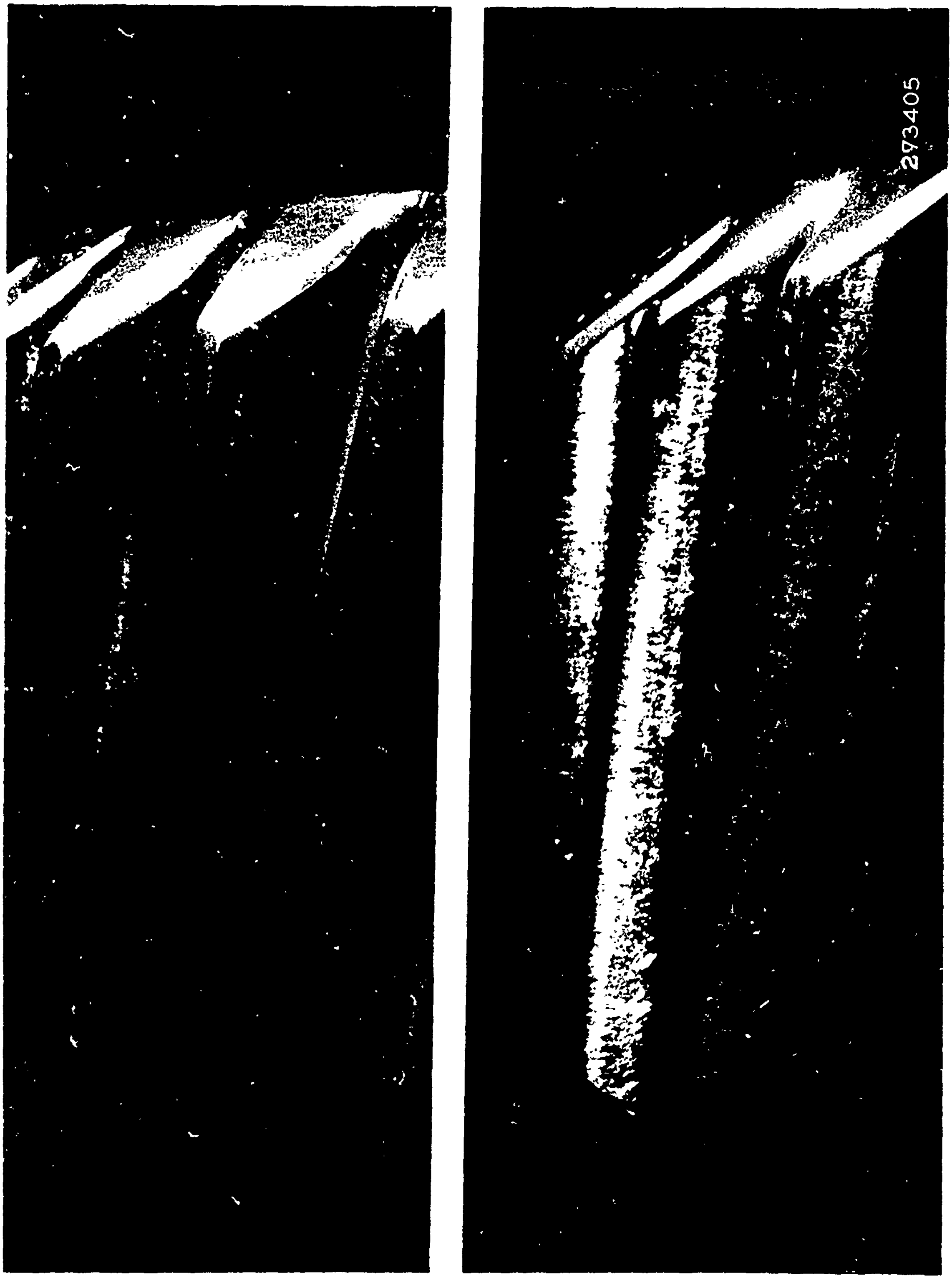


FIGURE 29 PLANET GEAR TEETH (BEFORE TEST)

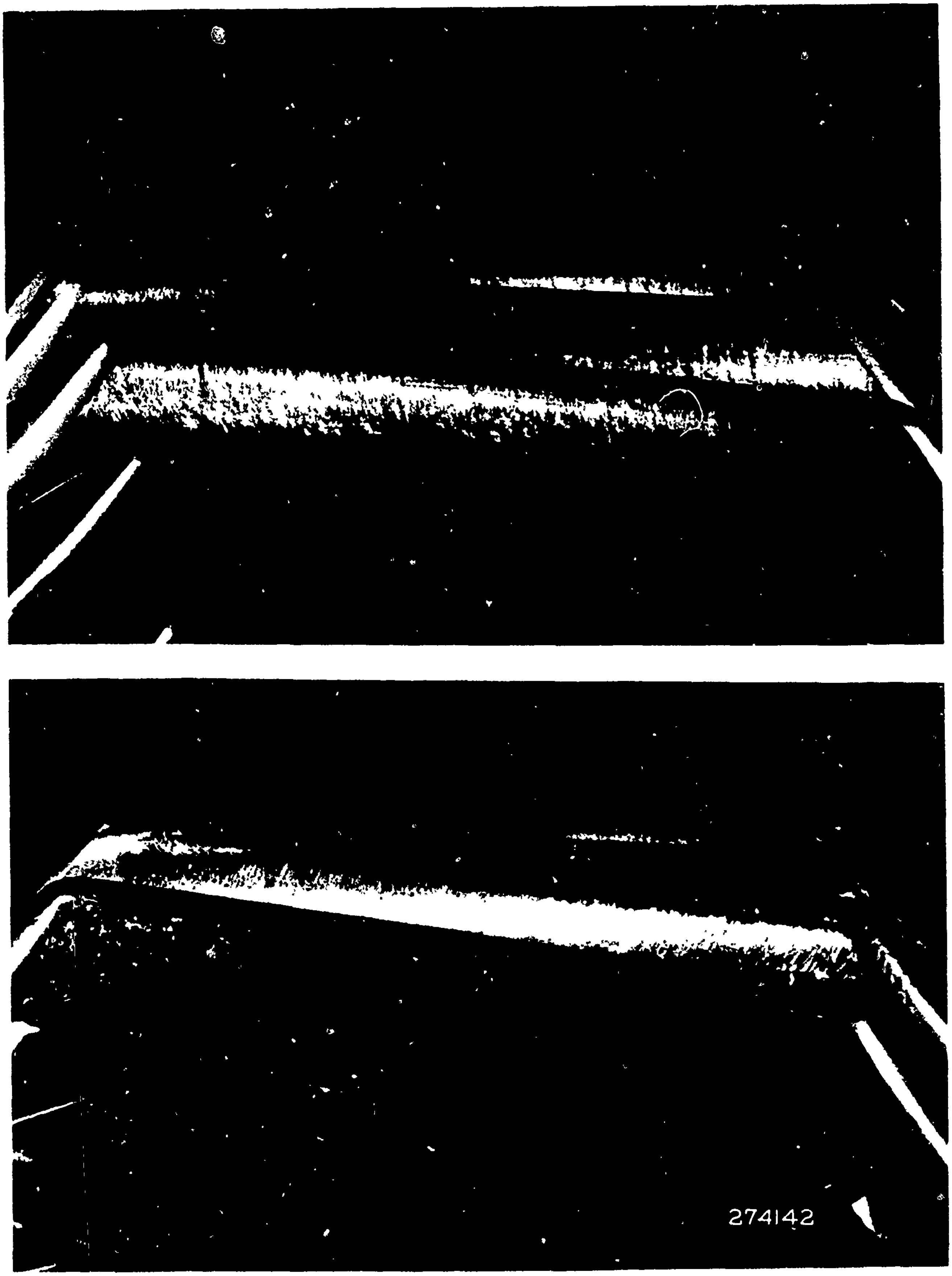


FIGURE 30 PLANET GEAR TEETH (AFTER TEST)

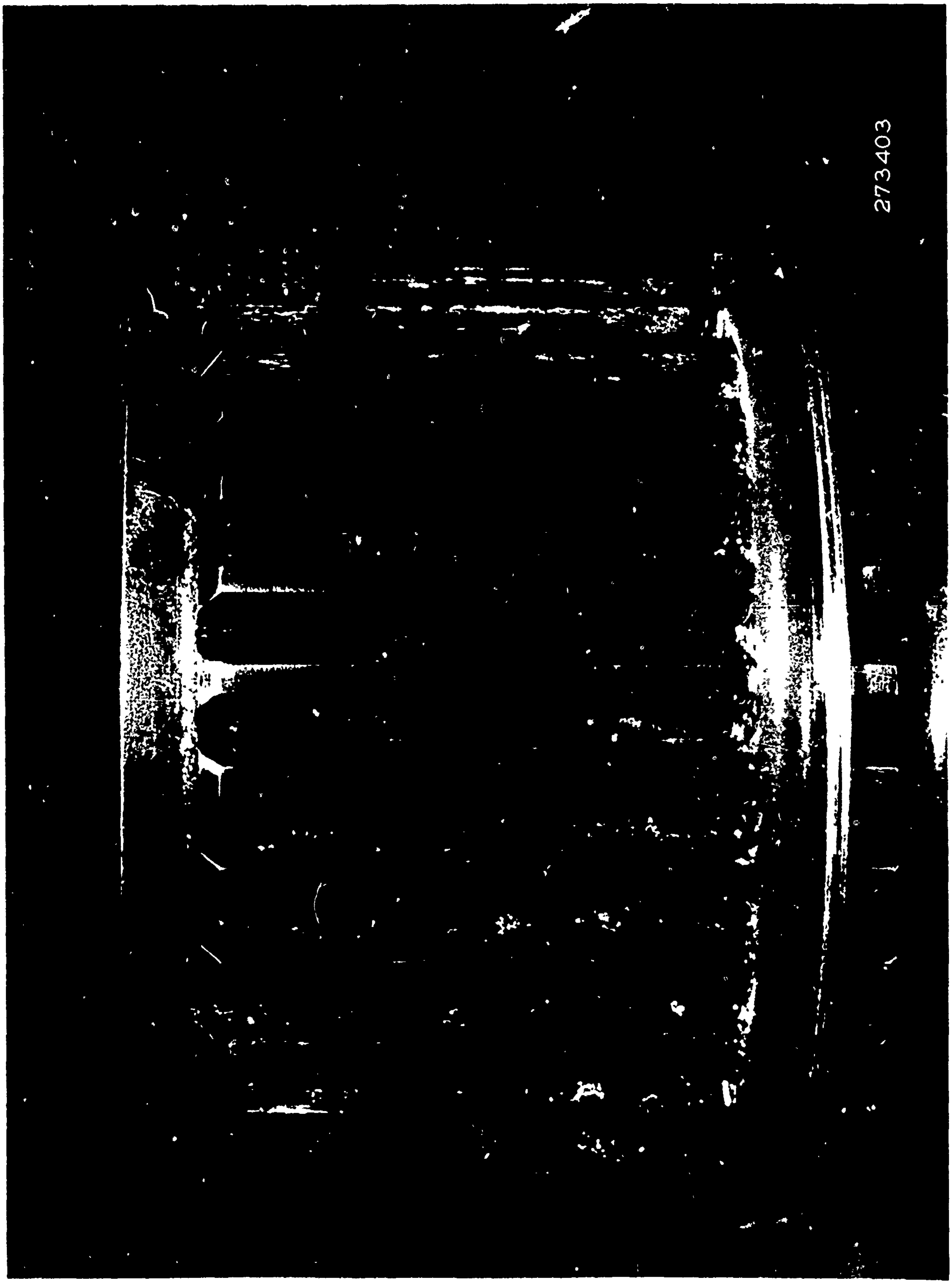


FIGURE 31 PINION DRIVE SHAFT SPLINE (BEFORE TEST)

