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Progress Report No. 3

EVALUATION OF "ZYTEL" AS A GEAR MATERIAL

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

A comprehensive fatigue testing program with "Zytel" spur gears was completed. The results are described, and a means of calculating the torque and horsepower capacity of "Zytel" gear teeth is presented.

Investigations of gear-tooth temperatures and of the energy required to cause tooth failure are also described.

I. INTRODUCTION

This project was undertaken for the Polychemicals Department of the E.I. duPont de Nemours and Co., Inc., Wilmington, Delaware. Its purpose is to evaluate molded "Zytel" as a gear material and to establish design data for gears made of "Zytel." This is outlined in detail in the proposal of May, 1953, entitled "Design Data for Nylon Gears," submitted to E.I. duPont de Nemours and Co., Inc., by the Engineering Research Institute of The University of Michigan.

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.

This report covers the work done from January, 1956, to the present time.

II. EXPERIMENTAL APPARATUS

The test machines and instrumentations were described in Progress Report No. 2, January, 1956. No significant changes have been made in the equipment since that time.

III. EXPERIMENTAL WORK AND TEST RESULTS

A. Introduction

The following experimental work was carried on during the period covered by this report:

1. Fatigue testing of spur gears to obtain more data to substantiate and refine the method for calculating the load-carrying capacity of hob-cut "Zytel" gears, as presented in Progress Report No. 2.

2. Investigation of gear-tooth operating temperatures to see how this temperature might influence the load-carrying capacity of the teeth.

3. Investigations of the possibility of establishing a relationship between the energy adsorbed by the teeth and tooth failure, to reduce if possible the need for extended fatigue testing.

Most time and effort were spent on the fatigue testing. Each of the above three phases of the experimental work is described in the paragraphs which follow.

B. Fatigue Testing of "Zytel" Gears

1. TEST PROGRAM

a. Objectives.—The main objectives of the fatigue testing program were as follows:

1. To obtain additional fatigue-life data to establish more firmly the load carrying capacity of hob-cut "Zytel" spur gear teeth.
2. To establish the effects of the following on the load-carrying capacity of the teeth:
   a. lubrication and lack of lubrication,
   b. pitch line velocity,
   c. pitch, or size, of teeth,
   d. different diameter gears meshing together,
   e. pressure angle and tooth form,
   f. face width.
3. To obtain some preliminary data on the load-carrying capacity of molded "Zytel" spure gear teeth.

b. Procedure.—The fatigue testing program was carried on very much as described in Progress Report No. 2. The test period was lengthened well beyond the 50 million cycle period originally used, to see if a definite endurance limit could be established. Since previous work had shown that tooth wear was not significant with "Zytel" gears, no wear measurements were taken.

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.

When a gear failed, that gear and the gear with which it meshed were both replaced, and this was counted as a single failure. An exception to this practice was made when a small diameter gear meshed with a much larger gear. In such a case, the teeth of the larger gear were so much stronger than the teeth of the smaller gear that it was felt that failure of the small gear teeth would not seriously impair the strength of the large gear teeth. One large gear was often used with several smaller gears before being replaced.

To obtain a reasonable smaple of results, at least four failures were obtained at most of the combinations of load and speed at which the gears were tested. It was recognized that a much larger number of failures for each test would be desirable to insure a more normal sample for statistical analysis of the results. However, the large amount of time required to obtain failures, especially at low speeds, limited the fatigue tests to a maximum of four failures at each load in the majority of cases.

c. Test Gears.—Various sizes of 16-, 32-, and 48-pitch spur gears were tested at velocities of 785 to 3730 ft/min, both with and without lubrication. One group of 16-pitch gears had molded teeth, but all others had hob-cut teeth.

The hob-cut gears were originally cut from molded blanks. As the teeth failed, or the test was completed, the gears were remachined to smaller diameters, and new teeth were cut. In this way each original blank was used several times in the fatigue testing program.

d. Backlash.—The 16-pitch teeth operated with 0.004- to 0.006-in. average backlash, the 32-pitch teeth with 0.002- to 0.004-in. average backlash, and the 48-pitch teeth with 0.001- to 0.002-in. average backlash, set when the gears were at room temperature.

