THE UNIVERSITY OF MICHIGAN
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LOAD-CARRYING CAPACITY OF SPUR GEAR TEETH HOB CUT FROM MOLDED DU PONT "ZYTEL" 101

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INTRODUCTION

Nonmetallic gears have been made and used for a long time. Various materials have been used successfully, and the particular qualities of these materials have been utilized in many installations. These qualities often include quietness of operation, ability to absorb shock loads, resistance to corrosion, and ability to operate with various fluids used as lubricants. When such gears can be molded to their final form, ease of manufacture becomes a very important quality of the nonmetallic gear.

In recent years many new plastic molding materials have been developed, some of them having characteristics which make them suitable for gears and similar mechanical parts. One such material is "Zytel" 101, a nylon resin manufactured by the du Pont Company.

At the present time a project is being carried on by the Engineering Research Institute of the University of Michigan, at Ann Arbor, to obtain design data on molded "Zytel" 101 as a gear material. This work is sponsored by the du Pont Company and is still in process. However, it is felt that some of the results obtained to date are conclusive enough to warrant publication at this time. A brief description of "Zytel" 101, and of the test work carried on so far, might also prove of interest.

MATERIAL

As previously mentioned, "Zytel" is a nylon resin. It is thermoplastic, and is supplied in the form of a granulated powder from which various products can be molded by conventional molding techniques. Molded "Zytel" is a relatively light-weight, rigid, but not brittle, material. "Zytel" is made in several varieties, of which "Zytel" 101 is the most heat resistant and best suited for mechanical parts. In the molded form it is easily machined.

Typical physical properties of this material are shown in Fig. 1. These physical properties are considerably affected by temperature. For example, Fig. 2 shows how the tensile strength decreases with higher temperature, while the impact strength increases.

The physical properties are further affected by moisture content of the molded material. Moisture content also affects the dimensional stability of the finished piece. When the material is conditioned after molding to have a moisture content of about 2.5% by weight, however, the material has good dimensional stability. All the physical properties shown in Fig. 1 are for this 2.5% moisture-conditioned, molded "Zytel" 101.

TEST MACHINES

To investigate the load-carrying capacity of molded "Zytel" 101 gears five identical test machines were designed and built. These machines operate c the "back-to-back" or "4-square" principle, with two sets of gears in each machine, loaded against one another. In this way the driving motor supplies only the power required to overcome friction in the test machines.

A schematic layout of the test machine is shown in Fig. 3. The gears to be tested are mounted on hollow shafts supported in ball bearings. Two stee torsion bars connect the hollow shafts. A friction coupling at the right end is used to twist the torsion bars, thus providing the desired load on the gear teeth. Figure 4 shows one of the test machines, with the friction coupling in the foreground and the driving motor behind. The bearing supports for the right-hand shaft of Fig. 4 are movable, thus allowing various sizes of gears to be tested. The gears shown in the machine have 2-1/2-in. pitch diameters, which might aid in visualizing the actual size of the test machine.

Load is applied to the gear teeth by twisting the friction coupling with a system of cables and weights operating on the pulleys shown in the foreground of Fig. 4. After clamping the friction coupling with the screws provided, the cables are removed to allow the coupling to rotate. Lead shot in buckets are used as weights; thus, any desired twisting moment can be applied to the coupling by changing the weight of shot in the buckets.

An overall view of the test-machine installation is shown in Fig. 5. Each machine is individually driven by a constant-speed motor through a speed variator, providing a 600 to 5000-rpm speed range. An individual oil-mist lubricating system is provided on each test machine to lubricate gears and bearings.

To allow measurement of the torque required to drive the test machine, a flexure-plate drive is incorporated. Electric strain gages on the flexure plate measure the strain on the plate, thus indicating the driving torque. As previously mentioned, this is the torque required to overcome friction and windage in the machine and is not the torque transmitted by the test gears.

TEST GEARS

Although it is realized that in the majority of cases gears made of "Zytel" 101 will have teeth formed by the mold, at the time of this writing all test work has been done with three sets of gears having hob-cut teeth rather than molded teeth. Cut teeth were used to allow greater flexibility in the manufacture of test gears at the start of the evaluation program. However, future testing will be concentrated on gears having molded teeth.

Each set of gears tested consisted of the 20 gears required to fill the five test machines, plus a few extra gears to be used as replacements when tooth failure occurred. The first and second sets of gears were identical, having 50 teeth, 20 pitch, 2-1/2-in. pitch diameter, 20-degree pressure angle, and 7/16-in. face width. The third set had 40 teeth, 16 pitch, 2-1/2-in. pitch diameter, 20-degree pressure angle, and 7/16-in. face width. All gears had full-depth teeth.

The test gears were machined from molded "Zytel" blanks with commercial class-4 hobs. Maximum total composite error was .0013 in. for all the gears taken together, while maximum tooth-to-tooth composite error was .0006 in.

It will be noticed that the test gear teeth were made with reasonable accuracy and have conventional profiles, pressure angles, and proportions. Since these have proven successful with metal gears, it seemed logical to start the evaluation with teeth made in this way, leaving such things as allowable error, optimum pressure angle, etc., for later investigation.

TEST WORK

At the start of this program it was felt that it might be possible to establish a correlation between the torque required to overcome friction in the gear teeth and the useful life of those teeth. If such a correlation could be soundly established, it would then be necessary only to measure the friction torque of the gear teeth to predict useful life, and much time-consuming testing could be eliminated. This is why the flexure-plate drive, mentioned earlier, was built into each machine.

As yet no such correlation has been established, but some interesting data have turned up along the way. Because the torque measured by the flexure-plate drive when the machine is in operation includes windage and bearing-friction torque, tests were first run to measure the bearing-friction torque only. This was done by assembling the test machines without gears and with the bearings from the two rear gear shafts mounted on the front shafts in suitable bearing rings. Figure 6 shows how this was accomplished. By applying a load to the bearings and measuring the torque required to drive the test machine, the friction torque of the bearings was established for various loads and speeds.

Having completed the bearing-friction tests, endurance testing of the gears was started. The general procedure was first to measure and record the composite errors of each gear with a Kodak Conju-Gage and to chart the profiles of three marked teeth on each gear with a Fellows Involute Checker. Periodically during the endurance test the gears would be removed from the machine and the inspections repeated. At each inspection the same three marked teeth on each gear were charted with the Fellows Involute Checker. In this way the rate of tooth wear, as indicated by composite-error changes and by profile changes, was determined.

Flexure-plate strain-gage readings were taken periodically during the test, and the torque thus recorded was used with the bearing-friction data to determine the friction torque in the gear teeth only.

Each test gear was numbered and occupied the same position in the test machine throughout a complete endurance test. The machines did not operate continuously during a period between tooth inspections, however. These periods were from 5 to 20 million cycles in duration, while the machines were operated only 8 to 16 hours per day. At the close of each day's operation, the load was removed from the gears to prevent possible permanent deformation of the heavily loaded teeth.

Since all the gears being used on any one test had the same number of teeth, it follows that a given tooth on any one gear would always mesh with exactly the same tooth on the mating gear during the entire period between inspections. Since this might not present a representative picture of tooth wear, two gears were removed from each machine at the start of the daily operation, and were reinstalled so as to mesh with other teeth on the same mating gear. In this way any one tooth would mesh with several mating gear teeth during any period of operation between inspections.

All five test machines operated simultaneously during the endurance tests, with various torque loads and sometimes various speeds on the different machines. The following endurance tests were completed:

First Endurance Test

Gears: 20 pitch, 50 teeth, 2-1/2-in. pitch diameter, 20-degree pressure angle, 7/16-in. face width, hob-cut teeth.

Torque and Speed: 58.8 lb-in. to 102.8 lb-in. at 1760 rpm and 58.8 lb-in. at 3450 rpm.

Duration: 20 million cycles at 1760 rpm and 40 million cycles at 3450 rpm.

Lubrication: oil-mist lubrication of teeth.

Second Endurance Test

Gears: 20 pitch, 50 teeth, 2-1/2-in. pitch diameter, 20-degree pressure angle, 7/16-in. face width, hob-cut teeth.

Torque and Speed: 58.8 lb-in. to 117.5 lb-in. at 2500 rpm.

Duration: 50 million cycles.

Lubrication: oil-mist lubrication of teeth.