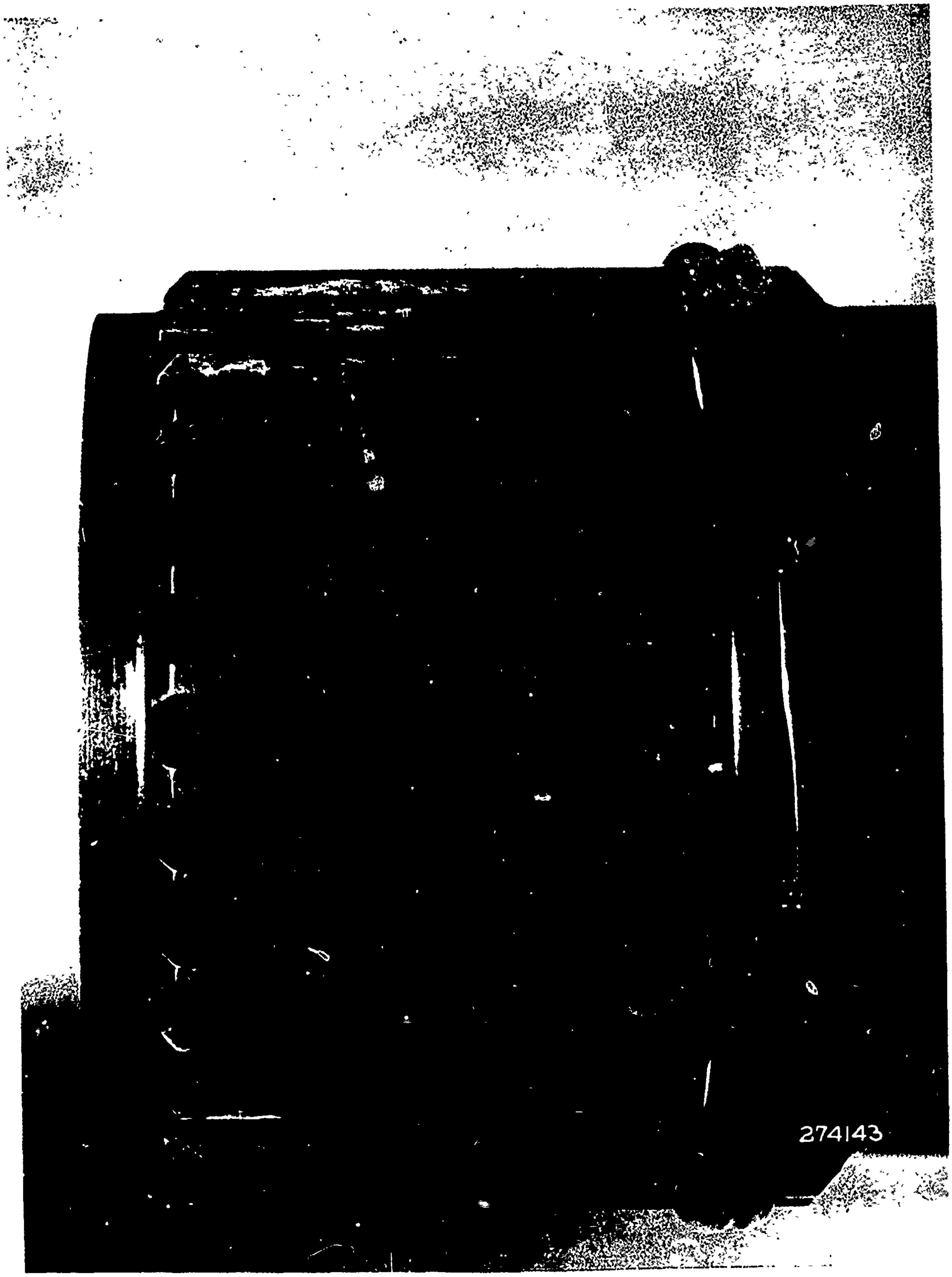


FIGURE 32 PINION DRIVE SHAFT SPLINE (AFTER TEST)

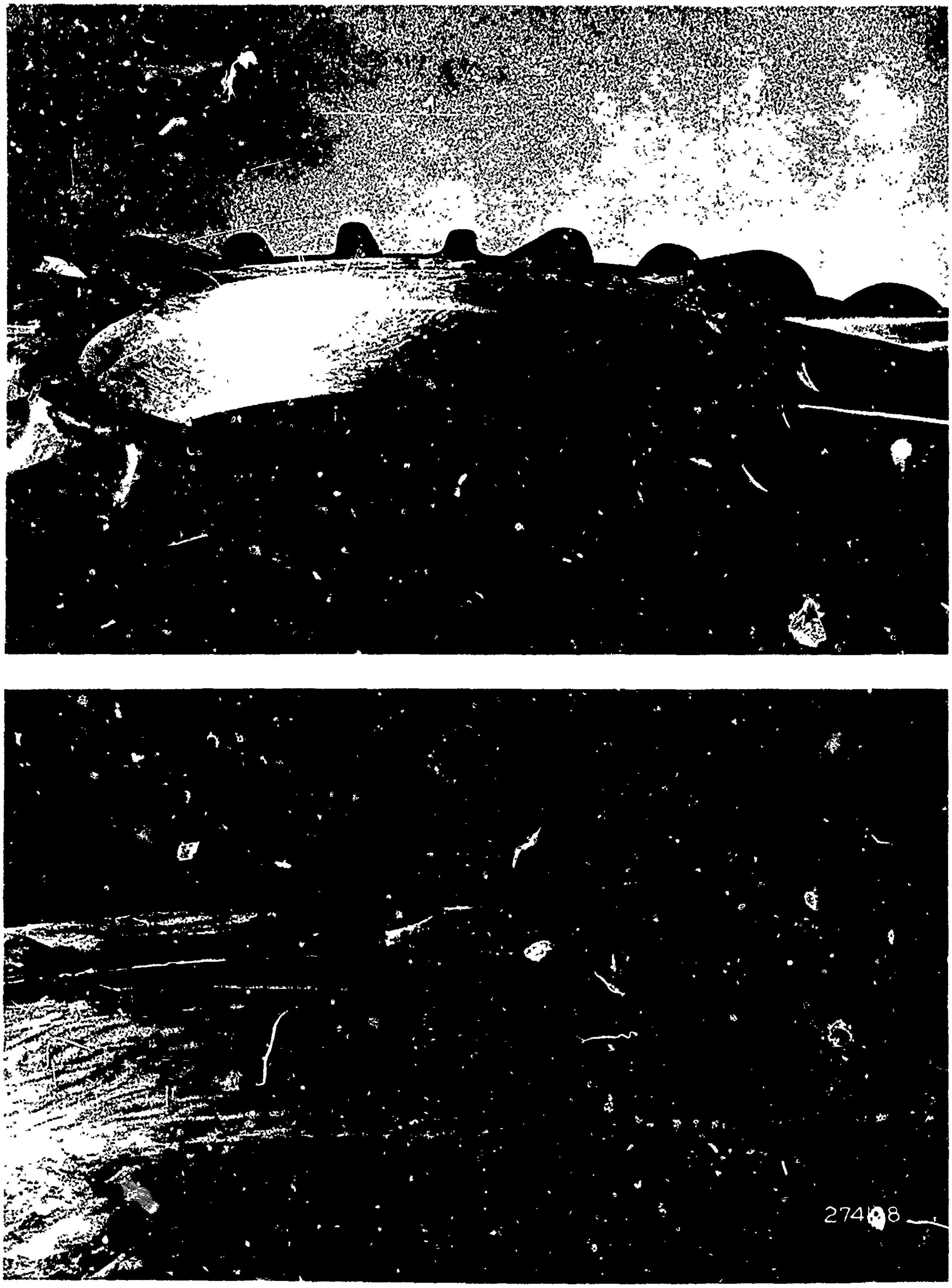


FIGURE 33 COMPOSITE CASE FOLLOWING DYNAMOMETER TEST

VI. VIBRATION SURVEY

A vibration survey was conducted on the composite case to determine if any case diaphragming (fore and aft) resonances existed in the engine operating range. This test was performed on the dynamometer immediately following the 10-hour dynamometer test.

Overall vibration levels were recorded during a slow dynamometer acceleration from 6,000 RPM to 15,200 RPM. A frequency analysis was taken where there was an indication of a possible resonance. The composite case did not exhibit diaphragming resonances in the engine RPM range of 6,000 to 15,200 RPM.

Test Set-Up

Vibration levels were measured using a CEC 4-128 vibration transducer. The integrated signal from this transducer was recorded on a B & K 1/3 octave analyzer and also converted to a DC level and recorded on an X-Y recorder. The RPM signal was taken from the dynamometer tachometer and converted from a frequency to a DC level for recording on the X axis of the X-Y recorder.

The vibration pickup was bonded to the case using Eastman's 910 cement and moved to the various locations as shown by Figures 34 and 35.

Test Procedure

After the vibration pickup was bonded to the case, the dynamometer was brought up to rated speed and 200 HP applied to the prop shaft. The dynamometer RPM was reduced to 6,000 RPM and then slowly accelerated to 15,200 RPM. Vibration and RPM were recorded on the X-Y recorder during the acceleration. Points of high vibration, indicating a possible resonant condition, were investigated using the 1/3 octave analyzer.

Test Results

Curves 1, 3, 7 and 12 present the overall vibration levels at various gear box locations during the RPM scans. All other curves are 1/3 octave analysis records of high vibration points.

Curve 1 presents the vibration at probe point 1, located on the rear case. Little difference is seen when the baseline Curve 1 is compared to the front case vibration shown in Curves 3, 7 and 12. A difference in vibration levels between the rear (baseline) case and the nose case would indicate a nose case resonance. A stand resonance would appear as a high vibration in both the nose and rear cases at any given RPM.

Analysis of the 1/3 octave data indicates that the primary vibration occurs at prop rotational, twice prop rotational, and engine rotational frequencies. The high vibration at 8,000 and 13,000 RPM appears to be a stand resonance occurring in the 31.5 Hz 1/3 octave band. The vibration levels at engine rotational frequency showed little variation at the RPM points checked.