A rather exacting method of setting the backlash was described in Progress Report No. 1. This proved to be too tedious for continued use,and with practice it was found that the backlash could be set with sufficient accuracy by the "feel" of the teeth.

No testing was done to determine the optimum backlash, if such a condition exists. Over a reasonable range, the variation in backlash seemed to have little or no effect on the performance of the gears. Excessive backlash, or no backlash at all, both seemed to reduce greatly the life of the teeth.

e. Lubrication.—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 doused with oil at the start of the tests with no lubrication, but were never lubricated again.

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

2. TESTS AND RESULTS WITH 16-PITCH TEETH

A large portion of the fatigue testing was done with 16-pitch spur gears as follows:

a. Hob-cut 40-tooth spur gears with full-depth teeth, 20° pressure angle, and 7/16-in. face were tested in pairs at velocities of 785, 1635, and 3270 ft/min, both with and without lubrication. Most of the testing was done at 1635 ft/min to establish a pattern of results. The testing at 785 and 3270 ft/min was done to determine the effects of velocity on the load-carrying capacity of the teeth.

b. Hob-cut 27-tooth spur gears with full-depth teeth, 20° pressure angle, and 7/16-in. face, meshing with either 57-tooth or 60-tooth gears were tested at velocities of 1635 and 3730 ft/min with lubrication and at 3730 ft/min without lubrication. The testing at 1635 ft/min was done to see if the results obtained with the 27-tooth gears would be comparable to that obtained with the larger 40-tooth gears described above. The testing at 3730 ft/min was done to evaluate further the effects of velocity on the load-carrying capacity of the teeth.

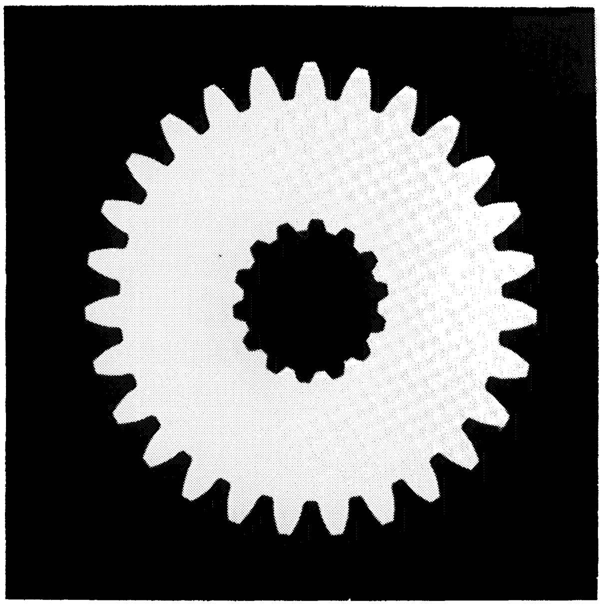
c. Hob-cut 27-tooth spur gears with full-depth teeth, 20° pressure angle, and 7/16-in. face, meshing with 53-tooth gears, were tested at 1635 ft/min with lubrication. <u>These gears were made with the outside diameter of the 27-tooth gear 1/16 in. larger than standard while the outside diameter of the 53-tooth gear was made 1/16 in. smaller than standard</u>. On the 27-tooth gear, this modification serves to increase the thickness of each tooth at the root of the tooth, thus increasing its strength as a beam. On the 53-tooth gear, however, this thickness is decreased, thus weakening the originally stronger 53-tooth gear to strengthen the weaker 27-tooth gear. This modification decreases the angle of approach, but increases the angle of recess, and increases the maximum sliding velocity.

   Illustration 1 shows this oversize 27-tooth gear, along with a standard gear and other gears described below.