Third Endurance Test

Gears: 16 pitch, 40 teeth, 2-1/2-in. pitch diameter, 20-degree pressure angle, 7/16-in. face width, hob-cut teeth.

Torque and Speed: 58.8 lb-in. to 117.5 lb-in. at 2500 rpm.

Lubrication: oil-mist lubrication of teeth.

Fourth Endurance Test

Gears: 16 pitch, 40 teeth, 2-1/2-in. pitch diameter, 20-degree pressure angle, 7/16-in. face width, hob-cut teeth.

Torque and Speed: 29.4 lb-in. to 88.1 lb-in. at 2500 rpm.

Duration: 50 million cycles.

Lubrication: gear teeth not lubricated during test.

It will be noticed that the first three tests were run with the teeth mist lubricated, while the teeth were not lubricated during the fourth test. So that no oil would reach the gears during the fourth test, the oil-mist lubricating system was removed from each machine and the bearings packed with grease. At the start of the fourth test all the gear teeth were doused with oil from a hand oil can, but no additional lubrication was provided during the test.

The same gears were used for the third and fourth tests. After completing the third test the gears were reversed in their machines to load the nonworn side of the teeth, and those teeth most heavily loaded in the third test were most lightly loaded in the fourth test.

In addition to the above endurance testing, a short test was run without lubrication, after cleaning the teeth with a solvent to remove completely all lubricating oil.

LOAD-CARRYING CAPACITY WITH LUBRICATION

Results of those tests showed that when the teeth are lubricated, the load-carrying capacity of the teeth is governed by the breaking strength of the teeth and not by the rate at which the teeth wear. This same thing is true of the nonlubricated teeth, although the life of the teeth prior to breaking is considerably reduced when no lubrication is provided. When the teeth are cleaned with a solvent prior to operation, however, load-carrying capacity is greatly reduced and is governed by the rate at which the teeth wear rather than by breaking strength.

When lubricated, the teeth fail at the root in a manner that has the appearance of a typical bending-fatigue failure, as shown in Fig. 7. At lower torque loads failure occurs at a successively greater number of cycles, again indicating fatigue failure.

Bending stress in the teeth was calculated by the basic Lewis equation:

$$S = \frac{F \cdot P}{f \cdot Y} \,, \tag{1}$$

where

- S = bending stress at root of tooth, psi,
- F = tangential force on tooth,
- P = diametral pitch,
- f = face width, inches, and
- Y = tooth form factor for load applied near pitch circle.

However,

$$F = \frac{2 \cdot T}{D} \,, \tag{2}$$

where

T = torque on gear, lb-inches, and

D = pitch diameter, inches.

Hence, by substitution, the Lewis equation can be rewritten in a more easily used form:

$$S = \frac{2 \cdot T \cdot P}{D \cdot f \cdot Y} \quad . \tag{3}$$

The above equation assumes that one tooth carries all the load when contact is near the pitch circle. Photographs taken of both the 16- and 20-pitch teeth under static load in various positions of engagement showed that this condition existed. Despite the relatively low modulus of "Zytel" 101, the teeth did not deform enough to distribute part of the load to other teeth at this position of engagement. Because of this low modulus, however, it seems very unlikely that the entire load ever can be concentrated on one tooth when contact takes place at the end of the tooth, even when considerable error in tooth form, size, and spacing might be present. Hence, the use of the above equation, with the form factor taken for contact near the pitch circle, seems to be reasonably correct.

Published values of the proper form factor were used with the above equation to calculate the bending stresses plotted in Fig. 8 for both the 16- and 20-pitch teeth. The line of Fig. 8 was drawn from the test results and is considered to show the bending failure stress for any given number of cycles, when the teeth are lubricated. It will be noticed that the failure stress continues to decrease with the number of cycles, indicating that this material has no definite flexural endurance limit, or stress below which the teeth can operate indefinitely without failure.

The data of Fig. 8 were obtained with the 16- and 20-pitch hob-cut teeth previously described, operating with pitch line velocities from 1150 to 2260 ft/min and with oil-mist lubrication. Alignment of the test gears was virtually perfect, inertia of the rotating parts was relatively small, and the load was constant. Nevertheless, it is felt that these same data should apply reasonably well to other installations where tooth size, velocities, and other conditions are not too much different than the conditions described. Because the ideal conditions of test-stand operation will seldom be duplicated in any particular installation, a reasonable margin of safety probably should be used with the stresses shown in Fig. 8. Allowable stresses up to 75% of the failure stress would seem reasonable in many installations, with still lower stresses used when warranted by more severe loading conditions or misalignment. These maximum recommended stresses are shown by the lower line of Fig. 9.

To calculate the load-carrying capacity of lubricated "Zytel" 101 hob-cut teeth, Equation 3 can be rewritten as follows:

$$T = \frac{S \cdot D \cdot f \cdot Y}{2 \cdot P} , \tag{4}$$

where

T = torque to be transmitted for any given life, lb-inches,

S = allowable bending stress, never exceeding the value shown by the lower line of Fig. 9, and

Y = tooth form factor for load applied near pitch circle.

LOAD-CARRYING CAPACITY WITHOUT LUBRICATION

When the teeth are not lubricated after being doused initially with oil, failure takes place just below the pitch circle rather than at the root of the tooth. Continued operation after a tooth has broken tends to mutilate the adjoining teeth when no lubrication is provided, with results as shown in Fig. 10.

Failure near the pitch circle may be caused by excessive surface temperatures weakening the material at this point, thus initiating a shear type of failure. Although the mechanics of this failure have not been thoroughly investigated, it is quite apparent that the bending-stress calculations used with the lubricated teeth are not applicable in this case. A comparison of the torque capacity of the same teeth, with and without lubrication, is shown in Fig. 11. It will be noted that for any desired life, the torque capacity is considerably reduced when no lubrication is provided.

Figure 12 shows this same comparison, plotted as a ratio against cycles of operation. This can be used with the lubricated-tooth data presented earlier to establish the load-carrying capacity of teeth that can be lubricated once prior to operation, but which are not to be lubricated thereafter.

GEAR-TOOTH WEAR

Although the load-carrying capacity of hob cut teeth of "Zytel" 101 appears to be governed by breaking strength rather than by wear, some of the information obtained on tooth wear is worth presenting. It was found that when the teeth were lubricated, the teeth wore at essentially the same rate regardless of the load, provided the tooth load was not so large as to cause bending-fatigue failure prior to about 50 million cycles. Figure 13 shows the rate of tooth wear as measured by the change in composite error and the profile change. Since the test gears were open to the atmosphere during all the test work, it was concluded that tooth wear might be further reduced, or even eliminated completely, if the teeth were totally enclosed to keep out all foreign particles.

Without lubrication the rate of wear increased with an increase in tooth load, as shown by Figure 14. For the same torque, tooth wear was considerably greater when no lubrication was provided, and this is illustrated in Fig. 15. However, it was previously shown that without lubrication the failure life of the teeth was reduced. When transmitting torques that permit about 50-million-cycles life for both the lubricated and nonlubricated conditions, tooth wear was essentially the same for both conditions. This is shown in Fig. 16, where the average tooth wear is approximately .001 for 50 million cycles of operation.

Calculated surface compressive stresses, or pressures, of approximately 5000 and 2600 psi were carried successfully for 50 million cycles by the lubricated and nonlubricated teeth, respectively, without excessive wear. These stresses were calculated by the Hertz equation for contact at the pitch point with all the load on one tooth. A modulus of elasticity of 160,000 psi at 100 degrees F was assumed for these calculations.

EFFECTS OF TEMPERATURE AND HUMIDITY

The ambient temperature and humidity were not controlled during the test work. Temperature ranged from 68 to 92 degrees F, while the relative humidity varied from 29 to 80%. The gears were not visibly affected by these temperature and humidity changes.

One gear which was never operated was checked with the other gears at each inspection to see if the gears were dimensionally stable. Virtually no change could be noted in these control gears, indicating that the dimensional stability was not affected by the day-to-day changes in temperature and humidity.

GEAR-TOOTH-FRICTION TORQUE

With or without lubrication, the torque lost due to friction in the teeth was small. This friction torque was about one percent of the transmitted torque when the teeth were lubricated, and not much greater than this without lubrication.

The friction torque tends to increase as the gear teeth wear with continued operation. This tendency was more pronounced when lubrication was omitted.

GENERAL REMARKS

Inaccuracies in tooth form and thickness caused by continued wear did not seem to affect the operation of the gears in any way, except for the tendency for the tooth friction to increase somewhat, as pointed out above. This may indicate that gears made of "Zytel" 101 do not need the same degree of tooth accuracy required by some other materials.

It is realized that this paper still leaves many questions unanswered so far as gears of molded "Zytel" are concerned. It is anticipated that more information will become available as the testing program continues. In the meantime, however, it is hoped that the information presented here will be useful to those who design gears of "Zytel" 101.

Tensile strength, 70°F	9,300 psi
Elongation, 70°F	220%
Modulus of elasticity, 73°F	200,000 psi
Shear strength	8,000 psi
Impact strength, Izod, 73°F	2.0 ft-lb/in.
Hardness, Rockwell	R108
Specific gravity	1.14

Fig. 1. Physical properties of "Zytel" 101 (E. I. du Pont de Nemours and Company).

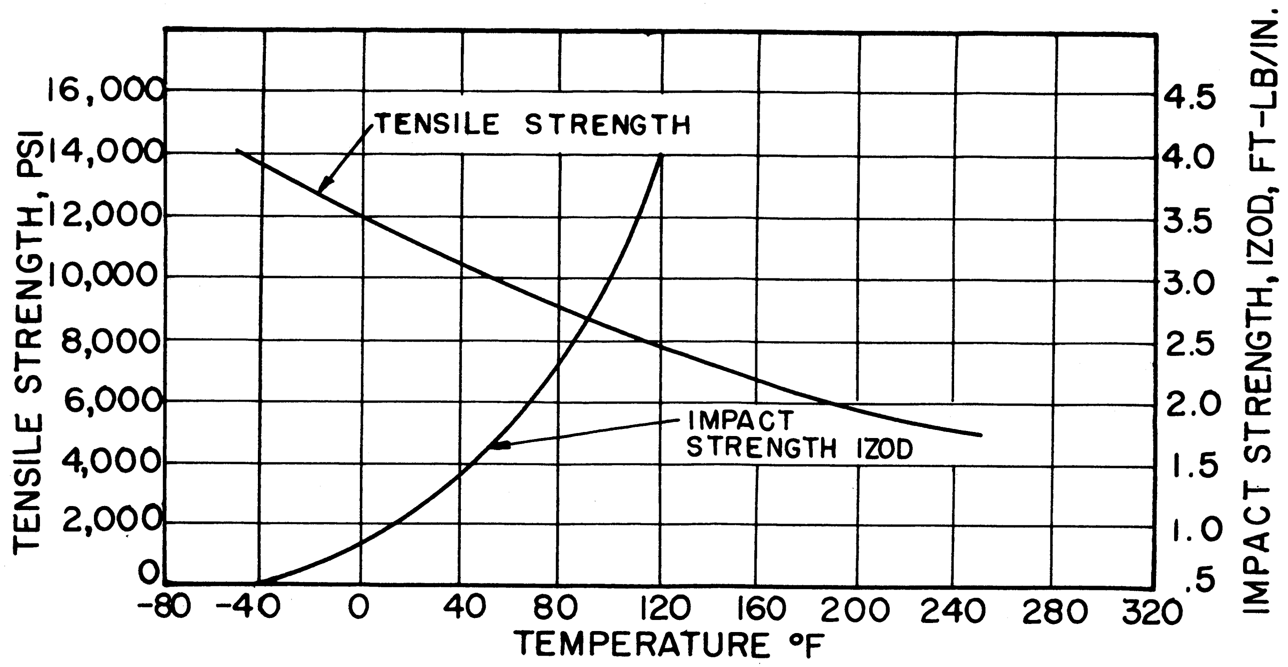

Fig. 2. Effects of temperature on physical properties of "Zytel" 101 (E. I. du Pont de Nemours and Company.)

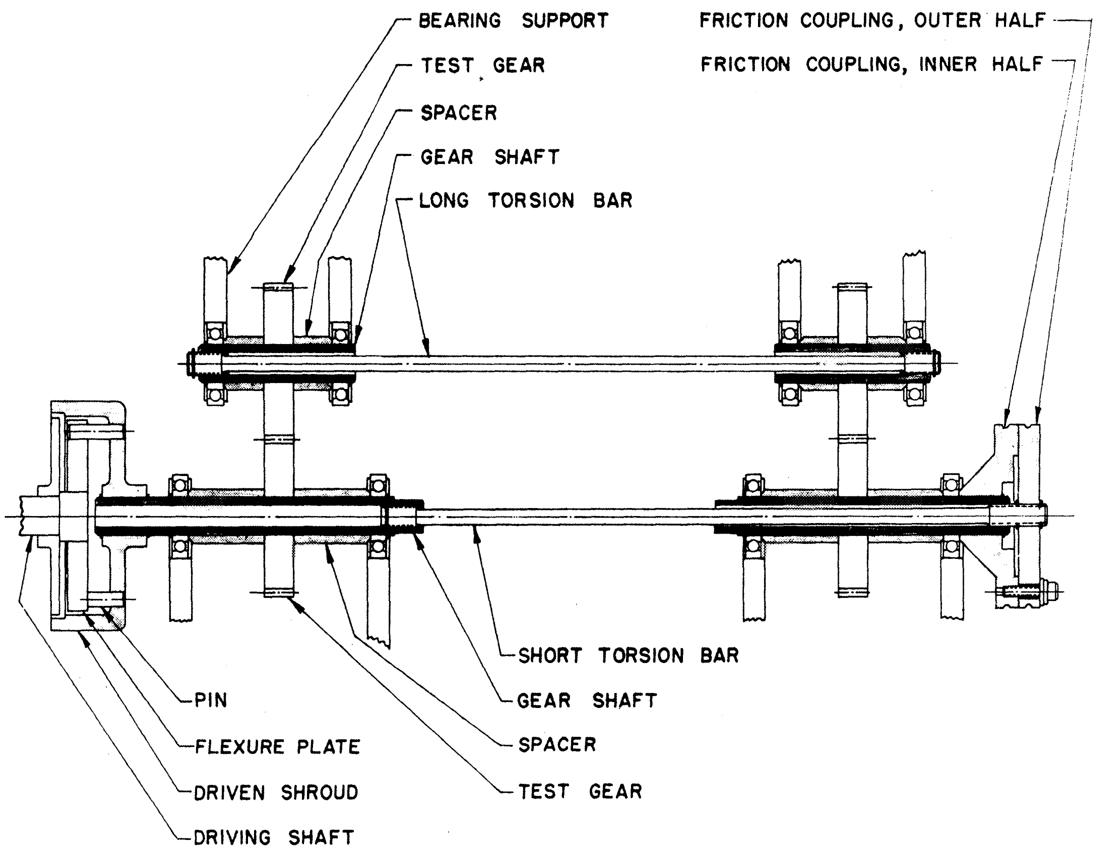

Fig. 3. Schematic layout of test machine.