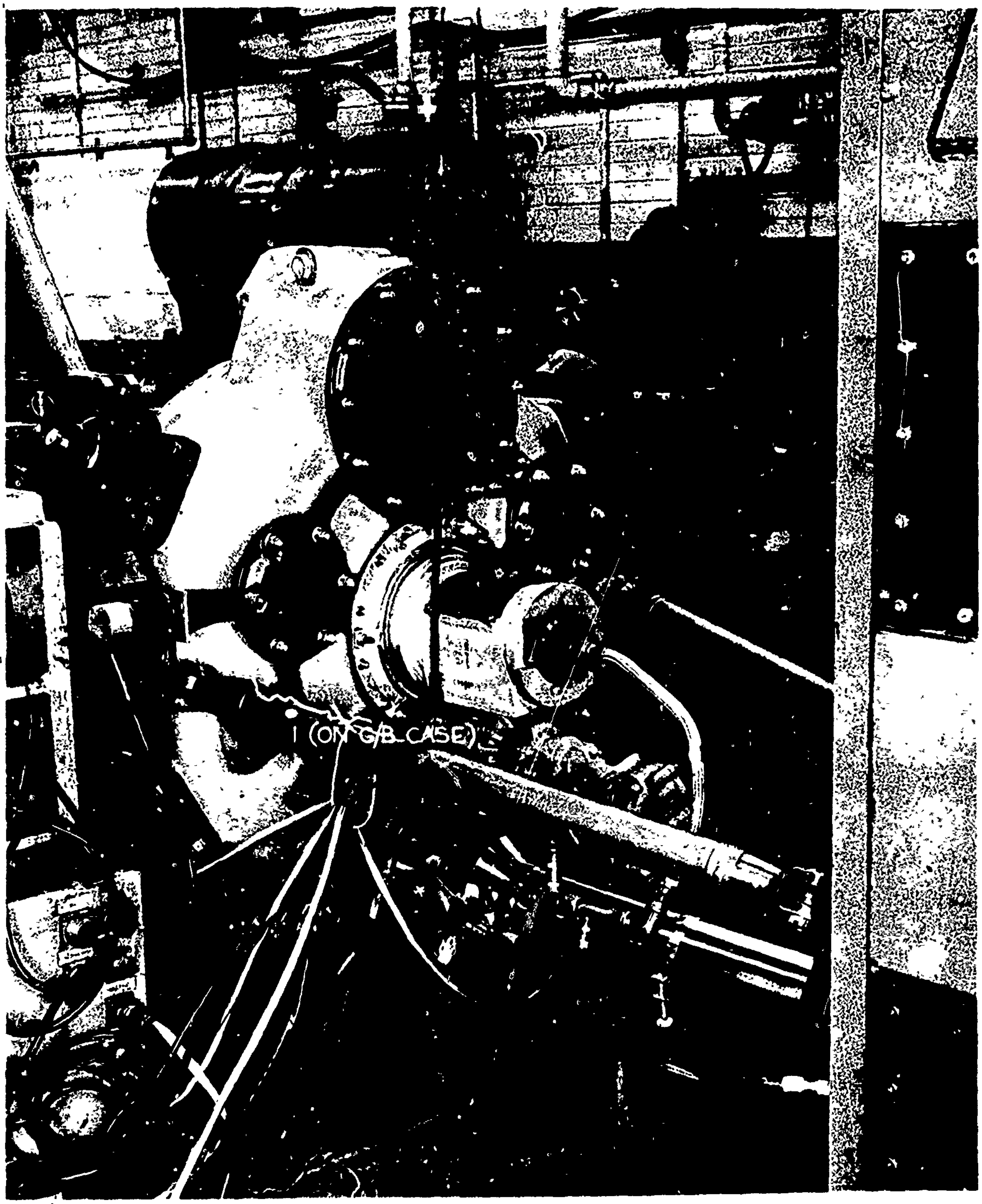
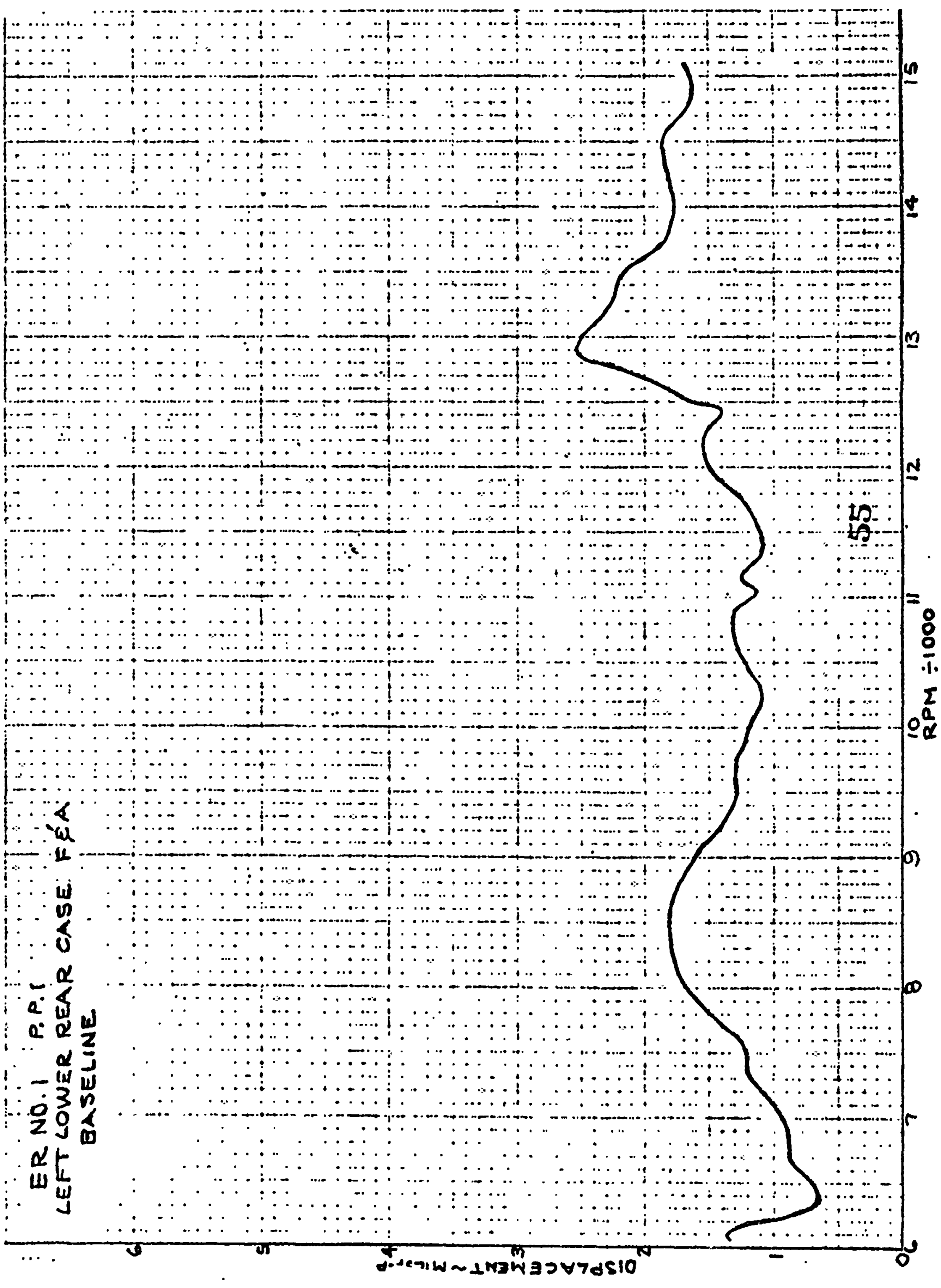


FIGURE 34 TRANSDUCER LOCATIONS-VIBRATION SURVEY COMPOSITE CASE

FIGURE 35 TRANSDUCER LOCATIONS-VIBRATION SURVEY COMPOSITE CASE

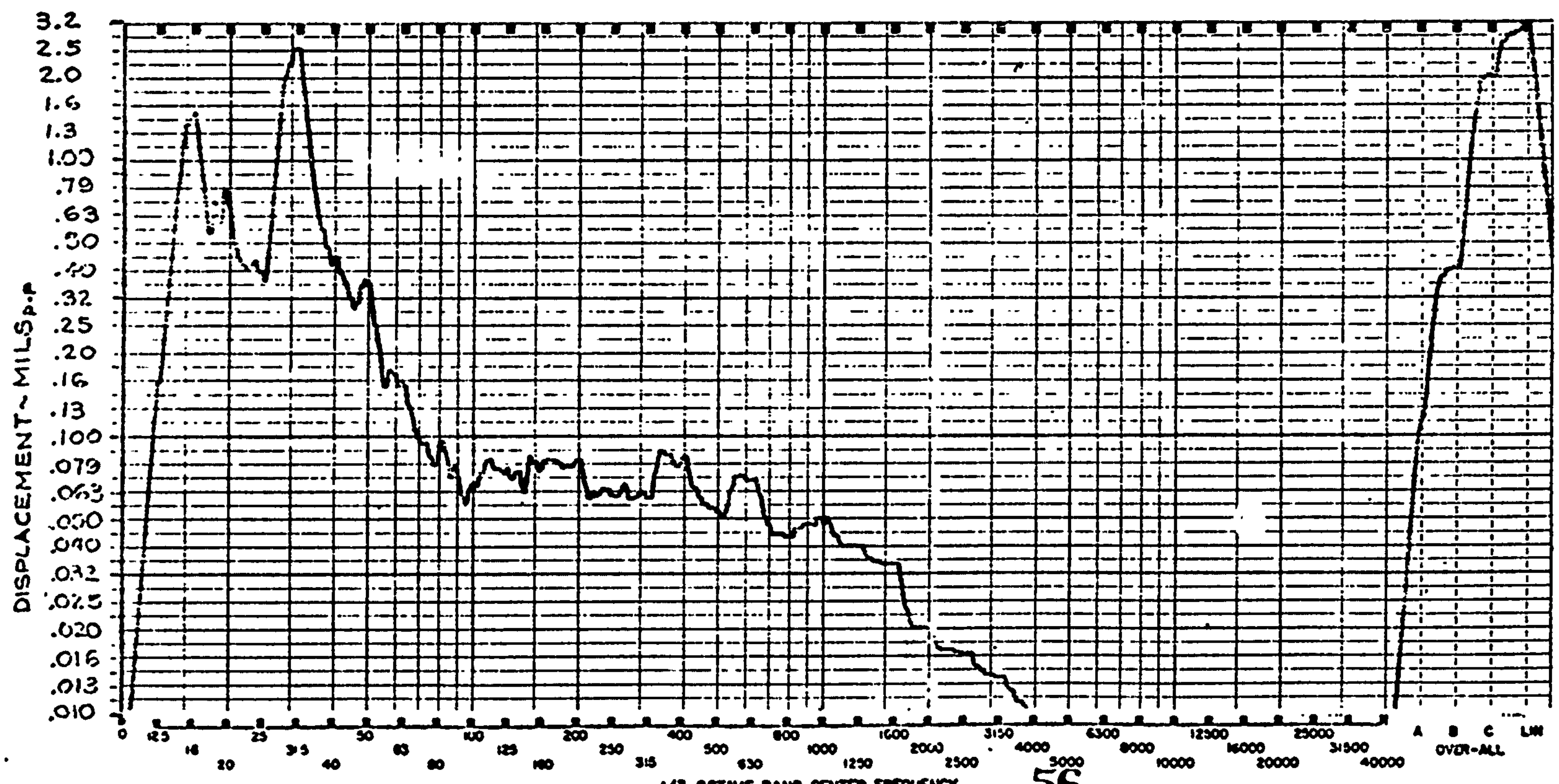
CURVE NO. 1



ER NO. 1 P.P. 1
LEFT LOWER REAR CASE F&A
BASELINE
DISPLACEMENT ~ MILS-P
RPM ÷1000
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15

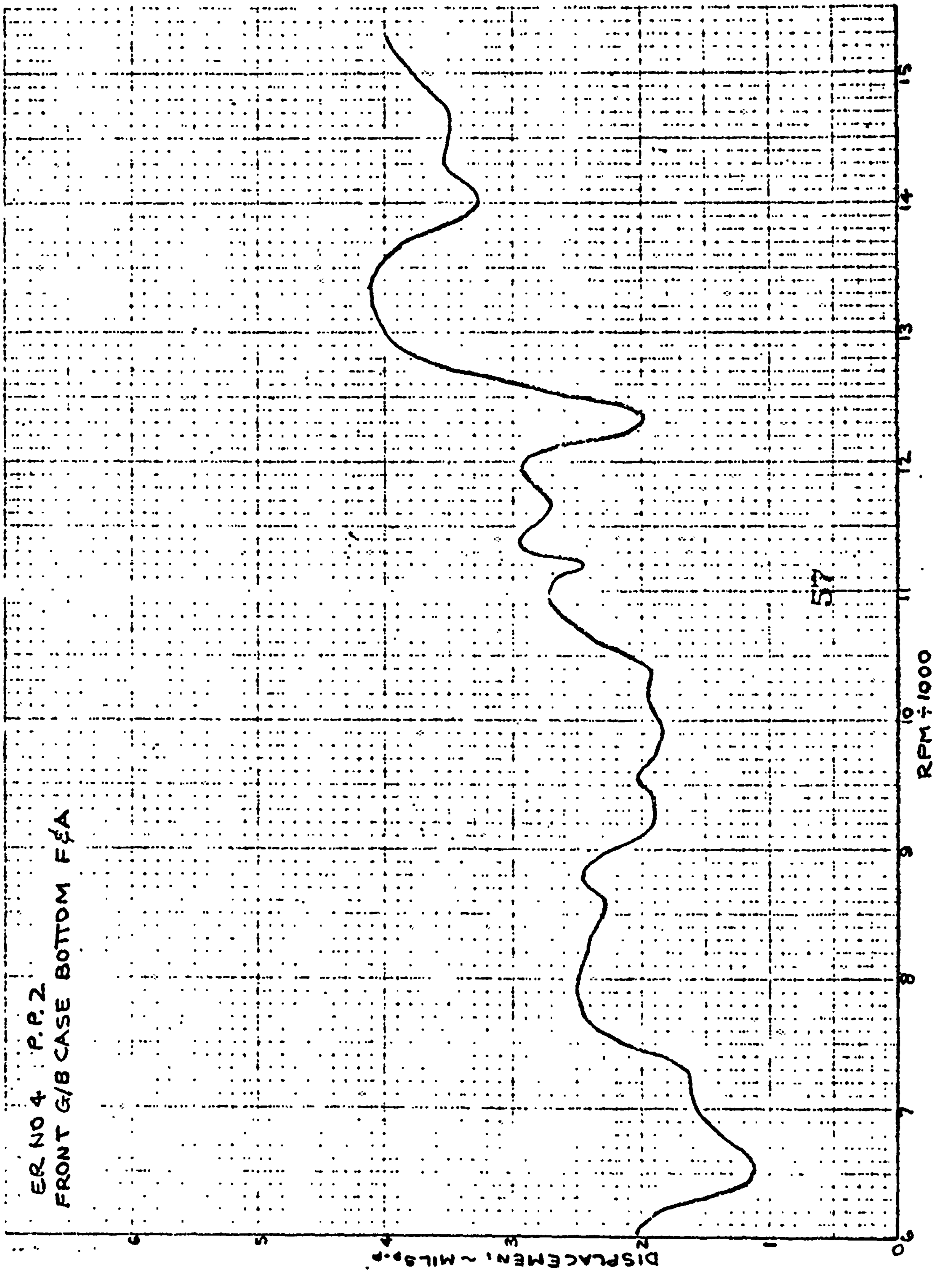
ER NO 3 P.P. 1

BASELINE
G/B REAR CASE LOWER LEFT F & 1
12,900 RPM = 215 HZ
PROP ROTATIONAL = 16 HZ



CURVE NO. 2

CURVE NO. 3



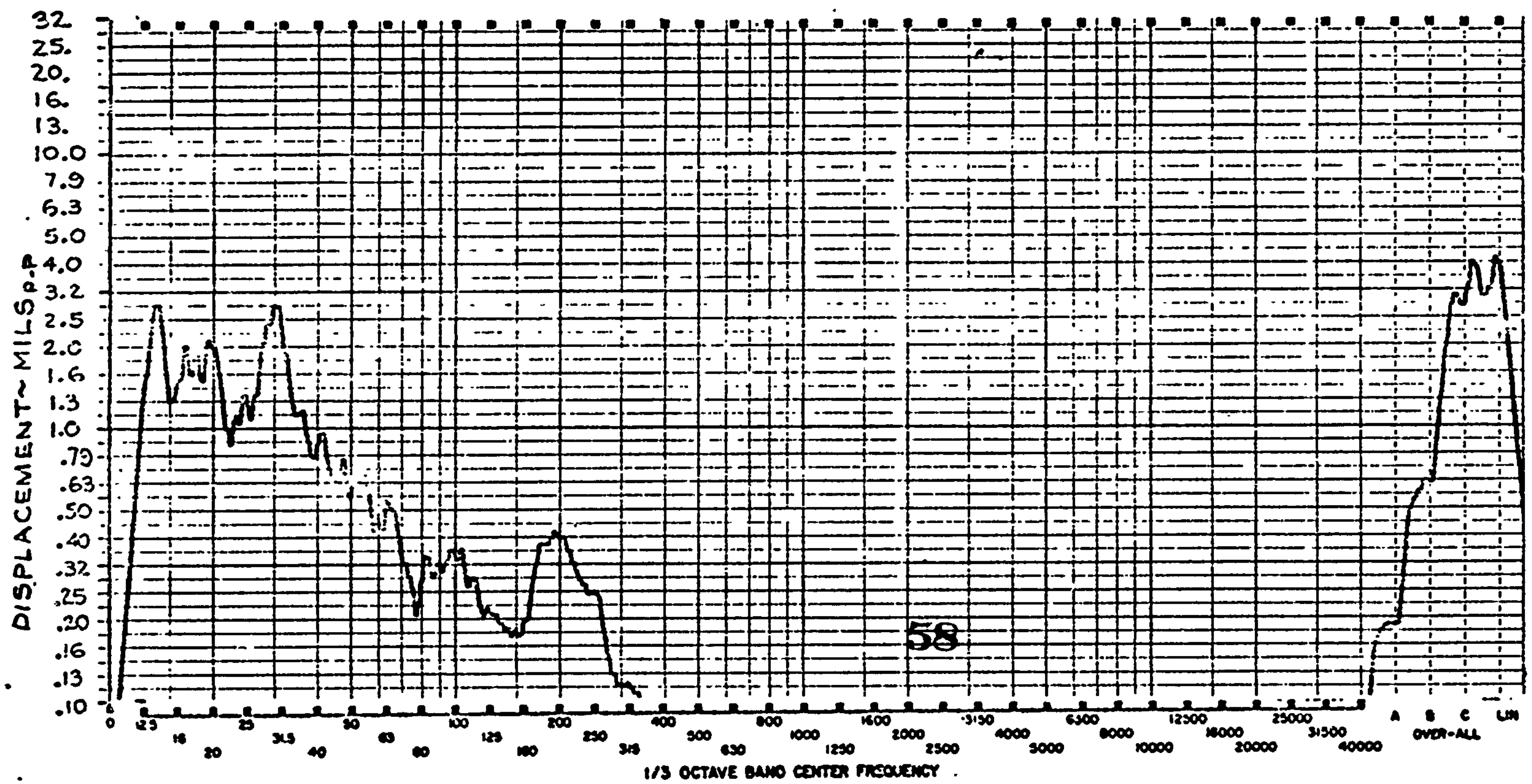
ER NO 4 P.P. 2
FRONT G/B CASE BOTTOM F&A
RPM ÷ 1000
DISPLACEMENT ~ MILS p.p
6
7
8
9
10
11
12
13
14
15
0
1
2
3
4
5
6

F NO 2

G/E FRONT CASE BOTTOM F&A

12,900 RPM = 215 Hz

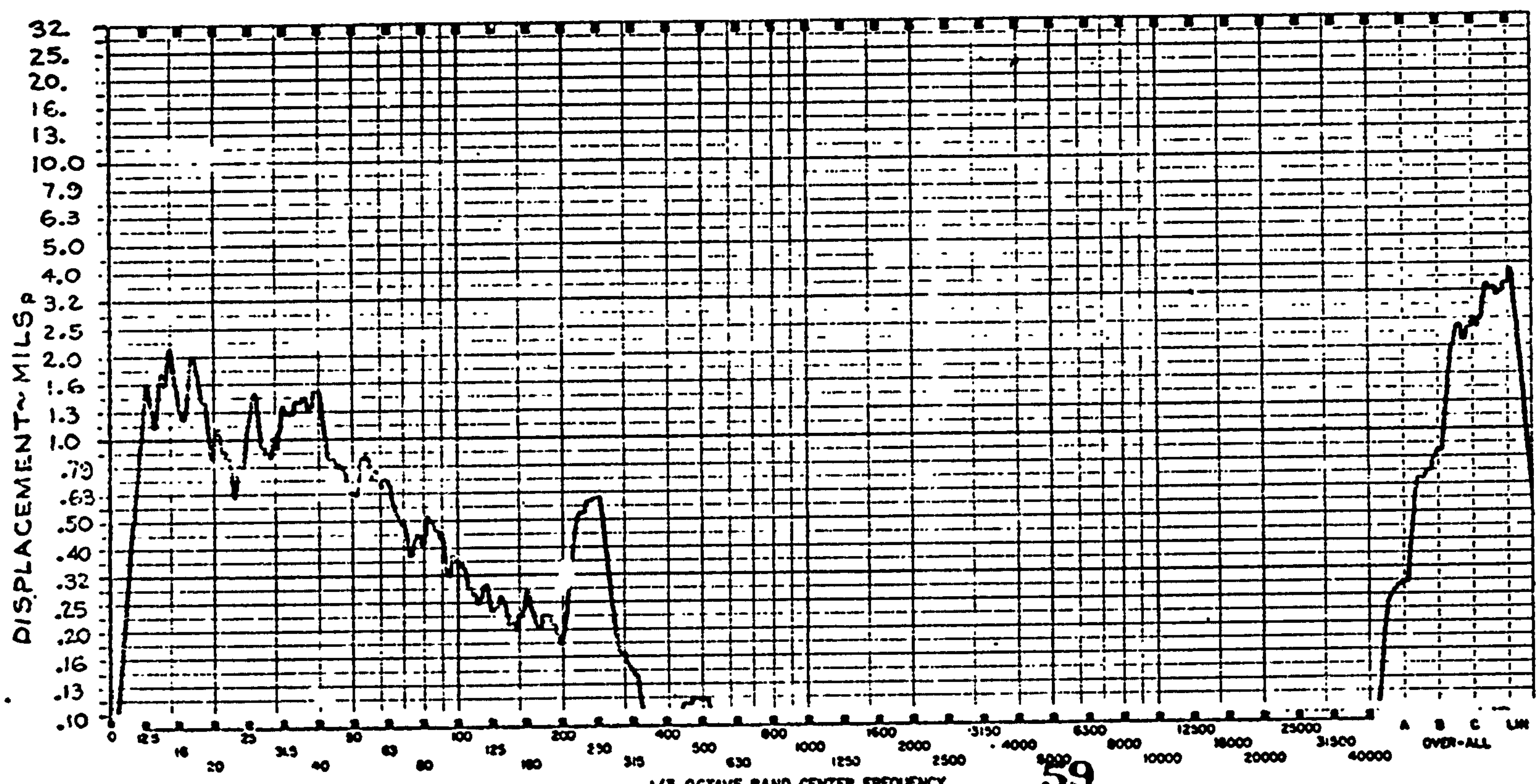
PROP ROTATIONAL = 16 Hz



CURVE NO. 4

ER NO. 6 P.P. 2

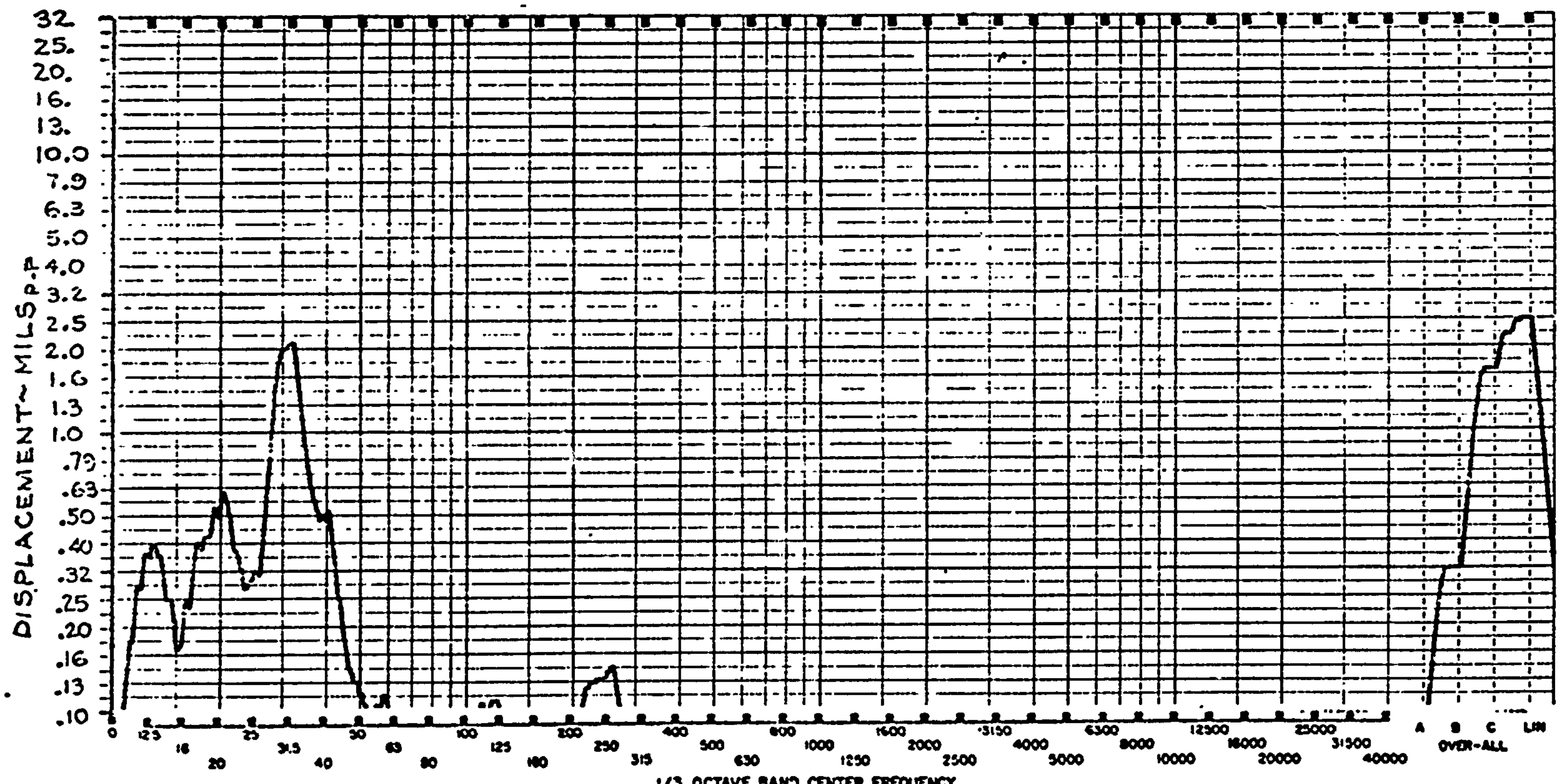
G/B FRONT CASE BOTTOM F & A
15,000 RPM = 250 HZ
PROP ROTATIONAL = 18 HZ



CURVE NO. 5

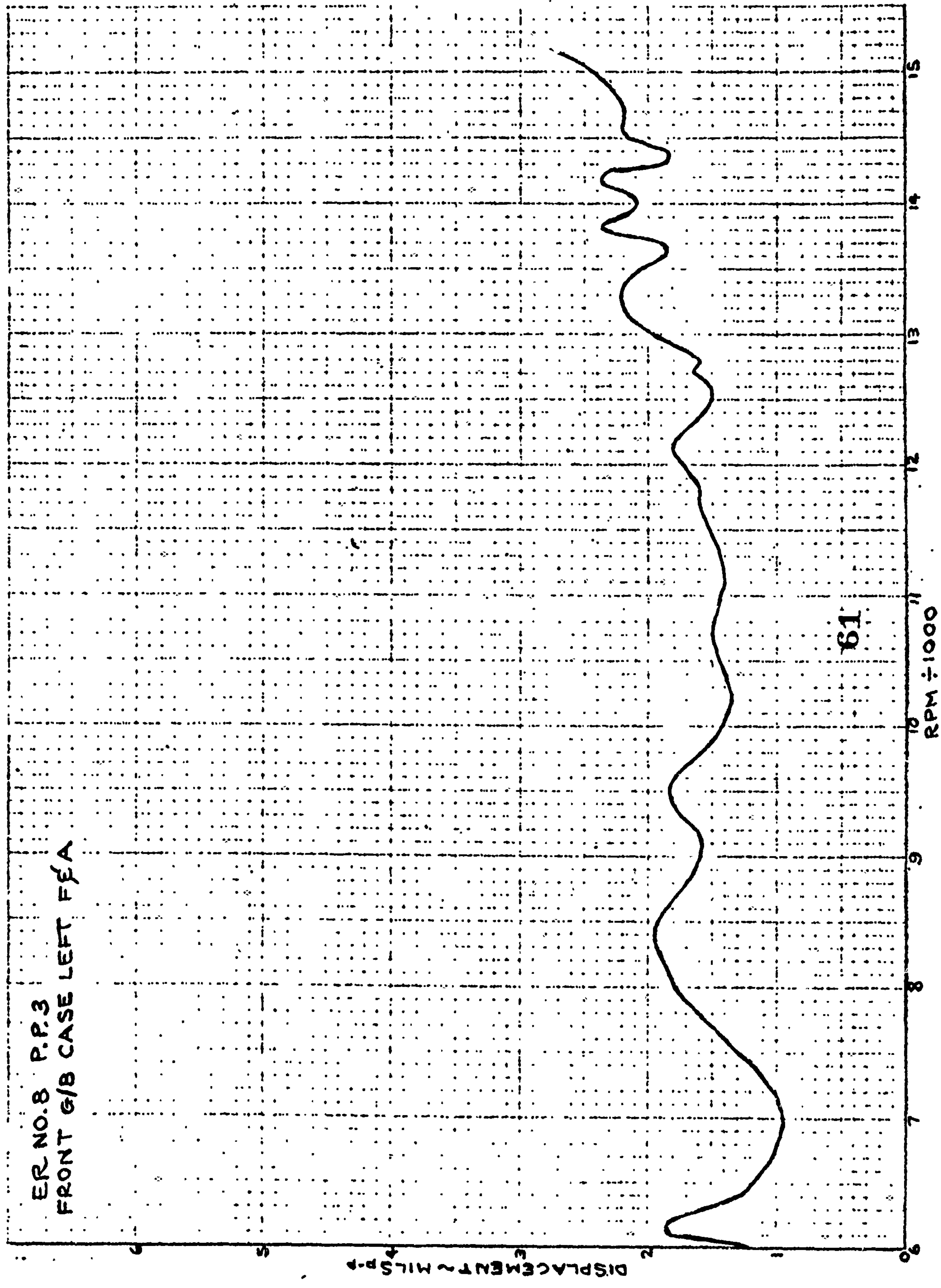
ER NO. 7 P.P. 2

G/B FRONT CASE BOTTOM F&A
8000 RPM = 133 HZ
PROP ROTATIONAL ≈ 10 HZ



CURVE NO. 6

CURVE NO. 7

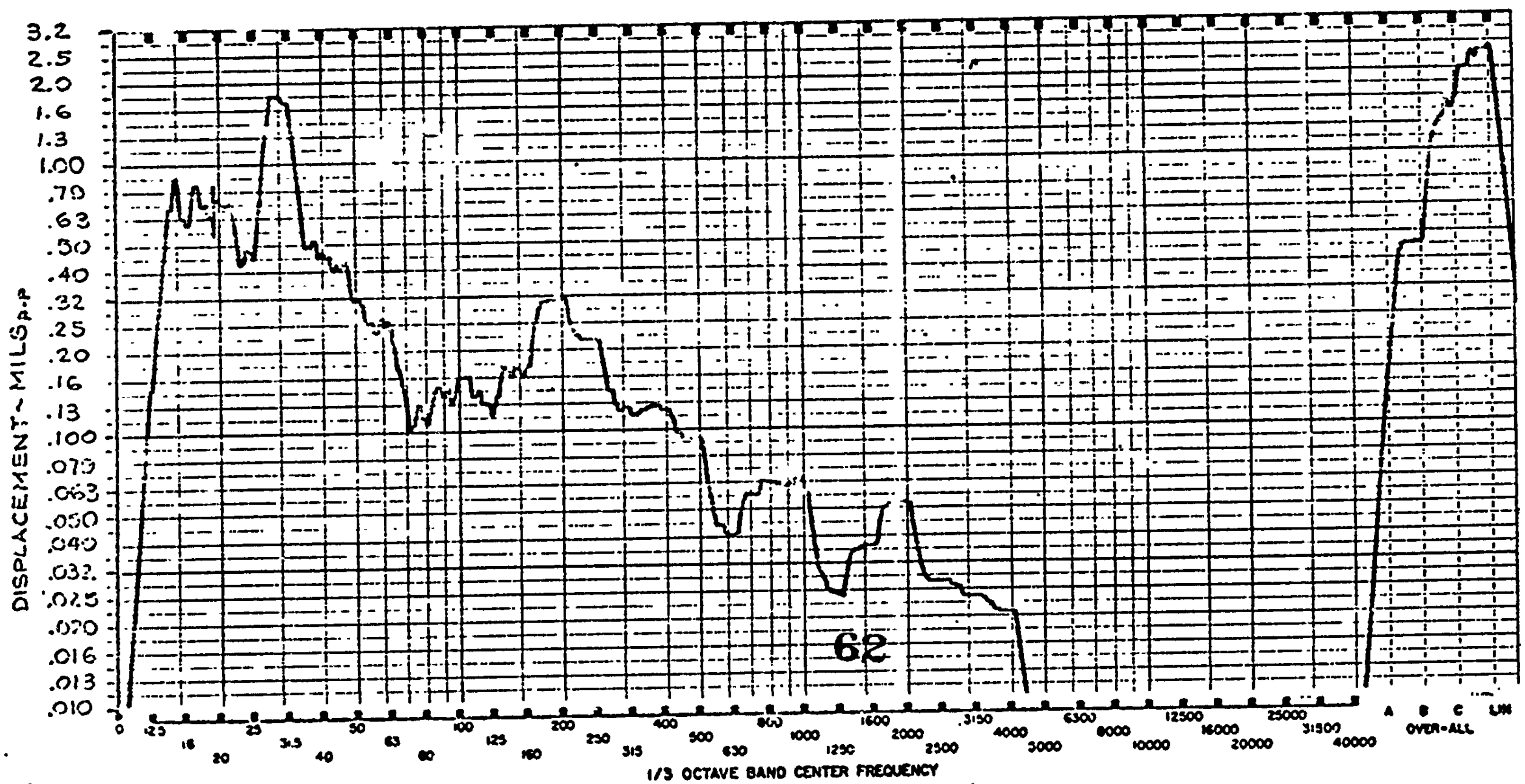

ER. NO.8 P.P.3
FRONT G/B CASE LEFT F&A
DISPLACEMENT ~ MILS p-p
0
1
2
3
4
5
6
RPM ÷ 1000
6
7
8
9
10
11
12
13
14
15

ER NO 9 P.r.s

G/B FRONT CASE LEFT F&A

13,200 RPM = 220 Hz

PROP ROTATIONAL = 16 Hz



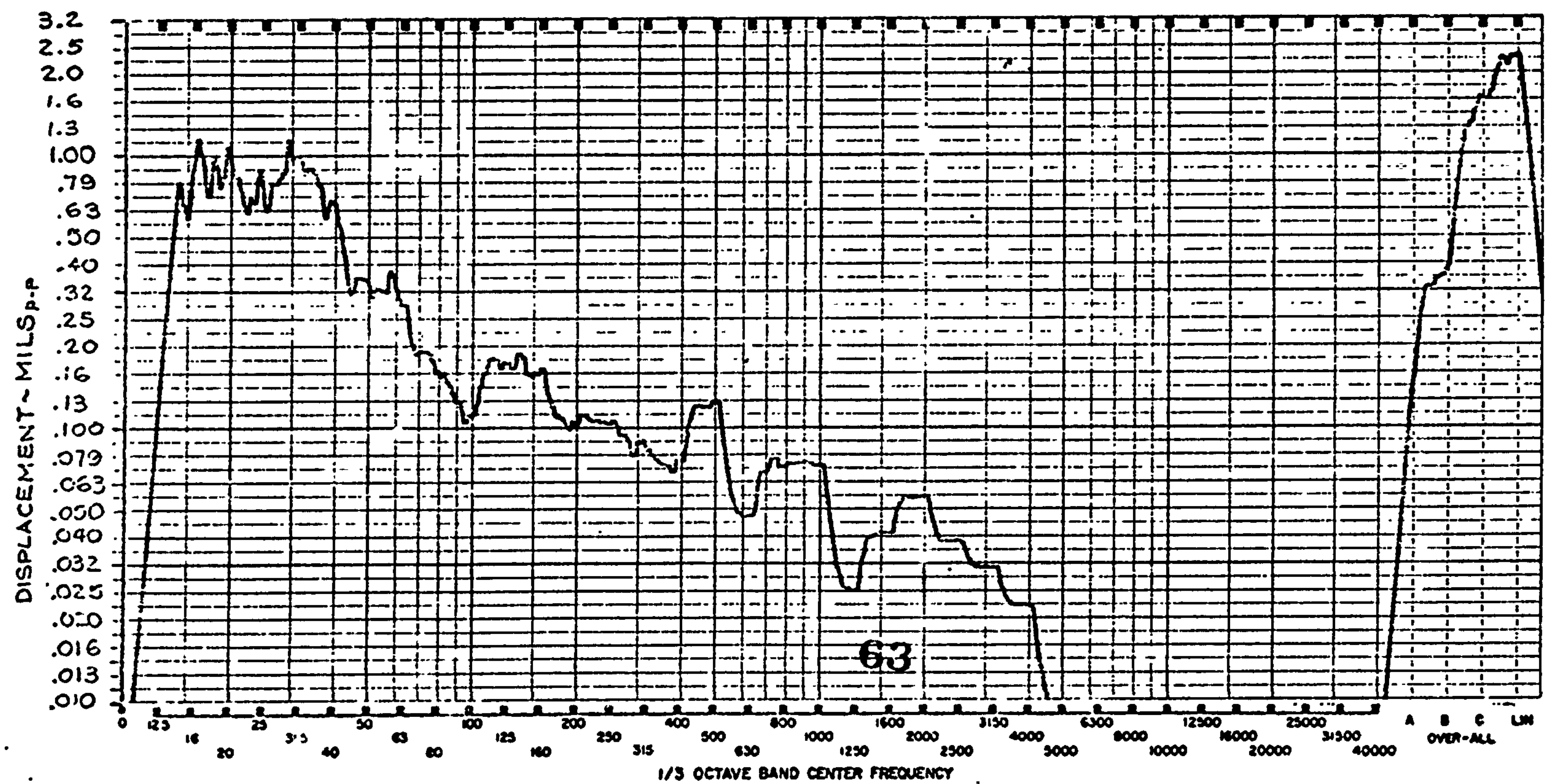
CURVE NO. 8

ER NO 10 P.P. 3

G/B FRONT CASE LEFT F&A

14,200 RPM = 237 HZ

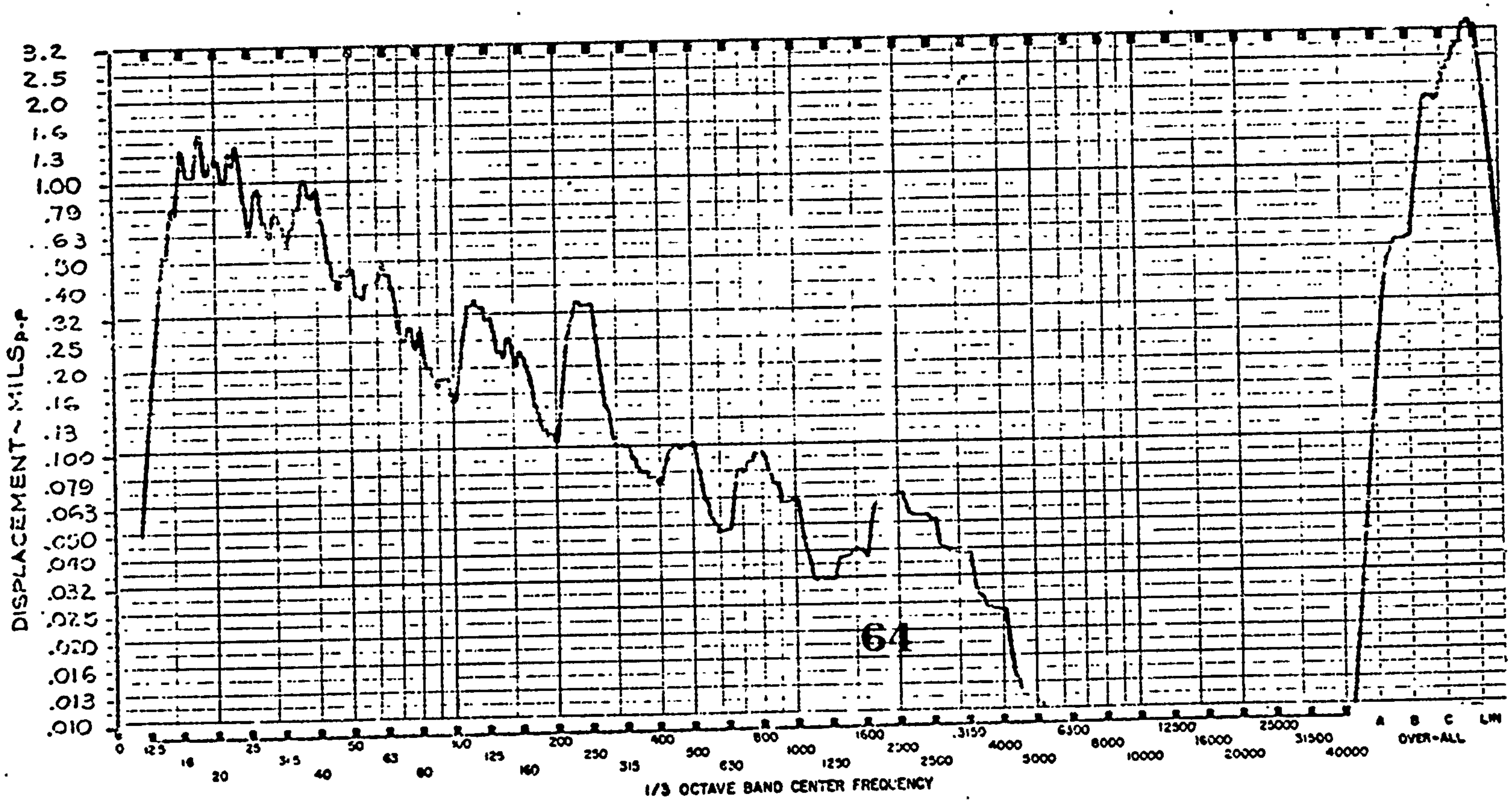
PROP ROTATIONAL = 18 HZ



CURVE NO. 9

E.R. NO. 11 F.F. 3

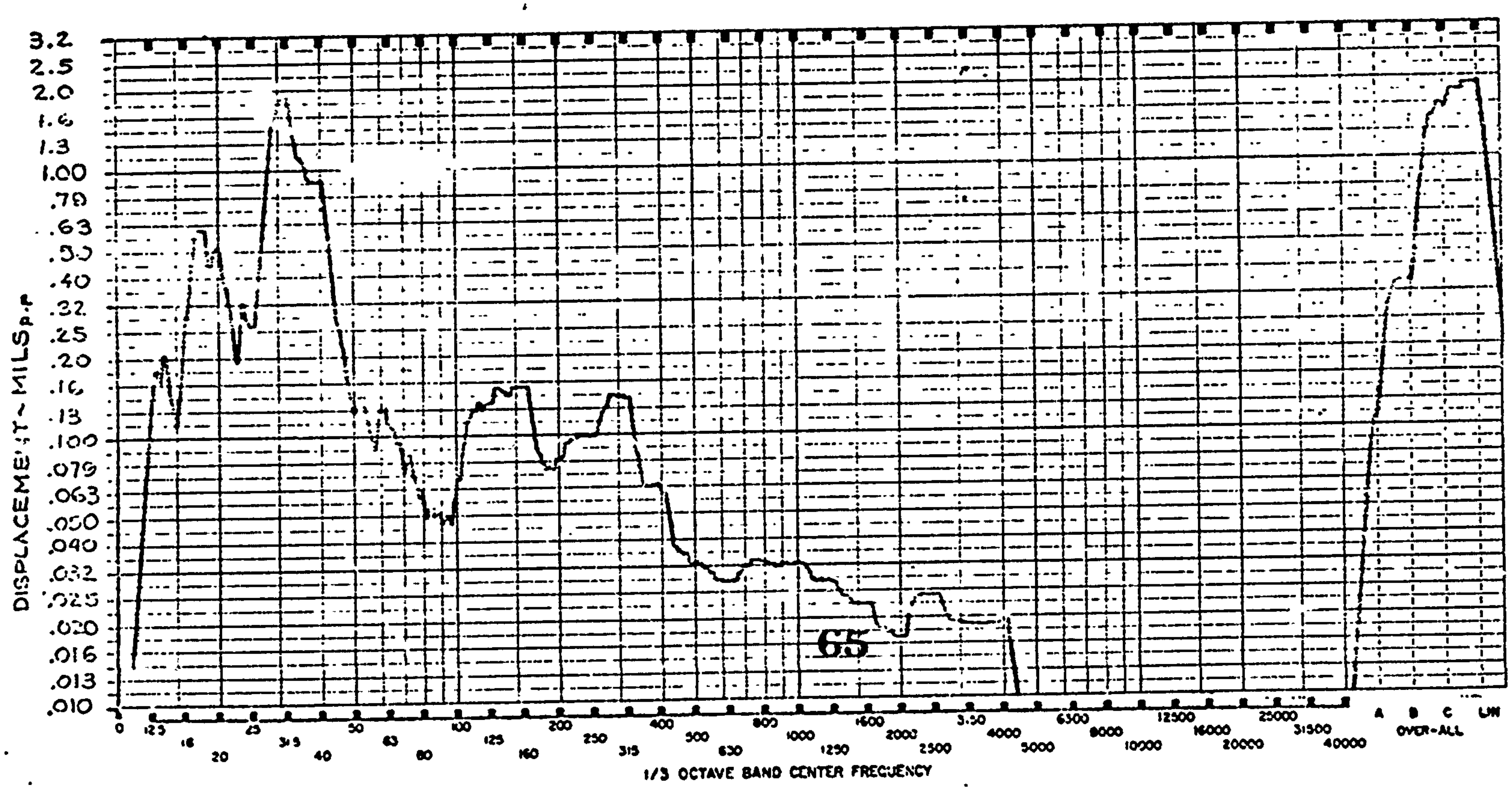
G/B FRONT CASE LEFT F&A
15,150 RPM = 252 HZ
PROP ROTATIONAL = 19 HZ



CURVE NO. 10

EK NO. 12 T.P. 3

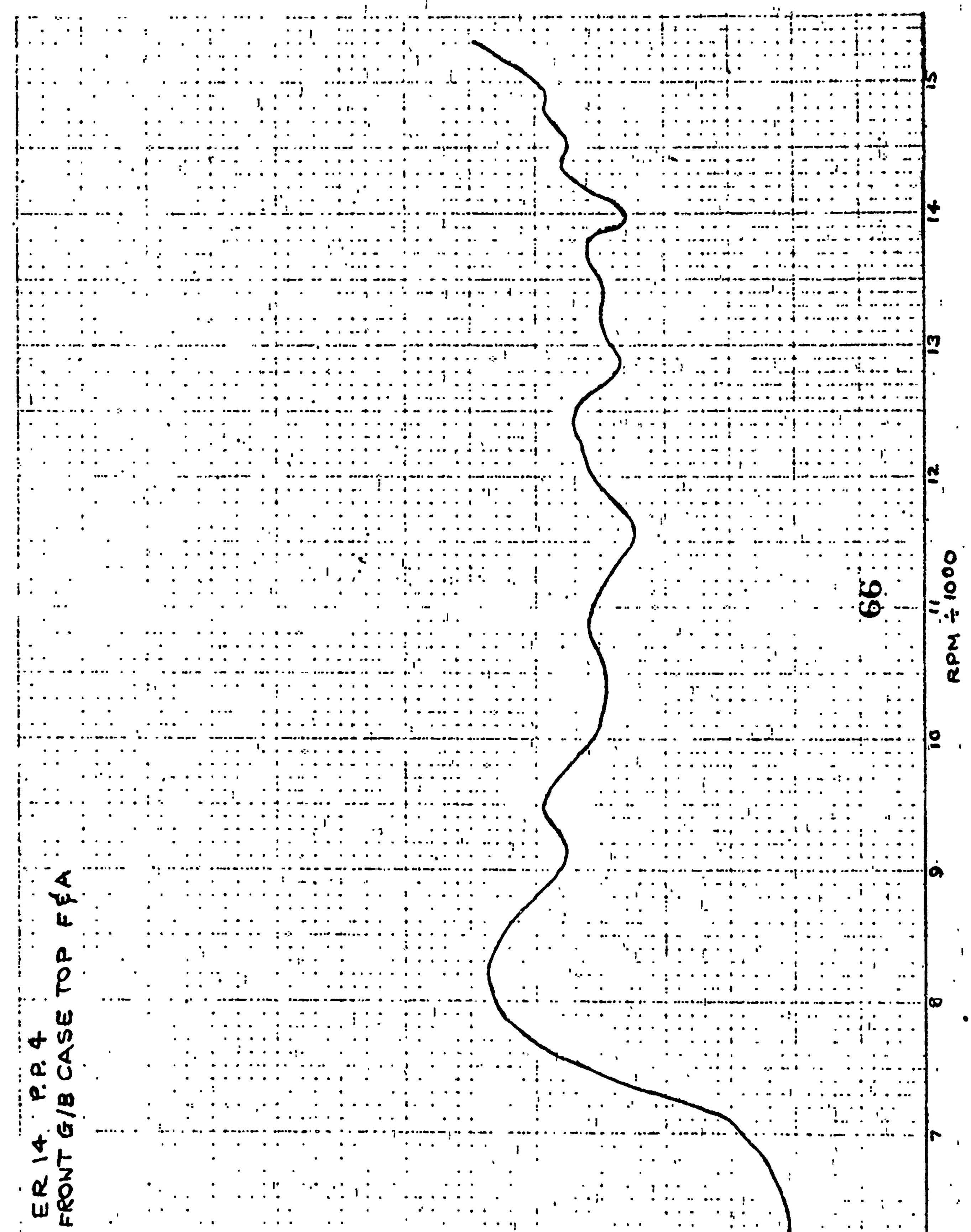
G/B FRONT CASE LEFT F/A
3480 RPM = 141 HZ
PROP ROTATIONAL = 10 HZ



65

CURVE NO. 11

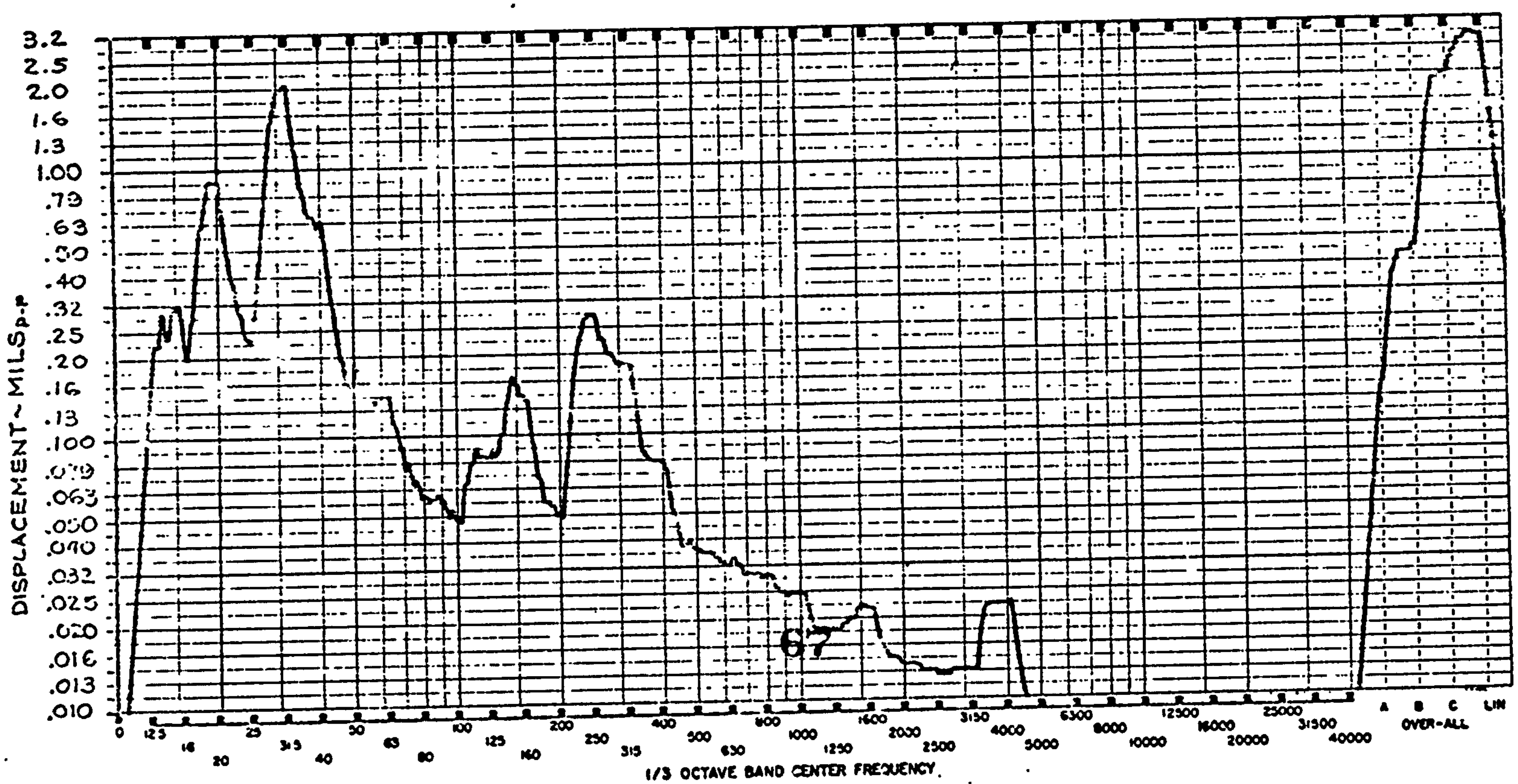
CURVE NO. 12



66

ER NO. 15 P.P. 4

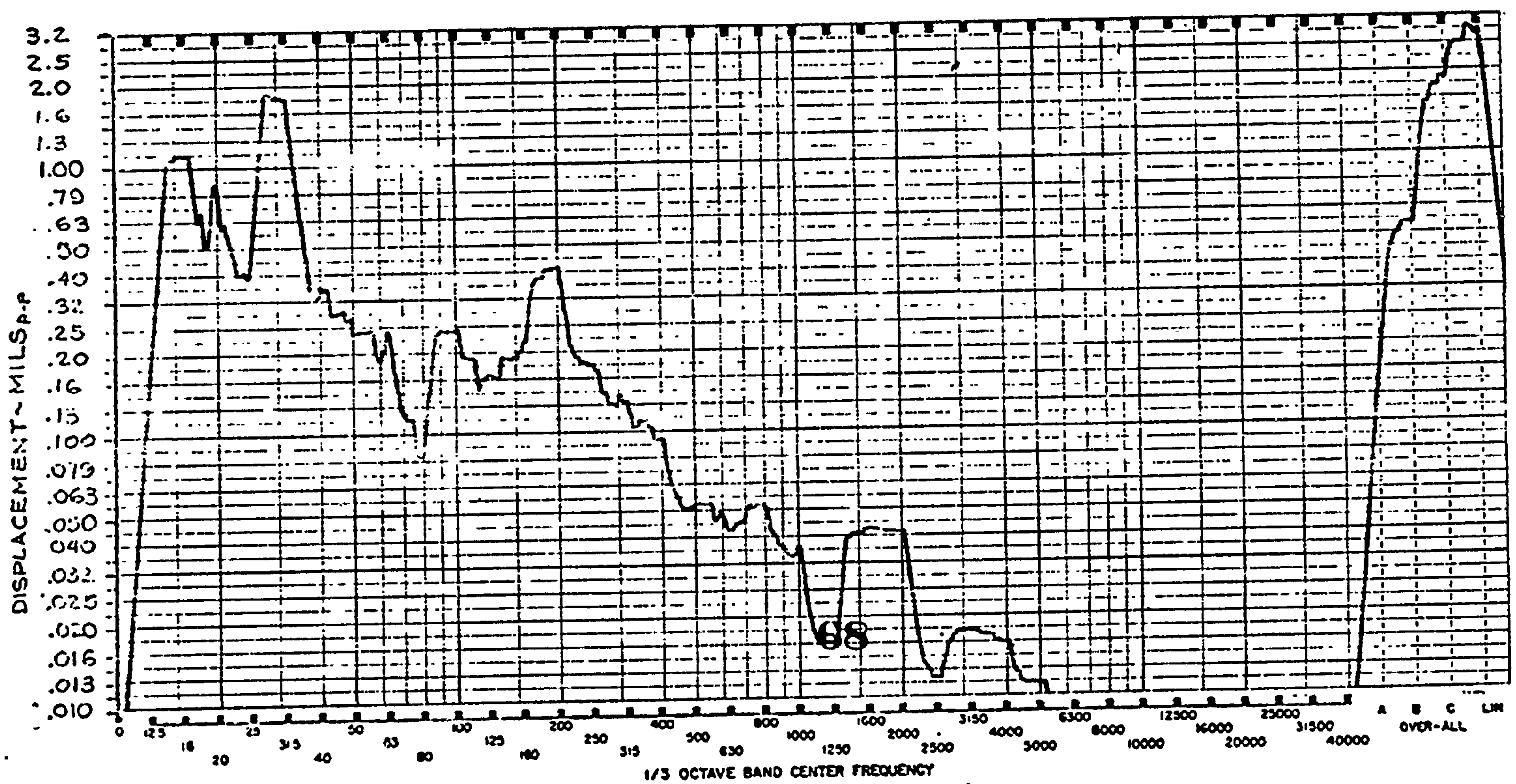
G/B FRONT CASE TOP F&A
8200 RPM = 137 Hz
PROP ROTATIONAL = 10 Hz



CURVE NO. 13

ER NO 16 P.P. 4

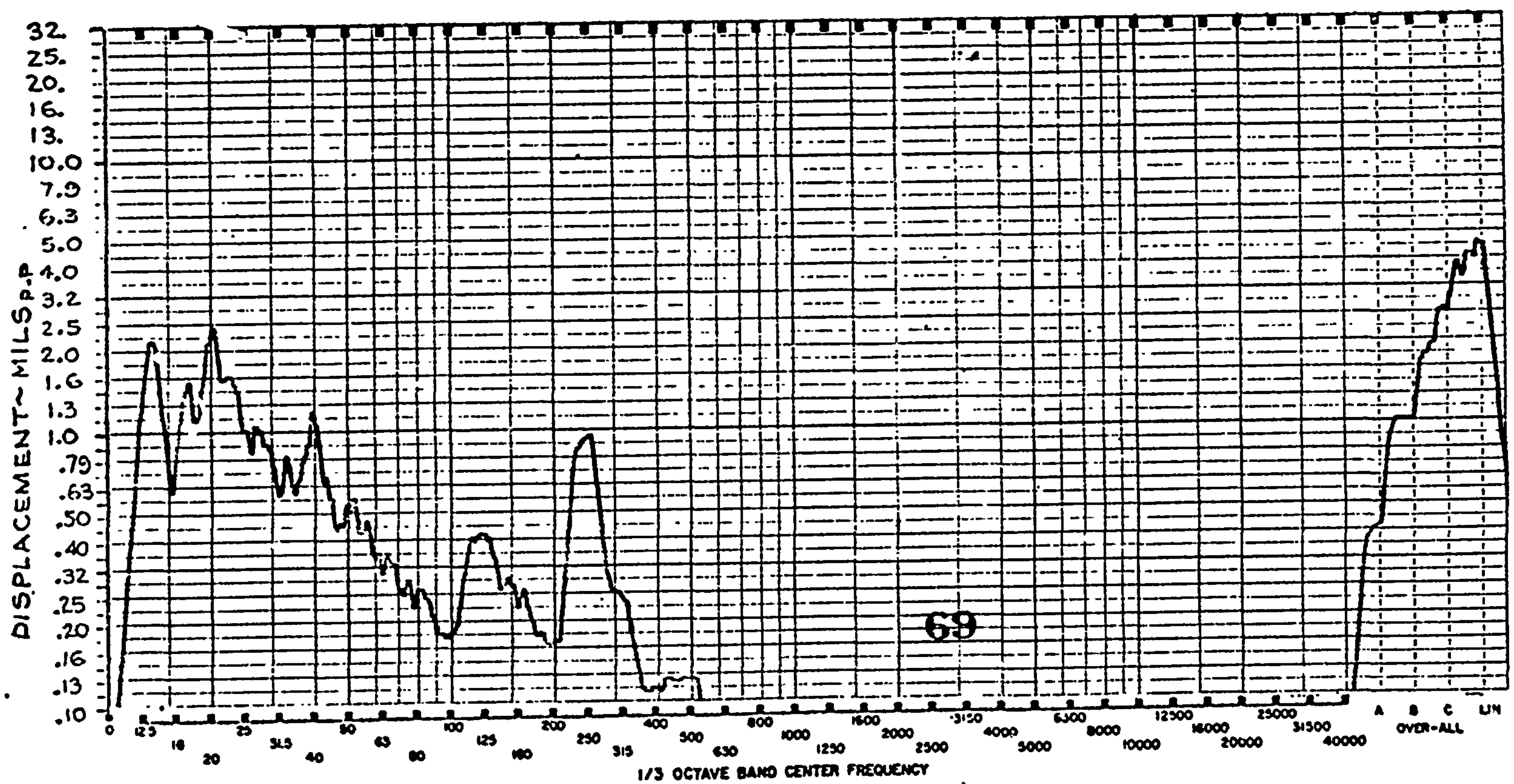
G/B FRONT CASE TOP F&A
12,530 RPM = 209 HZ
PROP ROTATIONAL = 15 HZ



CURVE NO. 14

ER NO. 17 P.P. 4

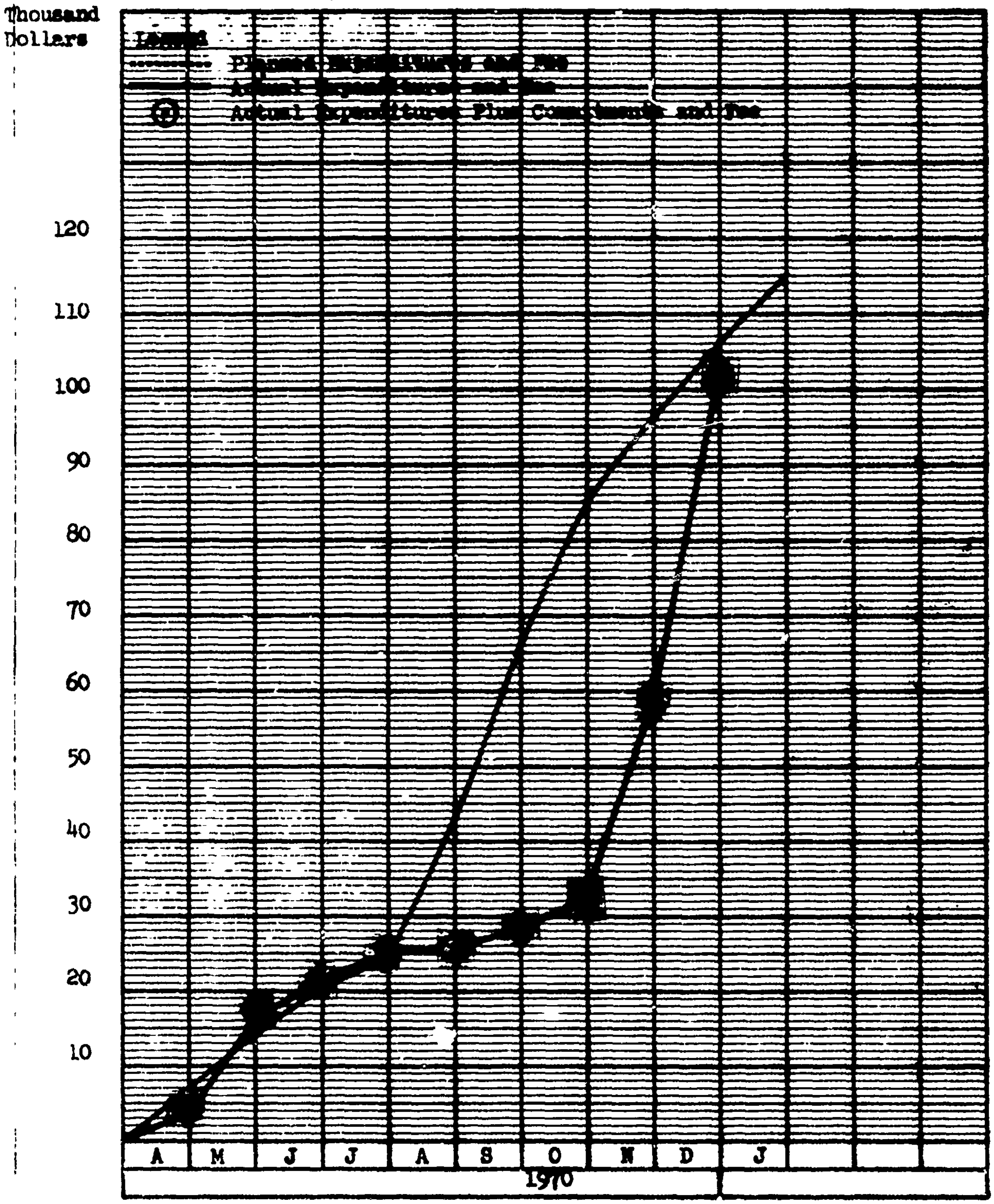
G/B FRONT CASE TOP F&A
15,180 RPM = 253 HZ
PROP ROTATIONAL = 19 HZ



CURVE NO. 15

EXPENDITURE CURVE
LIGHTWEIGHT GEARBOX COMPOSITE CASE STUDY

Contract: M00019-70-C-0244 Date 12 JAN. 1971



FORM 52

FORM 2381-2
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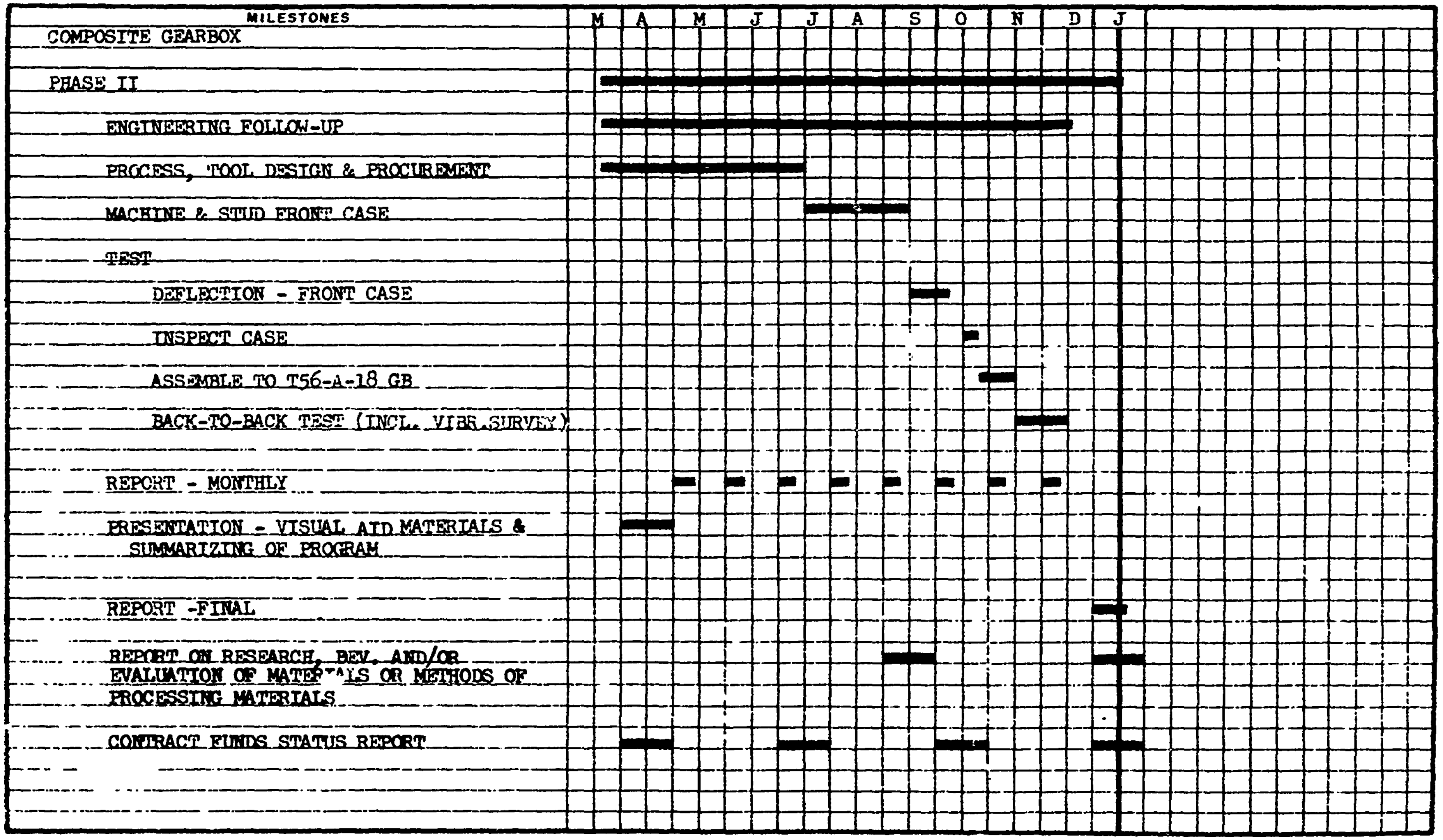
PROGRAM MASTER SCHEDULE



APPROVALS:
PROG. ADMIN SUPV. *(signature)*
CH. PROJ. ENGR. *(signature)*
CH. PROJ. ENGR. SUPV.
DRAFTING
PROCUREMENT

FABRICATION
TEST

SCHEDULE NO. P155-1 DATE 4-14-70
PREPARED BY: J. M. Duvall
REVISED: 1-12-71 PAGE 1 OF 1