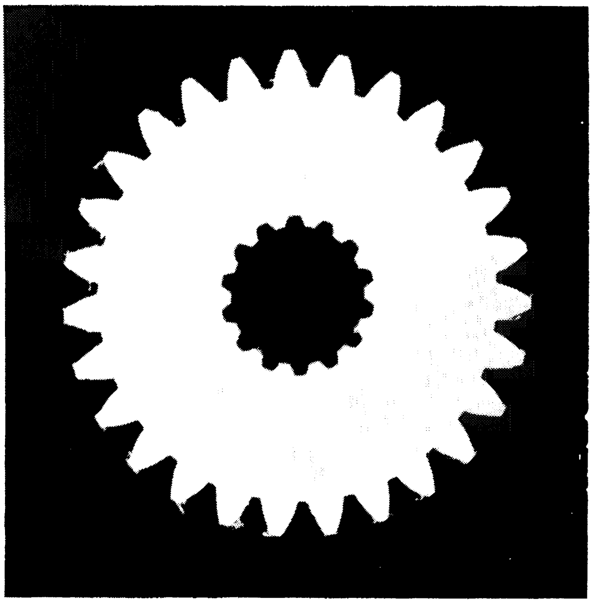
d. Hob-cut 27-tooth spur gears with <u>stub teeth</u>, 20° pressure angle, and 7/16-in. face meshing with 53-tooth gears were tested at 1635 ft/min with lubrication.

   Being shorter, the stub tooth should be stronger than the full-depth tooth. Stubbing the teeth decreases the length of the path of contact, the arc of approach and recess, and the magnitude of the maximum sliding velocity.

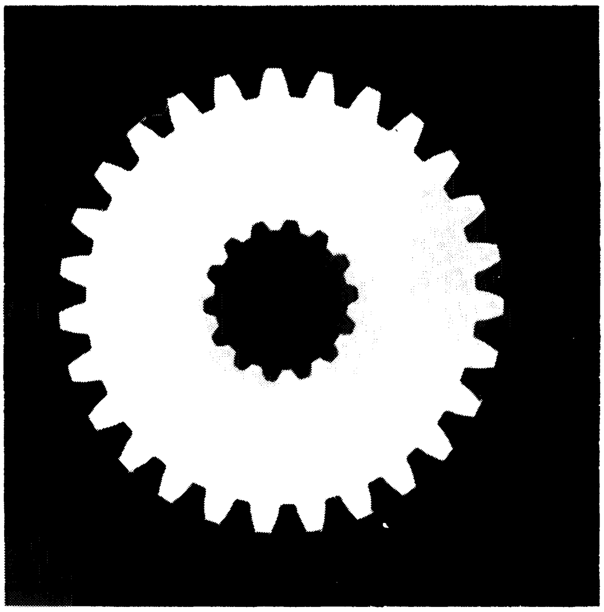
   A 20° stub-tooth gear is shown in Illustration 1.



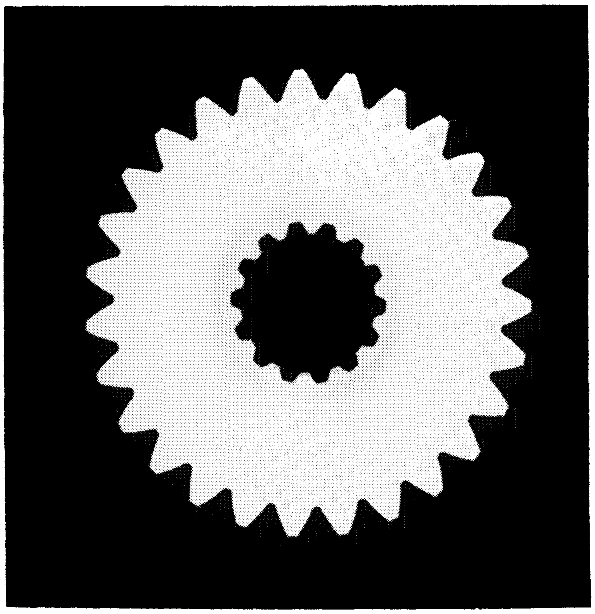
(a) 20° full-depth teeth



(b) 20° full-depth teeth with O.D. 1/16-in. oversize



(c) 20° stub teeth



(d) 30° stub teeth

Illustration 1. 16-pitch, 27-tooth gears.

e. Hob-cut 27-tooth spur gears with stub teeth, 30° pressure angle, and 7/16-in. face, meshing with 58-tooth gears, were tested at 1635 ft/min with lubrication.