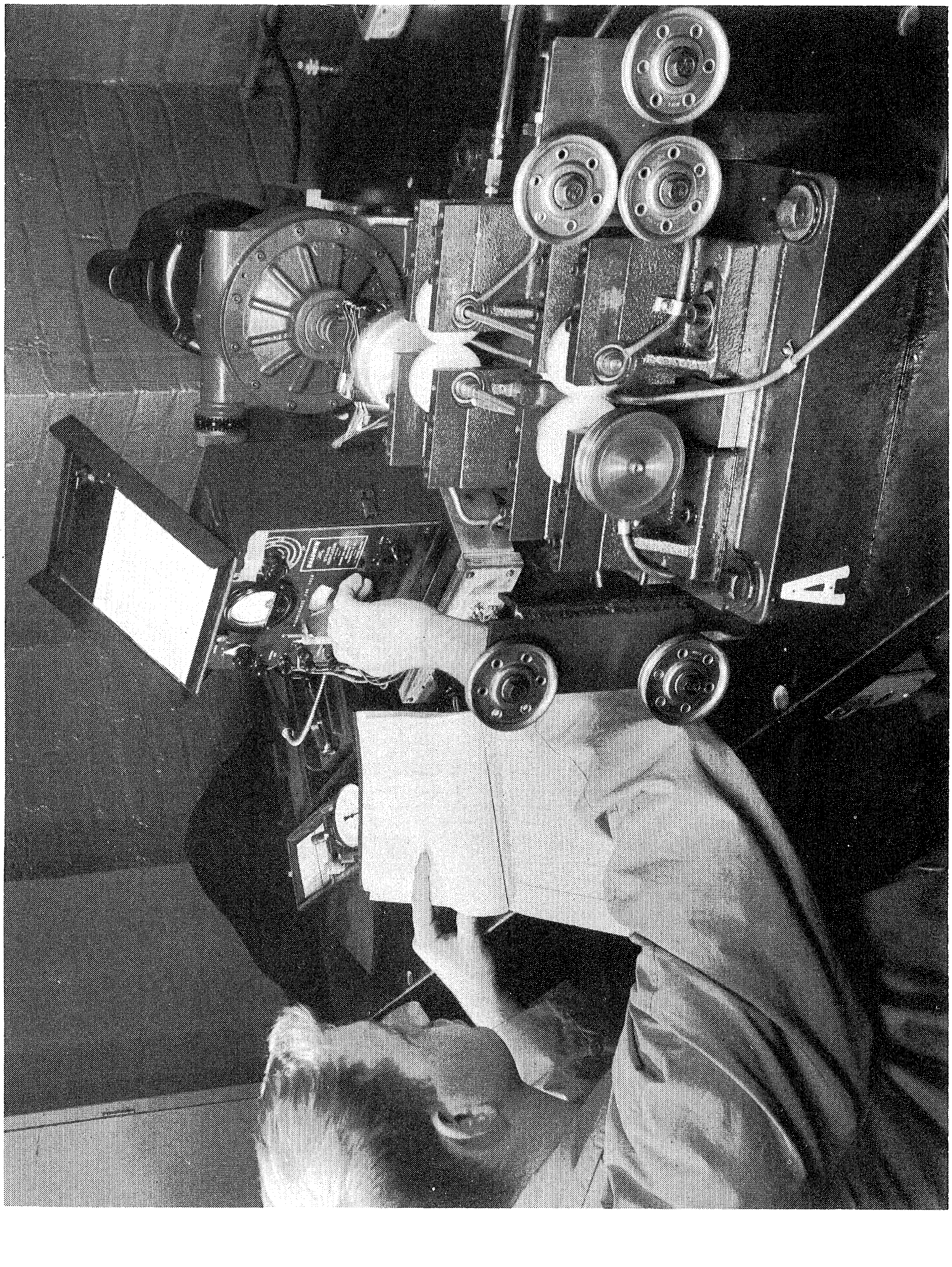

Fig. 4. Test machine.

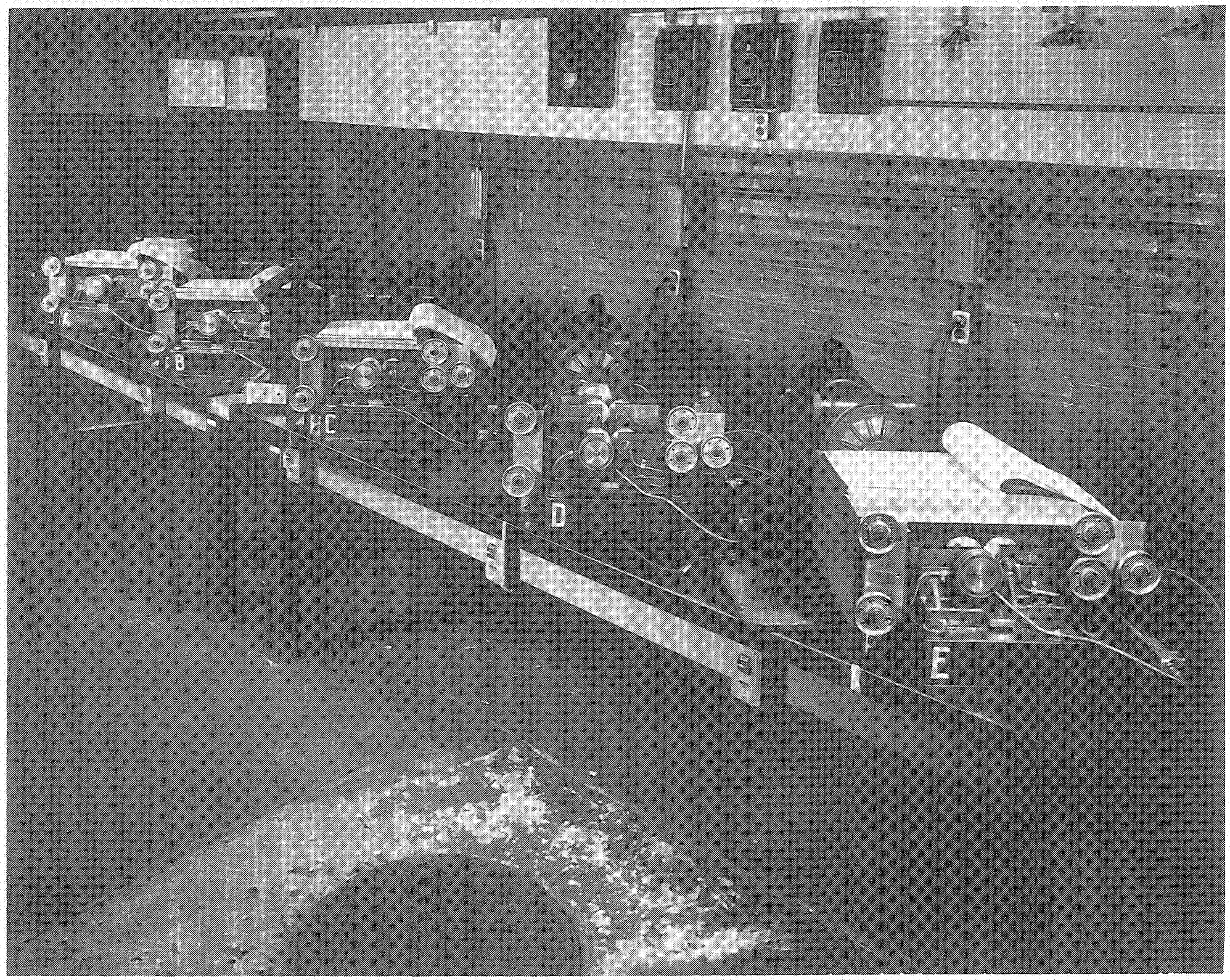

Fig. 5. Test-machine installation.

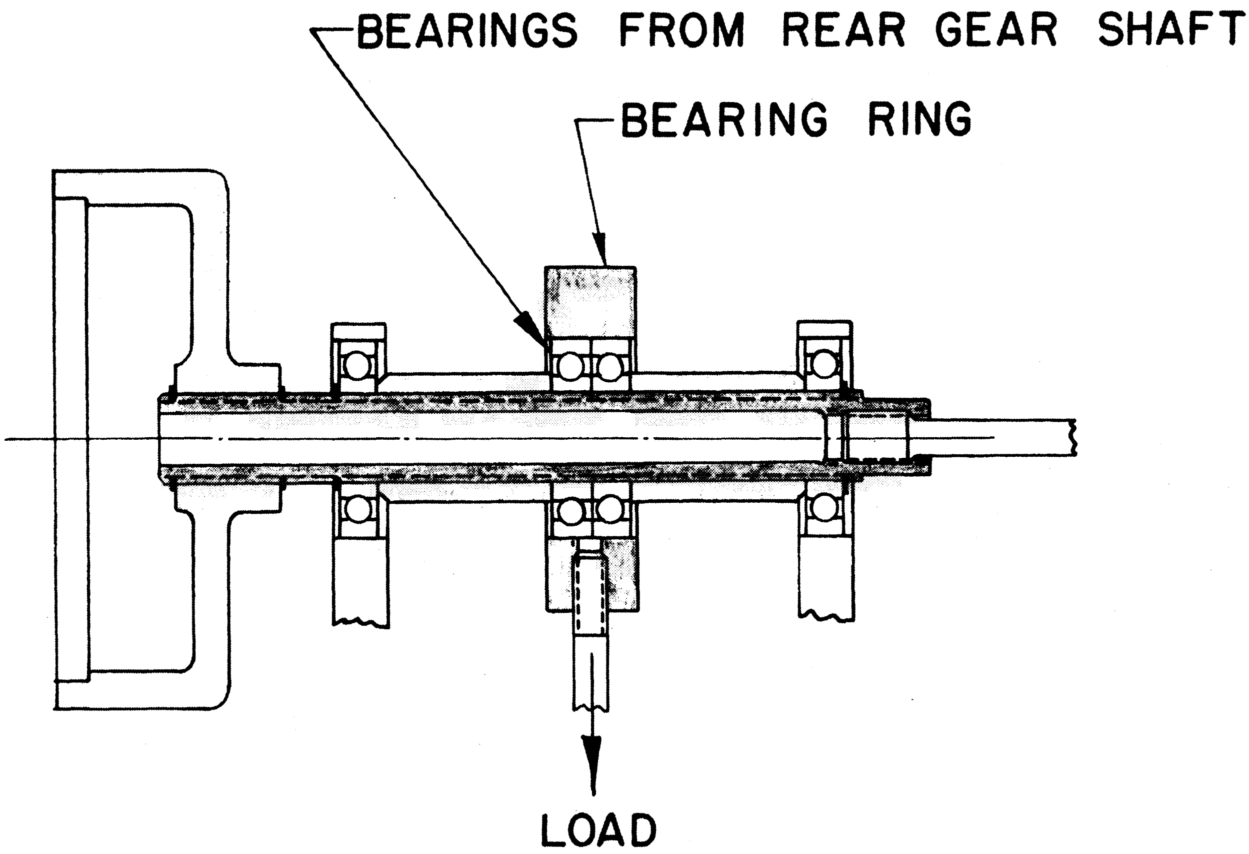

Fig. 6. Bearing test setup.

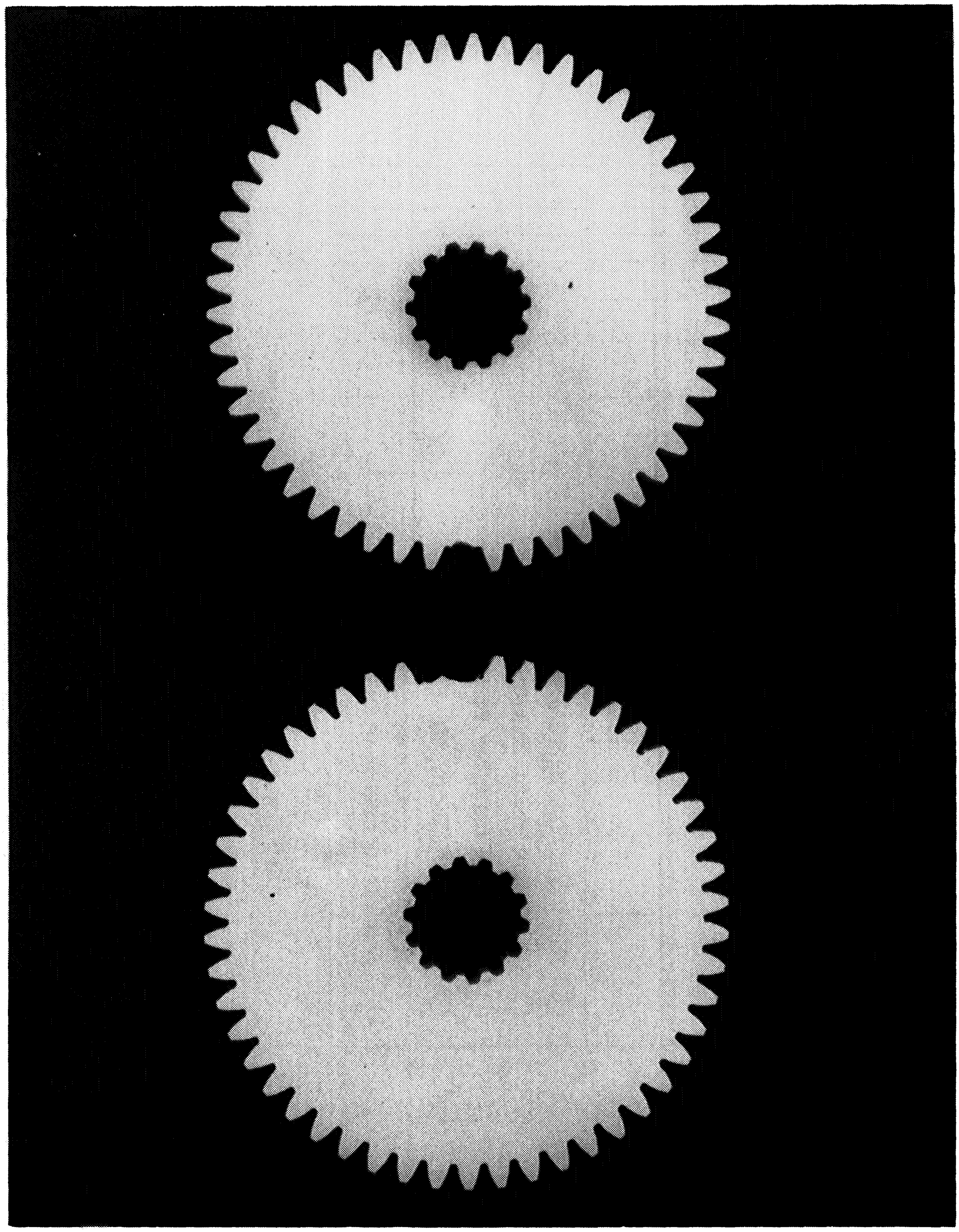

Fig. 7. Failure of lubricated gear teeth.

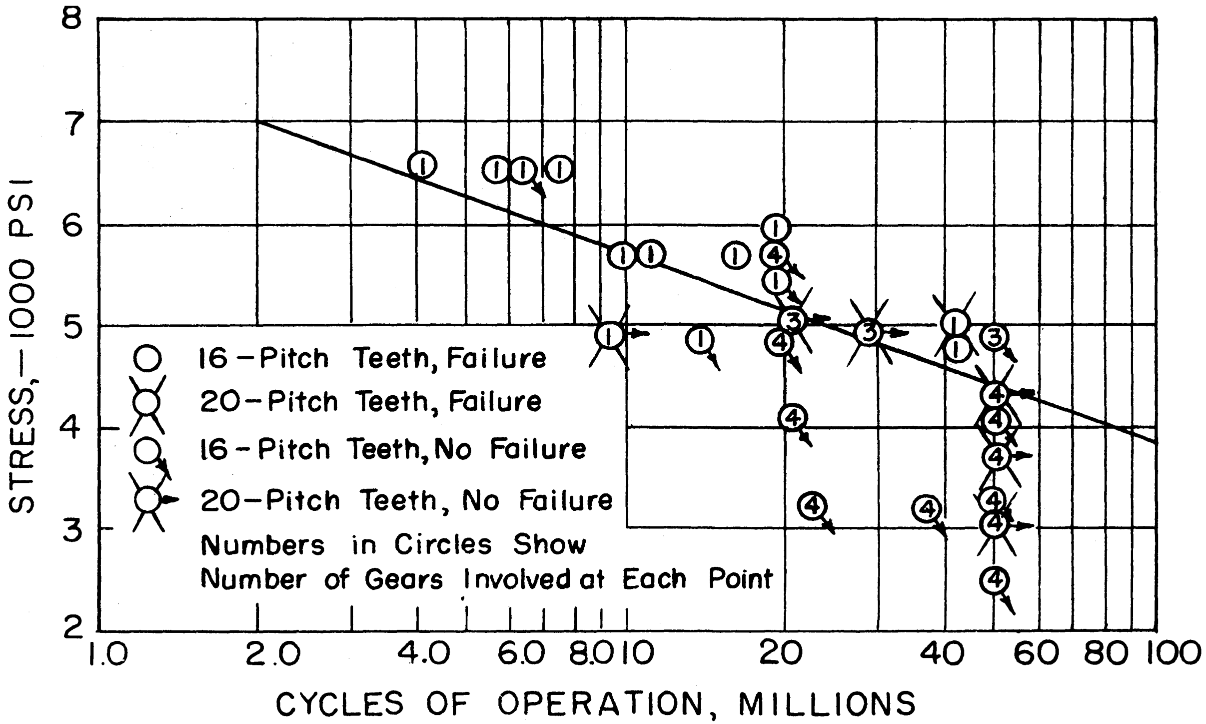

Fig. 8. Bending stress vs. cycles to cause failure, lubricated teeth.

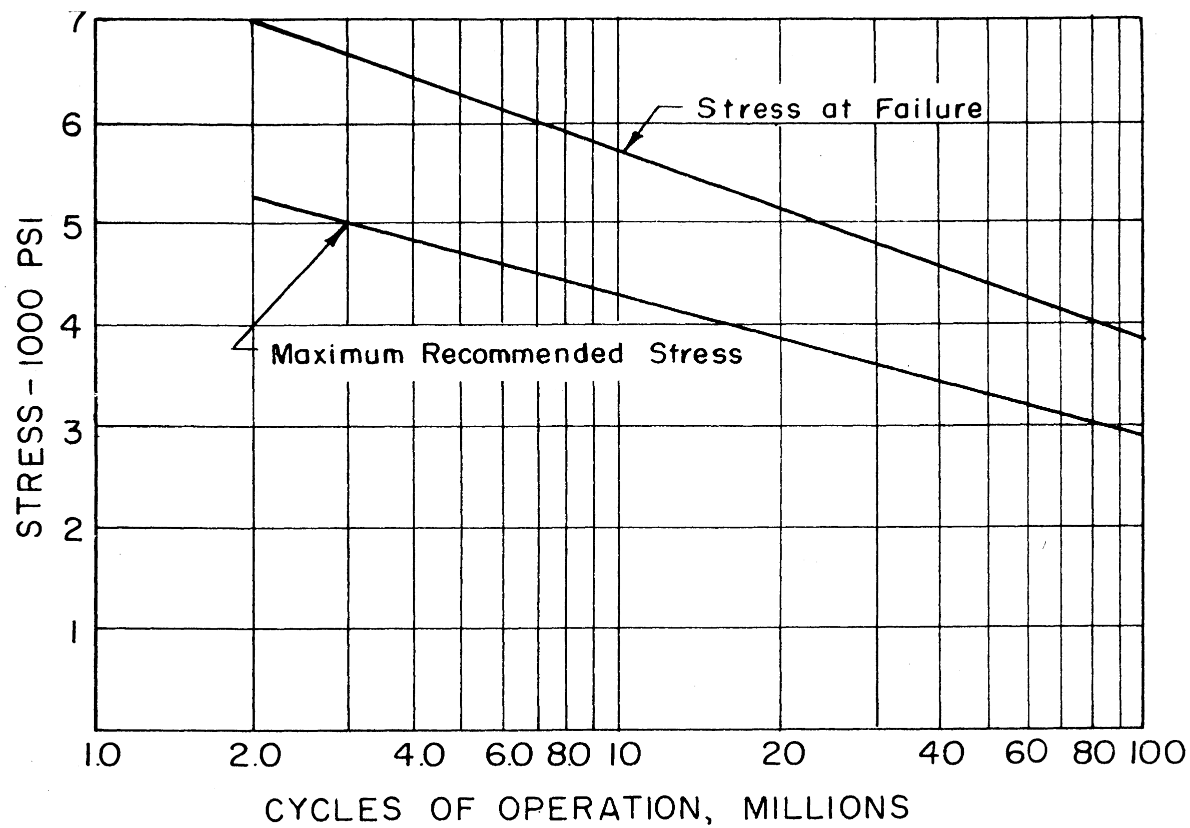

Fig. 9. Recommended stress for lubricated spur gear teeth hob cut from molded "Zytel" 101.

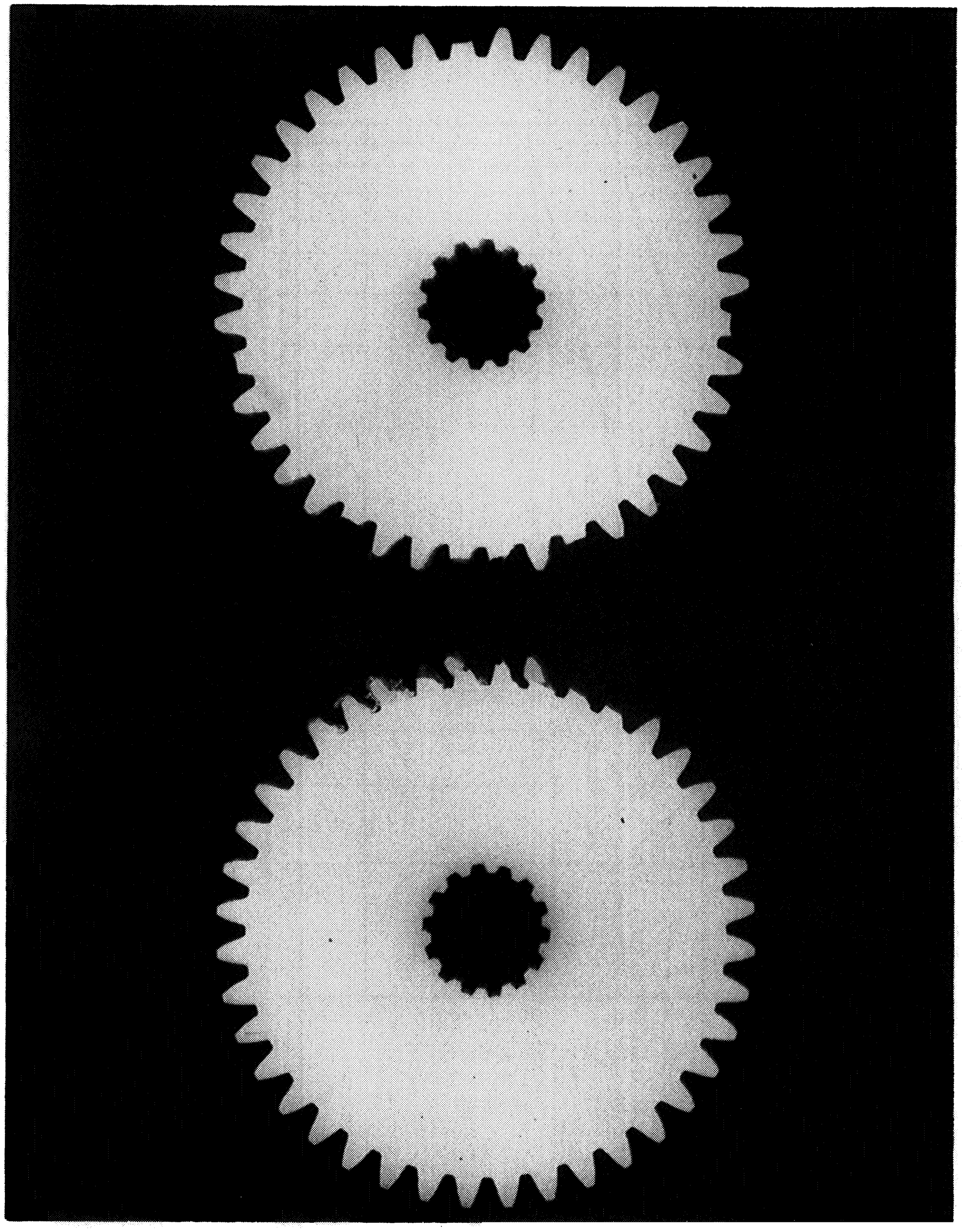

Fig. 10. Failure and mutilation of nonlubricated teeth.

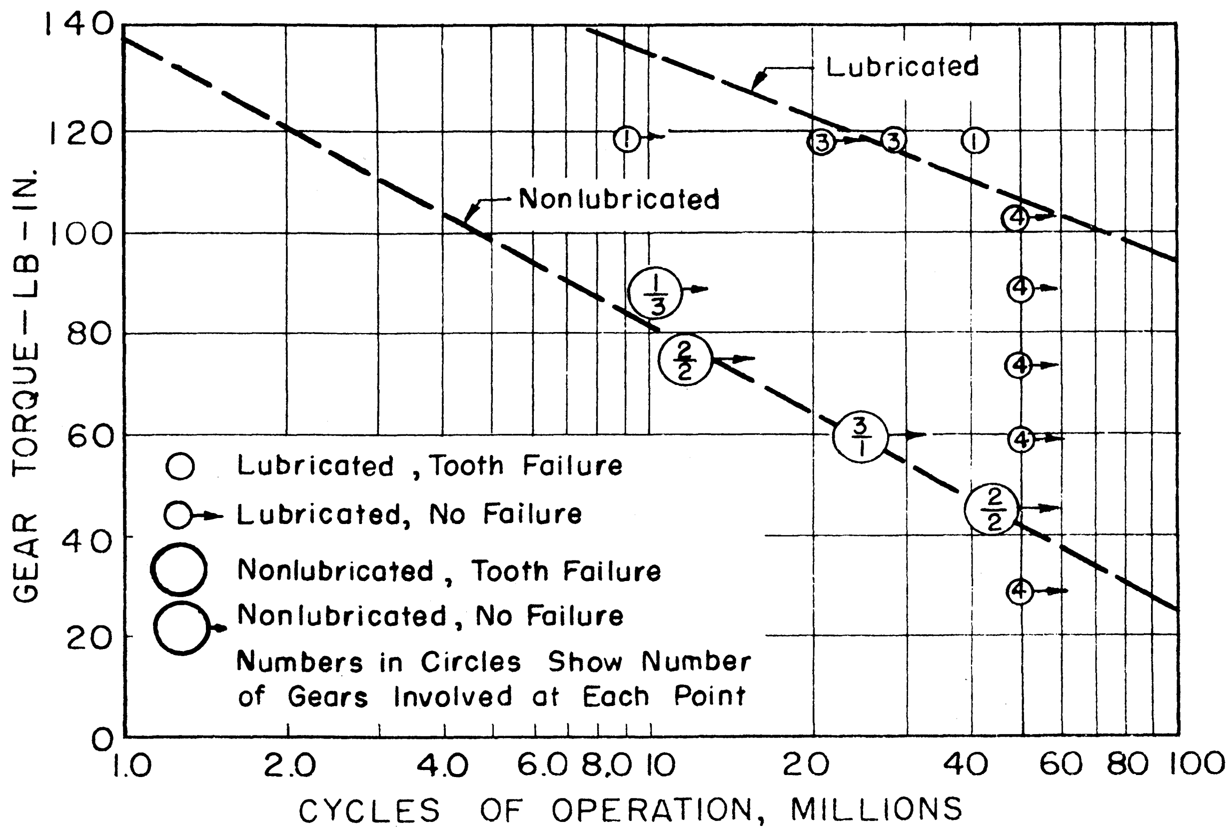

Fig. 11. Transmitted torque vs. cycles to failure, lubricated and nonlubricated teeth.

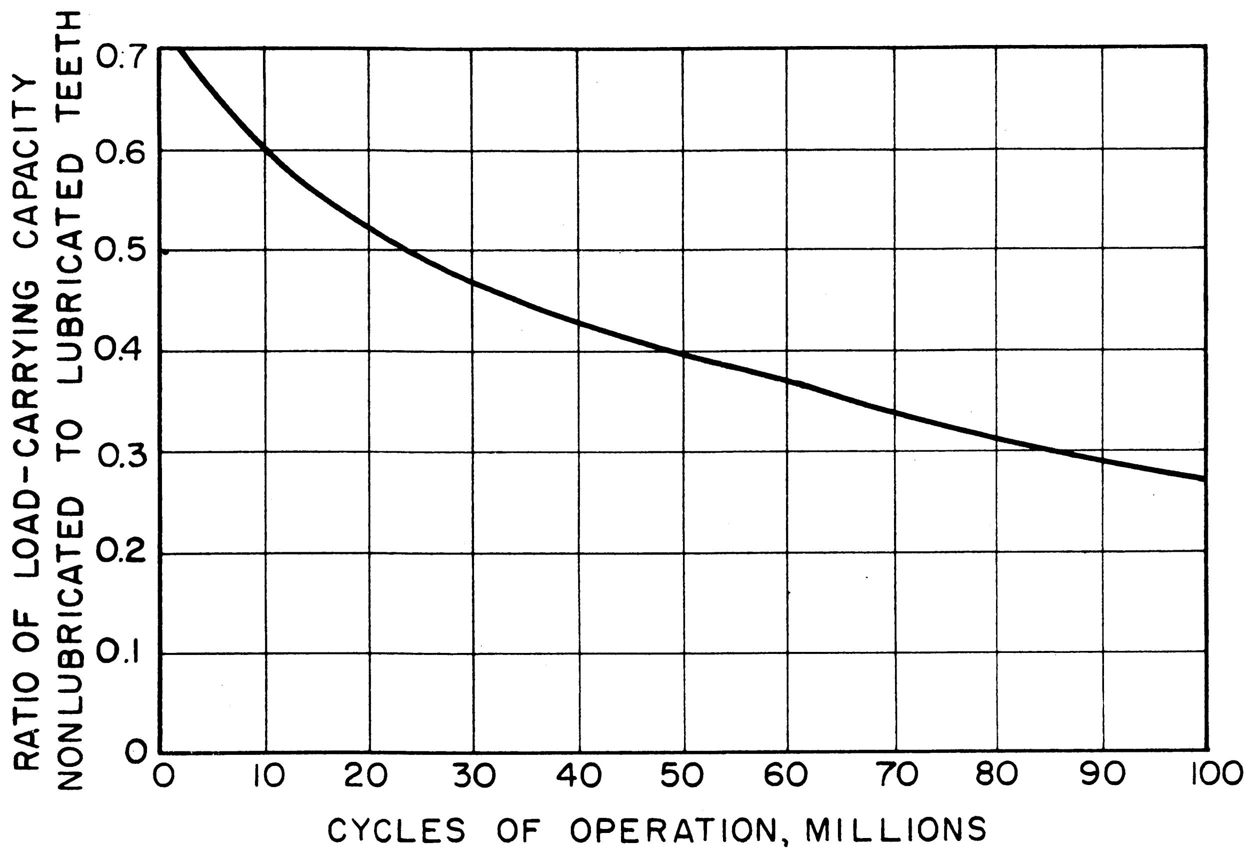

Fig. 12. Load-carrying ratio of lubricated and nonlubricated teeth.

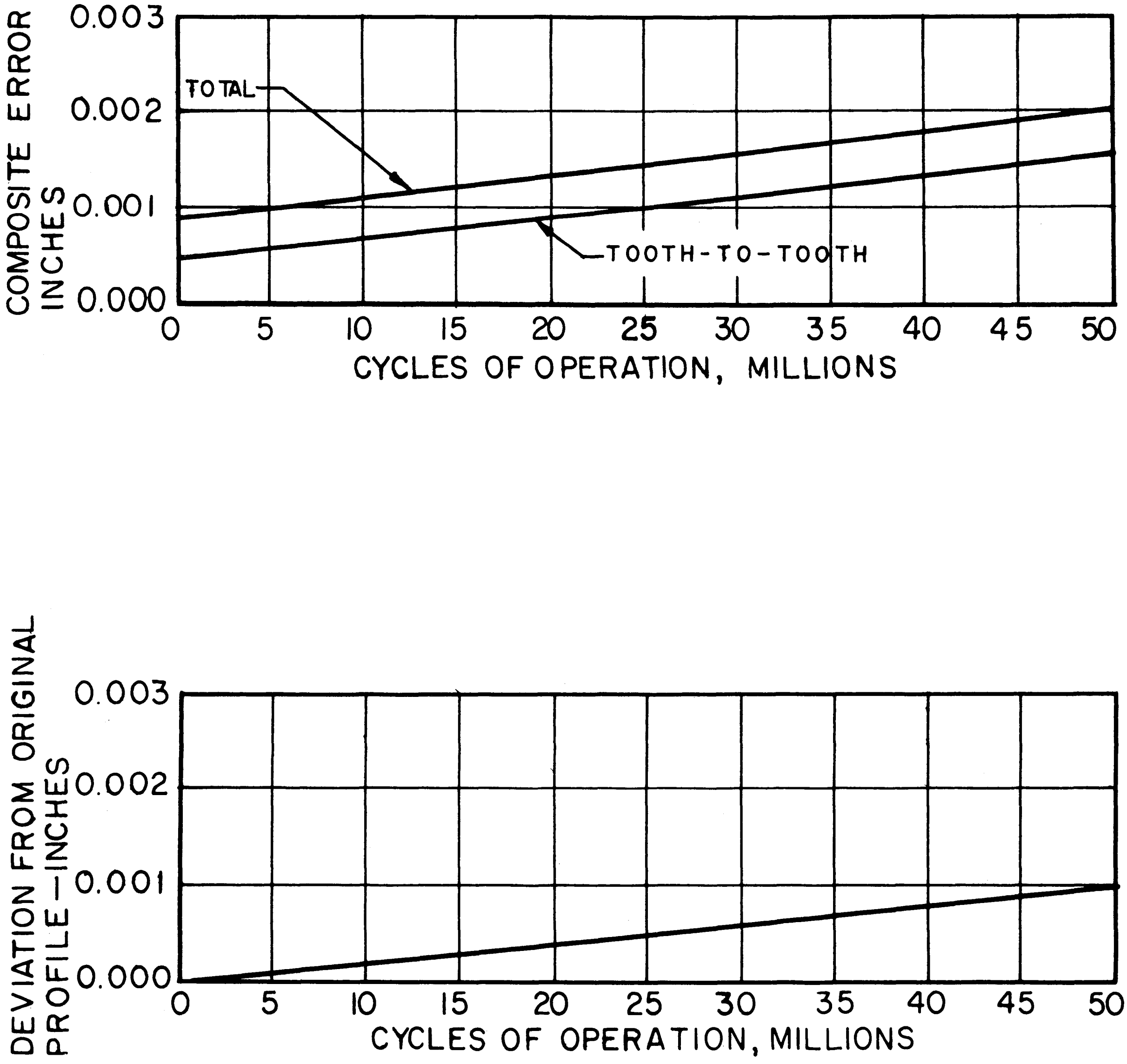

Fig. 13. Gear tooth wear, lubricated teeth (16- and 20-pitch teeth, torque 58.8 to 117.5 lb-inches).

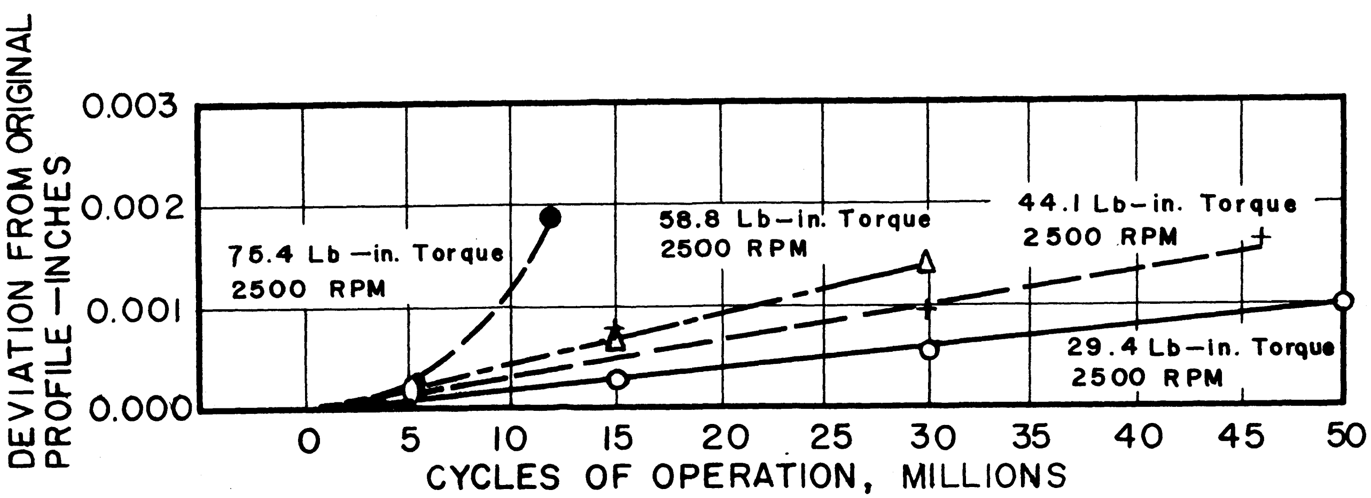

Fig. 14. Gear tooth wear, nonlubricated teeth.

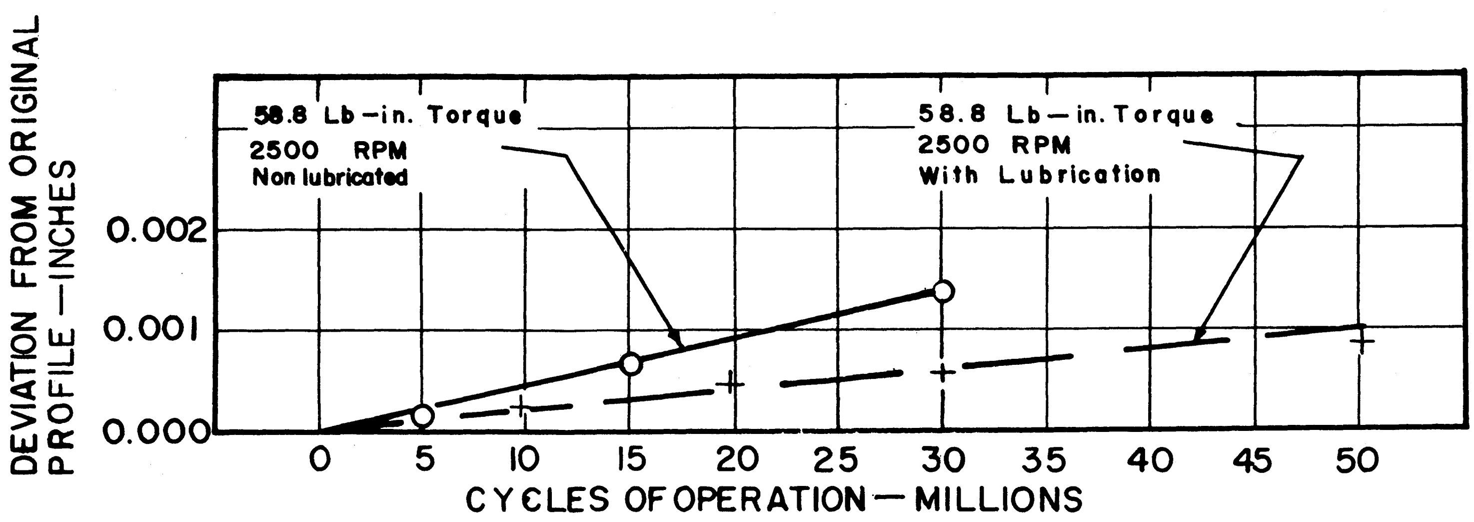

Fig. 15. Wear of lubricated and nonlubricated teeth with equal torque.

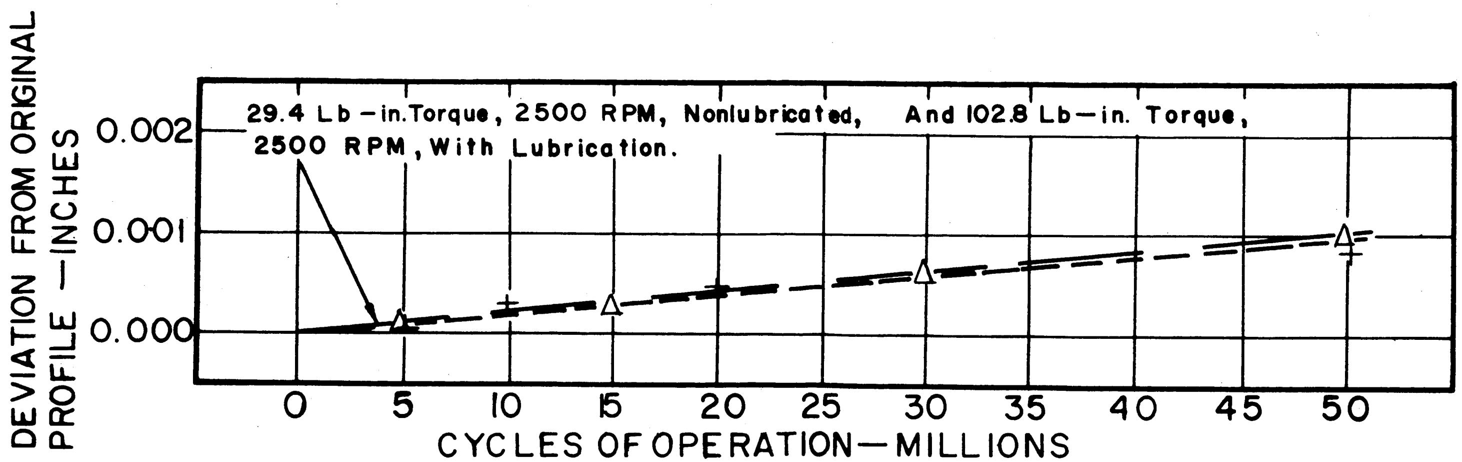

Fig. 16. Wear of lubricated and nonlubricated teeth with torque to provide equal life.