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Progress Report No. 4

EVALUATION OF ZYTEL AS A GEAR MATERIAL
AND
EVALUATION OF DELRIN AS A GEAR MATERIAL

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UMRI Project 2176

E. I. duPONT de NEMOURS AND COMPANY, INC.
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ABSTRACT

A comprehensive test program with ZYTEL spur gears has been completed. The test program and some of the equipment is described and test results are given. Two methods of calculating the load-carrying capacity of ZYTEL spur gears are presented, both derived from the analysis of the test results. Preliminary investigations of friction and wear, and of hysteresis characteristics of ZYTEL are discussed.

A new test program with DELRIN spur gears has been started. Some preliminary results of this work are presented.

I. INTRODUCTION

This project was undertaken for the Polychemicals Department of the E. I. duPont de Nemours and Company, Inc., Wilmington, Delaware. Its purpose was 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. duPont de Nemours and Company, Inc., by the Engineering Research Institute of The University of Michigan, now The University of Michigan Research Institute.

The preliminary work, consisting of designing and building five identical test machines and initial operation of these machines, was described in Progress Report No. 1, December, 1954. The test program was described and some test results and conclusions were presented in Progress Report No. 2, January, 1956. The results of an extensive test program and a method of calculating the load-carrying capacity of hob-cut teeth were presented in Progress Report No. 3, August, 1957.

This report covers the work done from August, 1957, to the present time.

The original scope of the project has been enlarged to include the evaluation of gears made of DELRIN as well as ZYTEL, and work done with both materials is covered in this report. ZYTEL 101 is the only ZYTEL material which has been used since the start of this project; hence the parts of this report which concern ZYTEL refer only to ZYTEL 101. Various DELRIN materials have been used, and these are mentioned in those parts of the report where the work with DELRIN is described.

II. EXPERIMENTAL APPARATUS

The gear testing machines and instrumentation were described in Progress Report No. 2, January, 1956. No significant changes have been made in the test machines since that time. However, some additional experimental apparatus has been built and is now being used. This additional apparatus, described in more detail later, is listed below.

1. Boston helical gear speed-reducer test assembly.
2. Rolling-sliding contact test machine.
3. Hysteresis investigating device.

III. EVALUATION OF ZYTEL AS A GEAR MATERIAL

1. EXPERIMENTAL WORK WITH ZYTEL

As mentioned in the Introduction, ZYTEL 101 has been used for all of the experimental work involving ZYTEL. The following experimental work has been carried on during the period covered by this report:

a. Fatigue-testing of molded spur gears to establish the load-carrying capacity of molded teeth, and to provide design data for ZYTEL gears.

b. Fatigue-testing of large helical gears in an industrial speed reducer to help verify the application of the design data.

c. Wear testing of hob-cut ZYTEL gears meshing with steel gears.

d. Investigation of the relationship of contact pressure, sliding velocity, and coefficient of friction, and the effects of the above on surface deterioration.

e. Preliminary investigations of hysteresis characteristics of ZYTEL under various conditions of stress.

Most of the time and effort was spent on the fatigue-testing of the molded spur gears. Each of the above phases of the experimental work is described in the paragraphs which follow.

2. FATIGUE-TEST PROGRAM WITH ZYTEL SPUR GEARS

The fatigue-testing program was carried on very much as described in Progress Report No. 3, August, 1957. The test machines ran continuously, 24 hours per day, for most of the fatigue-testing, being stopped only occasionally for visual inspection of the teeth or to install new gears.

When a gear failed, that gear and the gear with which it meshed were both replaced, and this was counted as a single failure. When the failure was not immediately detected, and the gears continued to run with several teeth missing all four gears were replaced, but this was still considered to be a single failure.

3. MOLDED ZYTEL TEST GEARS

Fatigue-testing was done with the following molded gears:

a. 20 P, 50 T, 20° pressure angle, full-depth teeth, 1/2-in. face.
b. 32 P, 80 T, 20° pressure angle, full-depth teeth, 1/2-in. face.

The details of these gears are shown in Figs. 1 and 2. A single-cavity mold is used, with a ring gate at the hub. Figure 3 shows one of the test gears sectioned to show the ring gate. The teeth on both of the above gears have a small chamfer rather than a smooth curve joining the tooth profile to the bottom land. However, the chamfer very nearly approximates the conventional shape at the root of generated teeth.

To determine the accuracy of the molded gears, six gears having 20-pitch teeth and six gears having 32-pitch teeth were inspected by measuring the composite error with a Kodak Conju-Gage, and tracing the profiles with a Fellows Involute Profile Checker. Both of these machines, and their principles of operation, are described in Progress Report No. 2, January, 1956.

The profiles on both sides of six teeth on each of the gears inspected were traced with the Fellows Involute Profile Checker. During this work, the base circle radius used as a reference in the machine was varied until the profiles traced out as a straight line on the inspection chart. In this manner the true base circle radius of the teeth could be found, and the true pressure angle could be calculated. It was found that the pressure angle was 19.17° for the 20-pitch teeth, and 19.57° for the 32-pitch teeth. The decreased pressure angles were probably caused by differential shrinkage which occurred during and after the molding process. The actual figures obtained were as follows:

Measured base radius, 20-pitch teeth = 1.1806 in.
Theoretical pitch radius, 20-pitch teeth = 1.250 in.
Cosine of pressure angle = 1.1806/1.250 = .9445
Pressure angle, 20-pitch teeth = 19.17°.

Measured base radius, 32-pitch teeth = 1.1778 in.
Theoretical pitch radius, 32-pitch teeth = 1.250 in.
Cosine of pressure angle = 1.1778/1.250 = .9422
Pressure angle, 32-pitch teeth = 19.57°.

Although the pressure angles were found to be less than the intended 20°, the profiles were found to be very nearly perfect involutes. On the teeth checked, the maximum variation from the true involute was 0.001 in., while many of the teeth varied no more than 0.0002 in. from the true involute profile. Average maximum variation was 0.0006 in. for the gears checked.

The results obtained (in inches) with the Kodak Conju-Gage were as follows.

	20-pitch	32-pitch
Total composite error, max.:	0.0034	0.0025
Total composite error, avg.:	0.0028	0.0016
Tooth-to-tooth composite error, max.:	0.0019	0.0008
Tooth-to-tooth composite error, avg.:	0.0011	0.0006

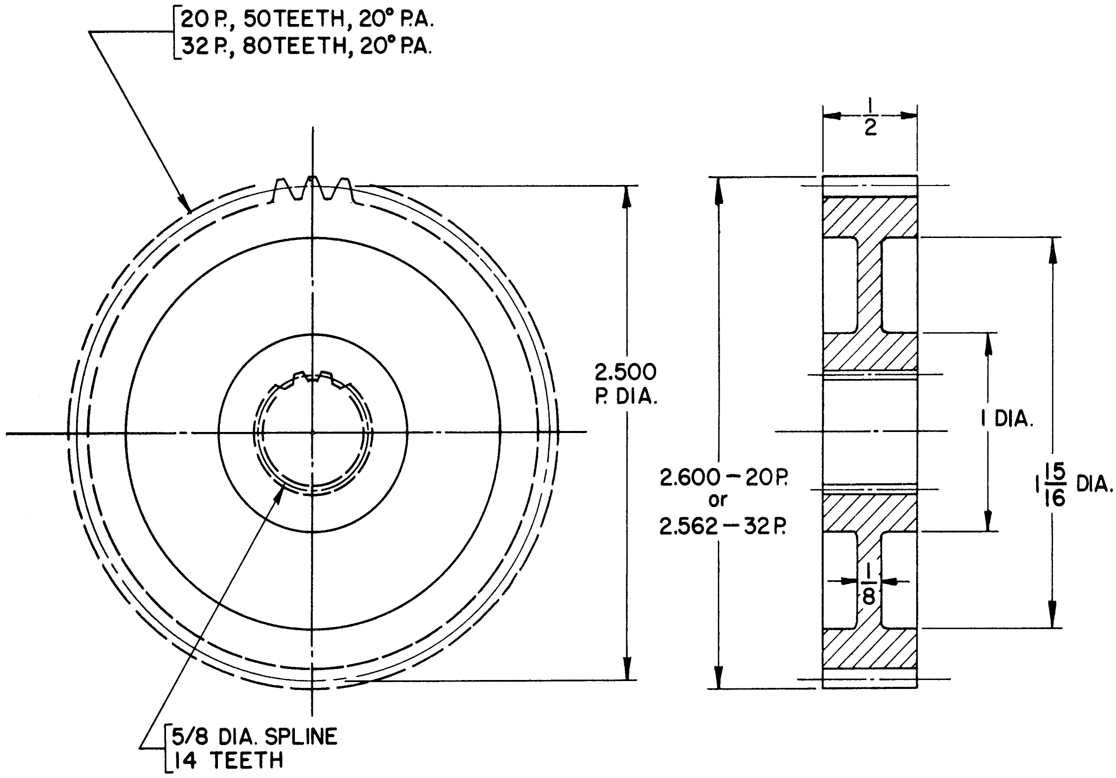


Fig. 1. Molded test gear.

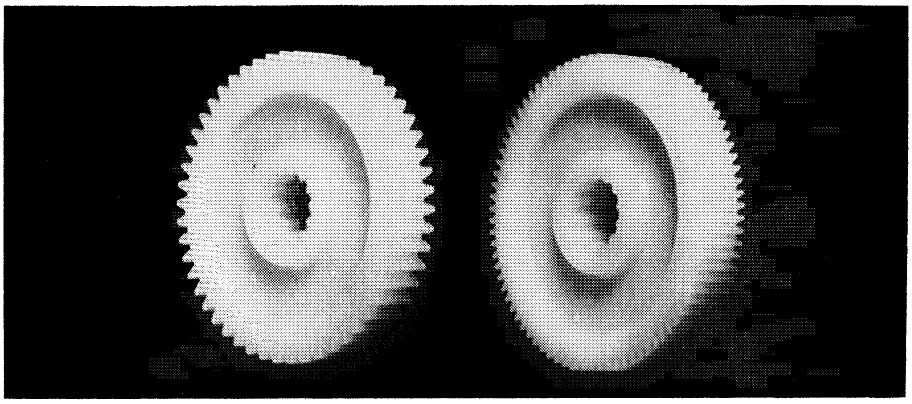


Fig. 2. Molded test gears.

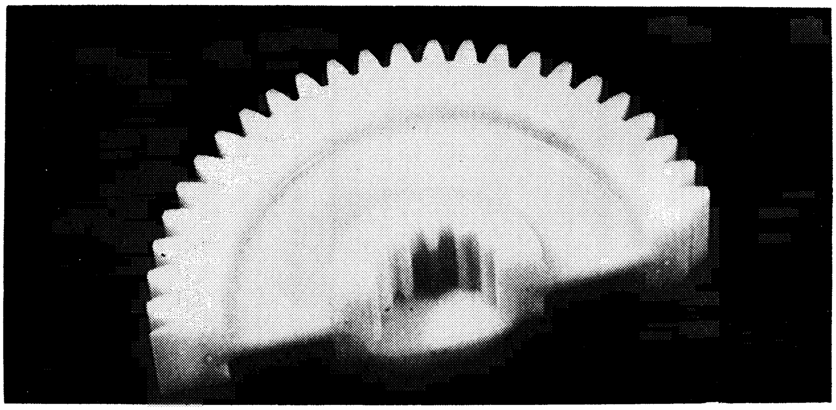


Fig. 3. Section through ring gate.

The tooth-to-tooth composite error combines errors in tooth form, tooth thickness, and individual tooth spacing. The total composite error combines all the above errors with eccentricity of the pitch diameter. The average values stated above are within the allowable error range of AGMA Commercial Class 1 for the 20-pitch teeth, and Commercial Class 2 for the 32-pitch teeth.

The twelve gears which were inspected were selected at random from the first batch of gears received. It was later noticed that some of the gears in this first batch appeared to be warped, and had a considerable wobble when rotated on a shaft, or when rolled across a flat surface. None of the twelve gears used for inspection had a pronounced wobble, however.

After discovering this distortion in some of the gears in the first batch, all the gears in the second batch which were not moisture-conditioned were checked for wobble in the plane of rotation. This was done by rotating the gear by hand and measuring the wobble with a dial indicator bearing on the rim of the gear. A total of 50 gears were checked in this manner. The wobble varied from a maximum of 0.0395 in. to a minimum of 0.0037 in. The average was 0.0123 in.

The warping which caused this wobble was probably caused by the thin web solidifying more rapidly than the heavier rim during the molding process. Shrinkage of the rim while cooling would then cause residual stress in the rim and the resulting distortion. Unless the wobble was rather severe, however, the gears were used in the test program.

4. BACKLASH, LUBRICATION, AMBIENT TEMPERATURE, AND HUMIDITY

The 32-pitch teeth operated with 0.002- to 0.004-in. average backlash, while the 20-pitch teeth operated with 0.003- to 0.005-in. average backlash. Over a reasonable range, variation in backlash seemed to have little or no effect, although no testing was done to establish an optimum backlash if such exists. Both excessive backlash and no backlash at all seem to reduce greatly the life of the teeth.

Oil-mist lubrication, as described in Progress Report No. 1, was used when lubrication of the teeth was desired. When no lubrication was desired, the oil-mist system was disconnected and the bearings were lubricated with grease. New gears were well lubricated with oil at the start of the test with no lubrication, but were not lubricated thereafter.

Ambient temperature and humidity were not controlled at any time during the test program. Temperatures ranged from approximately 68°F to 92°F, while the relative humidity varied from about 29 to 85% during the period covered by this report.

5. METHOD OF CALCULATING BENDING STRESS IN ZYTEL TEETH

The bending stress shown on the curves which follow was calculated by the original Lewis equation, as described in Progress Report Nos. 2 and 3. The entire load is considered to be carried by one tooth in contact near the pitch point. For easy reference the equation is reproduced here as follows:

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

where

- S = calculated bending stress, psi,
- F = tangential force on tooth, lb,
- P = diametral pitch,
- f = face width, in., and
- Y = form factor for load near pitch point.

An error was made in presenting values of the form factor Y in Progress Report No. 3. Correct values of the form factor for various numbers of teeth are shown in Table I of this report. These values of the form factor are from Buckingham's Analytical Mechanics of Gears, McGraw-Hill, page 477. The values in Table I are Buckingham's values multiplied by 3.14159 for use in Eq. (1). As stated above, the stress is calculated as though one tooth carries the entire load. A photographic investigation was made of the load distribution between the teeth while the gears were running, and this work is described in Section 10 of this report.

6. FATIGUE TESTS AND RESULTS WITH MOLDED ZYTEL GEARS

Fatigue-testing of the 20- and 32-pitch gears shown in Figs. 1 and 2 was done at 2500 rpm, both with and without lubrication. At this speed the pitch-line velocity is 1635 ft/min. Some of the gears were tested as molded, with no subsequent treatment. Others were moisture-conditioned in potassium acetate for 60 hours to 2.5% equilibrium. Still others were annealed at 300°F. The results obtained are shown in Figs. 4 and 5. For comparison, the lines established for 20- and 32-pitch hob-cut teeth from Progress Report No. 3 are also shown.

Many of the 20-pitch gears which had not been annealed failed by breaking through the rim and web, rather than by breaking of the teeth. It was not uncommon with the 20-pitch gears to find that cracks developed at the root of a large number of teeth after a short period of operation, but that the gear continued to carry the load for a long period of time after the cracks developed. During this continued operation, one or more of the cracks would progress through the rim until it finally reached the web, at which time a piece commonly broke out to end the test.

TABLE I

TOOTH FORM FACTOR "Y" FOR LOAD NEAR PITCH POINT

Number of Teeth	20° Full-Depth Form	20° Stub-Depth Form
14	-----	0.540
15	-----	0.556
16	-----	0.578
17	-----	0.587
18	-----	0.603
19	-----	0.616
20	0.544	0.628
21	0.553	0.638
22	0.559	0.648
24	0.572	0.664
26	0.588	0.678
27	0.592	0.684
28	0.597	0.688
30	0.606	0.698
34	0.628	0.714
38	0.651	0.729
40	0.660	0.733
43	0.672	0.739
50	0.694	0.758
53	0.700	0.762
56	0.706	0.764
57	0.707	0.765
60	0.713	0.774
64	0.719	0.778
66	0.721	0.782
74	0.734	0.791
75	0.735	0.792
100	0.757	0.808
120	0.766	0.817
150	0.779	0.830
164	0.781	0.832
300	0.801	0.855
Rack	0.823	0.881

Figures 6a, b, and c show typical failures of this type. The cracking mentioned above is clearly visible at the root of the teeth.

At high tooth loads virtually all the 20-pitch gears which failed did so by breaking through the rim. Most of the 20-pitch gears which failed at lower

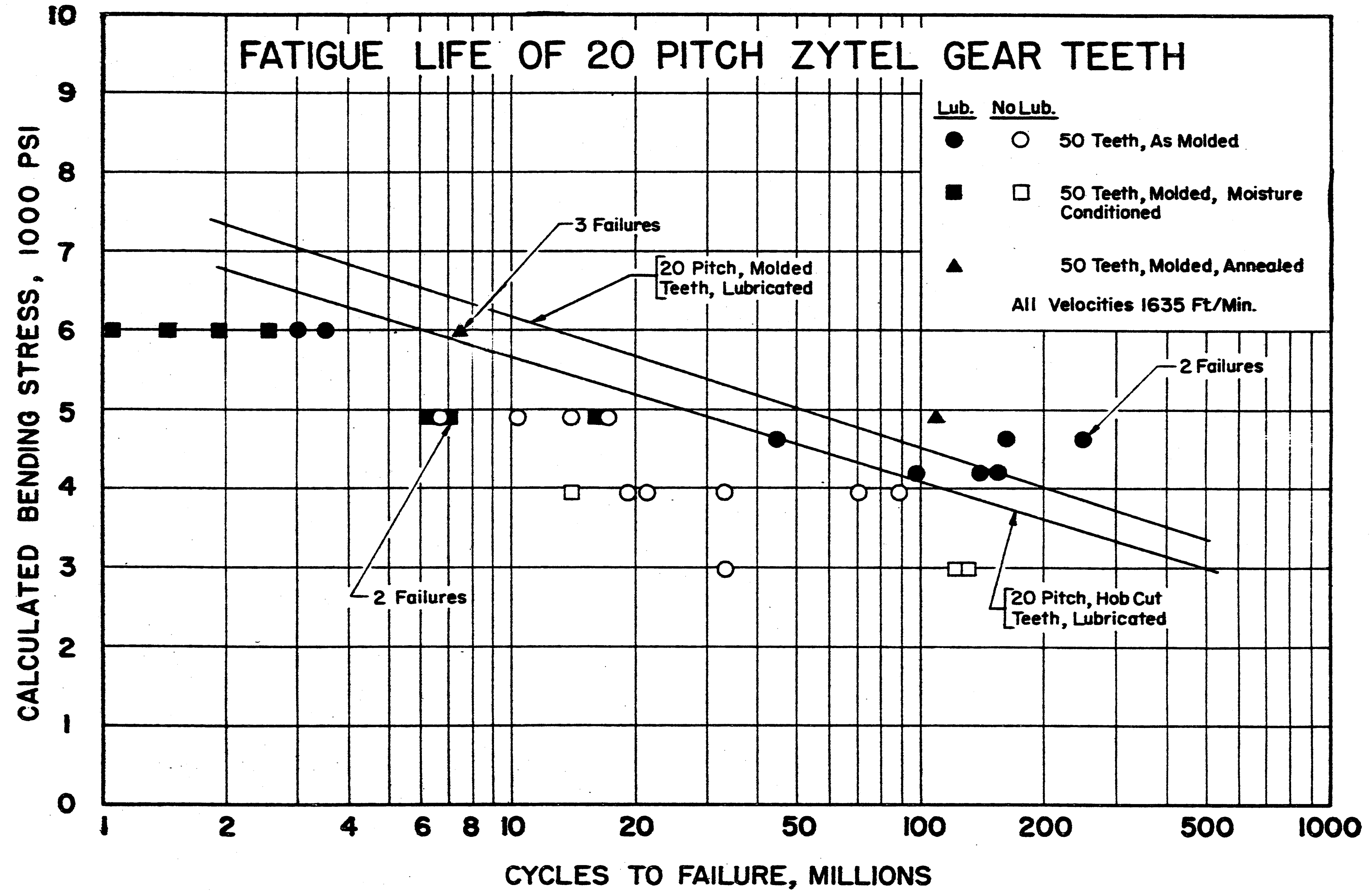


Fig. 4. Fatigue life of 20-pitch molded ZYTEL gear teeth.

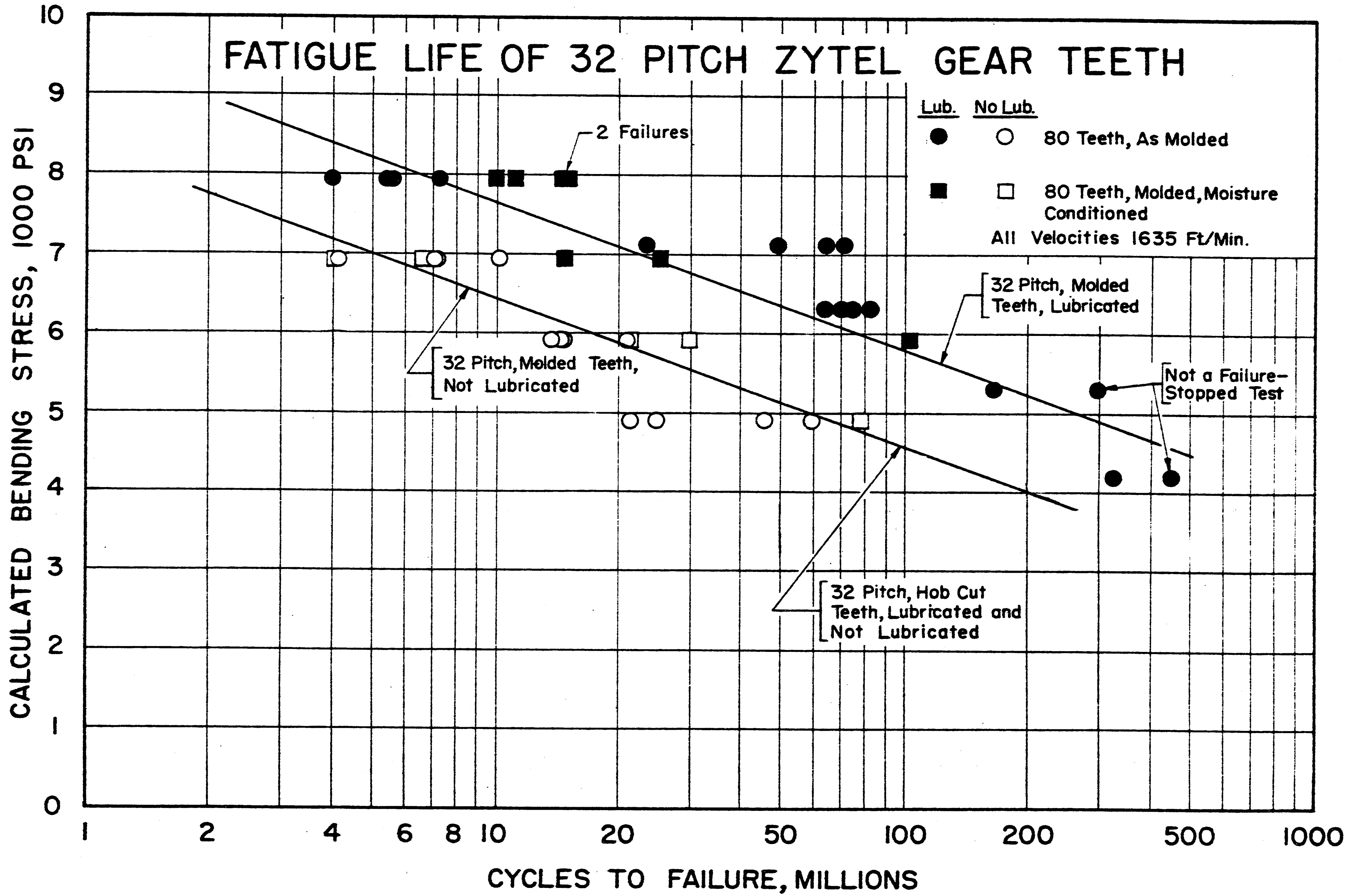
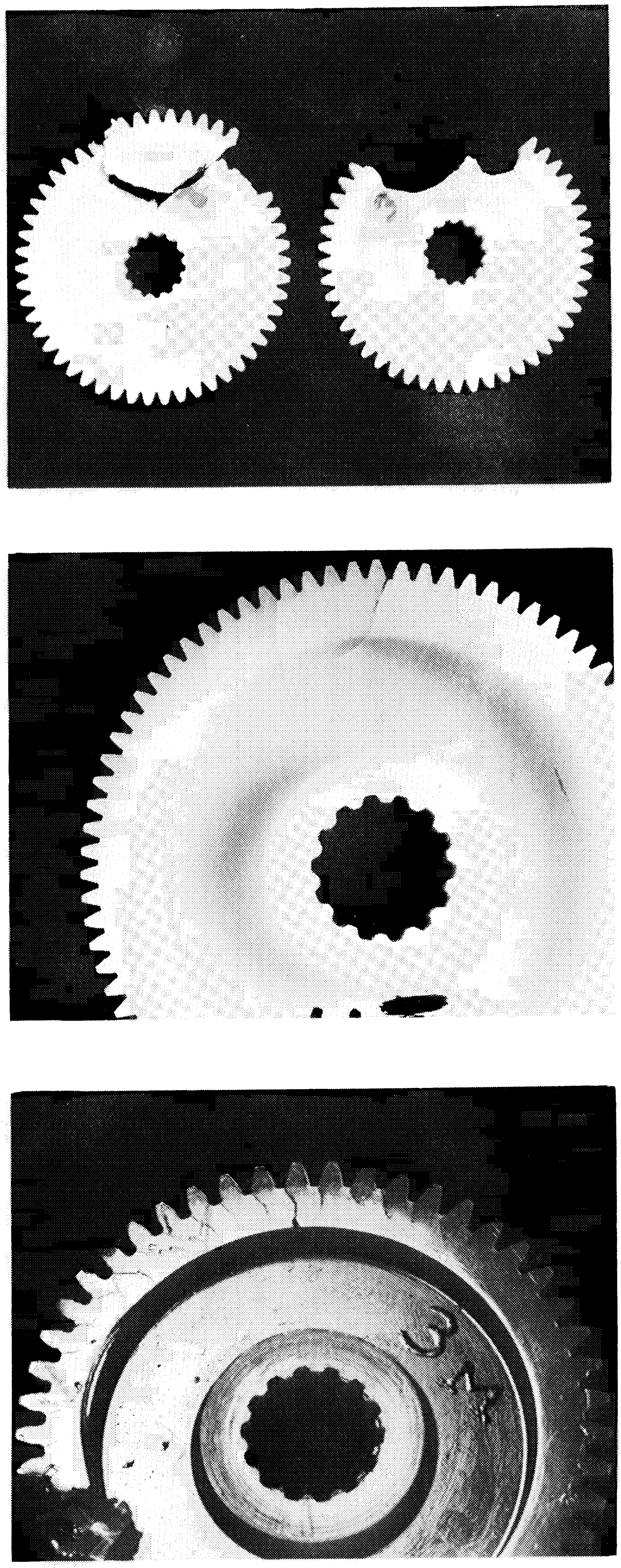


Fig. 5. Fatigue life of 32-pitch molded ZYTEL gear teeth.



a

b

c

Fig. 6. Failures of 20-pitch gears.

loads, however, did so by breaking through the teeth in a typical fatigue type of failure. It would seem apparent that the cracking at the root of the teeth, and the breaking through the rim, are caused in part by residual stresses set up in the rim during the molding process. That such stresses exist is illustrated by the width of the crack which developed in the gear shown in Fig. 6b.

Some investigating was done in an attempt to establish the approximate magnitude of this residual stress in the rim. The results of this work are described in Section 11 of this report. The presence of the residual stress in the rim appears to have considerably reduced the life of the 20-pitch teeth, especially at high stresses. The small number of annealed gears which have been tested to date indicate that at the high stresses the life can be materially increased by annealing to remove the residual stress. The results obtained with the annealed gears were used to establish the upper end of the line of Fig. 4 which represents the fatigue life of the 20-pitch molded teeth.

The residual stress in the rim did not seem to harm the 32-pitch teeth to the same degree that it did the 20-pitch teeth. Only a few of the 32-pitch gears failed by breaking through the rim. Figure 7 shows typical failures of the 32-pitch teeth.

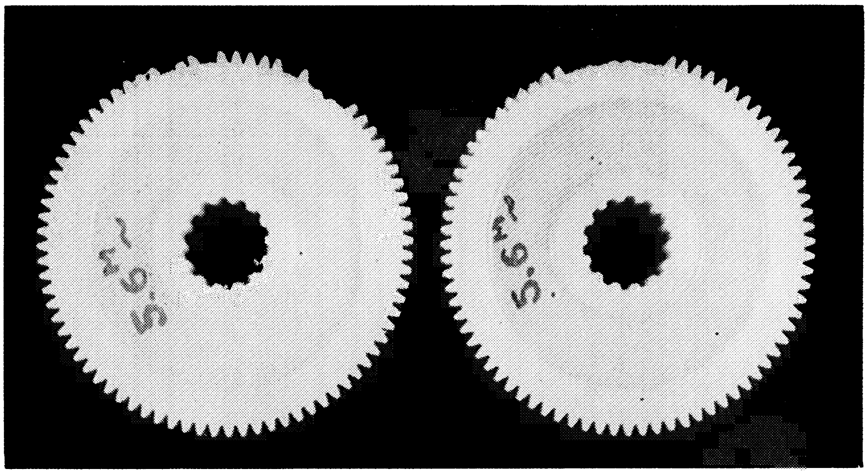


Fig. 7. Failures of 32-pitch gears.

It will be noticed from Figs. 4 and 5 that, when lubricated, the molded gears have a greater life, or can carry a greater load, than the cut gears. Without lubrication the 32-pitch molded teeth have about the same life as the cut gears. No fatigue data have ever been obtained for the 20-pitch cut gears with no lubrication; hence no comparison can be made of the performance of cut and molded 20-pitch teeth without lubrication.

Comparing the performance of the lubricated molded gears and the hob-cut gears on Figs. 4 and 5, it can be seen that the improvement shown by the 32-pitch molded gears over the hob-cut gears was considerably greater than the improvement shown by the 20-pitch molded gears over the hob-cut gears.

It is felt that the 20-pitch gears would have made a better showing at high loads if the web of the gear had been thicker to support the teeth better. This feeling was substantiated by observing the action of the 20-pitch gears and the 32-pitch gears when loaded statically until the teeth failed. With large tooth loads, the 20-pitch gears distorted noticeably. The teeth which were originally meshing with no backlash pulled radially away from one another as the rim and web distorted. This caused the teeth to mesh near the tips, increasing the bending stress and contributing to failure. In the plane of rotation the entire gear distorted enough to be apparent to the eye. The apparatus used to load the gears statically was such that it was not feasible to take measurements of the distortion, and the distortion became easily visible only after the load had exceeded the highest loads carried in the fatigue tests. However, it can be concluded that similar distortion, but of a lesser magnitude, takes place during the highest loads carried during the fatigue tests, and that this distortion could adversely affect the life of the teeth.

During the static-load tests described above, the 20-pitch teeth failed by plastic yielding, but did not break. The 32-pitch teeth, when tested in the same manner, tended to break at the root. Since the load required to break the 32-pitch teeth was much less than those applied to the 20-pitch teeth, no distortion of the 32-pitch gears was apparent visually.

It will be noticed from Fig. 4 that, at heavy loads, the molded 20-pitch gears which had been annealed to remove the residual stresses had only a slightly greater life than the hob-cut gears. At lighter loads the gears which had not been annealed had a considerably greater life than the hob-cut gears. If all the 20-pitch gears had been annealed and had a thicker web, the entire line would probably be moved upward, with the greatest improvement to be expected where loads are high and life is short. The improvement of the 20-pitch molded gears over the hob-cut gears might then approximate that obtained with the 32-pitch molded gears more closely.

With this in mind, the lines of Fig. 8 have been drawn to represent what are felt to be reasonable fatigue-life curves for molded 16-, 20-, 32-, and 48-pitch ZYTEL teeth. The 20-pitch line has been raised slightly from that of Fig. 4, anticipating some improvement from annealing the gears and providing thicker webs. Since no molded 16- or 48-pitch gears were tested, the lines for these teeth are drawn very slightly higher than the lines previously established for hob-cut gears. Since both the 20- and 32-pitch molded gears had greater life than the cut gears, it seems that some improvement in 16- and 48-pitch molded gears can also be anticipated.

Figure 9 shows another set of fatigue-life curves, in which tangential force per 1/2 in. of face width is plotted against cycles to failure for the 16-, 20-, and 32-pitch gears. These are actual test results obtained with the cut and molded teeth at 1635 ft/min pitch-line velocity. It is interesting to note from Fig. 9 that, with a 2-1/2-in. pitch diameter, the force which the teeth can carry does not vary a great deal as the pitch varies from 16 to 32. It must be kept in mind that the 16-pitch teeth were hob-cut, some of the

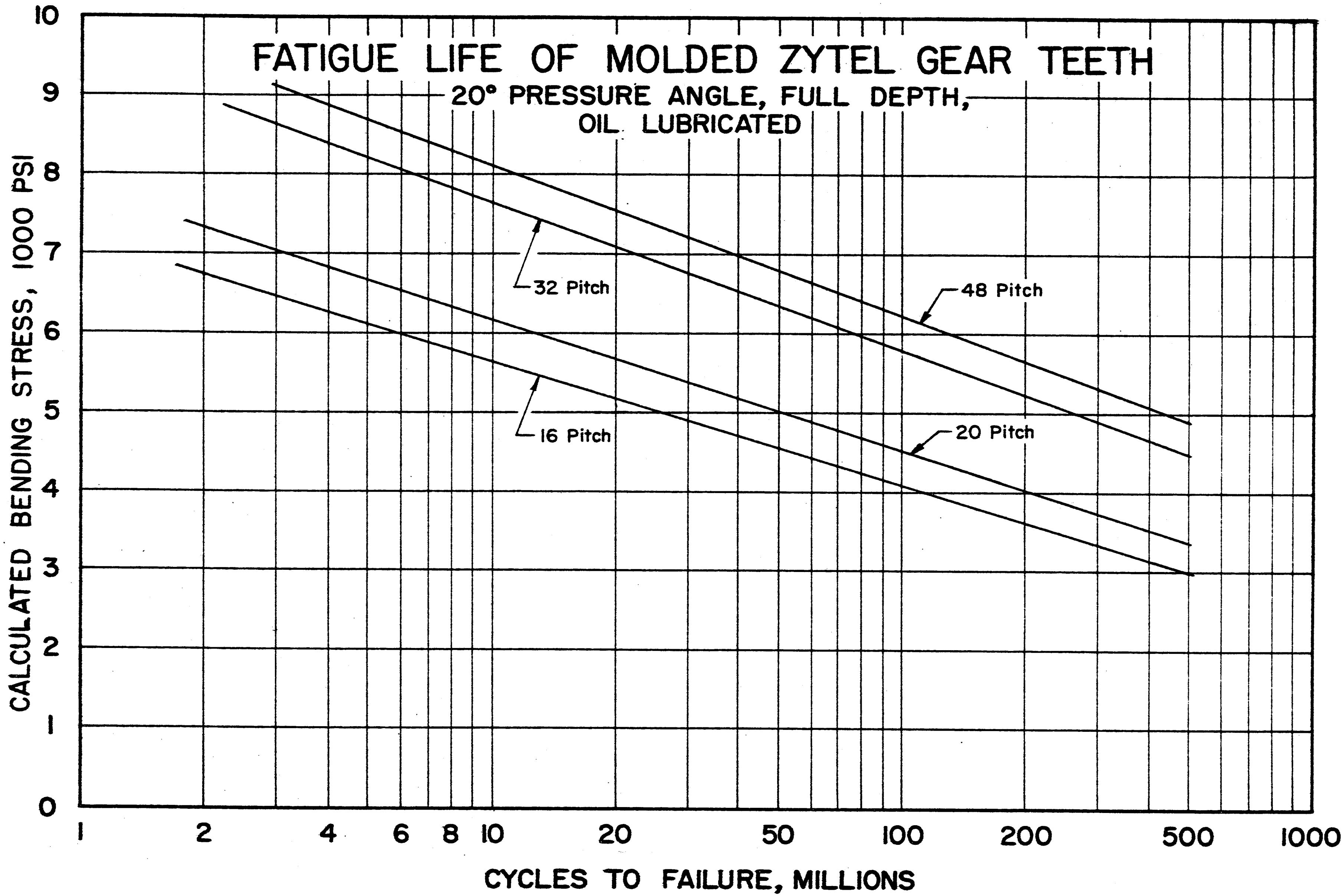


Fig. 8. Fatigue life of molded ZYTEL gear teeth.

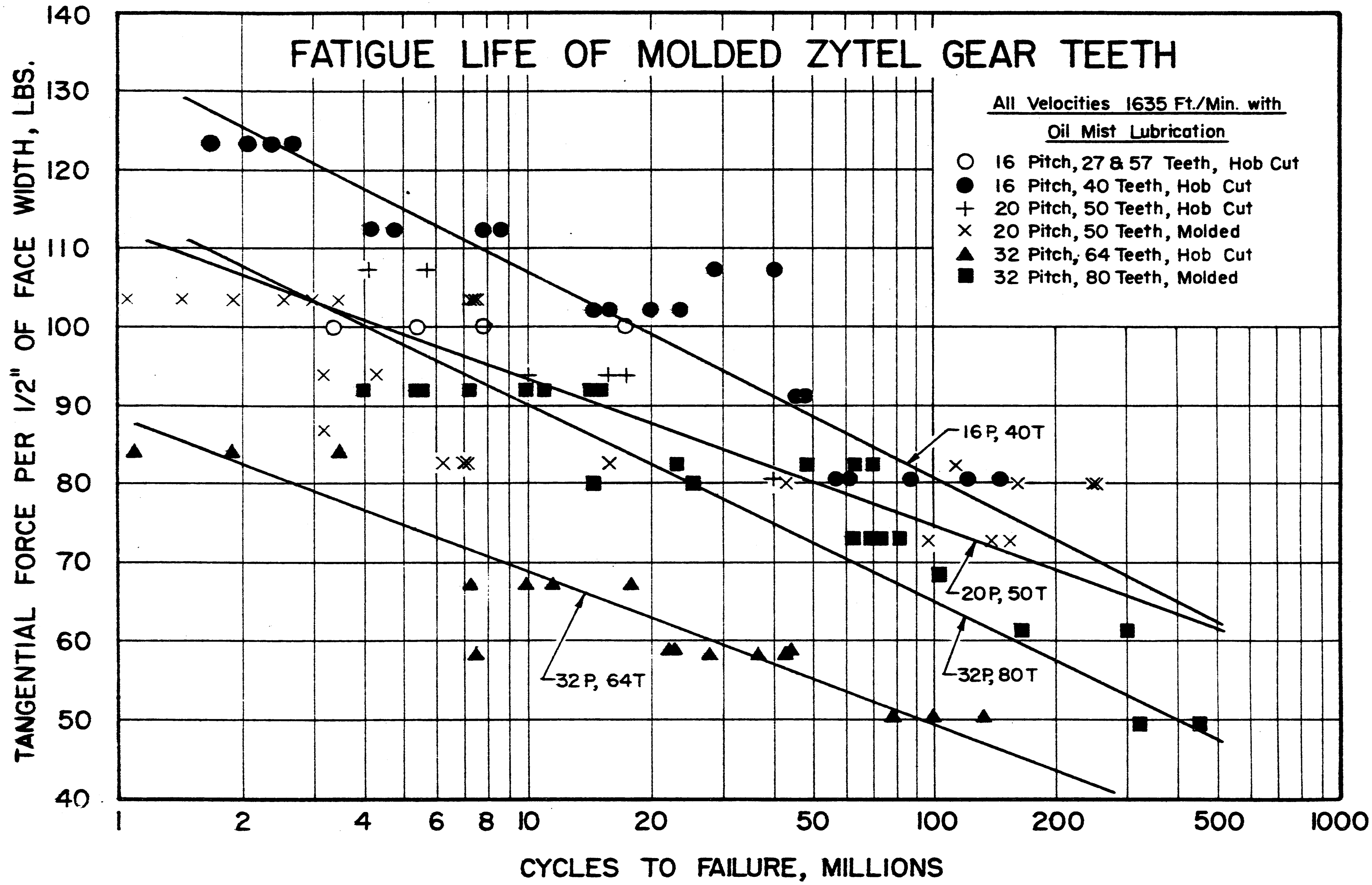


Fig. 9. Fatigue life of molded ZYTEL gear teeth.

20-pitch teeth were molded and some were cut, while all the 32-pitch teeth with a 2-1/2-in. pitch diameter were molded. Furthermore, both annealed and nonannealed 20-pitch molded-gear test results are shown in Fig. 9. As previously discussed, molded teeth, annealed, and with a suitably thick web, would probably raise the 16- and 20-pitch lines on Fig. 9. Even so, it is still evident that increasing the size of the teeth by decreasing the pitch does not bring about a corresponding increase in the load-carrying capacity of the teeth. This is most probably due to the load being shared by a larger number of teeth when small teeth are used, and to the lower operating temperature of the smaller teeth. The test results indicate, however, that this load-sharing and the operating temperature are not simple functions of the tooth geometry, such as the contact ratio, or length of the path of contact, but instead appear to be a function of the relative flexibility of the teeth. This type of analysis does not lend itself to a simple design procedure, and so has not been completely investigated to date. However, work on this analysis is continuing at the present time.

It is also interesting to note the lower line on Fig. 9 showing the results obtained with 32-pitch cut teeth with a 2-in. pitch diameter. It was previously mentioned that the molded 32-pitch teeth showed considerable improvement over the cut 32-pitch teeth. This, of course, is also shown in Fig. 9. It could be, however, that the better showing of the 32-pitch molded teeth was partly due to the difference in size of the cut and molded gears. The 2-in.-diameter gears rotate at a higher speed than the 2-1/2-in.-diameter gears to provide the same pitch-line velocity. The more frequent engagements at the higher speed should cause a higher temperature on the teeth of the 2-in.-diameter gear, and this might account for part of the difference in performance. At any rate, no simple geometric quantity, such as the contact ratio or length of path in contact, appears to influence directly this difference in the results obtained with the 2-in. cut gears and the 2-1/2-in. molded gears with the 32-pitch teeth.

The above difference in performance might also be influenced by the fact that the 2-in.-diameter cut gears were made from gears which had previously operated for some time as 16-pitch gears before being cut down to the 32-pitch gears. Some change in the material may have resulted from this previous operation.

7. FIRST METHOD OF CALCULATING LOAD-CARRYING CAPACITY OF ZYTEL SPUR GEARS

One of the main reasons for doing all the fatigue-testing was to establish a method of calculating the load-carrying capacity of ZYTEL spur gears. Since it has been reasonably well established that bending fatigue strength, rather than wear, determines the load-carrying capacity of ZYTEL teeth, any method used to calculate the load-carrying capacity need only to consider bending fatigue strength of the teeth. The first method of calculating the load-carrying capacity is the same as that presented in Progress Report No. 3. The

Lewis equation presented in Section 5 of this report can be rewritten to establish the torque or horsepower which a given gear can transmit, using an allowable stress obtained from the fatigue tests.

To establish the torque which a ZYTEL gear can transmit, the Lewis equation can be written as:

$$T = \frac{S D f Y K}{2 P} \tag{2}$$

where

- T = torque teeth can carry, lb-in.,
- S = allowable bending stress, psi, from Fig. 9 of this report, for molded and lubricated gears,
- K = design factor from Fig. 10 of this report,
- D = pitch diameter, in.,
- f = face width, in.,
- Y = form factor from Fig. 3 of this report, and
- P = diametral pitch.

To establish the horsepower which a ZYTEL gear can transmit, the Lewis equation can be written as:

$$hp = \frac{S D f Y N K}{126{,}000 P} \tag{3}$$

where

- hp = horsepower teeth can transmit, and
- N = gear speed, rpm.

For both of the above equations, the allowable stress can be determined from Fig. 10 for molded teeth which are lubricated. To provide a reasonable margin of safety, the lines of Fig. 10 are 25% lower than those which are considered to be representative of the test data in Fig. 8.

The stress factor "K" is selected from Table II and compensates for lack of lubrication, for teeth which have been cut rather than molded, and for velocities in excess of 4,000 ft/min. Other than those shown in Table II, no correction need be applied for velocity.

It must be kept in mind that Fig. 10 shows the maximum stresses which should be used to calculate the torque or horsepower capacity of molded ZYTEL teeth. These limiting values have been established for rather ideal conditions of operation, and for very accurately molded gear teeth. The designer must use reasonable judgment when applying these stresses to gears which are less accurately molded, or which operate under more severe conditions of impact, overload, elevated temperatures, etc. In such cases the stresses shown in Fig. 10 should be reduced to provide an additional margin of safety, before applying the factors of Table II.

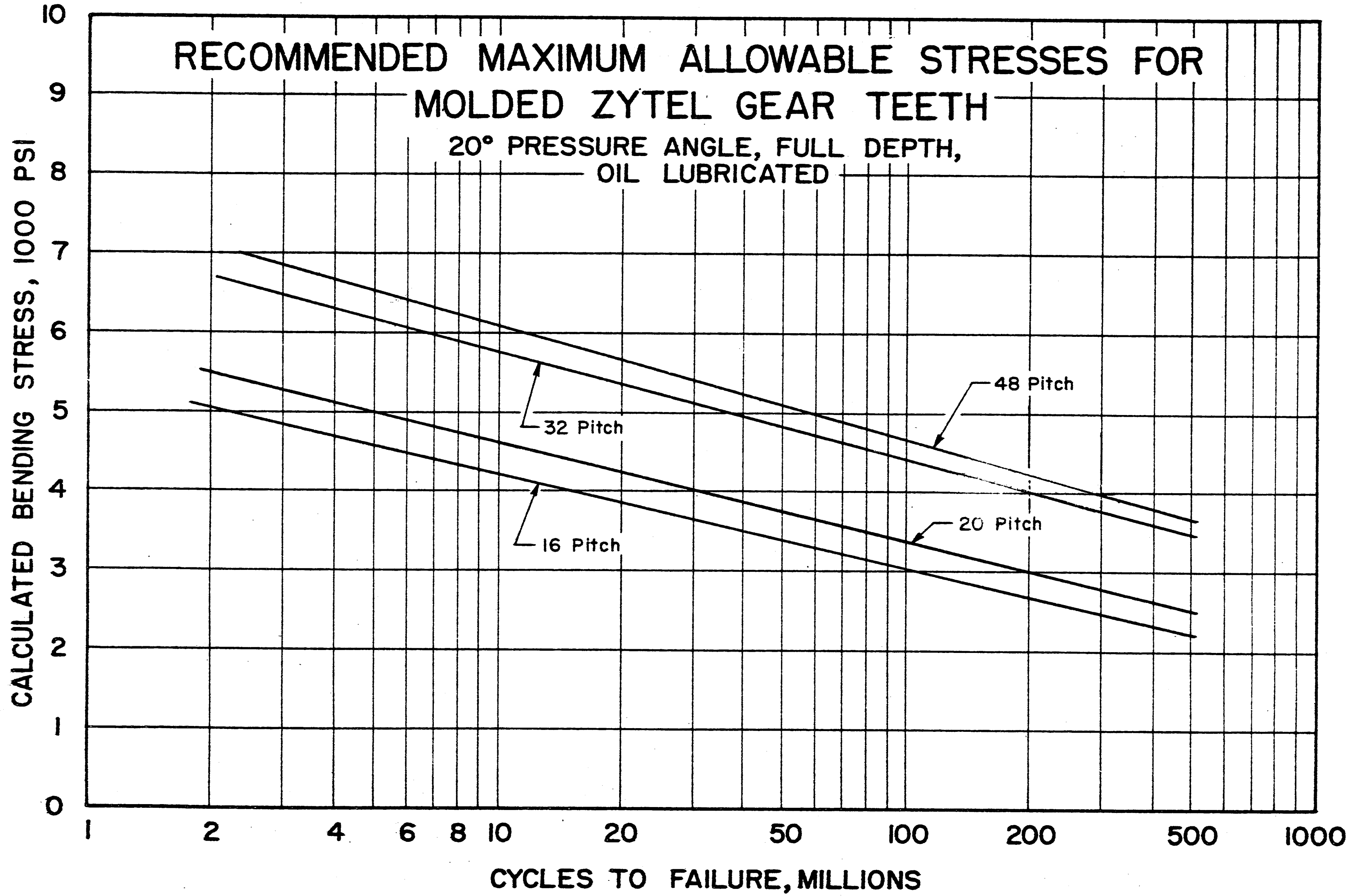


Fig. 10. Recommended maximum allowable stress for molded ZYTEL gear teeth.

TABLE II

VALUES OF DESIGN FACTOR K FOR USE
WITH LEWIS EQUATION AND DESIGN CHARTS

Teeth	Lubrication	Velocity, ft/min	Pitch	Factor K
molded	yes	below 4000	16-48	1.00
molded	yes	above 4000	16-48	0.85
molded	no	below 1635	16-20	0.70
molded	no	above 1635	16-20	0.50
molded	no	below 4000	32-48	0.80
cut	yes	below 4000	16-48	0.85
cut	yes	above 4000	16-48	0.72
cut	no	below 1635	16-20	0.60
cut	no	above 1635	16-20	0.42
cut	no	below 4000	32-48	0.70

To illustrate the use of this method of calculating the load-carrying capacity of ZYTEL teeth, the following examples are given:

Example I: A molded 20-pitch pinion and gear having 24 and 60 teeth, respectively, have a 1/2-in. face width. The pinion rotates at 1725 rpm. What horsepower can be transmitted for a life of about 100 million revolutions of the pinion if the teeth are lubricated and the loading is relatively steady?

From Fig. 10, S = 3350 psi for 20-pitch teeth at 100 million cycles.

From Table II, K = 1.0 for lubricated, molded teeth, with a velocity below 4000 ft/min.

The pitch diameter of the pinion = N/P = 24/20 = 1.20 in.

Then by Eq. (3), using the form factor Y for the 24-tooth pinion:

$$\text{hp} = \frac{3350 \times 1.20 \times 0.5 \times 0.572 \times 1725 \times 1.0}{126{,}000 \times 20} = 0.79 \text{ (answer).}$$

Example II: A torque of 25 lb-in. is to be carried by a 64-tooth, 32-pitch pinion meshing with a 100-tooth gear. The pinion turns at 3000 rpm, and some impact or shock loading is anticipated. These gears are to have cut teeth, and must operate without lubrication. Operation is to be intermittent, and so a total life of about 50 million revolutions of the pinion should be satisfactory. What face width is required?

From Fig. 10, S = 4800 psi for 32-pitch teeth at 50 million cycles. To allow for some impact, this should be reduced somewhat, possibly to about 3800 psi.

From Table II, K = 0.70 for cut teeth, not lubricated, and with a velocity below 4000 ft/min.

Then by Eq. (2), using the form factor Y for the 64-tooth pinion:

$$25 = \frac{3800 \times 2.0 \times f \times 0.719 \times 0.70}{2 \times 32}$$

$$f = \frac{25 \times 2 \times 32}{3800 \times 2.0 \times 0.719 \times 0.70} = 0.42, \text{ or about } 7/16 \text{ (answer)}.$$

Example III: The following helical gears were cut from ZYTEL bar stock, and were installed in an industrial speed reducer:

	Pinion	Gear
Diametral pitch	12	12
Number of teeth	19	57
Pitch diameter	1.750 in.	5.250 in.
Helix angle	23.556°	23.556°
Face width	2.25 in.	2.00 in.

To balance more closely the strength of the gear and pinion teeth, the outside diameter of the pinion was made 0.030 in. oversize and that of the gear 0.030 in. undersize. The gears are splash-lubricated, and the pinion rotates at 1760 rpm. Determine the horsepower the gears will be able to transmit for a life of approximately 200 million revolutions of the pinion.

Although the stresses of Fig. 10 are for spur gears with smaller teeth than those of the helical gears, it seems that a reasonable answer can be obtained for the helical gears. At 200 million cycles, Fig. 10 shows an allowable stress of 2700 psi for molded 16-pitch spur gears. Because the helical gears have cut teeth and a lower pitch, the allowable stress should be reduced below 2700 psi. However, helical gear teeth commonly tend to be stronger than spur gear teeth; hence the allowable stress is taken as 2700 psi, with a stress factor K of 1.0.

By laying out the teeth, the form factors for the oversize pinion and undersize gear were found to be 0.640 and 0.820, respectively. Then by Eq. (3):

$$hp = \frac{2700 \times 1.75 \times 2.25 \times 0.640 \times 1760}{126{,}000 \times 12} = 7.9 \text{ (answer)}.$$

To allow for some impact and overloading, this might be reduced to 7-1/2 hp (answer).

Although Example III shows the application of this method of design to 12-pitch teeth, which are larger than the teeth from which the test data were obtained, it seems unwise to apply the stresses shown in Fig. 10 to the design of teeth much larger than those tested. The combination of larger loads and greater sliding velocity on large teeth may result in much higher operating temperatures of the teeth. This in turn would reduce the strength of the ZYTEL, thus making the allowable stresses of Fig. 10 incorrect. Furthermore, the combination of large loads and high sliding velocity might well exceed some limiting value not reached with the smaller teeth, beyond which surface deterioration might limit the load-carrying capacity rather than bending strength. Because of the above reasons, it is felt that this design method, using the allowable stresses of Fig. 10, should not be applied to large teeth, such as those having a pitch of 8 or lower.

Caution should also be exercised if the allowable stresses of Fig. 10 are to be applied to teeth smaller than the 48-pitch teeth shown. Unless the dimensions are very carefully controlled, inaccuracies in such small teeth could greatly reduce the load-carrying capacity.

8. SECOND METHOD OF CALCULATING LOAD-CARRYING CAPACITY OF ZYTEL SPUR GEARS

This second method for calculating the load-carrying capacity of molded ZYTEL teeth is nothing more than the presentation of the Lewis equation and the allowable stresses in diagram form to simplify the procedure. Such diagrams are shown in Figs. 11-14 for the 16-, 20-, 32-, and 48-pitch teeth, respectively. With this method the factor K of Table II is used to increase the torque which the pinion must be designed to transmit, or to decrease the calculated torque capacity of the pinion.

The use of the diagrams is illustrated by the examples which follow.

<u>Example IV</u>: A 32-pitch molded pinion and gear are to transmit 1.5 hp at 1750 rpm of the pinion, and are to provide a 1.5:1 speed reduction. The gears will be lubricated, and a life of at least 100 million revolutions of the pinion is desired. What pitch diameter and face width should be used for the pinion?

It is first necessary to find the pinion torque:

$$\text{Pinion torque} = \frac{63{,}000 \times \text{hp}}{\text{rpm}} = \frac{63{,}000 \times 1.5}{1750} = 54 \text{ lb-in.}$$

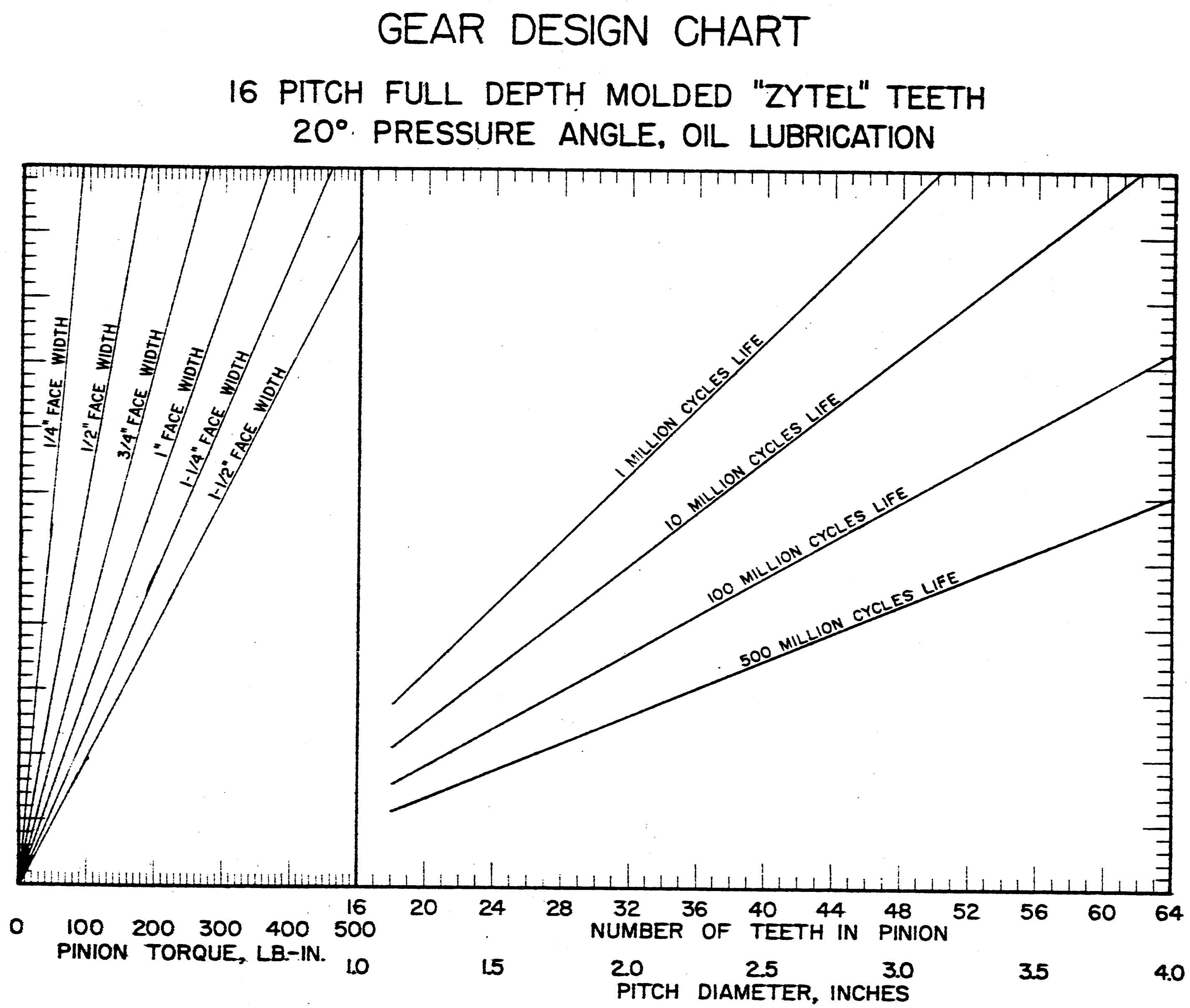


Fig. 11. Gear design chart, 16-pitch molded ZYTEL teeth.

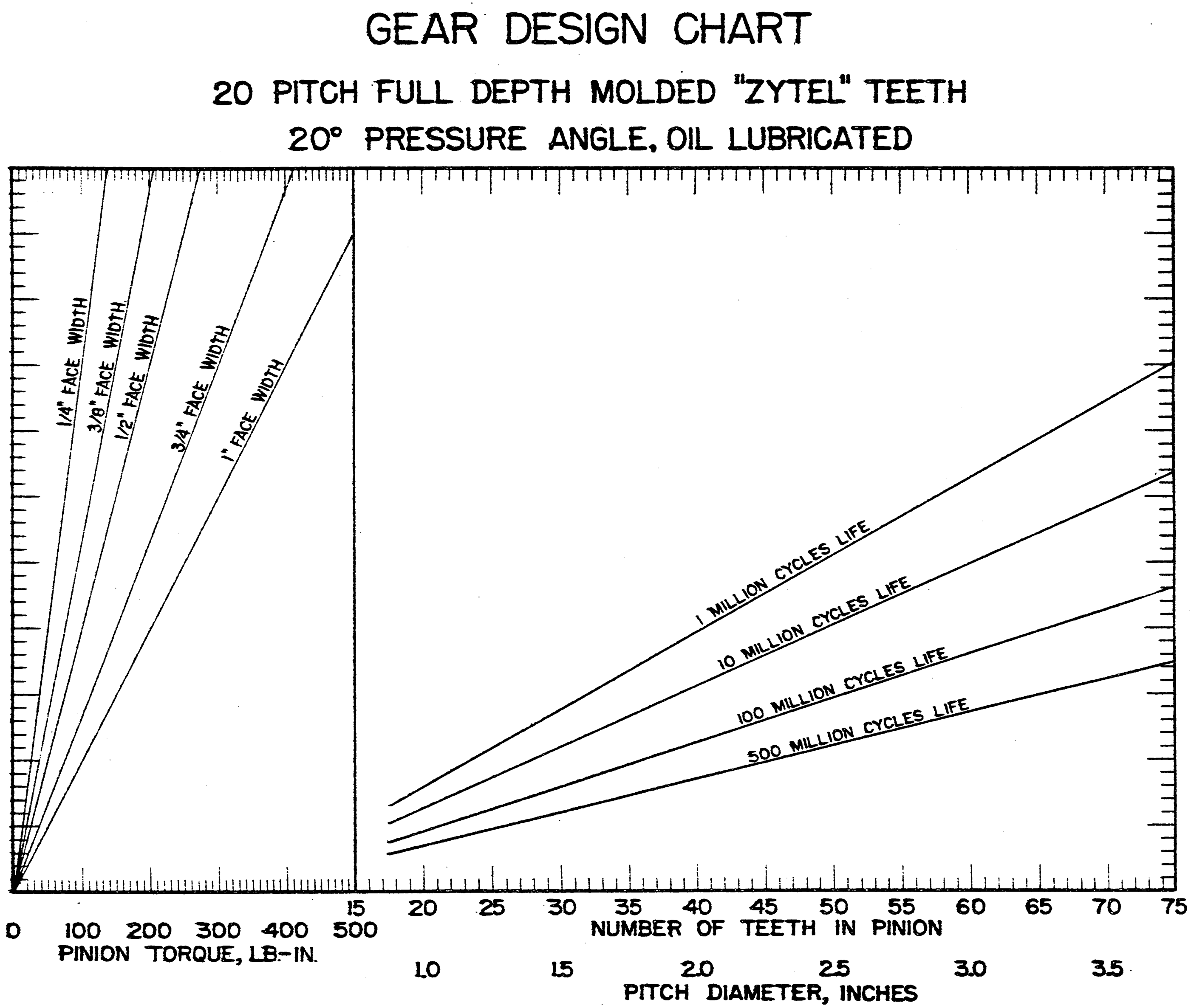


Fig. 12. Gear design chart, 20-pitch molded ZYTEL teeth.

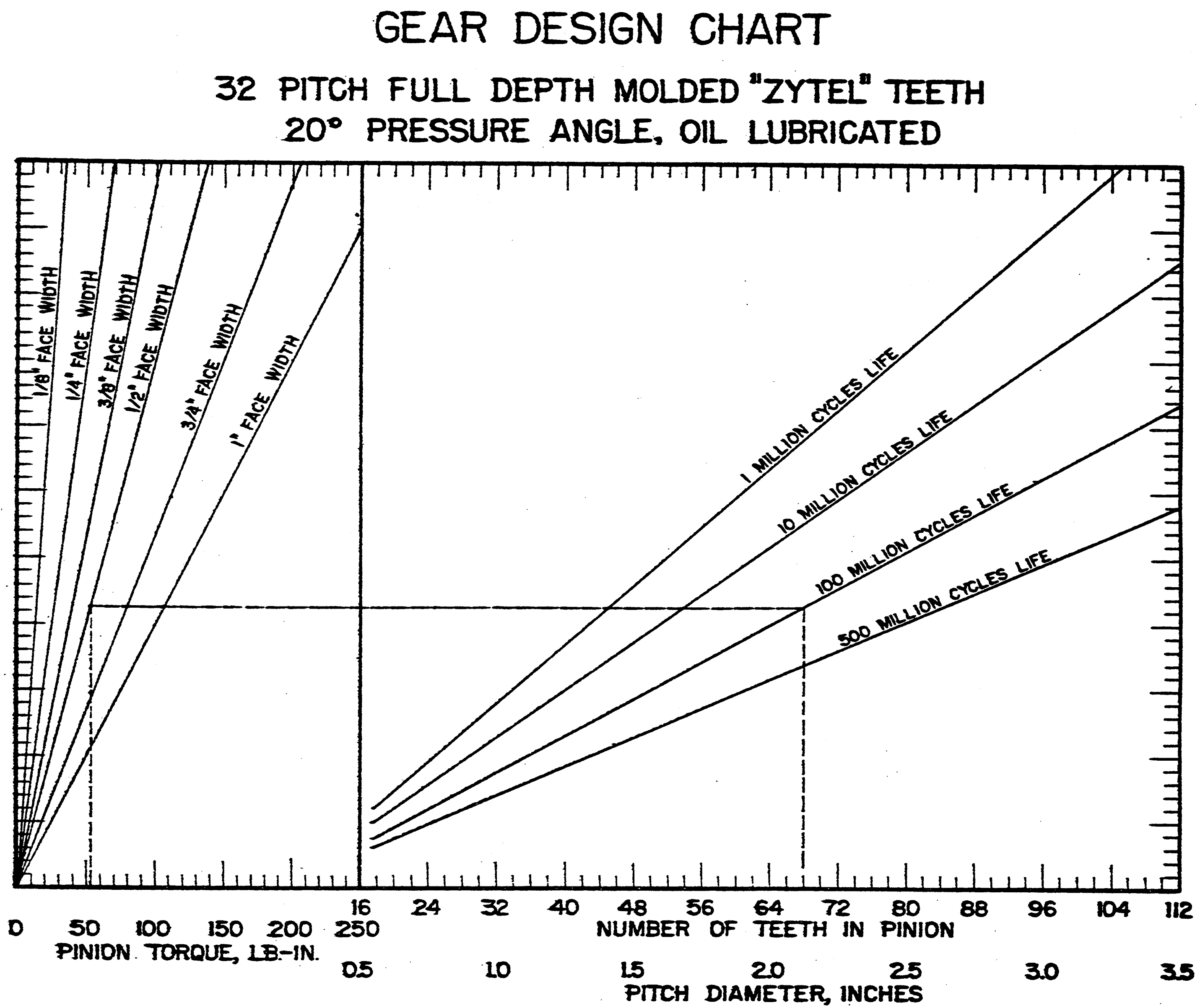


Fig. 13. Gear design chart, 32-pitch molded ZYTEL teeth.

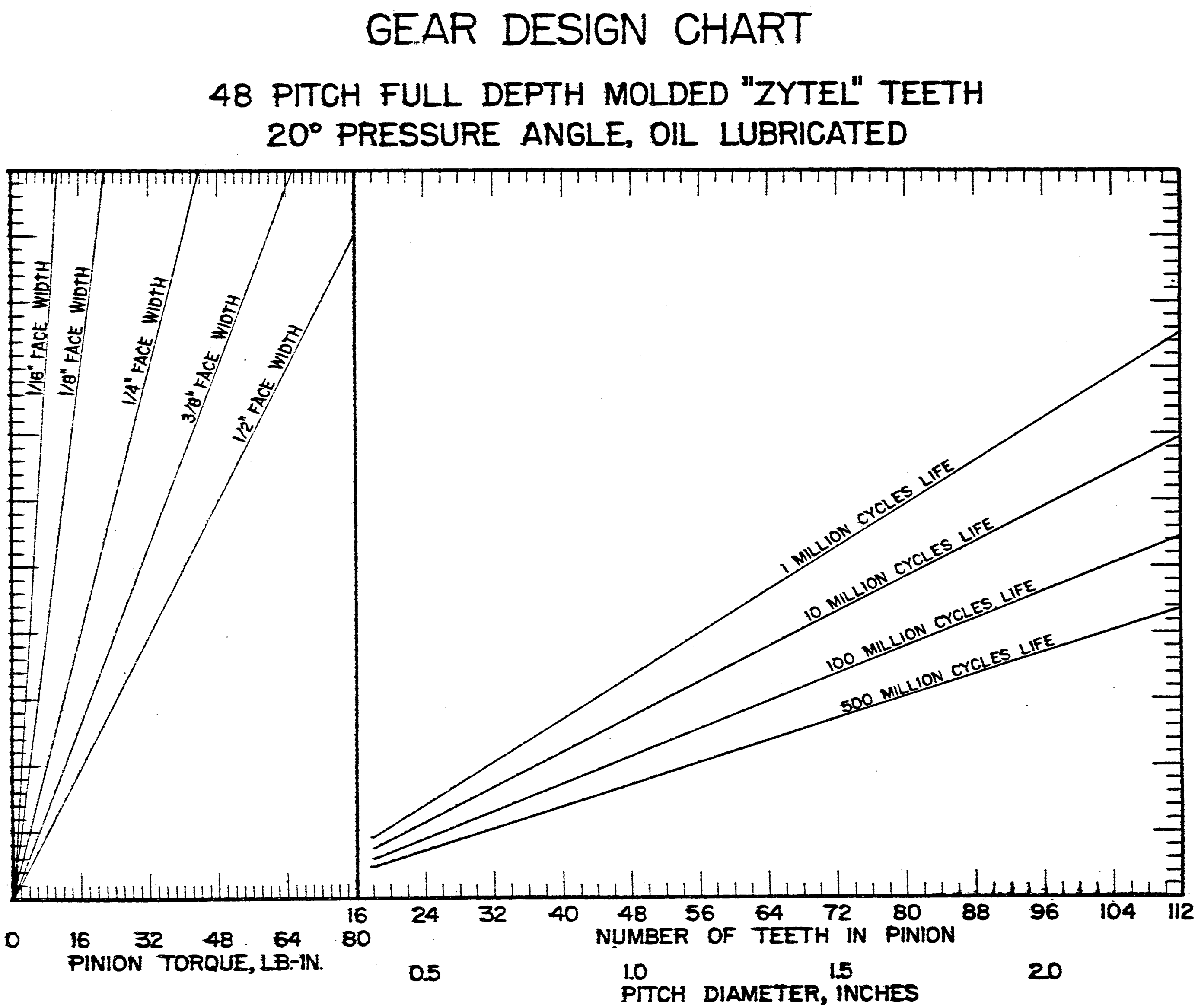


Fig. 14. Gear design chart, 48-pitch molded ZYTEL teeth.

Since these are molded and lubricated teeth, with a probable velocity of less than 4000 ft/min., the factor K from Table II is 1.0 and can be neglected.

Referring to Fig.13, the design chart for 32-pitch teeth, a vertical line is drawn at 54 on the horizontal torque scale. Trying a 1/2-in. face width, the vertical line is extended until it intersects the 1/2-in. face-width line. From this intersection a line is drawn horizontally to the right to intersect the 100-million-cycle life line, then vertically to establish the number of teeth in the pinion as 68, and the pitch diameter of the pinion as 2.125 in.

This solution is shown by dashed lines on Fig. 13. In a similar manner the pinion diameter required for other face widths could be determined, allowing the designer to make a choice of various combinations.

Example V: The gears of Example IV are to have a 1/2-in. face width, and are to operate without lubrication. What pitch diameter should be used for the pinion?

Table II shows a factor K of 0.80 for molded 32-pitch teeth not lubricated and with a pitch-line velocity below 4000 ft/min. Then:

$$\text{Design torque of pinion} = \frac{54}{0.80} = 67.5 \text{ lb-in.}$$

Using Fig. 13, the design chart for 32-pitch gears, a line is drawn vertically at 67.5 on the torque scale to intersect the 1/2-in. face-width line, then horizontally to the right to intersect the 100-million-cycle life line, then vertically to find the number of teeth to be 86, and the pitch diameter 2.688 in.

Example VI: A 16-pitch pinion with 40 teeth and a 3/4-in. face width is to be cut from ZYTEL bar stock. The teeth will be lubricated, and the loading is to be reasonably steady with occasional overloads. Service is to be intermittent, and a life of 10 million cycles will be sufficient. What torque can this pinion safely transmit?

Using Fig. 11, the design chart for 16-pitch teeth, a vertical line is drawn at 40 teeth to intersect the 10-million-cycle life line, is extended horizontally to the left to intersect the 3/4-in. face-width line, and downward to read 165 lb-in. of torque. Because the teeth are to be cut rather than molded, this must be reduced by a factor K of 0.85 from Table II, to 140 lb-in. To allow for some overloading, this might be further reduced to about 120 lb-in.

Example VII: A molded ZYTEL pinion is to have a 3/4-in. face width, and must transmit 100 lb-in. of torque for a life of 500 million cycles. The teeth will be lubricated. What pitch teeth and what pitch diameter should be used for this pinion?

Using the design charts for the 16-, 20-, 32-, and 48-pitch teeth, and a factor K of 1.0, the following results are obtained:

Pitch	No. of Teeth	Pitch Diam, in.
16	44	2.75
20	59	2.95
32	104	3.25
48	not feasible	not feasible

The above sizes are the minimum which should be used for the pitches shown, and should be increased at the discretion of the designer to take care of impact, overloads, elevated temperatures, etc. The most suitable choice of pitch and diameter can then be made so far as available space is concerned.

It will be noted from Fig. 14, the design chart for 48-pitch gears, that it is not feasible to carry a torque of 100 lb-in. with 48-pitch teeth, since a face width greater than 3/4 in. would be required. It is felt that the largest face widths shown on the design charts are the maximum which should be used.

In keeping with the discussion in Section 6 of this report concerning the relative load-carrying capacities of the various pitch teeth, it will be noted that there is not a great deal of difference in the size of the 16-, 20-, and 32-pitch pinions required to transmit the 100-lb-in. torque.

9. DESIGN RECOMMENDATIONS FOR ZYTEL SPUR GEARS

A number of design recommendations were listed in Progress Report No. 3. For easy reference, these are repeated here, plus some new suggestions. These design recommendations are in addition to the data on load-carrying capacity which are presented in Sections 7 and 8 of this report.

a. Size of Teeth.—As with metal gears, it is desirable to use the smallest teeth reasonably possible which are strong enough to carry the load. The discussion at the end of Section 7 concerning the limitations of the load-carrying calculations should be considered when determining the size of teeth to be used.

b. Pressure Angle.—The 20° pressure angle appears to be quite satisfactory. Increasing the pressure angle provides thicker teeth, but the resulting higher tooth temperatures tend to cancel any strength gained by the increased thickness.

c. Tooth Form.—The full-depth tooth form appears to be more desirable than the stub form. The full-depth teeth may be strengthened by making the blank oversized, but this should not be carried to extremes, since the strength gained by making the tooth thicker at its base will be more than offset by the weakening effect of higher temperature caused by the increased normal force and sliding velocity of the teeth.

d. Face Width.—Almost any reasonable face widths should be satisfactory, since the performance does not seem to be affected by the face width. What are felt to be reasonable maximum values for the face widths of 16-, 20-, 32-, and 48-pitch gears are shown on the gear design charts.

e. Accuracy of Manufacture.—Since the life of the gears seems to be shortened when the teeth are not accurately formed and spaced, it is desirable to have the teeth as accurately made as possible.

f. Backlash.—Backlash must be provided, but performance does not seem to be affected by reasonable variations in backlash. Recommended backlash, measured at room temperature, is approximately 0.004 to 0.006 in. for 16-pitch teeth, 0.003 to 0.005 in. for 20-pitch teeth, and 0.002 to 0.004 in. for 32- and finer pitch teeth. For high-speed or very heavy load operation, which may heat the teeth, the backlash should be somewhat more liberal than the values listed above.

g. Gear Proportions.—Rims, webs, and hubs should be rather generously proportioned to provide ample support for the teeth. This is especially important if the teeth are to be heavily loaded. In general, the rim thickness beneath the root of the teeth should be at least three times the thickness of the tooth at the pitch circle, the web thickness should be about equal to the rim thickness, the hub diameter at least 1-1/2 times the shaft diameter, and the hub length at least about equal to the shaft diameter. If the gear is large and the teeth are to be heavily loaded, it is better to mount the ZYTEL gear on a metal flange rather than to key or spline the gear directly to the shaft. The gear should be designed to reduce or eliminate the formation of residual stresses, and the mold should have a ring gate to maintain concentricity of the gear.

h. Treatment After Molding.—The gears should be fully annealed after molding if there is any evidence of residual stresses being present. It is better to design the gear to reduce or eliminate the formation of these stresses, as mentioned above, rather than to rely on the annealing to remove the stresses, since the annealing process may affect the accuracy of the teeth if the residual stresses are not uniformly distributed.

i. Lubrication.—Lubrication is desirable, especially for rather coarse teeth carrying heavy loads. A generous supply of lubricating oil can keep the gear-tooth temperatures down, and hence might increase the useful life of the teeth.

10. PHOTOGRAPHIC INVESTIGATION OF LOAD DISTRIBUTION

As previously mentioned, it is felt that the relatively greater load-carrying capacity of the smaller teeth is partly due to the load being carried by more than one tooth at all times. A photographic analysis made with the teeth loaded, but with the gears stationary, was described in Progress Report No. 2. This analysis indicated that when 16-pitch teeth are used, one tooth carries the entire load when in contact near the pitch point. With 20-pitch teeth, it appears that one tooth carries essentially all the load when in contact near the pitch point, with possibly a small part of the load being carried by the adjacent teeth.

During the period covered by this report, a second photographic analysis was made while the gears were running at various pitch-line velocities. When photographing the gears while they are running, it is not possible to set the gears in any desired position; hence a rather large number of pictures were taken to obtain a few that would show the load distribution when one tooth is in the position where it is most heavily loaded. It was found to be difficult to obtain pictures that were clear enough to show the load distribution when the teeth were moving with a very high velocity, and this prevented any worthwhile photographs of the 32-pitch teeth from being obtained.

The photographs of the 20-pitch teeth pretty well agreed with the photographs previously taken when the gears were not turning. When in contact near the pitch point, one tooth carries essentially all the load, with some small part being carried by the adjacent teeth. The tooth profile appeared to flatten under contact to distribute the load over a rather large surface area of the tooth, and there appeared to be some evidence of the tip of one tooth digging into the flank of the other. This is illustrated in Fig. 15. The flattened contact area of the number 14 teeth on each gear, and the tip of one of the number 15 teeth apparently digging into the flank of the other number 15 tooth, are visible. The calculated bending stress in the teeth was 6000 psi, and the pitch-line velocity was 785 ft/min, when the photograph (Fig. 15) was taken.

11. INVESTIGATION OF RESIDUAL STRESS IN MOLDED GEARS

Evidence of residual stress in the rim of the molded gears was described in Sections 3 and 6 of this report. Such stresses would result from the thin web solidifying more rapidly than the thicker rim after molding, thus restricting the contraction of the rim as it solidifies. An attempt was made to establish the approximate magnitude of this residual stress in the rim.

A small radial slot was milled completely through the rim of 7 gears, and the width of each slot was measured and recorded. The web and hub of the gear were then machined away, leaving only the rim with its slot. The width of the slot was again measured to see how much the rim had contracted when the web was removed. Figure 16 illustrates the way the gear was slotted and machined to remove the web. Care was taken to cut the ZYTEL slowly so that its temperature would not be sufficiently increased to influence the residual stresses.

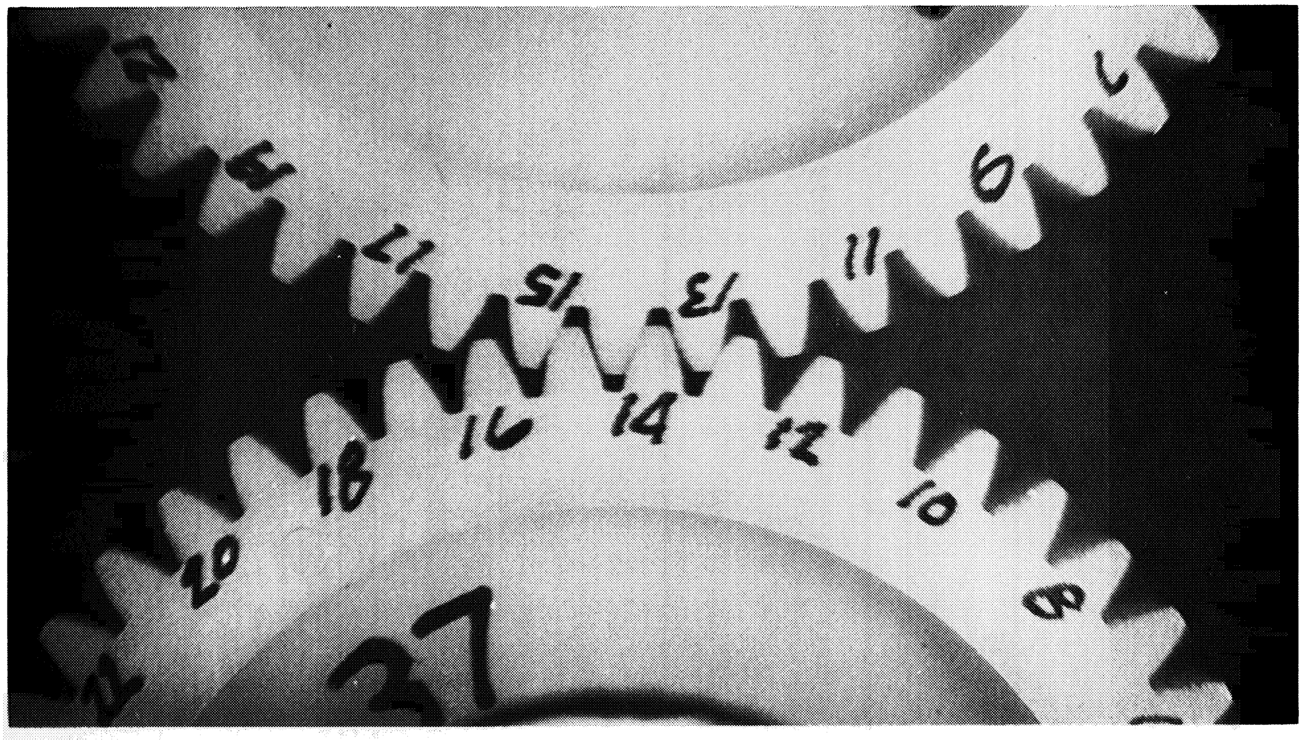


Fig. 15. 20-pitch ZYTEL teeth meshing under load.
ZYTEL 101, 1500 rpm, 44 lb/bucket (6000 psi).

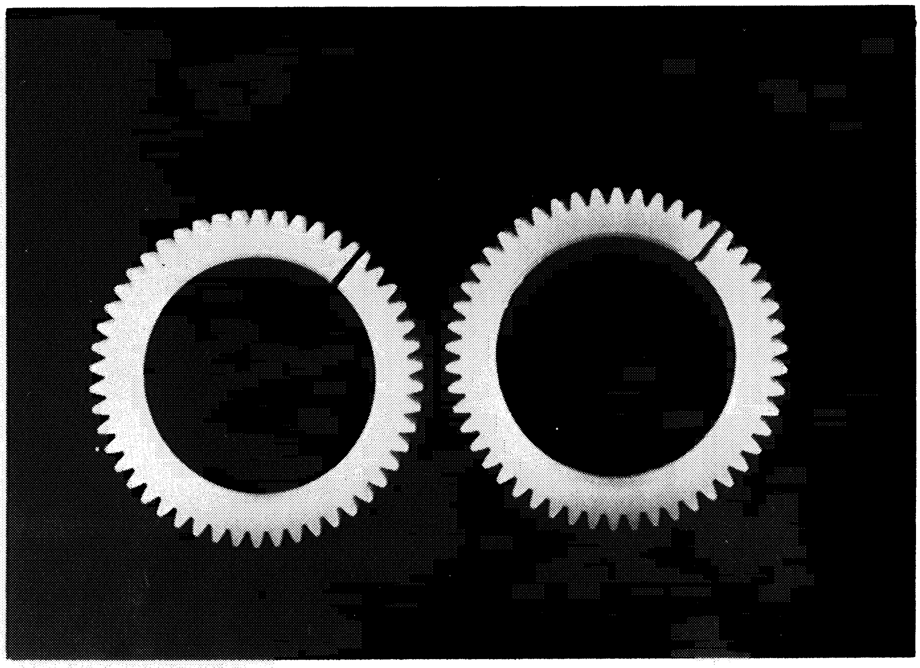


Fig. 16. Slotted rims of 20-pitch ZYTEL gears.

Table III shows the data obtained, and the calculated residual stresses for the 7 gears tested.

TABLE III

CALCULATED RESIDUAL STRESS IN RIMS OF 20-PITCH GEARS

Gear Number	Slot Width Before Web Was Removed, in.	Slot Width After Web Was Removed, in.	Change in Slot Width, in.	Calculated Residual Stress,psi
26	0.1010	0.0570	0.0440	1305
4	0.1002	0.0355	0.0647	1915
13	0.0960	0.0420	0.0540	1600
15	0.1009	0.0470	0.0539	1595
30	0.1001	0.0605	0.0396	1172
1	0.1007	0.0720	0.0287	848
17	0.0945	0.0630	0.0315	932

The stresses shown in Table III were calculated in the following manner. By definition,

$$E = \frac{S}{\text{unit strain}} ;$$

then

$$S = E \cdot \text{unit strain}, \tag{4}$$

where

$$S = \text{stress, psi,}$$

$$E = \text{modulus of elasticity, psi,}$$

$$\text{unit strain} = \frac{\text{change in slot width}}{3.14159 \times \text{mean diam of rim}} .$$

A modulus of elasticity of 200,000 psi and a mean rim diameter of 2.15 in. were used in Eq. (4) to calculate the stresses shown in Table III. These stresses should be a reasonable indication of the residual stresses which were present in the rims of the molded gears, and point out the need for careful mold design to prevent the formation of the residual stresses, or for annealing at a temperature sufficiently high to remove the residual stresses.

12. WEAR MEASUREMENTS OF MOLDED ZYTEL GEARS

The results of a rather extensive wear testing program with hob-cut ZYTEL gears were presented in Progress Report No. 2. These tests indicated that wear was generally insignificant during the life of the teeth. It was assumed that wear of the molded ZYTEL teeth would also be insignificant, and would not have to be considered in any method established to calculate the load-carrying capacity of the teeth. However, to be sure that this was so, a number of the molded gears were inspected at the completion of their fatigue-testing. Since these same gears had not been inspected when new, the results of the final inspection were compared to the average inspection data for new gears to obtain approximate values of wear. The average inspection data for new gears are shown in Section 3 of this report.

The results of this wear investigation are shown in Tables IV and V. Profile wear was determined from the profiles traced out by the Fellows Involute Profile Checker. The maximum profile wear shown is the average maximum variation of the profiles of three teeth on each gear from an average profile established for new gear teeth. The composite error data were obtained with the Kodak Conju-Gage.

The wear data of Tables IV and V confirm the conclusion that wear is not an important consideration in ZYTEL teeth, at least in the range of tooth sizes investigated to date.

13. WEAR MEASUREMENTS OF CUT ZYTEL AND STEEL GEARS

All the fatigue-testing previously described in this report was done with pairs of ZYTEL gears meshing together. In addition, some test work is being done with ZYTEL gears meshing with steel gears. The purpose of this test work is to investigate the rate of wear of the ZYTEL teeth, and to see how this rate of wear is influenced by the surface finish of the steel gear teeth. The 16-pitch gears with hob-cut teeth are being used for this investigation. The ZYTEL pinion has 27 teeth, and the steel gear has 40 teeth, both with 7/16-in. face width. The steel gears used so far have an average surface finish, measured radially, of 13.1 rms. The pitch-line velocity is 1635 ft/min, and the teeth are lubricated. Calculated bending stress in the ZYTEL teeth is 3050 psi.

The steel gears are mounted on the front shaft of a test machine, and the ZYTEL gears on the rear shaft. Thus one ZYTEL gear simulates the action of a driving gear, and the other the action of a driven gear. The ZYTEL teeth were inspected when new, and are inspected again after each 10 million cycles of operation. To increase the number of test data obtained, two pairs of ZYTEL gears are being alternately tested with the same pair of steel gears.

At the time of writing, the two pairs of ZYTEL gears have each completed 20 million cycles of operation, with results as shown in Figs. 17 and 18.

Definite wear patterns appear to be developing in the driven gears. However, the data indicate that the driving gears are growing in size. Additional test work may serve to clarify this behavior.

TABLE IV

WEAR DATA, 20-PITCH MOLDED ZYTEL GEARS

Gear No.	Moisture-Conditioned	Lubricated	Cycles of Operation, Millions	Bending Stress, psi	Maximum* Profile Wear, in.	Increase* in T.C.E.	Increase* in T.T.C.E.
1	no	yes	97	4200	0.0022	0.0012	-0.0001
2	no	yes	97	4200	0.0013	0.0012	0.0005
3	no	yes	154	4200	0.0019	0.0005	-0.0003
4	no	yes	154	4200	0.0029	-0.0008	-0.0001
5	no	yes	250	4620	0.0023	0.0000	-0.0004
6	no	yes	250	4620	0.0015	0.0012	0.0000
7	no	yes	250	4620	0.0020	-0.0008	-0.0004
8	no	yes	250	4620	0.0014	-0.0002	-0.0006
9	yes	no	122	2990	0.0018	0.0002	0.0000
10	yes	no	122	2990	0.0015	0.0011	0.0009
11	yes	no	131	2990	0.0018	0.0016	0.0003
12	yes	no	131	2990	0.0016	0.0008	-0.0005

*Profile wear is based on average profile of new neeth. T.C.E. = total composite error. T.T.C.E. = tooth-to-tooth composite error. Increase in composite errors is based on average composite errors of new teeth. Negative signs indicate decrease of error compared to average of new teeth.

TABLE V

WEAR DATA, 32-PITCH MOLDED ZYTEL GEARS

Gear No.	Moisture-Conditioned	Lubricated	Cycles of Operation, Millions	Bending Stress, psi	Maximum* Profile Wear, in.	Increase* in T.C.E.	Increase* in T.T.C.E.
1	yes	yes	165	5300	0.0008	-0.0003	-0.0001
2	yes	yes	165	5300	0.0007	0.0009	0.0001
3	no	yes	322	5300	0.0015	0.0018	0.0002
4	no	yes	322	5300	0.0016	0.0004	0.0000
5	yes	yes	103	5920	0.0009	0.0023	0.0004
6	yes	yes	103	5920	0.0002	0.0013	0.0004
7	no	yes	82.5	6300	0.0011	0.0005	0.0001
8	no	yes	82.5	6300	0.0008	0.0020	0.000
9	yes	no	78.8	4900	0.0004	0.0008	-0.0001
10	yes	no	78.8	4900	0.0006	0.0016	0.0001
11	no	no	59.6	4900	0.0003	0.0008	-0.0002
12	no	no	59.6	4900	0.0004	0.0009	0.0000

*Profile wear is based on average profile of new teeth. T.C.E. = total composite error. T.T.C.E. = tooth-to-tooth composite error. Increase in composite errors is based on average composite errors of new teeth. Negative signs indicate decrease of error compared to average of new teeth.

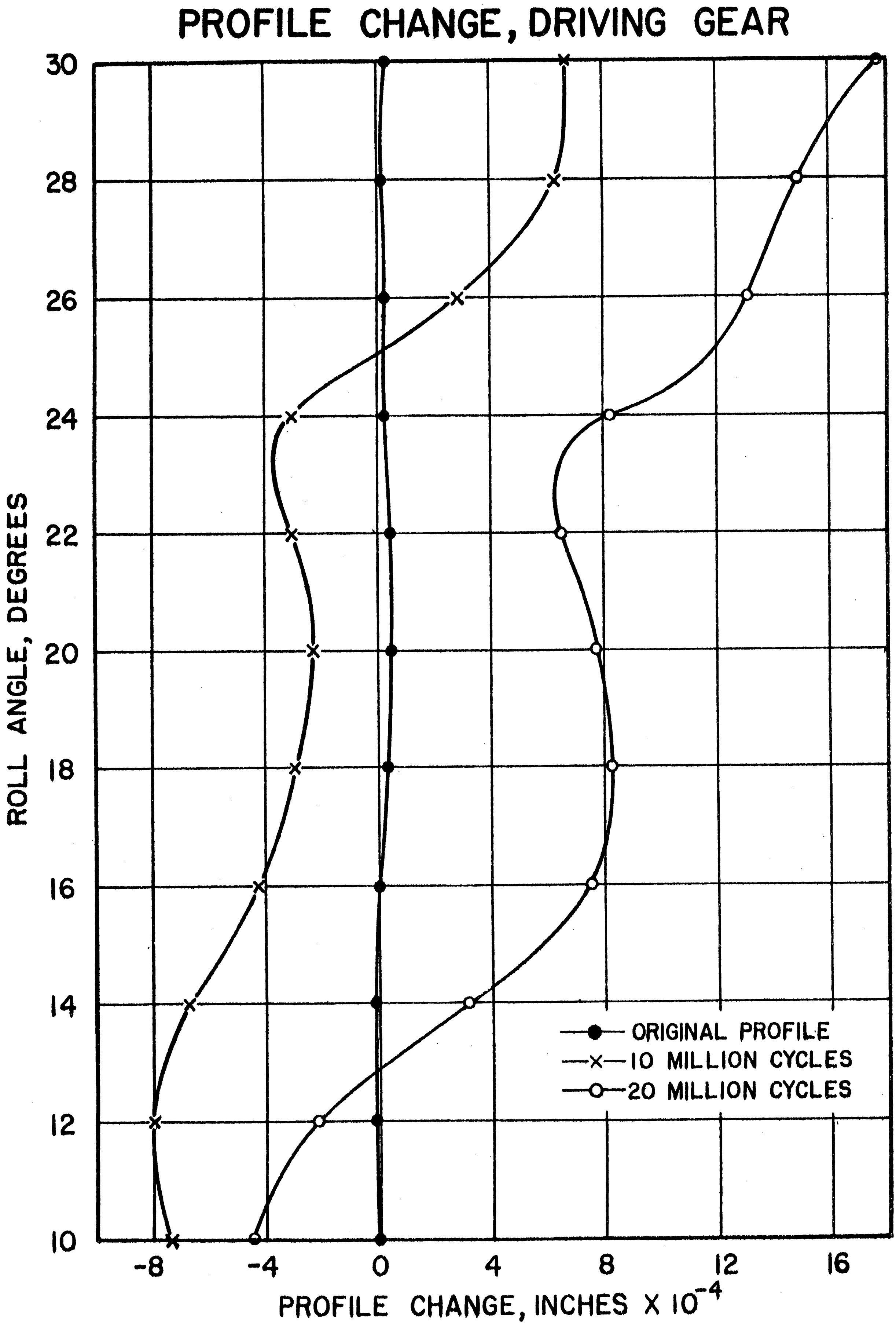


Fig. 17. Profile change of driving gear, ZYTEL and steel.

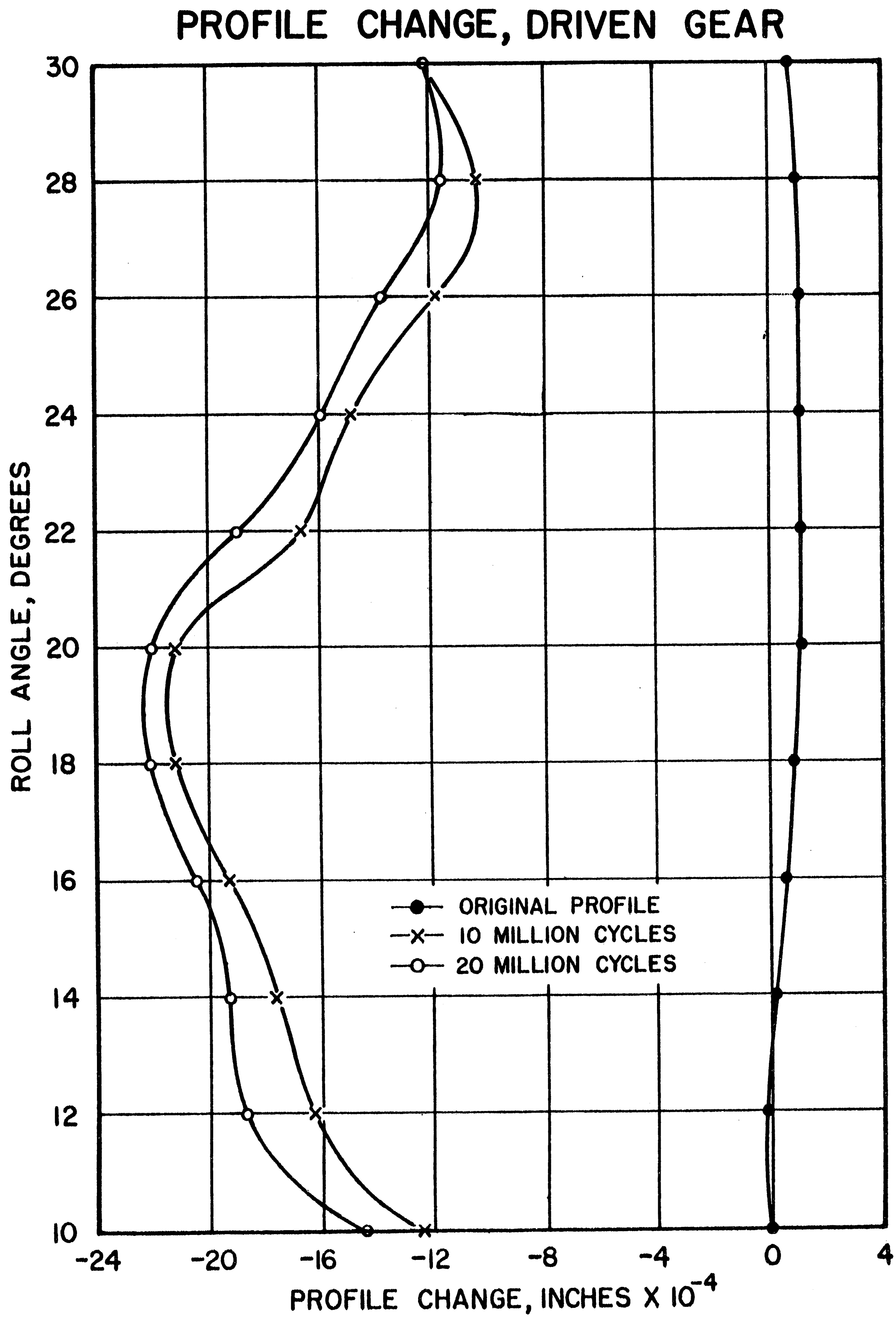


Fig. 18. Profile change of driven gear, ZYTEL and steel.

Some evidence of localized surface failure has started to show up on the ZYTEL gears. The surface compressive stress in the ZYTEL teeth is much higher when meshing with a steel gear rather than with a ZYTEL gear. This is due to the much larger modulus of elasticity of steel. Thus it may be that surface strength, rather than bending strength, will limit the load-carrying capacity of ZYTEL teeth meshing with steel. Testing of these same gears is being continued, and the surface will be closely watched for further deterioration.

Hob-cut teeth are being used for this test work because these gears were made before the molded gears became available. Similar work with the molded gears will follow. The present gears were made by cutting down broken 2-in. diameter 32-pitch cut gears; hence they are smaller than the steel gears. This has the advantage of having the ZYTEL gears turning faster than the steel gears, thus reducing the time required to obtain a large number of cycles of operation.

14. FATIGUE-TESTING OF LARGE HELICAL ZYTEL GEARS

To obtain some information on the application of the fatigue-test data to the design of gears other than the sizes tested, and operating in something other than the test machines, a Boston helical gear speed reducer, Model 1350, was purchased. The original speed reducer provided a 3.06:1 speed ratio, using 10-pitch helical gears made of steel, and had a 10-hp rating at 1760 rpm of the pinion shaft. The 17-tooth steel pinion had a 2-1/2-in. face width, while the face width of the 52-tooth steel gear was 2-1/4 in.

The steel pinion and gear were replaced by a gear and pinion cut from ZYTEL bar stock. The 19-tooth ZYTEL pinion and 57-tooth ZYTEL gear had 12-pitch helical teeth, and face widths of 2.25 in. and 2.00 in., respectively. The ZYTEL pinion is 0.030 in. oversize on the diameter, and the gear is 0.030 in. undersize. The center distance is 3.500 in. These gears are shown in Fig. 19. The pinion has two keyways 180° apart, while the gear is made to be flange-mounted to eliminate the high local stresses which would result if a key were used. Example III in Section 7 of this report shows how the load-carrying capacity of the helical pinion was established.

Figure 20 shows the gears installed but with the cover removed. The reducer is driven by a 10-hp, 1750-rpm induction motor, and is loaded by means of a Vickers vane pump discharging through a throttling valve. The motor was calibrated with a dynamometer to establish the torque-amperes curve; thus the torque delivered when driving the reduction gear can be determined by measuring the line current to the motor. A double-strand 1/2-in. pitch roller chain drive, visible in Fig. 20, is used to provide a pump speed of about 1200 rpm.

The unit, transmitting 7-1/2 hp at a pinion speed of 1760 rpm, operated almost continuously for 2090 hours before failure of several pinion teeth occurred. This represented 221 million revolutions of the pinion, or cycles of tooth-loading, at a calculated pinion tooth-bending stress of 2560 psi, and is in substantial agreement with the predicted life based on our "four-square"

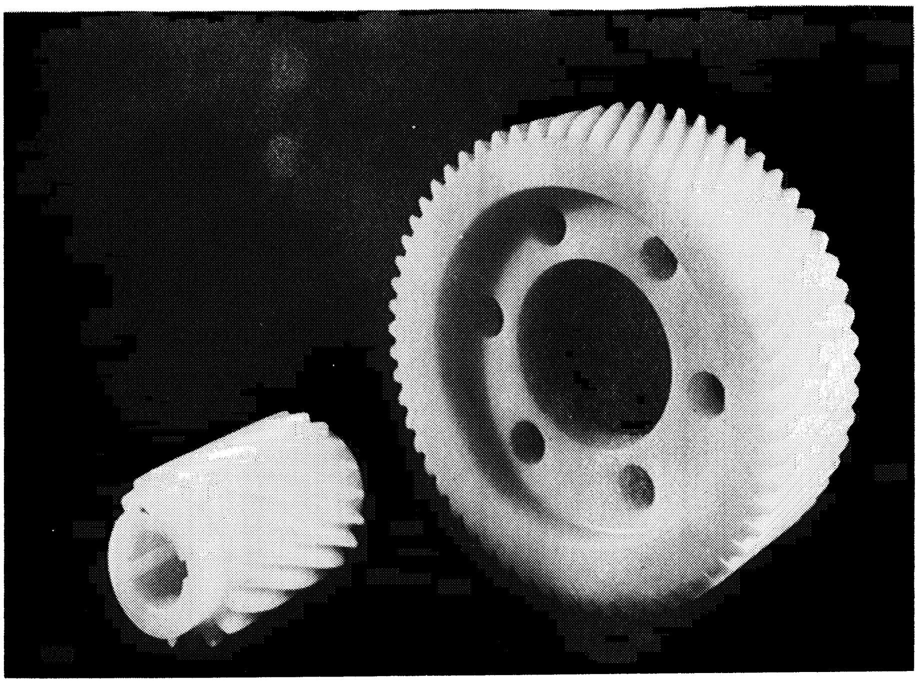


Fig. 19. ZYTEL helical pinion and gear.

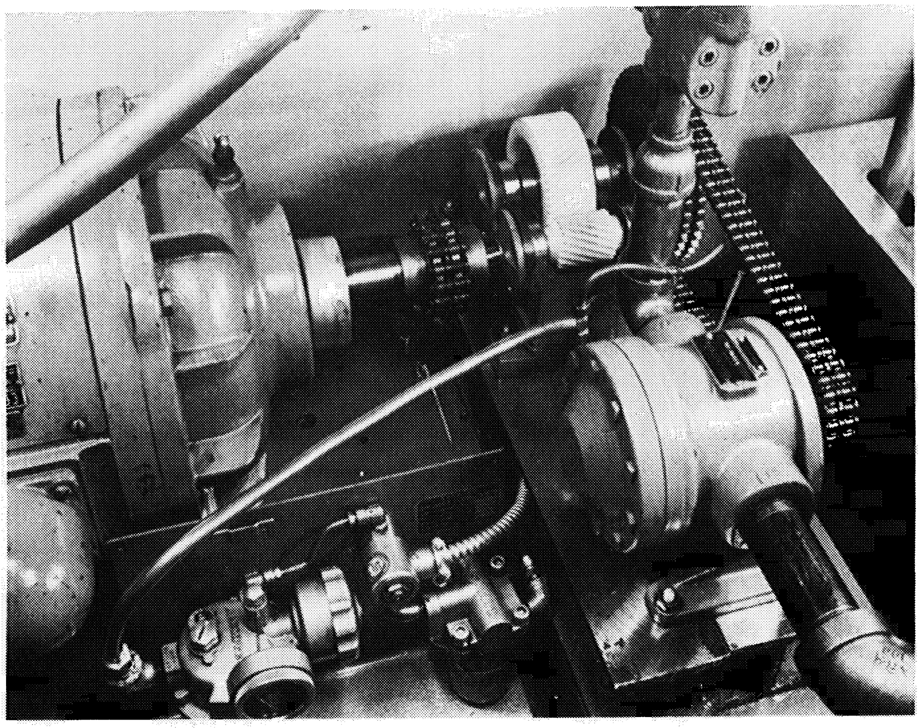


Fig. 20. Helical gear speed reducer.

test-machine fatigue data. Under the test conditions the calculated bending stress in the gear teeth was 2240 psi. The lubricating oil in the speed-reduction unit maintained an average temperature of approximately 140°F during the test period.

Improper lubrication of the roller chain during the early part of the test resulted in failure of the large cast iron chain sprocket mounted on the output shaft of the speed reducer. Wear allowed the chain to elongate, thus contacting the sprocket teeth near their outer end and causing the teeth of the sprocket to be sheared off. This failure took place at night, while the machines are not attended; hence the reducer continued to run for some time as the sprocket teeth failed. By the time it was detected, all the sprocket teeth were broken off, and the reduction gear was still running, but the entire test stand was vibrating violently.

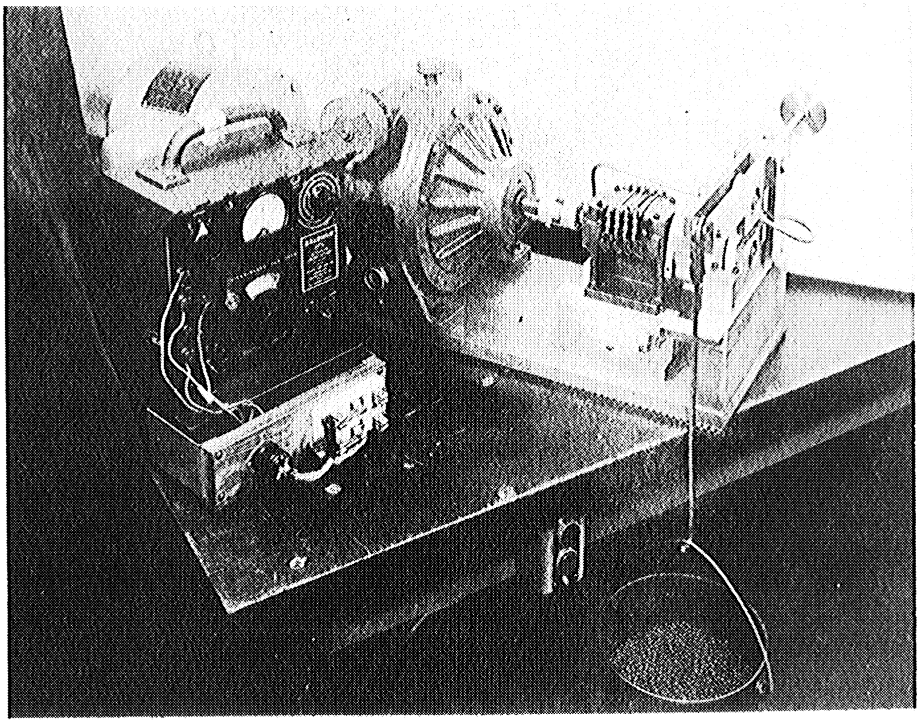
Visual inspection of the gears showed no damage to the teeth, so a new sprocket and chain were installed, and the test continued with the same gears. An oil-mist lubricator, also visible in Fig. 20, was installed to lubricate the new chain. After 1515 hours of operation, the cover of the reduction gear was removed to take the picture shown in Fig. 20. At this time it was found that some of the lubricating oil had leaked out of the reduction gear housing. This had lowered the oil level to such an extent that the gears and bearings were no longer lubricated, but instead were running perfectly dry, covered with a generous coating of red ferrous oxide formed by fretting corrosion of the tapered rolling bearings. Some of the teeth appeared to be slightly worn, and were permanently discolored, but were otherwise unharmed. The entire assembly was cleaned up, new lubricating oil was put in the housing, and the test continued with the same gears.

It can be readily seen that these adverse conditions have provided a more rugged test for these gears than had ever been anticipated.

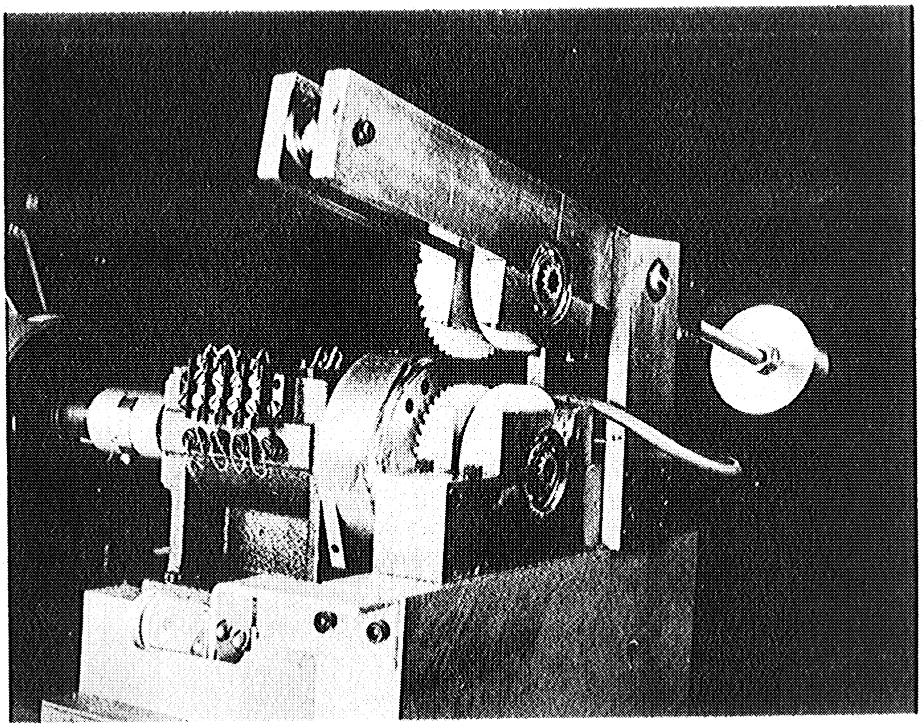
15. INVESTIGATION OF PRESSURE, SLIDING VELOCITY, FRICTION, AND WEAR

To investigate the relationship of contact pressure, sliding velocity, friction, and wear, a device was built which provides a combined rolling and sliding motion of two ZYTEL discs in contact under load. For lack of a better name, this has been called a rolling-sliding contact test machine.

The construction of this device is illustrated in Figs. 21a and b. Essentially, if consists of two ZYTEL discs mounted on parallel splined shafts, with two meshing ZYTEL 16-pitch 40-tooth gears on these same shafts. Thus the two shafts rotate at the same speed, driven by the meshing gears. If the two discs have different diameters, the surfaces of the discs must slide as they roll together. The sliding velocity depends on the difference in the disc diameters, and the speed of shaft rotation.



a



b

Fig. 21. Rolling-sliding contact test machine.

Load is applied to the discs by hanging a bucket of lead shot on the end of the pivoted arm as shown in Fig. 21a. The gears mesh with liberal backlash; hence all the load is carried on the discs.

A flexure plate drive, described in detail in Progress Report No. 1, was removed from one of the test machines and installed in this test device to measure the torque required to drive the sliding discs. A 1200-rpm motor and speed variator complete the setup. Lubrication for the discs can be supplied by an oil-mist lubricator.

Some preliminary tests have indicated that the test device performs satisfactorily, and some data have been obtained. Figure 22 shows a plot of the coefficient of sliding friction for various sliding velocities with three different conditions of lubrication. The coefficient of sliding friction is calculated as follows:

$$f = \frac{T}{F(r_1 - r_2)} \tag{5}$$

where

f = coefficient of sliding friction,
T = torque required to drive the device, lb-in.,
F = normal force on discs, lb, and
r_1 and r_2 = disc radii, in.

The dips in the curves may indicate the existence of an optimum combination of pressure and sliding velocity so far as friction is concerned, but the data obtained to date have not been sufficient to draw any conclusions. This test program is continuing at the present time, and shows a promise of producing results which may be applicable to the design of various types of gearing.

16. PRELIMINARY INVESTIGATION OF HYSTERESIS CHARACTERISTICS OF ZYTEL

The load-carrying capacity of ZYTEL teeth is to some extent limited by the temperature increase of the teeth under dynamic loading. The relationship of this temperature increase and bending stress for various gears was shown in Progress Report No. 3. It is difficult to compensate for anticipated temperature rise in any design procedure which might be established.

It has been felt that much of the temperature rise of the gear teeth might be due to hysteresis of the teeth when alternately loaded and unloaded during operation. To investigate the hysteresis characteristics of ZYTEL, which in turn may lead to further knowledge of anticipated temperature rise, a device shown in Fig. 23 is being constructed.

Three sections of 3/8-in.-diameter ZYTEL tubing are mounted in bearing plates and fastened together with friction couplings to form a complete circle

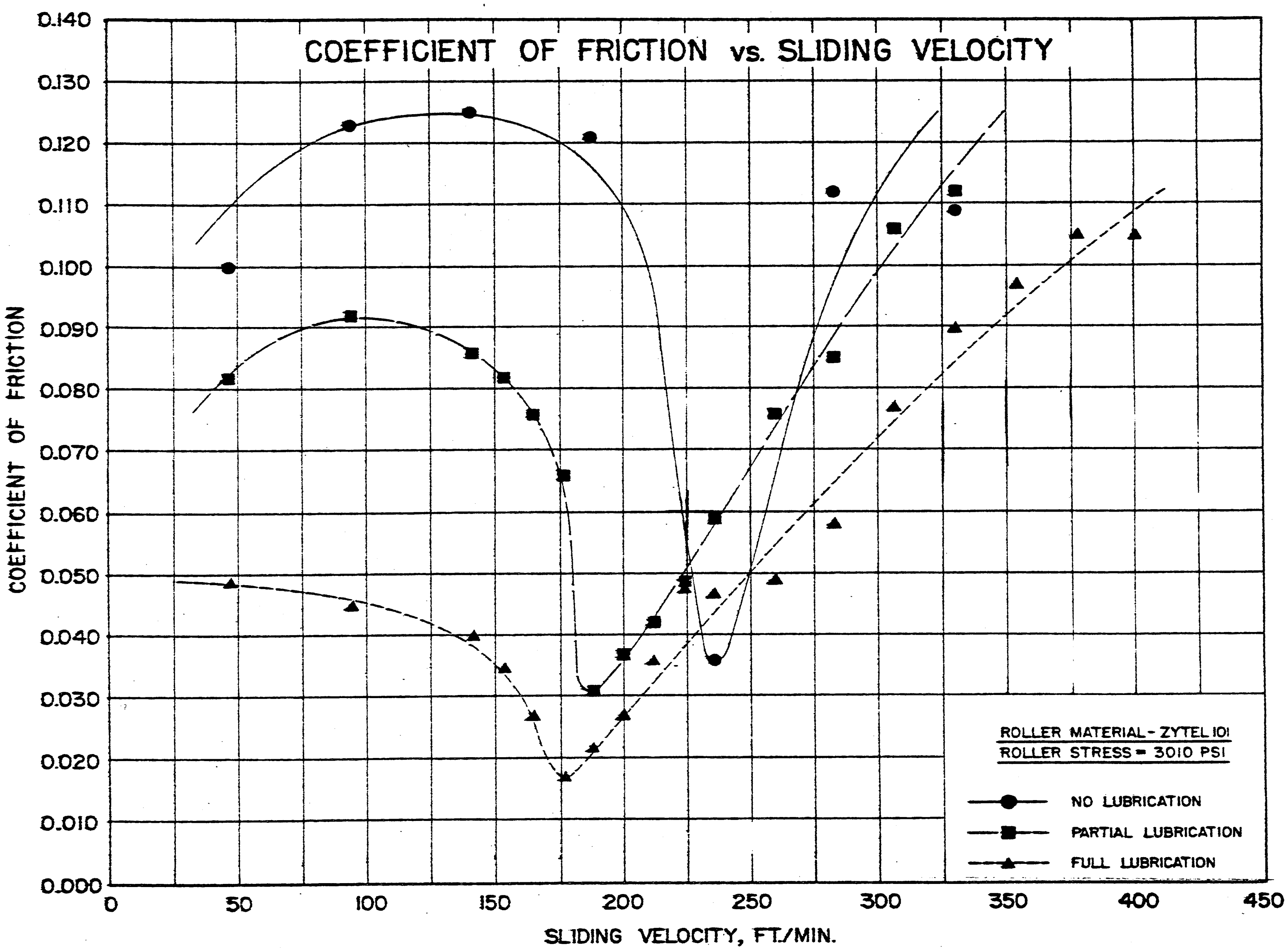


Fig. 22. Friction coefficients vs. sliding velocity.

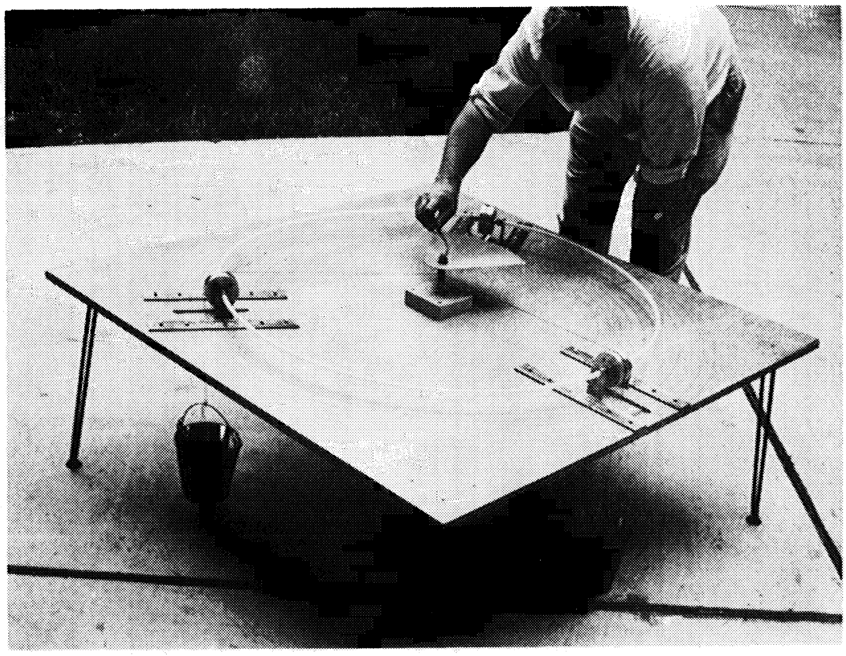


Fig. 23. Hysteresis investigating devices.

as shown. A bending stress is induced by twisting one-half of any coupling relative to the other half and clamping the two halves together. If the three couplings are rotated by wrapping up onto the central mast the cords which have been wrapped around the couplings, the energy required to rotate the coupled tubing should be a measure of the hysteresis of the tubing, since no external work is done.

This device is nearly completed, and some preliminary investigation has been started. However, at the time of this writing no results have been obtained.

IV. EVALUATION OF DELRIN AS A GEAR MATERIAL

17. EXPERIMENTAL WORK WITH DELRIN

To date the experimental work with DELRIN has been confined to fatigue testing of 16-, 20-, and 32-pitch spur gears, some with molded teeth, and some with hob-cut teeth, with and without lubrication.

The 16-pitch gears with hob-cut teeth were machined from 4-in.-diameter, 1/2-in.-thick molded DELRIN blanks. The 32-pitch gears with hob-cut teeth were made by turning down the 16-pitch gears after failure, and cutting new teeth.

The 20- and 32-pitch molded gears were the same as the molded ZYTEL test gears shown in Figs. 1 and 2 of this report. Inspection of a number of the molded DELRIN gears with the Fellows Involute Profile Checker showed that the 20-pitch teeth and the 32-pitch teeth have 20° pressure angles, and that the profiles are nearly perfect involutes. Variations from the true involute profiles were of the same order of magnitude as those listed for the molded ZYTEL gears in Section 3 of this report.

These same gears have been inspected with the Kodak Conju-Gage, with results as follows:

	20-pitch, in.	32 pitch, in.
Total composite error, max.:	0.0023	0.0038
Total composite error, avg.:	0.0019	0.0031
Tooth-to-tooth composite error, max.:	0.0007	0.0009
Tooth-to-tooth composite error, avg.:	0.00045	0.00073

As previously mentioned, the tooth-to-tooth composite error combines errors in tooth form, tooth thickness, and individual tooth spacing. The total composite error combines all the above errors with eccentricity of the pitch diameter.

The 20-pitch teeth which were inspected had been annealed at 300°F, while the 32-pitch teeth had not been annealed. It is interesting to note that the 20-pitch DELRIN teeth are more accurate than the 32-pitch DELRIN teeth. The opposite is true for the molded ZYTEL gears, as shown in Section 3 of this report. Annealing the 20-pitch gears may have had some influence on the accuracy of the teeth. The greater shrinkage of DELRIN as compared to ZYTEL may also have indirectly affected the total composite error of the DELRIN gear teeth as compared to the teeth of the ZYTEL gears. Because of this greater shrinkage, the splined shaft onto which the gears are pressed for inspection or for test must ream away some of the DELRIN material in the splined hole. Thus it is possible that the shaft could be slightly off-center if the shaft did not ream out the splines uniformly. When the gear is inspected, this would tent to increase the total composite error recorded by the Conju-Gage.

The splined holes in the ZYTEL gears were of such a size that they pressed snugly onto the shaft, with no ZYTEL removed in the process, thus lessening the chance of having the shaft and the gear eccentric to one another.

Test procedure, gear backlash, lubrication, ambient temperature and humidity, and method of calculating the bending stresses are all the same as for ZYTEL gears, described in Sections 2, 4, and 5 of this report.

18. FATIGUE TESTS AND RESULTS WITH DELRIN GEARS

The following fatigue tests have been completed at the time of this writing:

a. 16-pitch, 40-tooth hob-cut gears made of 91X DELRIN, with 20° pressure angle and 7/16-in. face were tested with pitch-line velocities of 1635 and 785 ft/min, with the teeth lubricated.

b. 32-pitch, 64-tooth hob-cut gears made of 91X DELRIN, with 20° pressure angles and 13/32-in. face were tested with a pitch-line velocity of 1635 ft/min, with the teeth lubricated.

c. 20-pitch, 50-tooth molded gears made of 100X and 500X DELRIN, with 20° pressure angles and 1/2-in. face were tested with pitch-line velocities of 1635 ft/min. The 500X DELRIN gears have been tested with and without lubrication, while the 100X DELRIN gears have been tested with the teeth lubricated.

d. 32-pitch, 80-tooth molded gears made of 500X DELRIN, with 20° pressure angle and 1/2-in. face were tested with a pitch-line velocity of 1635 ft/min, with and without lubrication.

The results of these tests are shown in Figs. 24-26. For comparison, the lines previously established for similar ZYTEL gears are also shown. It will be noticed that the 16- and 32-pitch gears with cut teeth, made of DELRIN 91X, performed about the same as the ZYTEL gears with cut teeth when lubricated. However, the 20- and 32-pitch gears with molded teeth made of DELRIN 500X, have not yet exhibited the improvement over cut gears that was found with ZYTEL. At the time of writing, there are not enough test data on the DELRIN gears to establish any definite conclusions, and the fatigue-test program is continuing.

The 16-pitch hob-cut DELRIN gears which were tested with 5450 and 4940 psi bending stress at 1635 ft/min were made from the first set of DELRIN blanks received. After these gears had been completely finished, including the hobbing of the teeth, they warped, forming a dish. Despite this "dished" condition, however, the gears were tested.

These same gears were later remachined to have 64 teeth, 32 pitch. At this same time, the face width was cut to 13/32 in. from the original 7/16 in. to remove the "dishing." These 32-pitch gears were tested with stresses of 6300 and 5020 psi at 1635 ft/min.

The 16-pitch gears which were tested with the bending stress of 4030 psi, both at 1635 and 785 ft/min, were made from the second batch of DELRIN blanks received. Care was taken to have the same amount cut from each face of the blank when these gears were machined, and no "dishing" took place. However, a pattern of small pits, running circumferentially around the gear, was found in each of these gears. These pits appeared on all the teeth slightly to one side of the center of the face. These pits were not discovered until some of the gears had operated for some time, after which oil had worked into the pits and made them visible.

None of the teeth failed by breaking at this pitted section. However, it will be noted from Fig. 24 that these gears seem to have a shorter operating

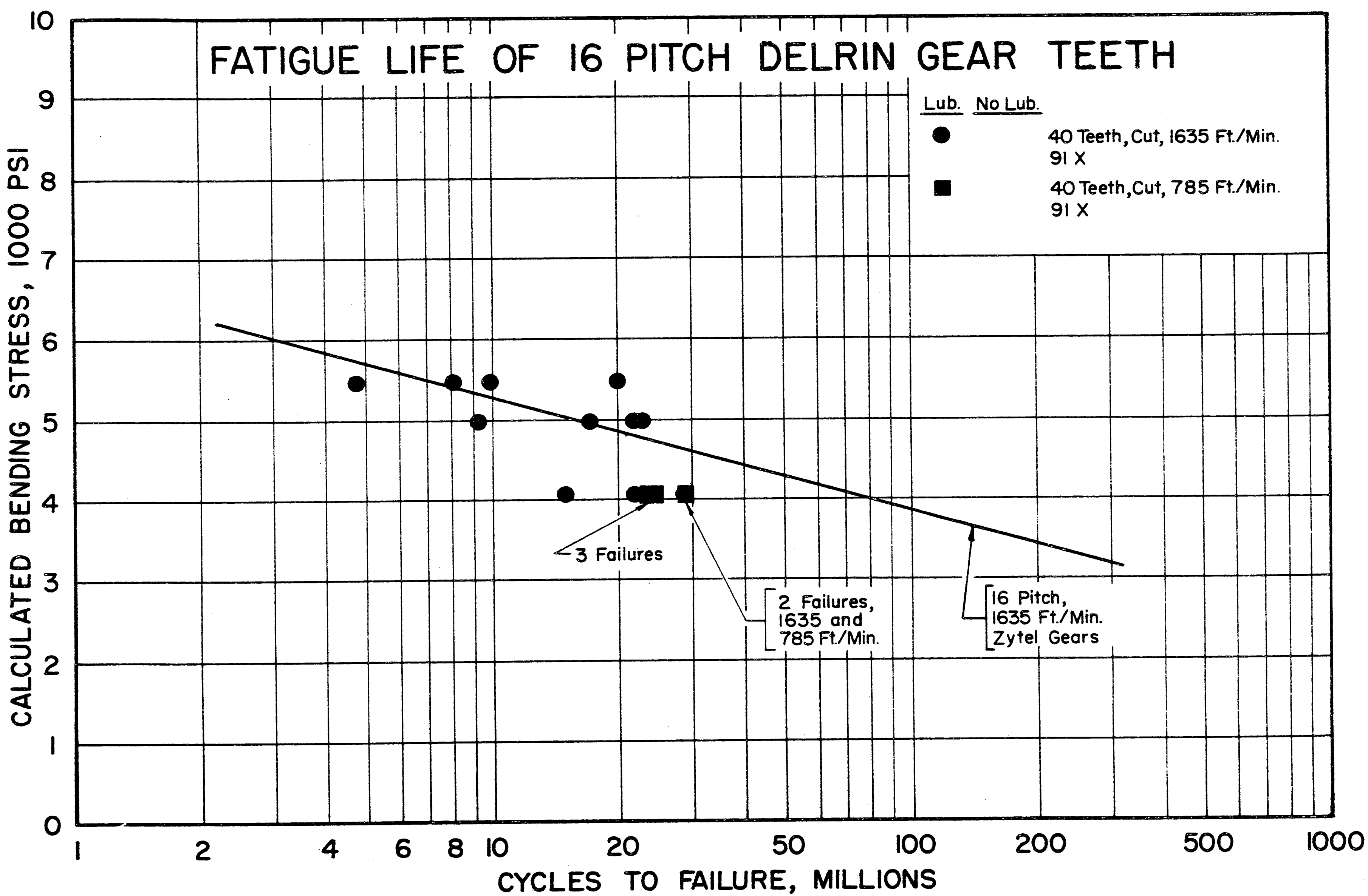


Fig. 24. Fatigue life of 16-pitch DELRIN gear teeth.

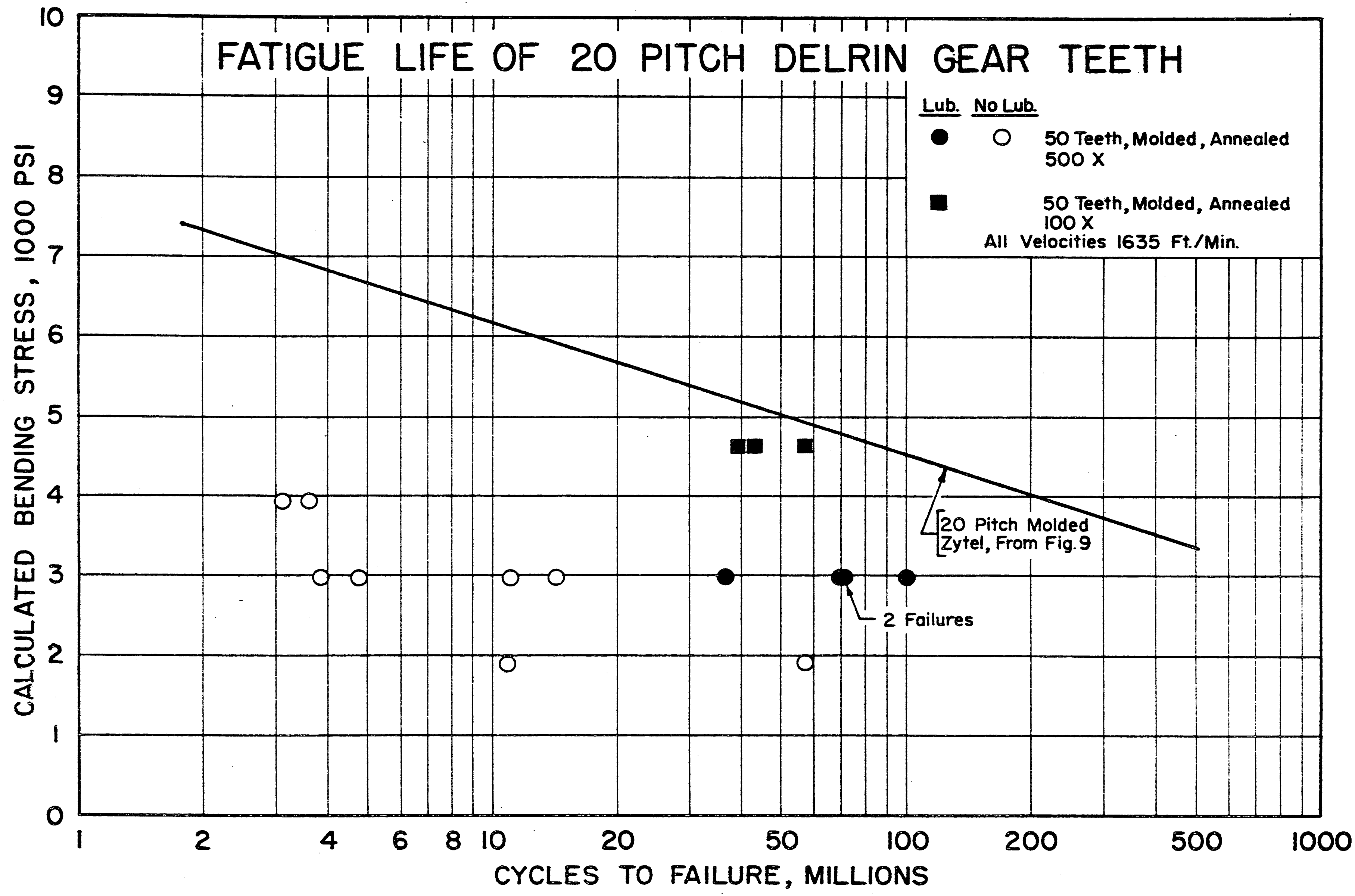


Fig. 25. Fatigue life of 20-pitch DELRIN gear teeth.

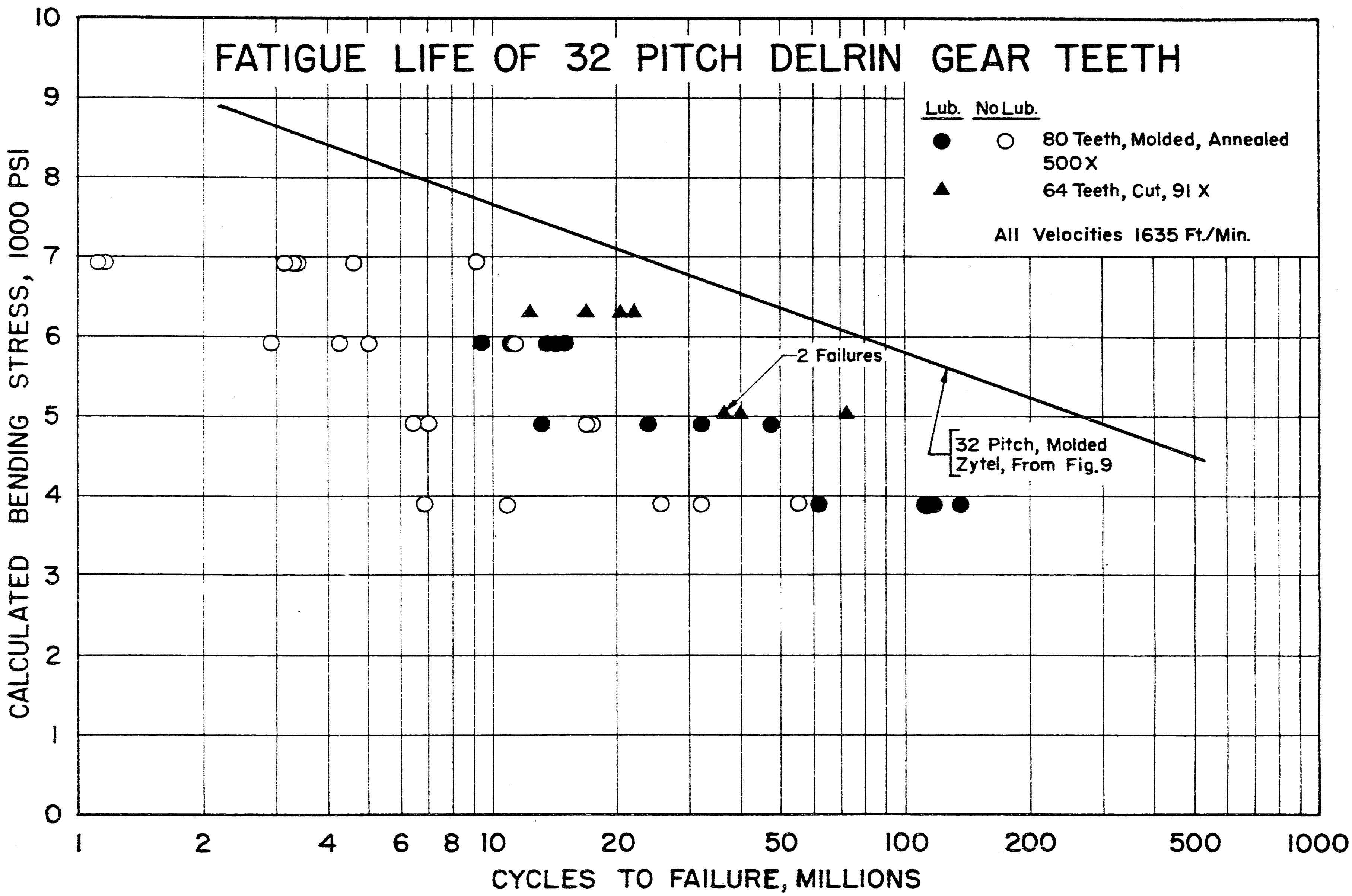


Fig. 26. Fatigue life of 32-pitch DELRIN gear teeth.

life than would be expected from the results obtained with the first set of hob-cut DELRIN gears. It is not known whether or not the molding defect which produced the pits contributed to the shortened life of the teeth. However, the presence of such a defect tends to cast doubt on the reliability of the results obtained with these gears.

All the molded DELRIN gears tested so far have been annealed at 300°F. None of these gears have failed by breaking through the rim. The lubricated gears which have failed have broken at the root in a typical fatigue failure as shown in Fig. 27.

Without lubrication, the performance of the molded teeth has been inconsistent, as shown in Figs. 25 and 26. Failure takes place by a process of rapid wear or disintegration, which seems to start after the gears have been running satisfactorily for a period of time. The gears are well lubricated with oil at the start of the tests, but are not lubricated again. The initial period of operation before the teeth start to disintegrate may be the time required to wear away this original lubrication.

Figure 28 shows some of these badly worn teeth. The material disintegrates into a fine white powder. The gears have been allowed to run until the teeth are so badly worn away that they break and drop the load. However, this generally takes place very shortly after the disintegration starts.

19. METHOD OF CALCULATING THE LOAD-CARRYING CAPACITY OF DELRIN SPUR GEARS

The methods for calculating the load-carrying capacity of ZYTEL spur gears, described in Sections 7 and 8 of this report, will also be applicable to DELRIN when allowable stresses have been established by sufficient fatigue-test data. However, it is felt that it would not be wise to try to establish allowable stresses from the limited number of data which have been obtained so far.

20. PROPOSED FUTURE WORK

The following proposals for future work have been previously stated in Mr. H. Poehle's letter of June 5, 1958, to Mr. R. N. Peterson at the Sales Service Laboratory. For purposes of record and reference they are repeated here:

a. The fatigue life testing program will be continued to obtain design data for molded DELRIN gears. Wear data will be obtained during much of the life testing to establish rates and wear patterns of the DELRIN gears. This shall constitute the major part of the work to be done, and shall include the following:

    1. Tests of gears formed by various molding cycles and annealing treatments,

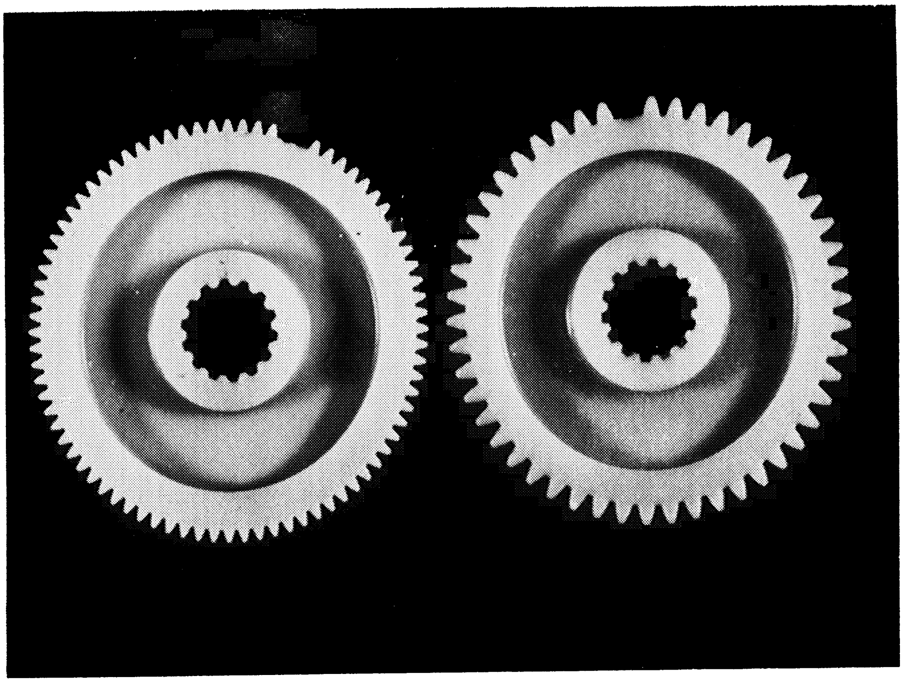


Fig. 27. Fatigue failure of molded DELRIN gear.

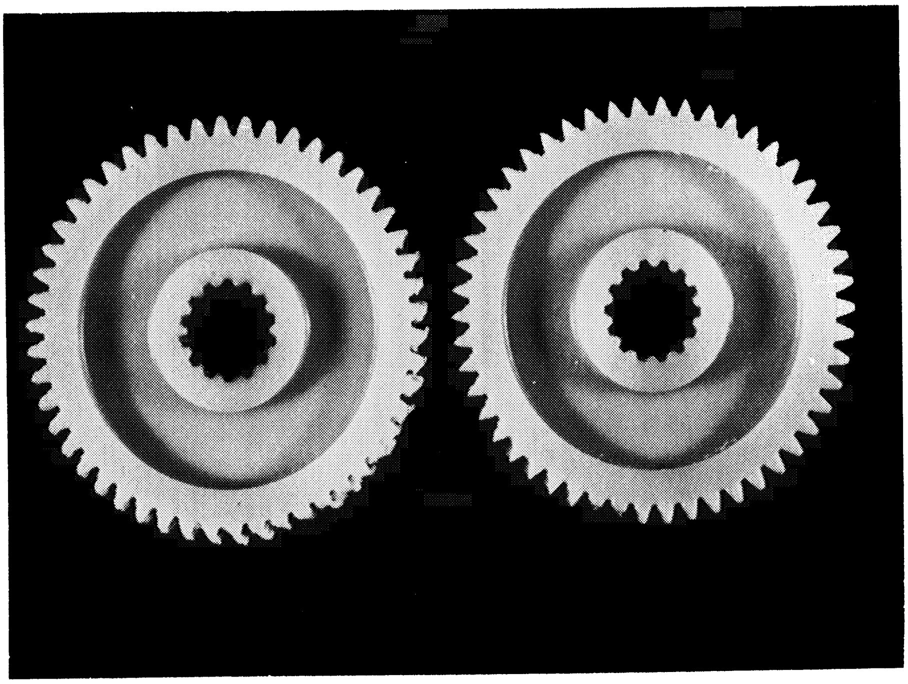


Fig. 28. Wear failure with no lubrication.

2. Tests with pitch-line velocities of 785, 1635, and 3000 ft/min,

3. Life and wear tests of DELRIN gears meshing with steel gears, and with ZYTEL gears,

4. Perhaps some testing at elevated ambient temperatures.

b. The effects of increasing the pressure angle to strengthen the teeth shall be investigated to determine the optimum pressure angle for DELRIN gears. This work shall be done with hob-cut gears.

c. The effects of contact pressure, sliding velocity, and various lubricants on the surface compressive endurance limit, the rate of wear, and the coefficient of friction shall be investigated with the rolling-sliding contact machine.

d. Operation of the Boston helical gear speed reducer with ZYTEL gears, and with DELRIN gears if the material is available, shall be continued throughout the year.

e. Investigation of hysteresis losses in DELRIN and ZYTEL shall be carried on, with the device described in Section 16 of this report.