The 30° stub tooth is very short and thick; hence should have high beam strength. The normal force and the maximum sliding velocity are greater than with the 20° stub tooth.

A 30° stub-tooth gear is shown in Illustration 1.

f. Molded 28-tooth spur gears with stub teeth, 20° pressure angle, and 1/2-in. face, were tested in pairs at 785 ft/min with lubrication. These gears were not molded specifically for this test program. After molding, each gear was bored and splined to allow mounting in the test machines. The tooth form, thickness, and spacing were very inaccurate compared to the hob-cut gears. Furthermore, there was a sharp corner, rather than a fillet, at the base of each tooth, as though an internal gear had been used as a mold for these gears.

It was felt that these gears were not a very good example of molded teeth, but they were the only molded gears available at this time, and so were tested.

The results obtained with the full-depth, standard 20° pressure angle hob-cut teeth are shown in Fig. 1. The results obtained with the oversize full-depth teeth, the 20° and 30° hob-cut stub teeth, and the molded stub teeth are shown in Fig. 2. The methods used for calculating the bending stresses shown are discussed in Section III-B-9 of this report.

3. DISCUSSION OF RESULTS WITH 16-PITCH TEETH

a. Hob-Cut Teeth, Lubricated.—When the 16-pitch full-depth teeth were lubricated, those teeth which failed did so by breaking at the root in a typical fatigue type of failure. Tooth breakage at the root was described and illustrated in Progress Report No. 2.

When lubricated, there was no significant difference in the test results obtained with the full-depth hob-cut teeth at pitch line velocities of 1635 and 3730 ft/min. These results are represented by a common line on Fig. 1.

The test results obtained with these same teeth at 785 ft/min are represented by a line having somewhat less slope than the 1635- and 3730-ft/min line, indicating that the load-carrying capacity of the teeth is favored by the lower velocity.

No ready explanation can be found for the difference in slope of the 785 ft/min line and the line representing 1635 and 3730 ft/min. Both lines were

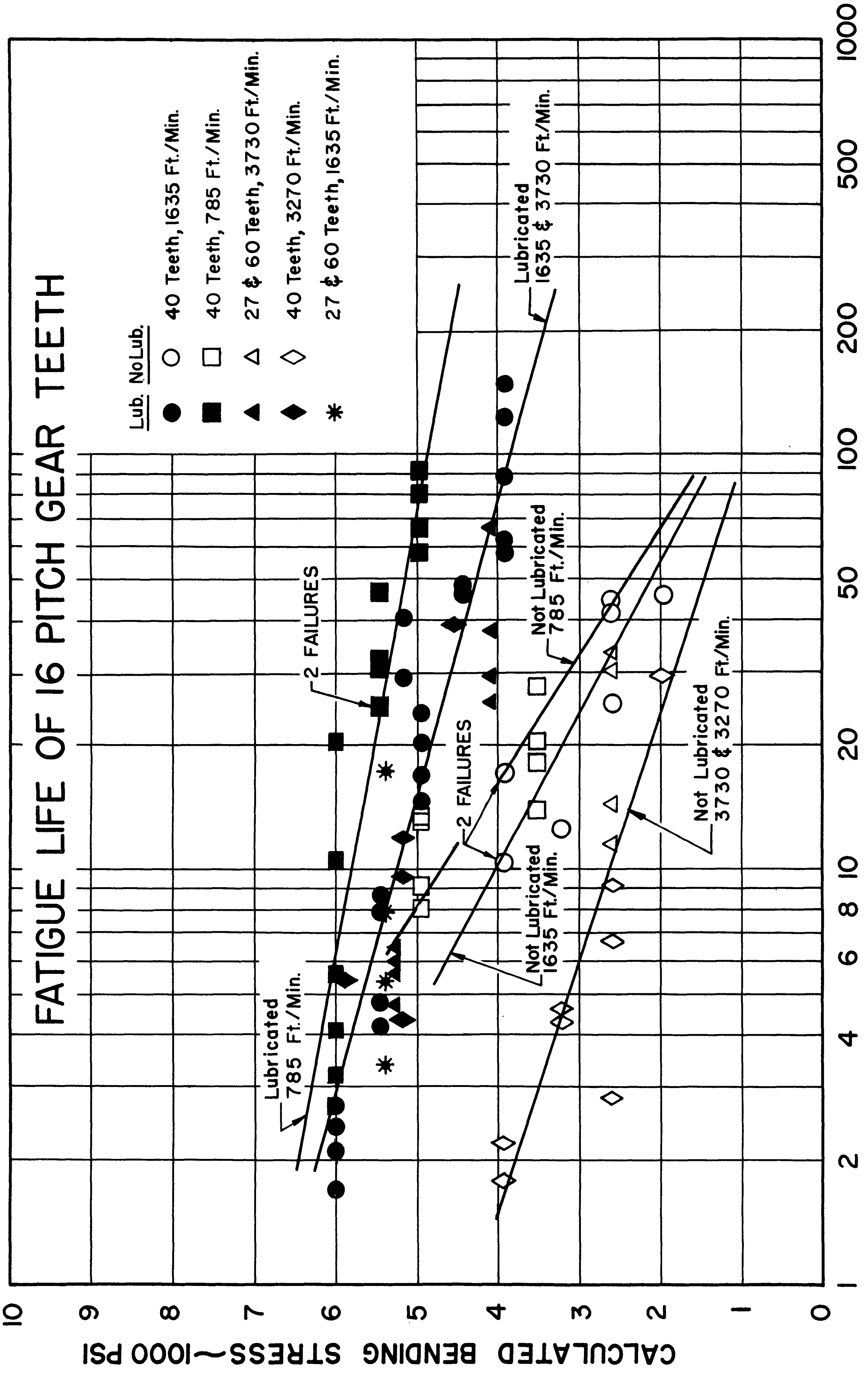


Fig. 1. Fatigue life of 16-pitch gear teeth.

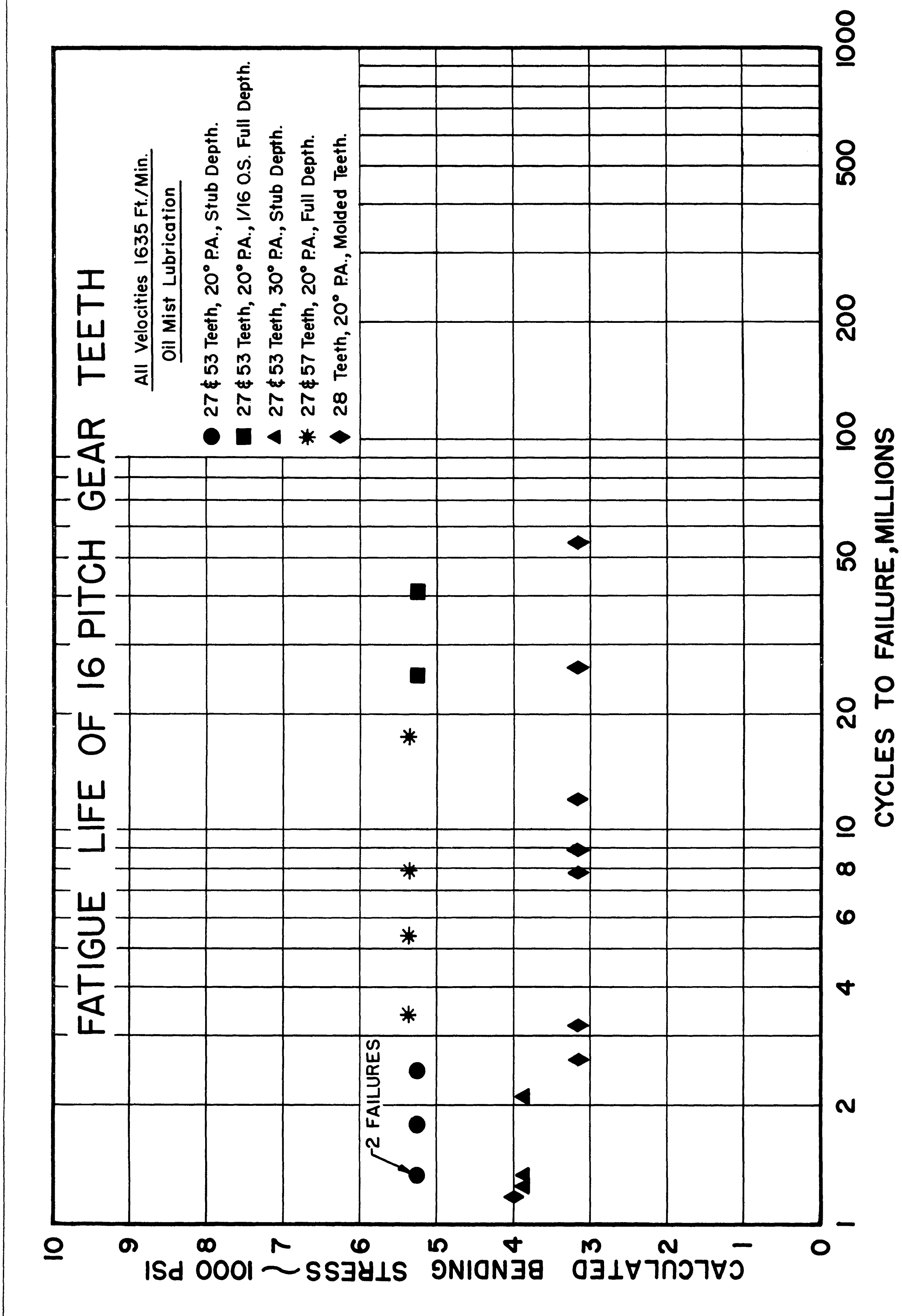


Fig. 2. Fatigue life of 16-pitch gear teeth.

obtained by statistical analysis from a reasonable number of failures; hence both should be considered to be valid. Although the difference between the two lines is enough to be significant, this difference is not very large numerically until a large number of cycles has been reached.

It is interesting to note that changing the velocity from 785 to 3730 ft/min, a range of almost 5:1, produced relatively little change in the stress at which the teeth fail when they are lubricated.

Good agreement was found between the test results obtained at 1635 ft/min with pairs of 40-tooth gears meshing together, and 27- and 57-tooth gears meshing together. This would indicate that the numbers of teeth in a pair of meshing gears do not affect the stress at which the teeth will fail.

Figure 2 seems to indicate that the 20° and 30° stub teeth are not as satisfactory as the 20° full-depth teeth, but that an improvement was made by making the full-depth teeth 1/16-in. oversize. It should be pointed out, however, that although the stress is essentially the same for the 20° stub, full-depth, and oversize teeth, the pound load on the stub and oversize teeth was greater than on the full-depth teeth. In fact, the 20° stub, 30° stub, and 20° oversize teeth all carried the same load in pounds, while the 20° full-depth teeth were less heavily loaded. As explained in Section III-B-9 of this report, the differences in the form factor for these teeth accounts for the stress differences at the same load.

It does appear, however, that the full-depth form is preferable to the stub-tooth form, and that the 20° pressure angle is preferable to the 30° pressure angle. It also appears that strengthening the teeth by making them oversize is desirable. However, it must be remembered that with this system the smaller gear is strengthened, while the larger gear is weakened, and that the gears are no longer interchangeable with other gears of the same pitch.

During the test, the temperature of the 30° stub teeth averaged 202°F, the 20° stub and oversize teeth averaged 180°F, and the 20° full-depth teeth averaged 160°F. The higher temperature of the 30° stub teeth was probably due to the greater force on the tooth surface, and the greater sliding velocity, both resulting from the larger pressure angle. Apparently the greater strength obtained with the thicker 30° tooth was more than offset by the weakening effect of the temperature increase.

The method of measuring tooth temperatures is described in Section III-C of this report.

b. <u>Hob-Cut Teeth, Not Lubricated</u>.—When the 16-pitch full-depth teeth were not lubricated, about 50% of the teeth which failed did so by breaking at the root, while the remaining 50% broke near the center of the tooth. Tooth breakage near the tooth center was described and illustrated in Progress Report No. 2.

As shown by Fig. 1, lack of lubrication greatly reduces the load-carrying capacity of the 16-pitch teeth. This is especially noticeable at the larger number of cycles of operation. Futhermore, the effect of pitch line velocity appears to be more pronounced when the teeth are not lubricated.

The three lines representing the nonlubricated test data appear to converge at about 200 million cycles. No particular significance seems to be attached to this, however. Since none of the tests with no lubrication ran more than 50 million cycles, the validity of extending the lines to 200 million cycles appears to be questionable.

c. Molded Teeth, Lubricated.—The molded teeth commonly failed by breaking at the root of the tooth. However, an exception to this took place with that pair of molded gears which ran 54.5 million cycles. These teeth did not break, but instead the surface of several teeth became very severely pitted and worn away, as shown in Illustration 2. The teeth continued to carry the load, but the test was stopped before any of the teeth actually broke.

Illustration 2. Surface failure of molded teeth.

As previously mentioned, the teeth on these molded gears were very inaccurate compared to the hob-cut teeth. It was felt that this inaccuracy contributed to the rather poor performance of these gears and to the large variation in the test results, shown in Fig. 2. This seems to indicate that tooth accuracy has a considerable effect on the load-carrying capacity of "Zytel" teeth.

It is also interesting to note that when a pair of the molded gears ran 54.5 million cycles, the tooth surface failed. This is the only time this type of failure took place in the entire test program, and might indicate that molded teeth do not have as durable a surface as hob-cut teeth.

4. TESTS AND RESULTS WITH 32-PITCH TEETH

A considerable amount of fatigue testing was done with 32-pitch spur gears as follows:

a. Hob-cut 64-tooth spur gears with full-depth teeth, 20° pressure angle, and 7/16-in face were tested in pairs at velocities of 785 and 1635 ft/min, both with and without lubrication. These same 64-tooth gears, meshing with 120-tooth gears, were also tested at 3730 ft/min with no lubrication.

b. Gears the same as those described above, but with a 7/32-in. face width were tested at 1635 ft/min with lubrication, to see how the smaller face width would affect the load-carrying capacity of the teeth.

c. Hob cut 56-tooth spur gears, with full-depth teeth, 20° pressure angle, and 7/16-in. face, meshing with 120-tooth gears, were tested at 3730 ft/min, both with and without lubrication.

The results obtained with the 32-pitch teeth are shown in Fig. 3.

5. DISCUSSION OF RESULTS WITH 32-PITCH TEETH

a. <u>Hob-Cut Teeth, Lubricated</u>.—As with the 16-pitch teeth, those 32-pitch teeth which failed while operating with lubrication did so by breaking at the root.

When lubricated, there was no significant difference in the test results obtained at pitch line velocities of 785, 1635, and 3730 ft/min. These results are all represented in Fig. 3 by a common line determined by a statistical analysis.

Good agreement was found between the results obtained with gears having 7/32-in. face width instead of the 7/16-in. face width common to all other hob-cut gears.

b. <u>Hob-Cut Teeth, Not Lubricated</u>.—The 32-pitch teeth which failed while operating without lubrication generally did so by failing at the root. Some failed by breaking near the middle of the tooth, but this was not common.

It will be noted from Fig. 3 that lack of lubrication appears to have virtually no effect on the performance of the 32-pitch teeth, whether operating at 785, 1635, or 3730 ft/min.

6. TESTS AND RESULTS WITH 48-PITCH TEETH

a. Hob-cut 74-tooth spur gears with full-depth teeth, 20° pressure angle, and 7/16-in. face were tested in pairs, and with 66-tooth gears, at 785 ft/min with lubrication.

b. