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EVALUATION OF "ZYTEL" AS A GEAR MATERIAL

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TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF ILLUSTRATIONS

ABSTRACT

The object of this project is to evaluate molded "Zytel" as a gear material. Five identical testing machines were designed and built. Preliminary tests, aimed at checking the performance of the machines, have been completed. These tests show the machines to be consistent with one another, and capable of measuring very small values of losses in the teeth of the test gears.

Further study will endeavor to correlate gear losses and gear wear.

EVALUATION OF "ZYTEL"AS A GEAR MATERIAL

I. INTRODUCTION

This project was undertaken for the Polychemicals Department of E. I. du Pont de Nemours and Company, Wilmington, Delaware.

The purpose of the project is to evaluate molded "Zytel" as a gear material and to establish design data for gears made of "Zytel". This is outlined in detail in the proposal of May, 1953, entitled "Design Data for Nylon Gears," submitted to E. I. du Pont de Nemours and Company by the Engineering Research Institute of the University of Michigan.

The evaluation of any gear material requires a good deal of testing to obtain necessary data. Testing, however, can be very time-consuming and expensive. This is especially so when obtaining data on useful life and wear rates of gear teeth over a wide range of gear sizes and operating conditions.

It was therefore decided that in the testing work, an attempt would be made to correlate the power losses in the gear teeth with the rate of wear or other type of deterioration of the teeth and the useful life of the gears under various operating conditions. If such a correlation can be soundly established it will then only be necessary to measure the power losses in the teeth of the gears being tested in order to accurately predict the rate of wear and useful life of these gears. Thus a lot of time-consuming test work can be eliminated.

All of the work described, covered by this report, was done with the above thoughts in mind.

This report covers the work done from August 1, 1953, when the project started, to the present time, a period of 16-1/2 months. All of this work was financed by the initial grant of $25,000.

II. EXPERIMENTAL APPARATUS

A. Test Machines

Five identical test machines were designed and built to carry out the proposed testing program. Illustrations 1 and 2 show the completed installation of these five test machines.

The test machines are of the "4 square" or "back-to-back" type in which two pairs of gears are tested simultaneously by loading one pair of gears against the other pair of gears. Each of the four gears are mounted on 5/8-inch, outside-diameter, splined, hollow shafts. These hollow shafts are supported in ball bearings and are joined together by steel torsion bars. Illustration 3 shows a schematic layout of the test machine. Gear tooth loading is obtained by applying a twisting moment to the outer half of a friction coupling while the inner half of the coupling is anchored. This is shown in Illustration 4.

The outer half of the coupling is splined to the torsion bar which in turn is splined to the hollow shaft of the upper left-hand gear in Illustration 3. The inner half of the coupling is splined to the hollow shaft of the lower left-hand gear in Illustration 3. Thus the loading shown imposes a twisting moment, or torque, on the entire gear system, putting the same load on the teeth of all four gears being tested.

The two halves of the coupling are clamped together by the four screws shown in Illustration 4, thus keeping the torque in the system when the loading and anchoring cables are removed to permit rotation of the gears.

Movable bearing supports allow the center distance between the gear shafts to be varied from a minimum of 1 inch to a maximum of 4 inches, thus the machines can test gears of various sizes. The maximum torque which can be exerted on the gears is approximately 300 lb-in.

The driving motor on each machine supplies only the power needed to overcome the losses in the system due to friction, windage, etc. The power supplied to overcome these losses is measured by use of a flexure-plate drive unit located between the motor and the gears being tested. Details of this drive are shown in Illustrations 5, 6, and 7.

Illustrations 5 and 6 show the driving half of the unit containing the flexure plate which is directly coupled to the driving motor. The driven half of the unit is an aluminum shroud containing two steel pins 180° apart. The flexure plate contacts these pins, thus driving the aluminum shroud. The

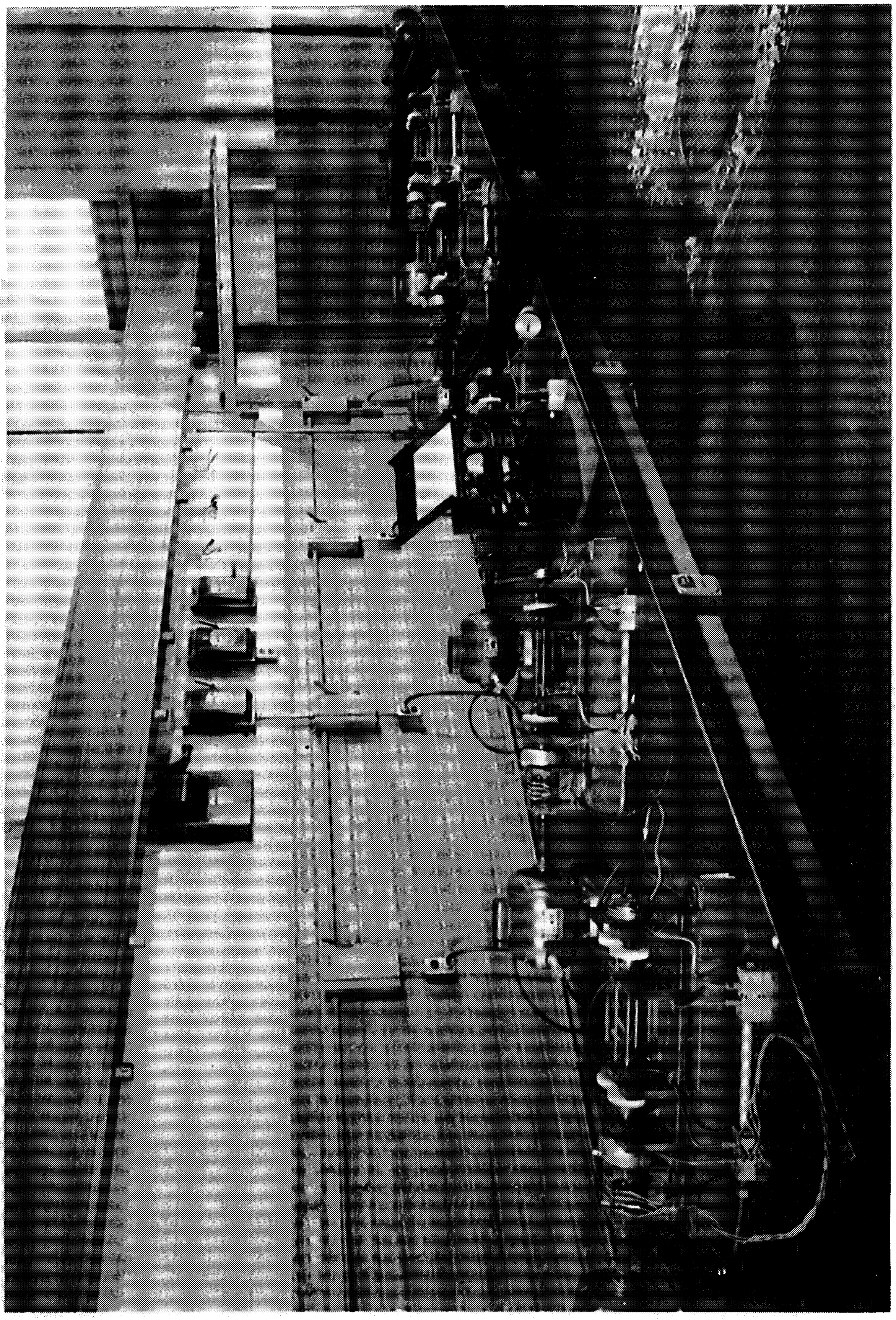

Illustration 1. Test Machine Installation Viewed from Left.

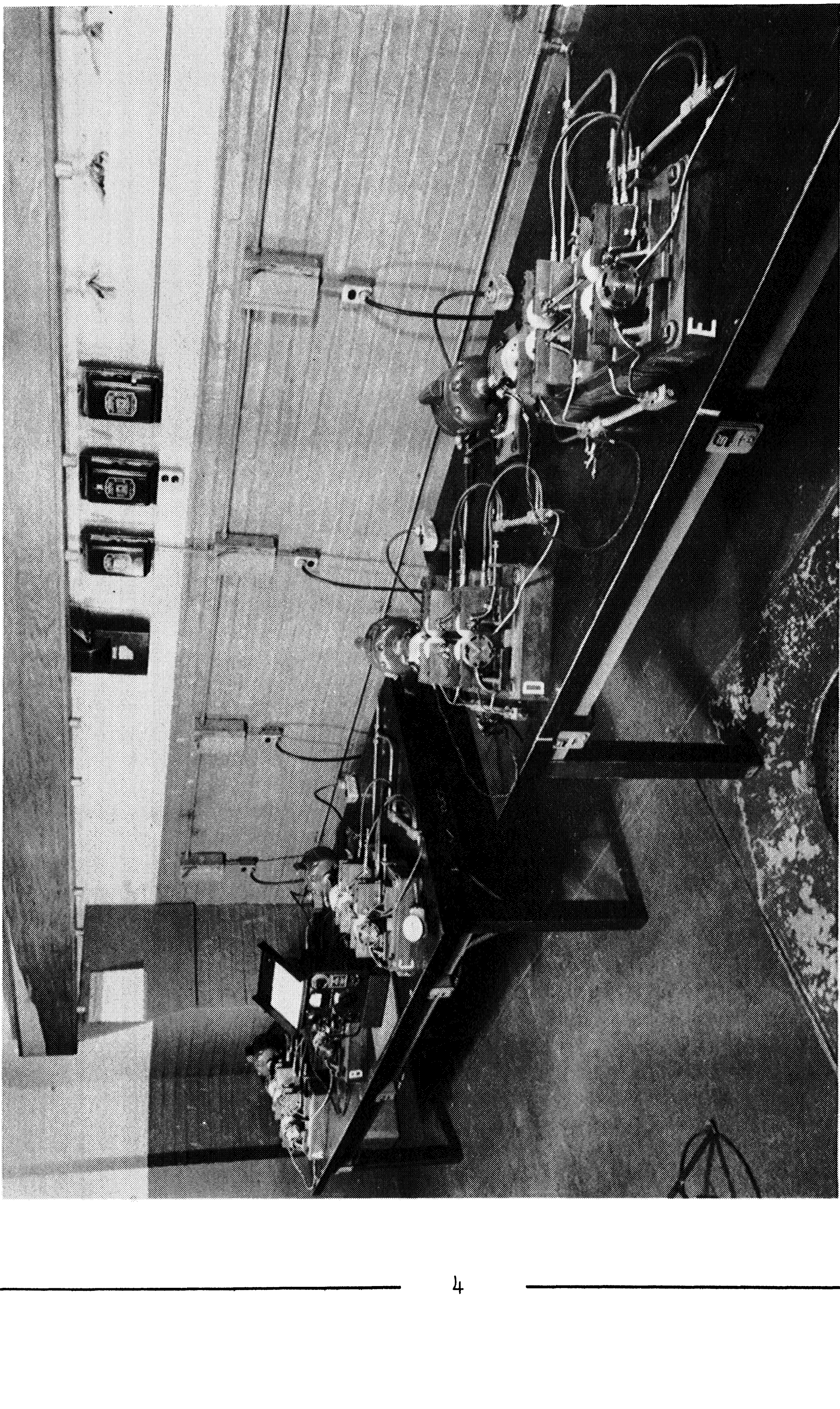

Illustration 2. Test Machine Installation Viewed from Right.

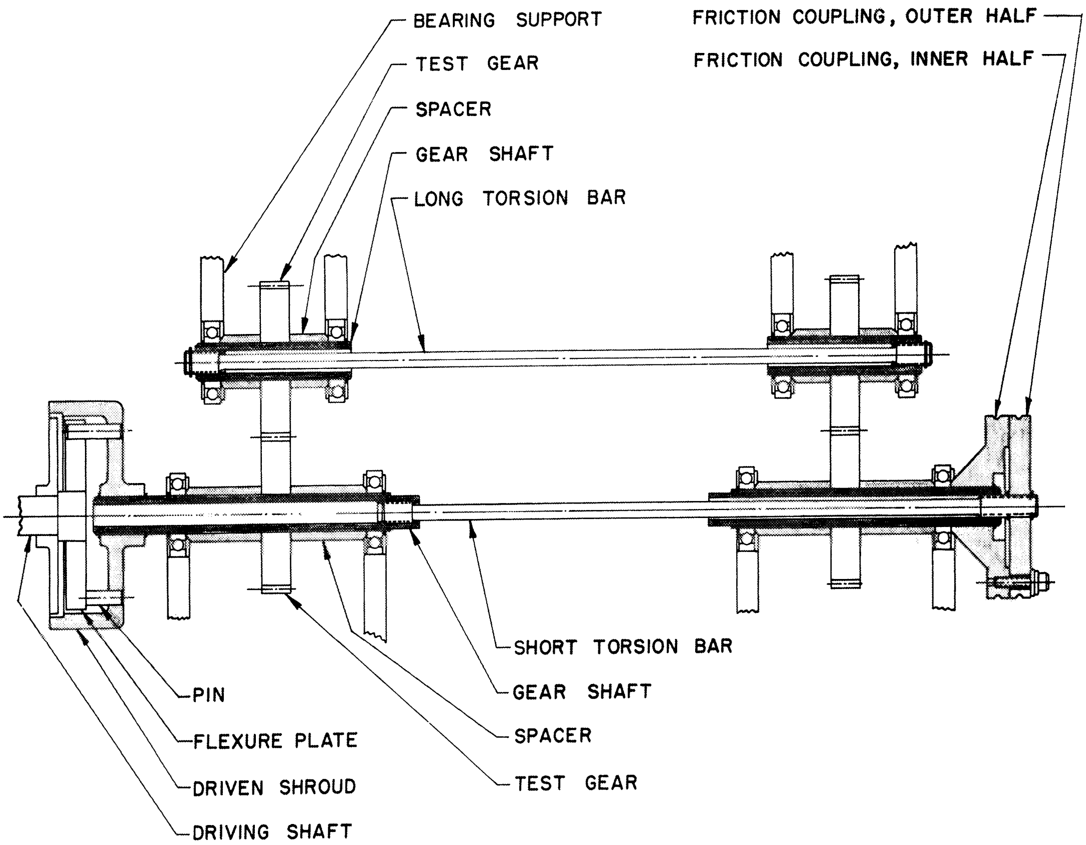

Illustration 3. Schematic Layout of Test Machine.

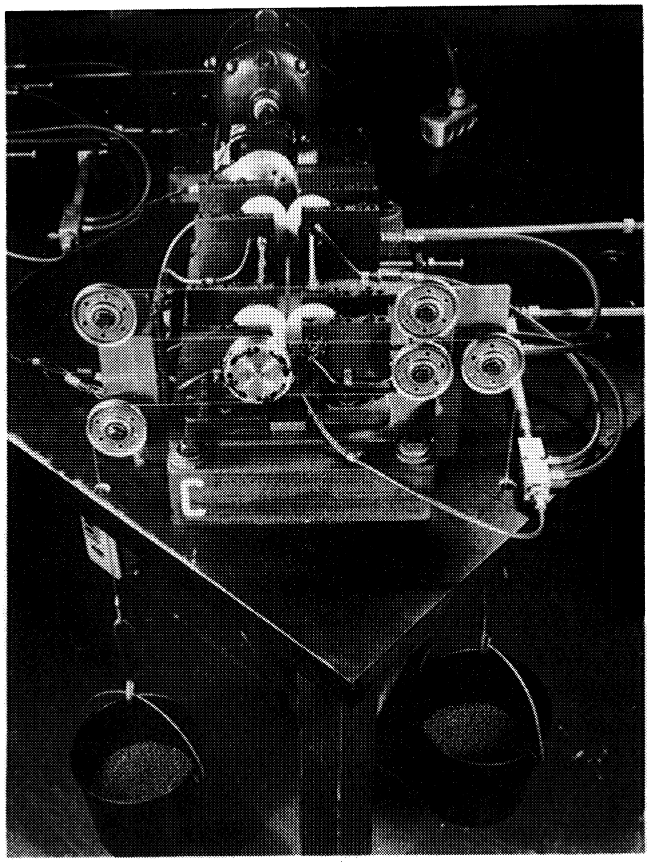

Illustration 4. Gear Tooth Loading Setup.

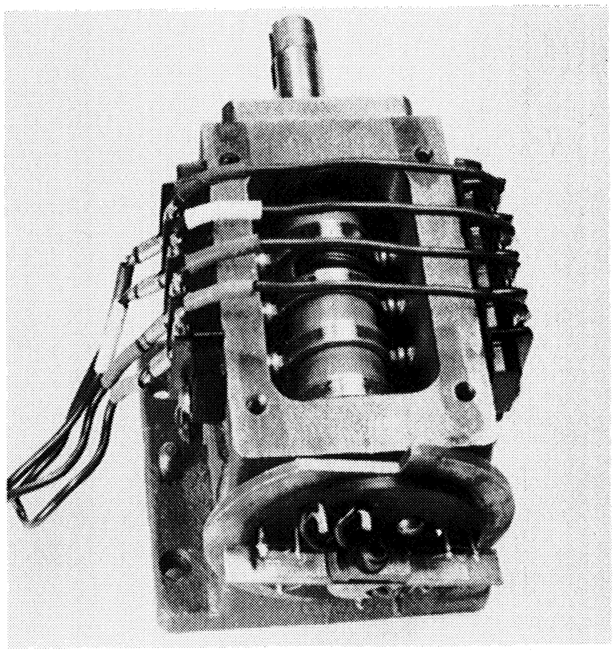

Illustration 5. Flexure-Plate Assembly, Top View.

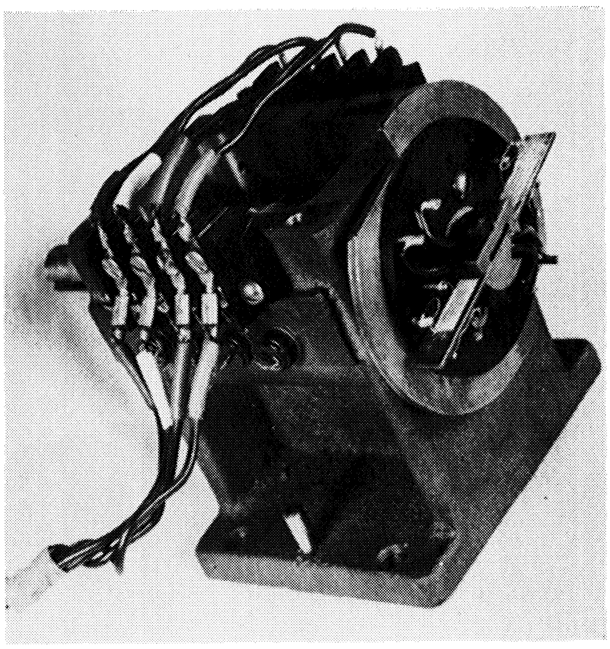

Illustration 6. Flexure-Plate Assembly, Oblique View.

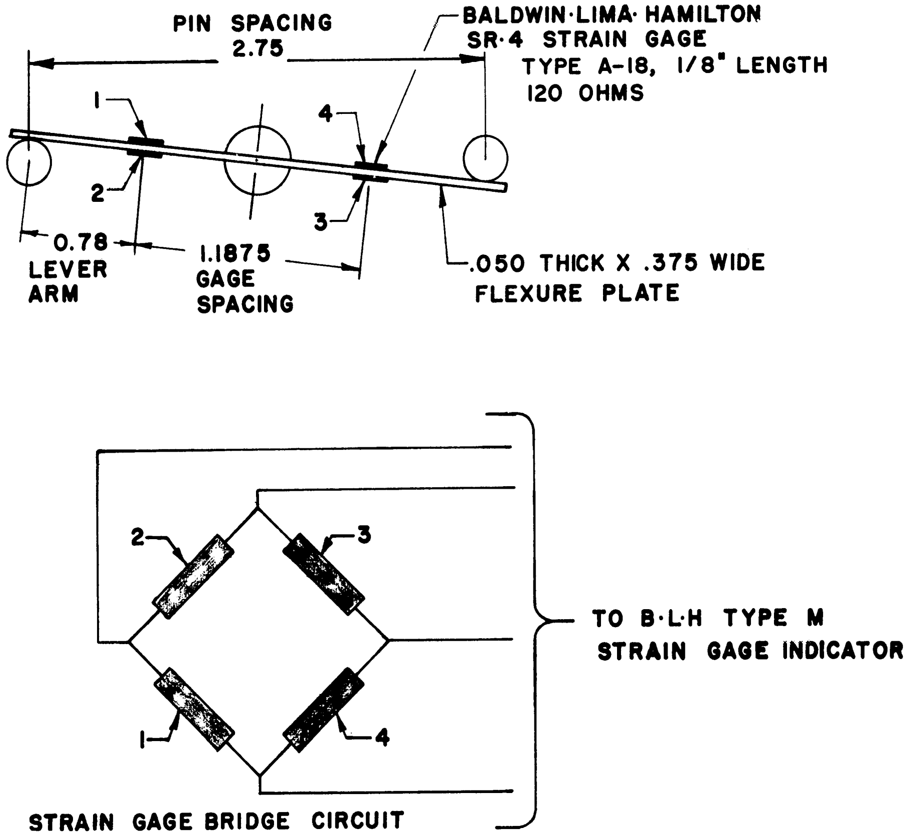

Illustration 7. Sketch of Flexure-Plate Wiring Diagram.

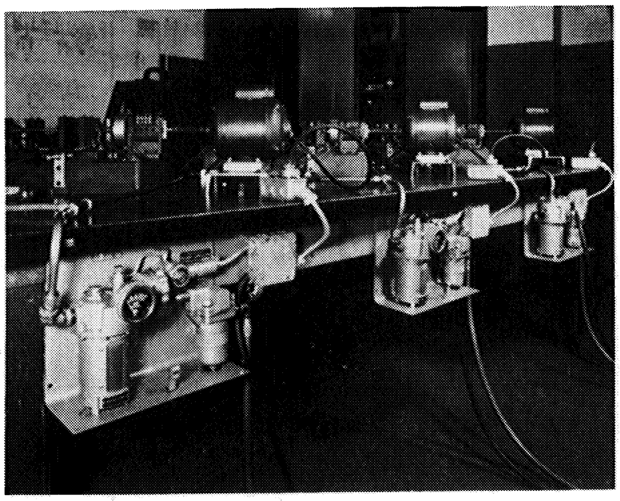

Illustration 8. Alemite Oil-Mist Lubricators.

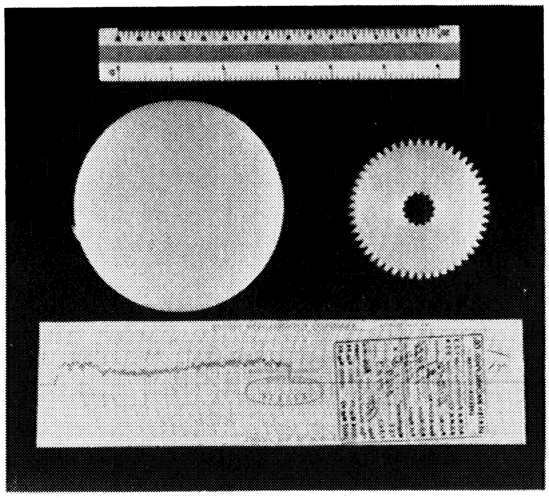

Illustration 9. Gear, Blank, and Chart.

shroud in turn is splined to a gear shaft. Power is transmitted through the flexure plate, and bonded wire strain gages mounted on the flexure plate permit measurement of flexure-plate load. Four gages on the flexure plate form a complete Wheatstone bridge. The bridge signal is taken out through four silver slip rings, each in contact with two carbon brushes. Illustration 7 shows the flexure-plate circuit diagram and the position of the pins in the driven member.

Lubrication for the gears and bearings of each machine is provided by an Alemite oil-mist lubricator as shown in Illustration 8. Individual oil-mist tubes go to each bearing from common headers, and a separate tube carries the oil mist to the gears.

When it becomes desirable to run the gears without lubrication, the tube carrying the oil mist to the gears can be removed. A sheet-metal housing is then slipped over each pair of gears to keep the oil mist supplied to the bearings from reaching the gears. As a further precaution the oil mist leaving the bearings is drawn off through a suction hose to an overhead venting duct.

During the work covered by this report, Socony-Vacuum turbine oil was used with each lubricator 1/2 turn open and with 5 psi air pressure. Both gears and bearings were lubricated.

B. Instrumentation

To date the following instrumentation has been used:

1. Baldwin SR-4 strain gage indicator, model M
2. James G. Biddle No. K-0 continuous reading tachometer, range 30 to 12,000 rpm
3. General Radio Strobotac, type 631-B, range 600 to 14,400 rpm
4. Pratt and Whitney hoke precision gage blocks
5. Cenco sling psychrometer
6. Buffalo platform scale

C. Test Gears

All work to date has been done on 50-tooth, 20° pressure angle, 20-pitch spur gears 7/16-inch wide. These gears have hob-cut teeth rather than molded teeth. The teeth are of the AGMA full height proportions, cut to have .002 to .004 inch backlash when operating at the theoretical center distance of 2.500 inches.

Illustration 9 shows one of the test gears, one of the molded "Zytel" blanks from which the gears were machined, and an inspection chart supplied by the vendor who made the gears. The molded "Zytel" blanks were supplied by du Pont. This material was formerly designated du Pont nylon FM-10001. Whether or not the gear blanks were moisture conditioned is not known.

The gear teeth were machined with AGMA commercial class 3 hobs. Actual inspection of the gear teeth after manufacture showed total composite errors of about .001 inch and tooth-to-tooth composite errors of about .0004 inch, thus meeting the requirements for AGMA precision class 1.

III. EXPERIMENTAL WORK AND TEST RESULTS

A. Machine Assembly

The five test machines were assembled without difficulty. Considerable trouble was encountered with the drive setup from the motor to the flexure-plate unit, however.

As originally designed, each test machine was mounted on an individual table and was driven by V-belts from the motor underneath the table. A Speedmaster variable speed unit was incorporated in the belt drive to allow speed adjustments over a relatively wide range. When put in operation it was found that the belts vibrated so violently that the entire output of the 1/4 horsepower, 3450 rpm a-c motor was absorbed by the drive unit leaving no power to drive the test machine. The belt vibrations also vibrated the entire table and test machine.

Experimentation failed to improve this situation, so the Speedmaster variable speed units were discarded. At this same time the individual tables were discarded and the test machines mounted on the large tables shown in Illustrations 1 and 2.

With the machines on the large tables, a V-belt drive was again tried but without the Speedmaster units. The motors were mounted on the tables, and a single belt was used; this also proved to be unsatisfactory. Small variations in the cross section of the belt caused the flexure-plate unit to have small but continuous variations in speed, making strain gage readings from the flexure plate impossible. With this setup the indicating needle on the Baldwin strain gage indicator fluctuated continuously over a wide range.

Adding inertia to the flexure-plate unit failed to help the situation so the belts were discarded. The motors were directly coupled to the flexure-plate units by means of a 6-inch length of rubber hose. This provided a fairly vibrationless drive and to date has proven to be very satisfactory.

The ball bearings which support the gear shafts are mounted with a slight interference fit in split cast iron bearing supports. This construction is readily seen in Illustration 4.

Although the castings had been stress relieved prior to machining, the bored holes in the bearing supports distorted slightly several weeks after the machines had been assembled. This distortion resulted in excessive clamping forces that prevented free rotation of bearings. The condition was corrected by hand scraping of bearing support holes. Fortunately, further distortion has not been detected.

B. Flexure-Plate Calibration

Illustration 10 shows the method used to calibrate the flexure plates. The motor shaft was held by a pipe wrench to prevent rotation while a known torque was applied by hanging weights on the cords attached to the friction coupling. The strain reading, in microinches per inch, was then read on the Baldwin strain indicator. During the calibration a motor of one of the other machines on the same table was kept running to provide a slight vibration to the table, thus minimizing the effects of friction.

Figures 1 through 5 show the flexure-plate calibration curves for the five test machines. The data from which the curves are plotted are shown in Tables I through V. It will be noted that the flexure plates are calibrated statically but are used dynamically. It is felt that very little error is thus introduced, however.

C. Bearing Tests

The test machines were set up as shown in Illustrations 11 and 12 for determination of the friction in the ball bearings. The two bearings from each of the two rear gear shafts were installed in a steel ring and then mounted in place of the gears on the two front gear shafts. Weights hanging on rods fastened to the rings provided equal loads on all four bearings of each assembly. The machines were run at 1770 and 3540 rpm with various loads on the bearing assemblies. The torque required to overcome bearing friction was determined by the flexure-plate strain readings.

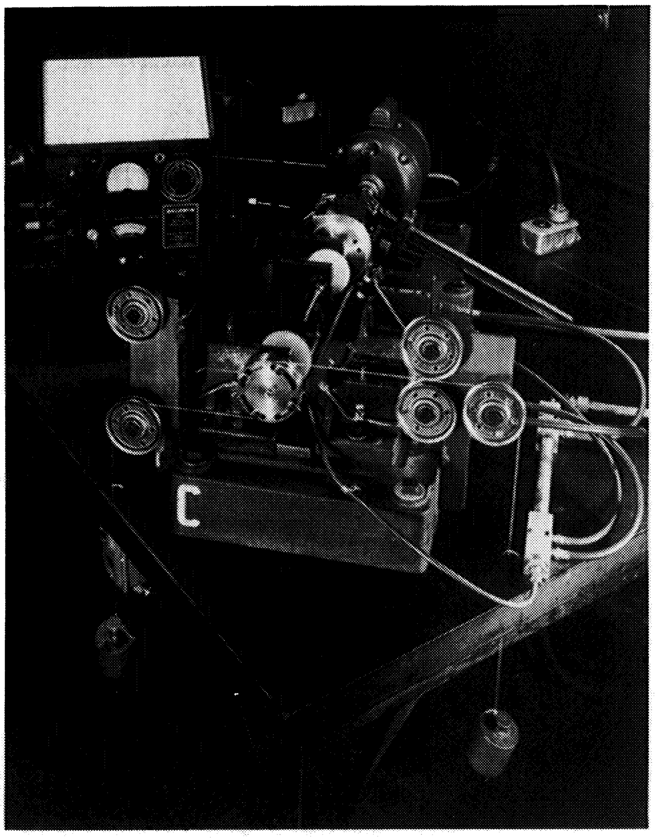

Illustration 10. Flexure-Plate Calibration.

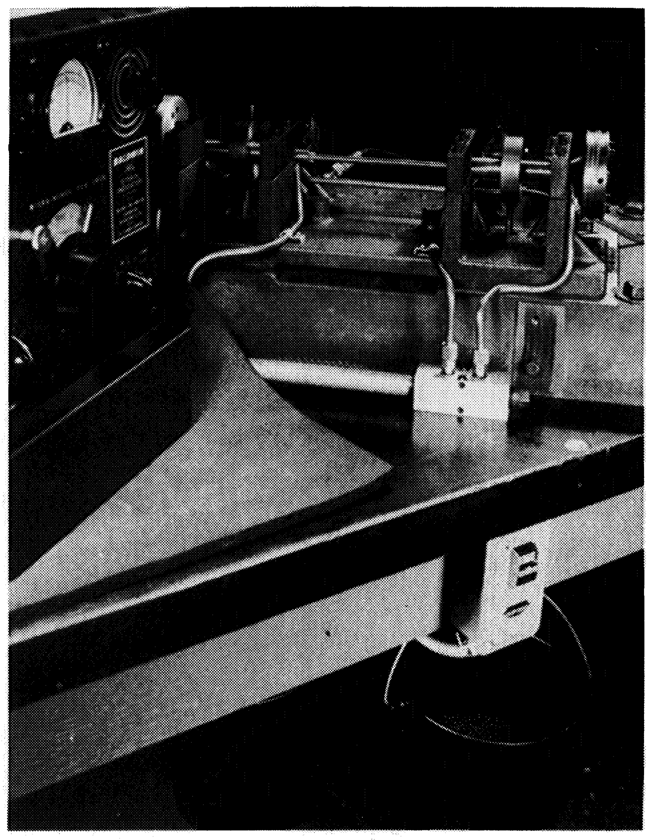

Illustration 11. Bearing Test Setup.

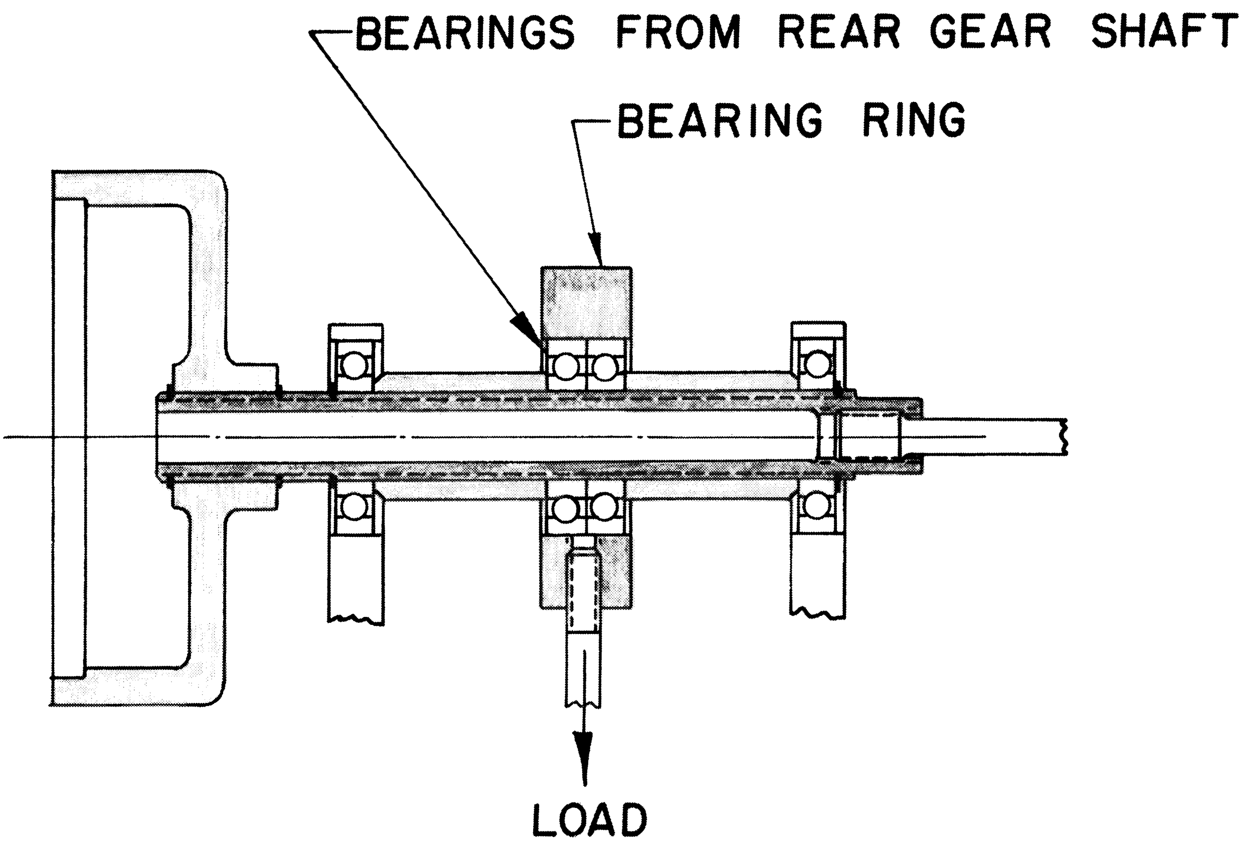

Illustration 12. Bearing Test Setup.

Figures 6 through 10 show the results of these tests at 1770 and 3540 rpm, respectively. The data from which the curves are plotted are shown in Tables VI through X.

D. Gear Tests

Illustration 13 shows a machine in operation testing the gears made of "Zytel". Preliminary tests run on the five machines showed a rather wide variation in the amount of torque required to drive the various machines under supposedly identical load and speed conditions. Investigation showed there was considerable variation in backlash between the gears in the five machines, and it was felt that this was a contributing factor.

The way in which the backlash was measured is shown in Illustrations 14 and 15. The rather elaborate setup shown was required because it was found that the teeth tended to deflect so readily that it was difficult to tell just where backlash stopped and deflection started.

In Illustrations 14 and 15, the clamp on the left-hand gear prevents that gear from rotating, while the clamp on the right-hand gear is free to rotate with that gear so far as the backlash allows. The weight of the dial indicator produces a small clockwise torque on the right-hand gear, thus turning that gear as far as the clearance between the gear teeth will allow. The above test condition is shown by Illustration 14 where counterweights are shown inactive, due to manual support. After reading the dial indicator in this position the weights are applied as shown in Illustration 15. The weights produce a counterclockwise torque on the righthand gear approximately equal to the clockwise torque originally applied. The difference between the indicator readings for the two positions, multiplied by the ratio of the levers involved, is taken as the backlash between the teeth.

So that all the gears would be operating with essentially the same conditions in regard to backlash, the center distances were adjusted to be within 2.4994 to 2.5001 inches on all pairs of gears. Illustration 16 shows how this was done.

The rear gear shafts and bearings were removed and a dummy shaft ground to the same diameter as the outside diameter of the bearings was installed in the rear bearing supports. By the use of gage blocks, the distance between the dummy shaft and the outer diameter of the front shaft bearings was carefully gaged to provide the desired center distance. A spacer block visible in the lower right corner of Illustration 16, was used with shims to establish the position of the rear bearing supports. Jack screws, visible in Illustrations 14 and 15 forced the rear bearing support casting against the shimmed spacer block while the bearing support casting was tightened to the base casting.

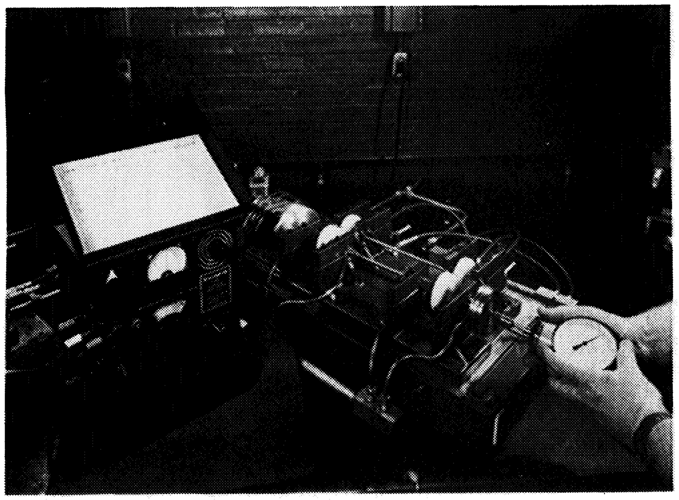

Illustration 13. Gear Test.

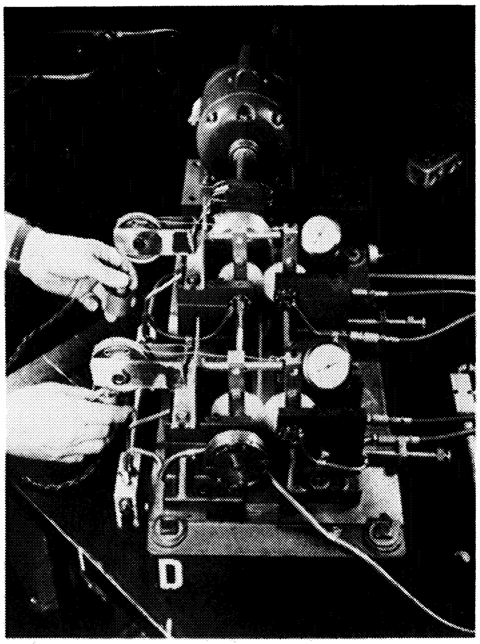

Illustration 14. Backlash Measurement Setup.

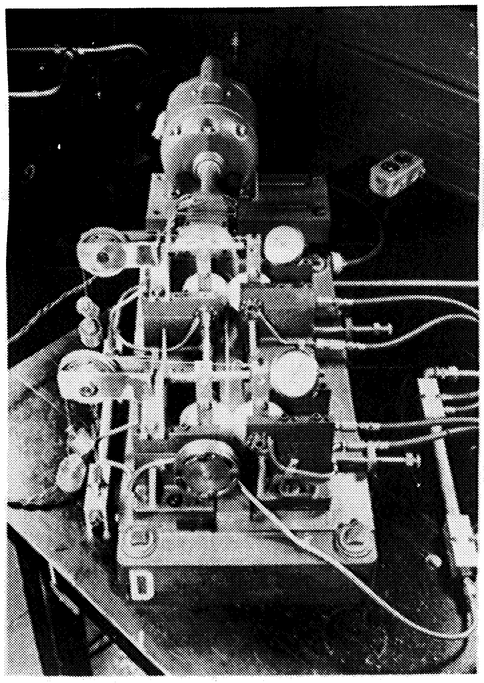

Illustration 15. Backlash Measurement Setup.

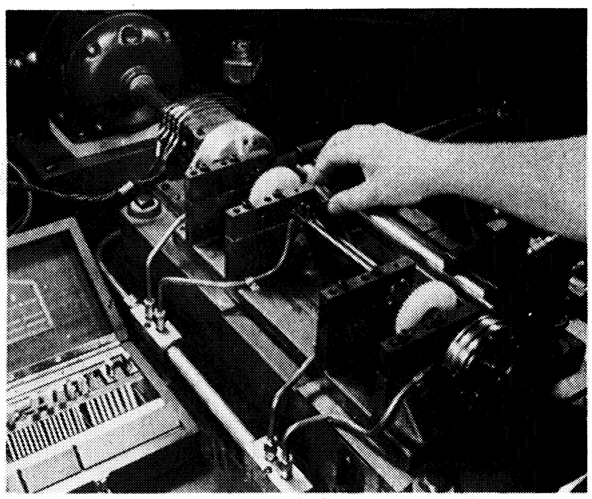

Illustration 16. Setting Center Distance.

After setting the gear center distances as described, variable gear torque tests were run on each of the five test machines at speeds of approximately 1760 and 3530 rpm. These were not intended to be life tests but were short duration tests to establish the relative performance of the five test machines. These speeds were selected because a-c motors having said characteristics were available.

The results of these tests are shown in Figs. 11 through 17. Tables XI through XVII show the data from which the curves are plotted.

The curves of Figs. 11 through 17 show drive torque, plotted vertically, against gear torque, plotted horizontally. The gear torque is that torque supplied to the system by the loading device shown in Illustration 4, and hence is the torque transmitted by each of the gears being tested.

The drive torque is measured by the flexure plate and is the torque required to overcome losses in the gear teeth and bearings.

IV. DISCUSSION OF TEST RESULTS

A. Bearing Tests

Figures 6 through 10 show that the bearing losses are reasonably consistent for the five machines, with the exception of machine E. The losses in machine E were considerably higher than in the other machines. The reason for this was not investigated but it is felt that the bearings in this machine might be a tighter fit in their supports than those of the other machines.

B. Gear Tests

The curves of Figs. 11, 12, 13, 14, and 15 showing the total loss in the gear teeth and bearings all have essentially the same slope, but the displacement from zero is not the same for all curves. This variation in the zero displacement may be due to the fact that the bearings of the two rear gear shafts are mounted differently when testing gears than when testing bearings. This may introduce some error into the test results since these bearings are slipped into the ring used for bearing testing, but are clamped into their supports when mounted in the test machine to determine the total loss. All of the total loss curves shown in these same figures are consistent in deviating upward from a straight line at high values of gear torque.

The 3530 rpm test on machine D was run twice with results as shown in Fig. 14a. Prior to the start of the first test the gears in this machine had not operated under load. Comparison of the two curves shown suggests that losses in machine-cut gears made from "Zytel" reduce somewhat after a short period of operation under load.

Five gear tests were made on machine B on different dates under different atmospheric conditions and at varying gear center distances. Results shown in Table XVI and plotted on Fig. 16 indicate that the above variables may not appreciably influence gear losses.

Figure 17 shows the composite or average results of the gear tests on all five machines. Table XVII contains the data from which the curves are plotted. The curves show the loss, expressed as driving torque, for one pair of gears, plotted against the torque transmitted by one pair of gears.

The displacement of the two curves of Fig. 17 from one another can be reasonably attributed to differences in windage loss due to the differences in speed. The upward deviation of the curves from a straight line at high values of gear torque may be due to interference brought about by tooth deflection.

As previously mentioned, the object of these tests was to establish the relative performance of the five test machines. The results show that drive torques of a very small order of magnitude can be successfully measured, indicating that the flexure plates are sensitive but at the same time consistent. The five machines seem entirely capable of producing comparable results.

V. PROPOSED FUTURE WORK

The work done to date completes the first phase of the project, namely, the procuring of the test machines and the preliminary testing necessary to establish the performance of the test machines. It is felt that a sound basis has now been established from which to proceed to future work.

The next phase should logically be directed toward obtaining the evaluating and design data for gears made of "Zytel". With this in mind, the following proposals for future work are set forth:

A. Tests will be run to establish wear rates and useful life of the gear teeth. These tests will be conducted with gears and teeth of various sizes and various tooth proportions operating with different conditions of speed and load. Information from such tests will be the basis for establishing the desired design data.

B. In conjunction with the wear and life tests will be investigations of the effects of backlash and interference, and of lubrication and absence of lubrication. With regard to lubrication it might be added that all future tests will be conducted using the same kind of lubricating oil as is used in the du Pont Company laboratories.

C. Tests will be conducted on steel gears to evaluate the losses and noise level as compared to gears made of "Zytel".

D. A photomicrographic study of the gear teeth under static load will be made to determine clearance and interference conditions of the deflected teeth. This will be helpful in determining the most desirable tooth form and proportions.

E. Methods of determining gear tooth temperatures may be investigated. Since power lost in the gear teeth must be dissipated in the form of heat and the heat in turn may affect the wearing properties of the teeth, such investigations may prove to be worthwhile.

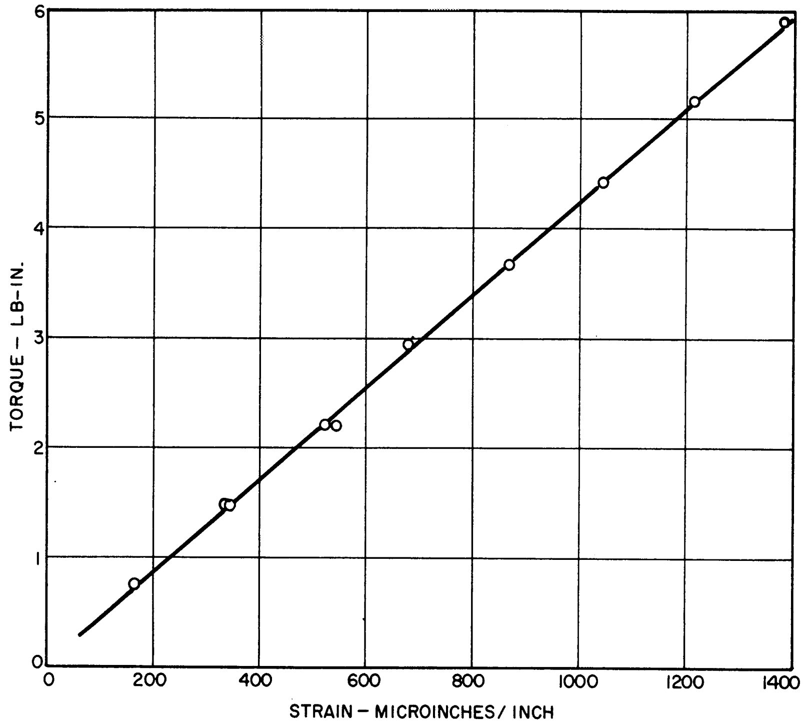

Fig. 1. Flexure-Plate Calibration, Machine A

TABLE I

FLEXURE-PLATE CALIBRATION, MACHINE A

Couple Force, oz	*Torque, lb-in.	Strain, μin./in.
8	1.47	334
4	0.73	163
8	1.47	341
12	2.20	521
16	2.94	680
20	3.67	868
24	4.41	1042
28	5.14	1212
32	5.88	1380
12	2.20	548

*Couple Distance = 2.9375"

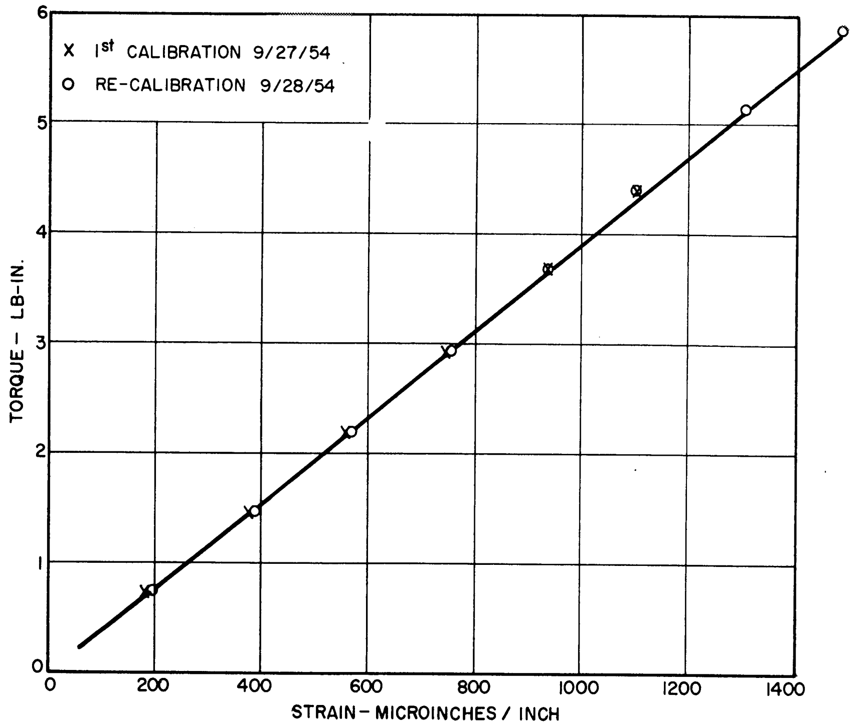

Fig. 2. Flexure-Plate Calibration, Machine B

TABLE II

FLEXURE-PLATE CALIBRATION, MACHINE B

Couple Force, oz	*Torque, lb-in.	Strain, µin./in.
4	0.73	179
8	1.47	373
12	2.20	555
16	2.94	740
20	3.67	936
24	4.41	1094
4	0.73	195
8	1.47	386
12	2.20	570
16	2.94	753
20	3.67	933
24	4.41	1098
28	5.14	1302
32	5.88	1480
16	2.94	753

*Couple Distance = 2.9375"

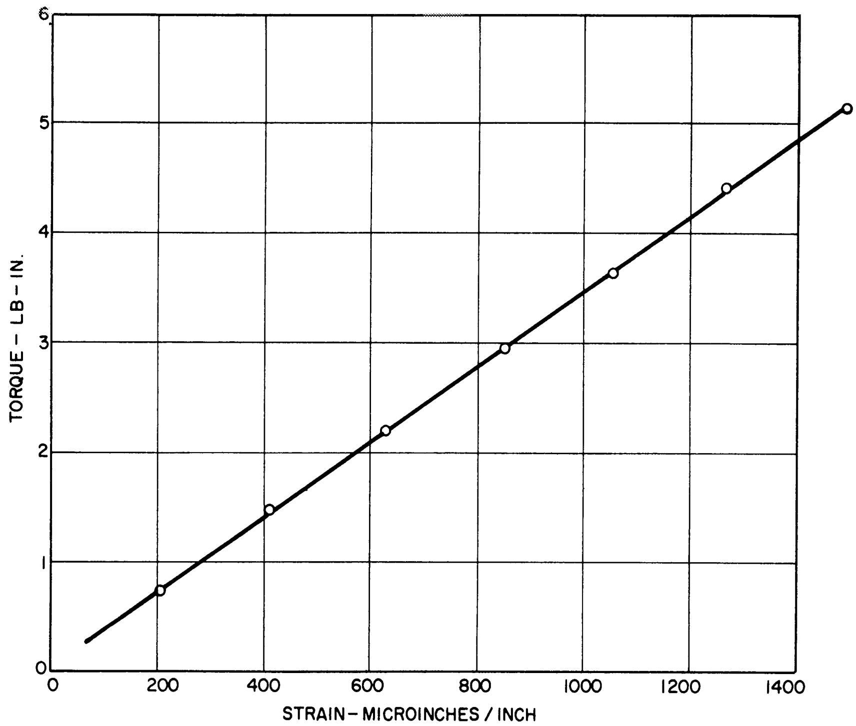

Fig. 3. Flexure-Plate Calibration, Machine C

TABLE III

FLEXURE-PLATE CALIBRATION, MACHINE C

Couple Force, oz	*Torque, lb-in.	Strain, μin./in.
4	0.73	203
8	1.47	410
12	2.20	629
16	2.94	848
20	3.67	1055
24	4.41	1267
28	5.14	1489
32	5.88	1688

*Couple Distance = 2.9375"

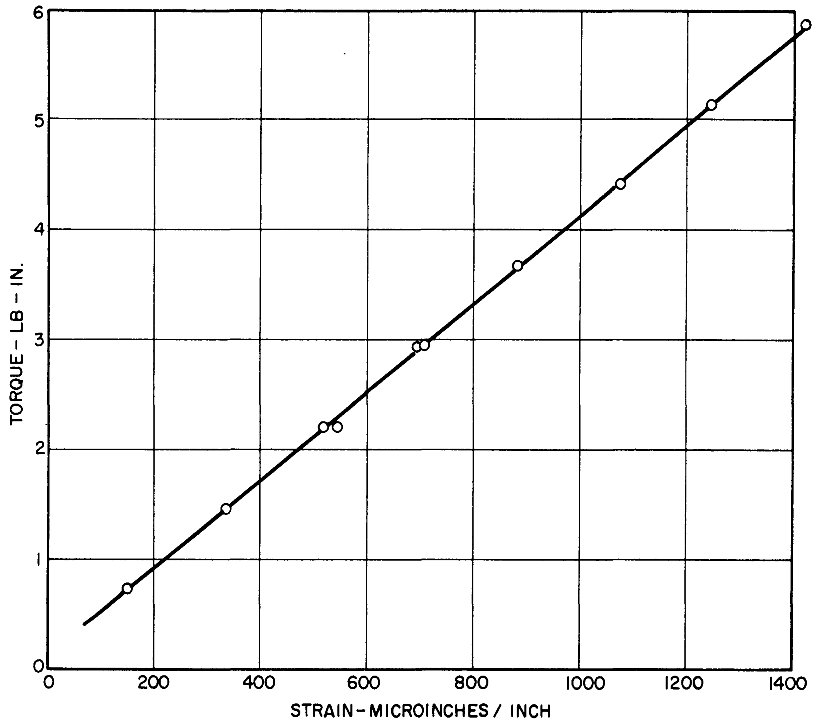

Fig. 4. Flexure-Plate Calibration, Machine D

TABLE IV

FLEXURE-PLATE CALIBRATION, MACHINE D

Couple Force, oz	*Torque, lb-in.	Strain, μin./in.
4	0.73	154
8	1.47	337
12	2.20	544
16	2.94	695
20	3.67	882
24	4.41	1076
28	5.14	1242
32	5.88	1421
12	2.20	518
16	2.94	709
16	2.94	706

*Couple Distance = 2.9375"

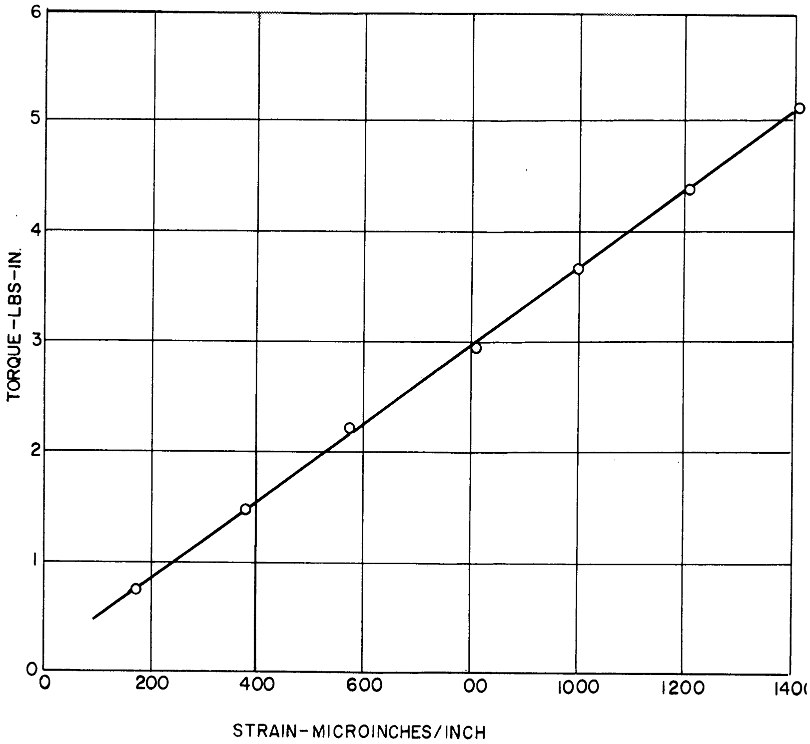

Fig. 5. Flexure-Plate Calibration, Machine E

TABLE V

FLEXURE-PLATE CALIBRATION, MACHINE E

Couple Force, oz	*Torque, lb-in.	Strain, μin./in.
4	0.73	172
8	1.47	377
12	2.20	570
16	2.94	804
20	3.67	994
24	4.41	1203
28	5.14	1409
32	5.88	1612

*Couple Distance = 2.9375"

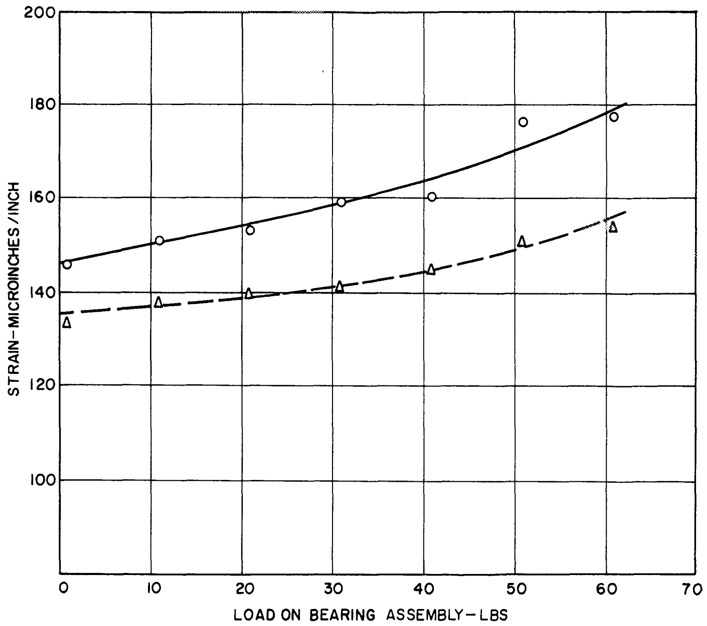

Fig. 6. Bearing Test, Machine A

TABLE VI

BEARING TEST, MACHINE A

Speed, rpm	Load, lb/brg assb	Strain, μin./in.
1770	0.8	134
1770	10.8	138
1770	20.8	140
1767	30.8	141
1765	40.8	145
1765	50.8	151
1769	60.8	154
3530	60.8	177
3522	50.8	176
3532	40.8	160
3545	30.8	159
3548	20.8	153
3550	10.8	151
3550	0.8	146

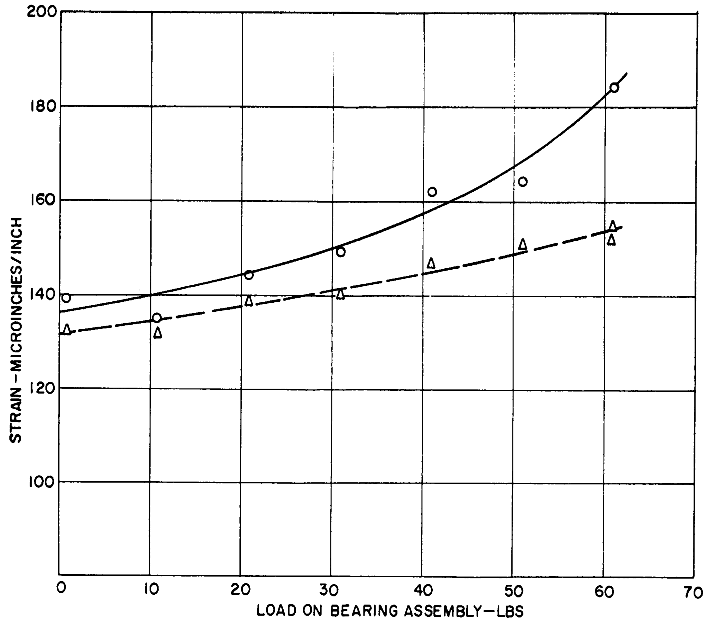

Fig. 7. Bearing Test, Machine B

TABLE VII

BEARING TEST, MACHINE B

Speed, rpm	Load lb/brg assb	Strain, μin./in.
1795	0.8	133
1795	10.8	132
1790	20.8	139
1790	30.8	140
1778	40.8	147
1774	50.8	151
1774	60.8	153
3588	60.8	184
3588	50.8	164
3581	40.8	162
3570	30.8	149
3570	20.8	144
3570	10.8	135
3570	0.8	139

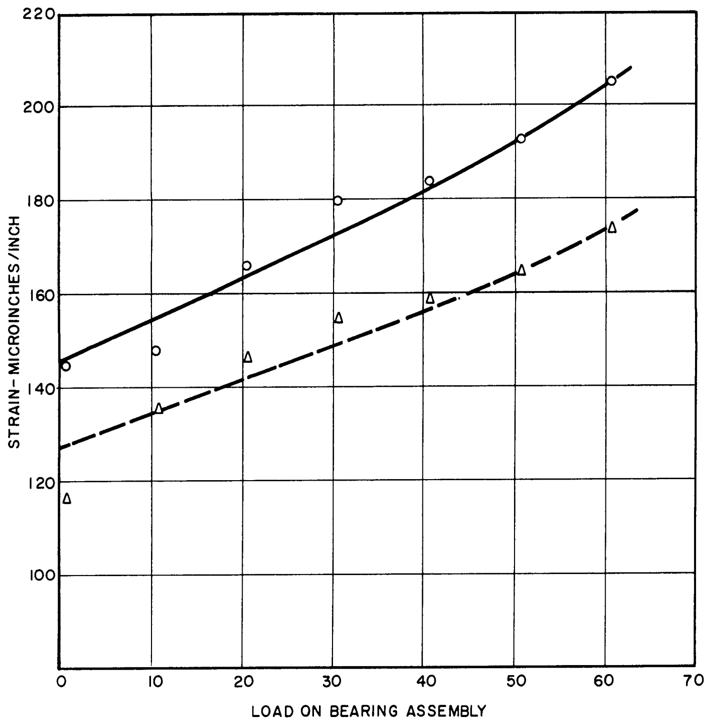

Fig. 8. Bearing Test, Machine C

TABLE VIII

BEARING TEST, MACHINE C

Speed, rpm	Load, lb/brg assb	Strain, μin./in.
1778	0.8	117
1778	10.8	136
1775	20.8	147
1775	30.8	155
1773	40.8	159
1772	50.8	165
1772	60.8	174
3550	60.8	205
3550	50.8	193
3545	40.8	184
3550	30.8	180
3550	20.8	166
3550	10.8	148
3570	0.8	145

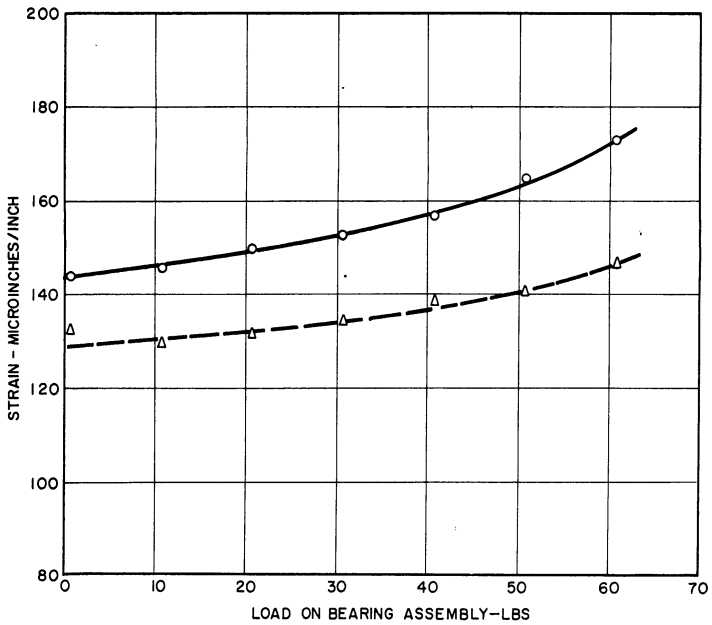

Fig. 9. Bearing Test, Machine D

TABLE IX

BEARING TEST, MACHINE D

Speed, rpm	Load lb/brg assb	Strain, μin./in.
1775	0.8	133
1775	10.8	130
1775	20.8	132
1775	30.8	135
1775	40.8	138
1780	50.8	141
1775	60.8	147
3550	60.8	173
3540	50.8	165
3550	40.8	157
3545	30.8	153
3550	20.8	150
3550	10.8	146
3550	0.8	144

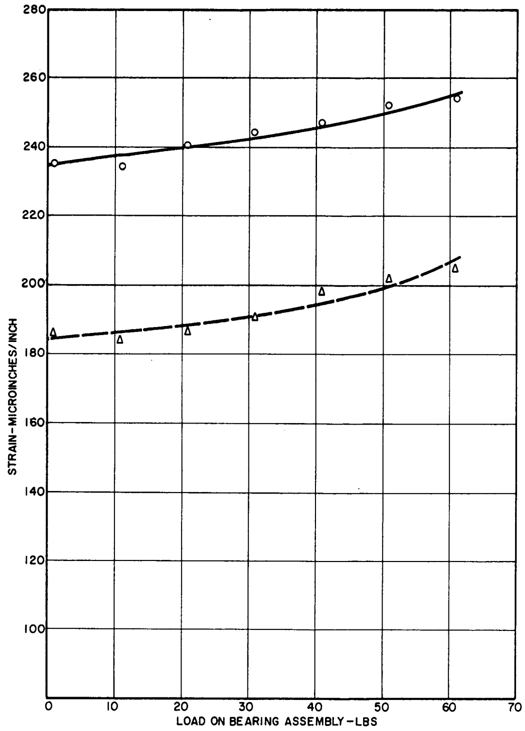

Fig. 10. Bearing Test, Machine E

TABLE X

BEARING TEST, MACHINE E

Speed, rpm	Load lb/ brg assb	Strain, µin./in.
1775	0.8	186
1775	10.8	184
1775	20.8	187
1775	30.8	191
1775	40.8	198
1775	50.8	202
1775	60.8	205
3550	60.8	254
3550	50.8	252
3550	40.8	247
3550	30.8	244
3550	20.8	240
3550	10.8	234
3550	0.8	235

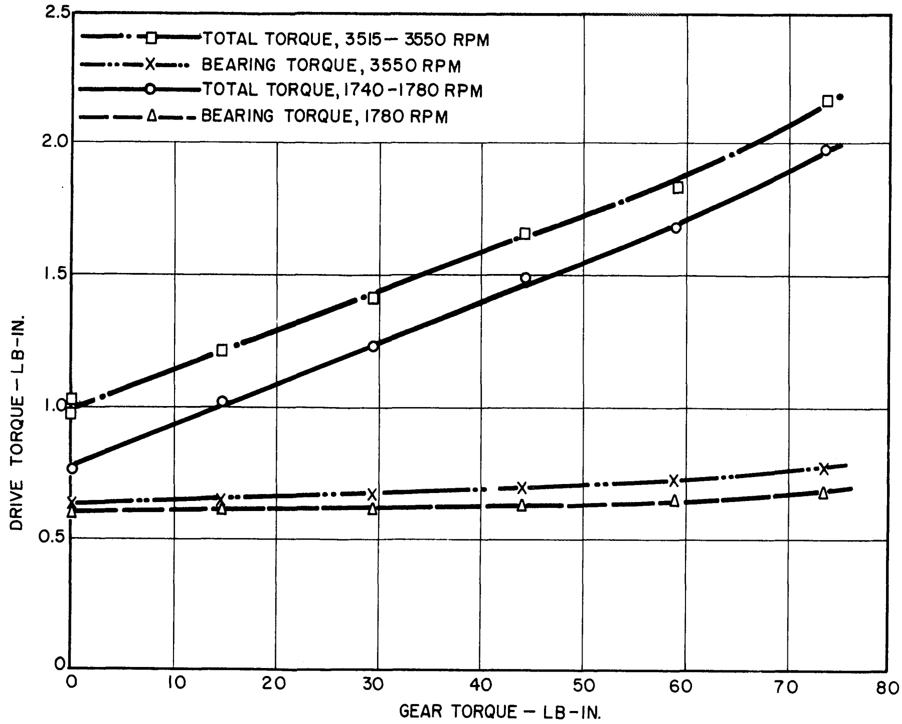

Fig. 11. Gear Test, Machine A

TABLE XI

GEAR TEST, MACHINE A

Gears: Nos. 5 and 6, 7 and 8
Shaft C.D.: 2.4995" to 2.4996"
Backlash: 5 and 6, 0.0025" to 0.0053"
7 and 8, 0.0038" to 0.0060"

Speed, rpm	Gear Torque, lb-in.	Load, lb	Bearing Drive [1]Strain, μin./in.	Bearing Drive [2]Torque, lb-in.	Total Drive Strain, μin./in.	Total Drive [3]Torque, lb-in.	Gear Drive [4]Torque, lb-in.
1780	0	0	135	0.60	176	0.76	0.16
1775	14.7	12.45	137	0.61	241	1.02	0.41
1770	29.4	24.9	140	0.62	292	1.23	0.61
1765	44.1	37.4	143	0.63	352	1.49	0.86
1755	58.8	49.9	149	0.65	401	1.68	1.03
1740	73.4	62.4	157	0.68	471	1.98	1.30
3550	0	0	146	0.63	243	1.02	0.39
3530	14.7	12.45	151	0.65	285	1.21	0.56
3530	29.4	24.9	156	0.67	337	1.41	0.74
3525	44.1	37.4	162	0.70	394	1.66	0.96
3525	58.8	49.9	170	0.73	436	1.84	1.11
3515	73.4	62.4	180	0.77	514	2.17	1.40
3550	0	0	146	0.63	224	0.97	0.34

[1] Strain at gear load from Fig. 6
[2,3] Torque at given strain from Fig. 1
[4] 3 minus 2

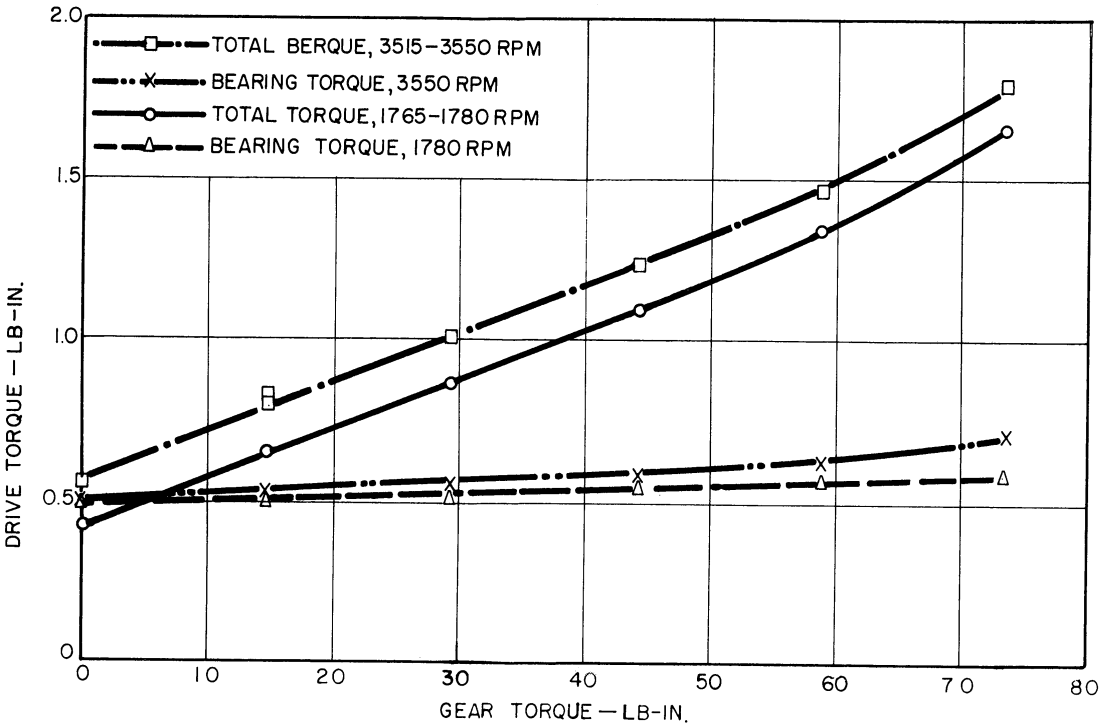

Fig. 12. Gear Test, Machine B

TABLE XII

GEAR TEST, MACHINE B

Gears: Nos. 9 and 10, 11 and 12
Shaft C.D : 2.5001" to 2.4994"
Backlash: 9 and 10, 0.0013" to 0.0053"
11 and 12, 0.0032" to 0.0040"

Speed, rpm	Gear Torque, lb-in.	Load, lb	Bearing Drive [1]Strain, μin./in.	Bearing Drive [2]Torque, lb-in.	Total Drive Strain, μin./in.	Total Drive [3]Torque, lb-in.	Gear Drive [4]Torque, lb-in.
1780	0	0	131	0.50	112	0.42	-0.08
1780	14.7	12.45	135	0.51	169	0.65	0.14
1775	29.4	24.9	139	0.53	228	0.87	0.34
1775	44.1	37.4	143	0.55	284	1.10	0.55
1765	58.8	49.9	149	0.57	350	1.35	0.78
1765	73.4	62.4	155	0.59	432	1.66	1.07
3550	0	0	136	0.51	147	0.56	0.05
3530	14.7	12.45	141	0.53	216	0.82	0.29
3525	29.4	24.9	147	0.56	265	1.01	0.45
3520	44.1	37.4	155	0.59	321	1.24	0.65
3520	58.8	49.9	167	0.63	385	1.47	0.84
3515	73.4	62.4	188	0.71	470	1.80	1.09
3550	0	0	136	0.51	131	0.50	-0.01

1 Strain at gear load from Fig. 7
2,3 Torque at given strain from Fig. 2
4 3 minus 2

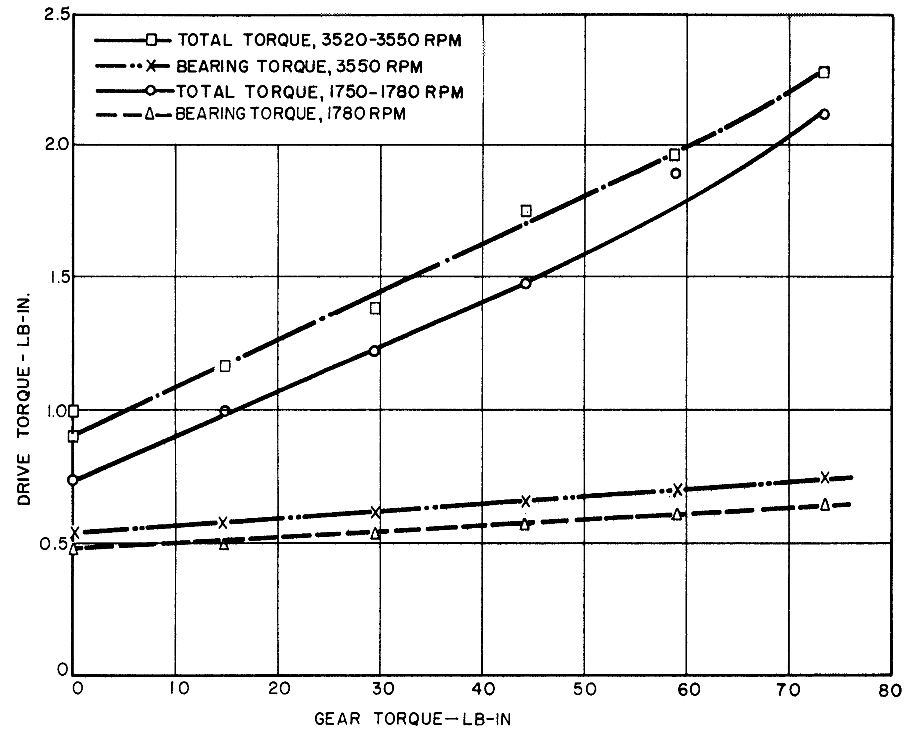

Fig. 13. Gear Test, Machine C

TABLE XIII

GEAR TEST, MACHINE C

Gears: Nos. 1 and 2, 3 and 4
Shaft C.D.: 2.4996" to 2.4998"
Backlash: 1 and 2, 0.0040" to 0.0056"
3 and 4, 0 0021" to 0.0040"

Speed, rpm	Gear Torque, lb-in.	Load, lb	Bearing Drive [1]Strain, μin./in.	Bearing Drive [2]Torque lb-in.	Total Drive Strain, μin./in.	Total Drive [3]Torque, lb-in.	Gear Drive [4]Torque, lb-in.
1780	0	0	127	0.48	203	0.73	0.25
1775	14.7	12.45	131	0.50	278	0.99	0.49
1770	29.4	24.9	145	0.54	345	1.21	0.67
1765	44.1	37.4	154	0.57	421	1.47	0.90
1760	58.8	49.9	164	0.61	541	1.89	1.28
1750	73.4	62.4	176	0.65	608	2.11	1.46
1780	0	0	127	0.48	197	0.71	0.23
3550	0	0	146	0.54	279	0.99	0.45
3550	14.7	12.45	157	0.58	330	1.16	0.58
3540	29.4	24.9	168	0.61	394	1.38	0.77
3535	44.1	37.4	179	0.65	451	1.75	1.10
3525	58.8	49.9	192	0.70	561	1.96	1.26
3520	73.4	62.4	207	0.75	652	2.27	1.52
3550	0	0	146	0.54	250	0.90	0.36

1 Strain at gear load from Fig. 8
2,3 Torque at given strain from Fig. 3
4 3 minus 2

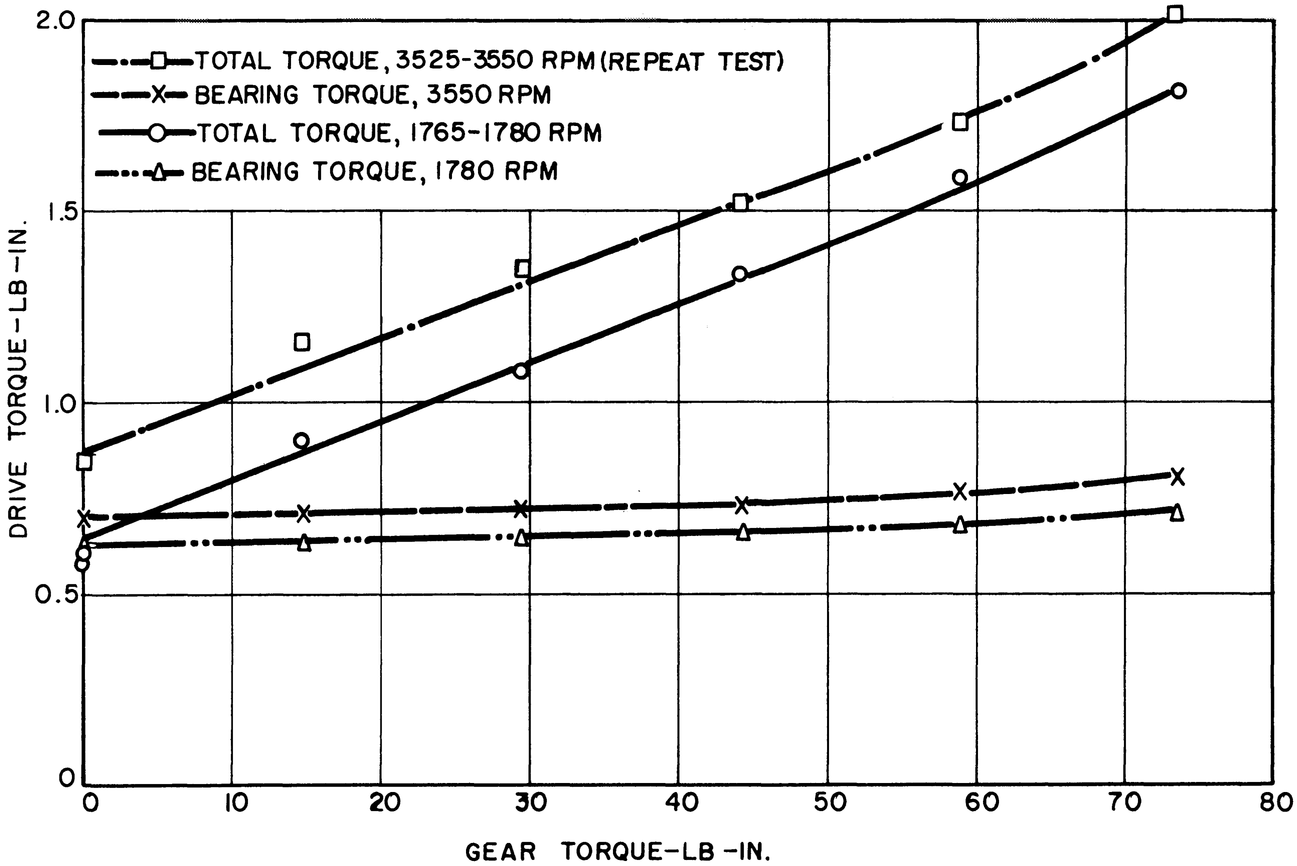

Fig. 14. Gear Test, Machine D

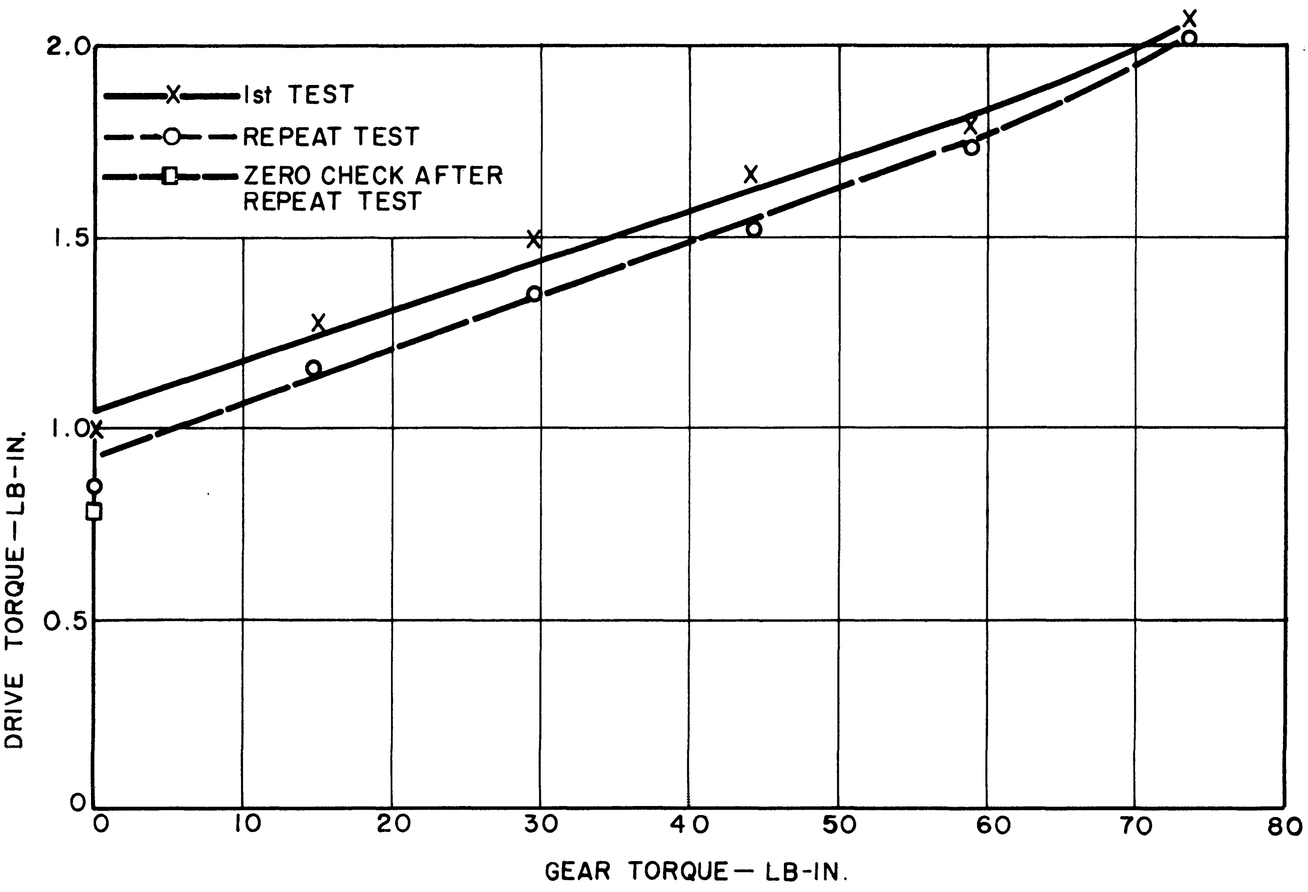

Fig. 14a. Gear Test, Machine D

TABLE XIV

GEAR TEST, MACHINE D

Gears: Nos. 13 and 14, 15 and 16
Shaft C.D.: 2.4995" to 2.500"
Backlash: 13 and 14, 0.0015" to 0.0044"
15 and 16, 0.0045" to 0.0056"

Speed, rpm	Gear Torque, lb-in.	Load, lb	Bearing Drive [1]Strain, μin./in.	Bearing Drive [2]Torque, lb-in.	Total Drive Strain, μin./in.	Total Drive [3]Torque, lb-in.	Gear Drive [4]Torque lb-in.
3550	0	0	144	0.70	220	1.00	0.30
3550	14.7	12.45	147	0.71	292	1.27	0.56
3540	29.4	24.9	151	0.72	346	1.50	0.78
3535	44.1	37.4	156	0.73	388	1.66	0.93
3520	58.8	49.9	163	0.77	419	1.79	1.02
3520	73.4	62.4	175	0.81	492	2.07	1.26
Gear test repeated for 3520 to 3550 rpm							
3550	0	0	144	0.70	184	0.85	0.15
3540	14.7	12.45	147	0.71	260	1.16	0.45
3540	29.4	24.9	151	0.72	308	1.35	0.63
3535	44.1	37.4	156	0.73	351	1.52	0.79
3525	58.8	49.9	163	0.77	405	1.73	0.96
3525	73.4	62.4	175	0.81	479	2.02	1.21
3550	0	0	144	0.70	167	0.78	0.08
1780	0	0	129	0.63	116	0.58	-0.05
1775	14.7	12.45	131	0.64	196	0.90	0.26
1775	29.4	24.9	133	0.65	243	1.09	0.44
1775	44.1	37.4	136	0.66	305	1.33	0.67
1770	58.8	49.9	141	0.68	371	1.59	0.91
1765	73.4	62.4	148	0.71	425	1.81	1.10
1780	0	0	129	0.63	120	0.60	-0.03

[1] Strain at gear load from Fig. 9
[2,3] Torque at given strain from Fig. 4
[4] [3] minus [2]

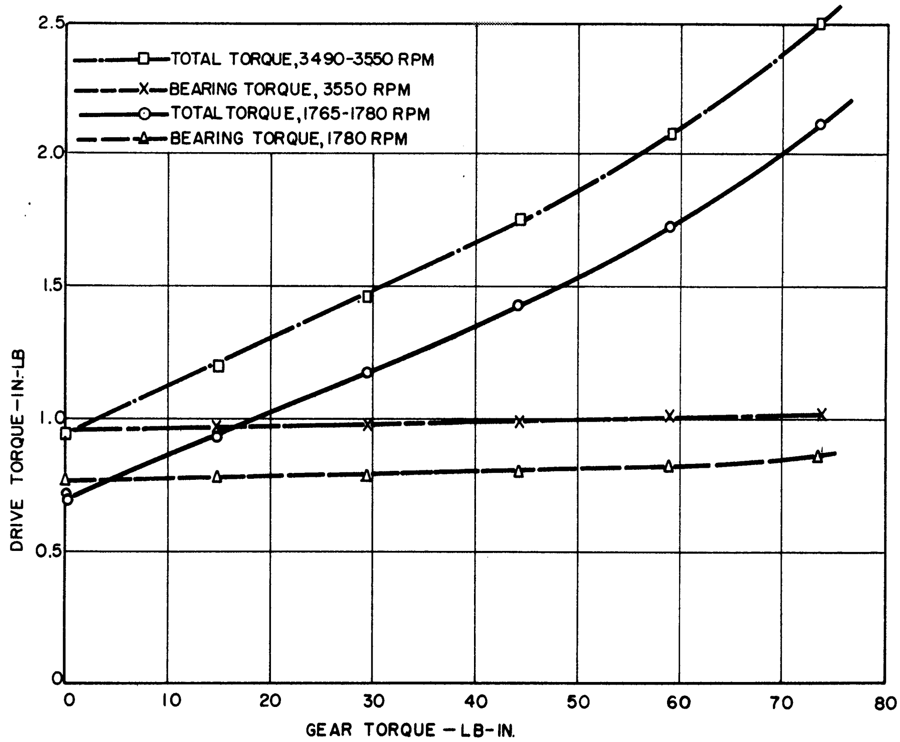

Fig. 15. Gear Test, Machine E

TABLE XV

GEAR TEST, MACHINE E

Gears: Nos. 17 and 18, 19 and 20
Shaft C.D.: 2.5001" to 2.5000"
Backlash: 17 and 18, 0.0010" to 0.0037"
19 and 20, 0.0031" to 0.0042"

Speed, rpm	Gear Torque, lb-in.	Load, lb	Bearing Drive [1]Strain, µin./in.	Bearing Drive [2]Torque, lb-in.	Total Drive Strain, µin./in.	Total Drive [3]Torque, lb-in.	Gear Drive [4]Torque lb-in.
3550	0	0	235	0.96	230	0.95	-0.01
3545	14.7	12.45	238	0.97	304	1.20	0.23
3530	29.4	24.9	241	0.98	379	1.46	0.48
*3490	44.1	37.4	245	0.99	460	1.75	0.76
3535	58.8	49.9	250	1.01	555	2.08	1.07
3525	73.4	62.4	252	1.02	671	2.50	1.48
3550	0	0	235	0.96	233	0.96	0.00
1780	0	0	184	0.77	158	0.69	-0.08
1780	14.7	12.45	186	0.78	228	0.93	0.15
1775	29.4	24.9	189	0.79	301	1.18	0.39
1775	44.1	37.4	193	0.81	371	1.43	0.62
1765	58.7	49.9	199	0.83	454	1.73	0.90
1750	73.4	62.4	209	0.87	563	2.11	1.23
1780	0	0	184	0.77	172	0.72	-0.05

* Hose coupling slipping, wire clamp added
1 Strain at gear load from Fig. 10
2,3 Torque at given strain from Fig. 5
4 3 minus 2

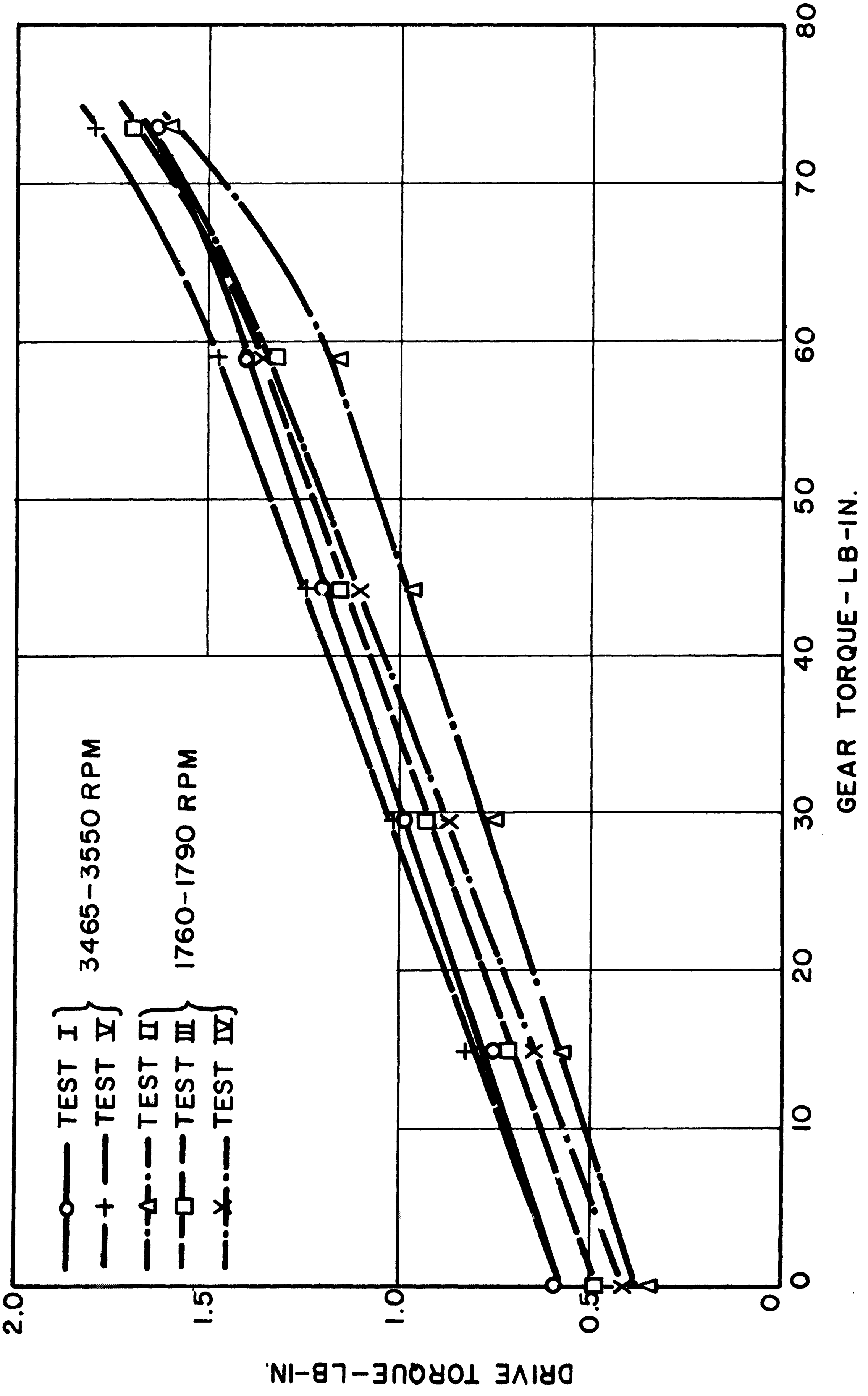

Fig. 16. Five Gear Tests, Machine B

TABLE XVI

FIVE GEAR TESTS, MACHINE B

Gears: Nos. 9 and 10, 11 and 12

No.	Date	Gear Speed, rpm	Gear Torque, lb-in.	Total Drive Strain, µin./in.	Total Drive [1]Torque, lb-in.	Shaft C.D., in.	Air Conditions D.B., °F	Air Conditions W.B., °F	Air Conditions R.H., %
		3545	0	154	0.59		81.5	68.0	53
		3530	14.7	195	0.75	2.499			
I	9/28/54	3525	29.4	255	0.98	to	82.5	69.0	55
		3510	44.1	311	1.20	2.494			
		3500	58.8	366	1.40				
		3465	73.4	424	1.63		82.5	69.0	55
		1790	0	92	0.35				
		1788	14.7	149	0.57				
		1786	29.4	194	0.75	2.499			
II	9/28/54	1786	44.1	251	0.96	to			
		1778	58.8	303	1.16	2.494			
		1772	73.4	413	1.60		83.0	69.2	55
		1788	0	109	0.41				
		1775	0	128	0.49		74.5	57	32
		1770	14.7	190	0.72	2.499			
*III	10/7/54	1770	29.4	242	0.93	to	76.0	56.0	26
		1770	44.1	303	1.16	2.494			
		1765	58.8	343	1.32				
		1760	73.4	442	1.70		77.0	56.5	26
		1780	0	112	0.42				
		1780	14.7	169	0.65	2.5001			
IV	10/21/54	1775	29.4	228	0.87	to			
		1775	44.1	284	1.10	2.4994	74.0	56.7	32
		1765	58.8	350	1.35				
		1765	73.4	432	1.66				
		3550	0	147	0.56		69.7	52.2	29
		3530	14.7	216	0.82				
		3525	29.4	265	1.01	2.5001	69.8	52.3	29
V	11/2/54	3520	44.1	321	1.24	to			
		3520	58.8	385	1.47	2.4994			
		3515	73.4	470	1.80		69.8	52.3	29
		3550	0	131	0.50				

[1]Torque at given strain from Fig. 2
*Gears picked up dirt particles during test

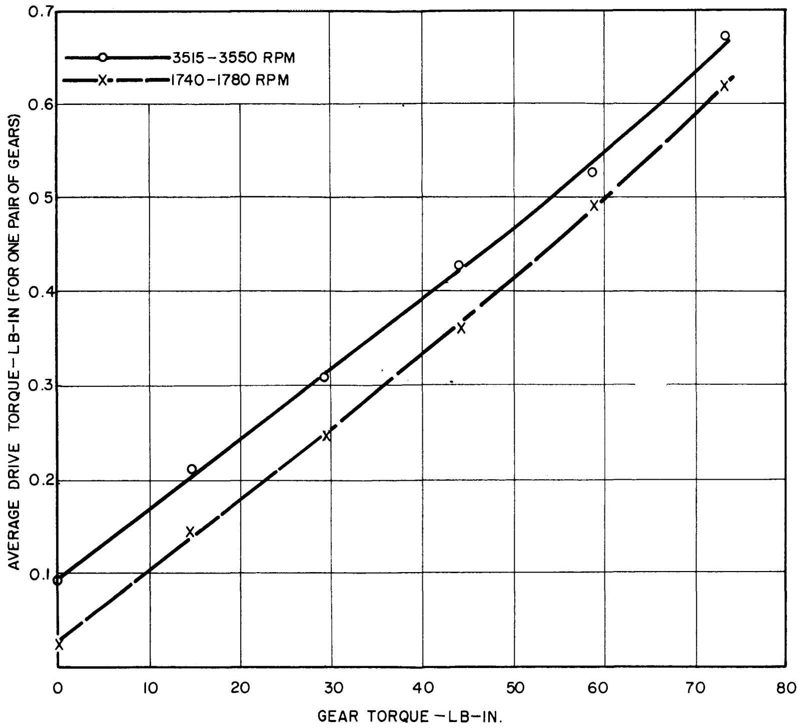

Fig. 17. Gear Losses, Average

TABLE XVII

COMPOSITE GEAR TEST DATA, MACHINES A, B, C, D, AND E

Gear Speed, rpm	Gear Torque, lb-in.	Gear, hp	*Drive Torque for One Pair of Gears A lb-in.	B lb-in.	C lb-in.	D lb-in.	E lb-in.	Average, lb-in.	Drive, hp
1780	0	0	0.080	-0.040	0.120	-0.020	-0.032	0.022	0.0006
1775-1780	14.7	0.414	0.205	0.070	0.245	0.130	0.075	0.145	0.0041
1770-1775	29.4	0.827	0.305	0.170	0.335	0.220	0.195	0.245	0.0069
1765-1775	44.1	1.239	0.430	0.275	0.450	0.335	0.310	0.360	0.0101
1755-1770	58.8	1.645	0.515	0.390	0.640	0.455	0.450	0.490	0.0137
1740-1765	73.4	2.040	0.650	0.535	0.730	0.550	0.615	0.616	0.0171
3550	0	0	0.182	0.010	0.202	0.057	-0.002	0.090	0.0051
3530-3550	14.7	0.825	0.280	0.145	0.290	0.225	0.115	0.211	0.0118
3525-3540	29.4	1.647	0.370	0.225	0.385	0.315	0.240	0.307	0.0172
3490-3535	44.1	2.460	0.480	0.325	0.550	0.395	0.380	0.426	0.0237
3520-3525	58.8	3.282	0.555	0.420	0.630	0.480	0.535	0.524	0.0293
3515-3525	73.4	4.100	0.700	0.545	0.760	0.605	0.740	0.670	0.0374

*Gear drive torque, Tables XI to XV, divided by two.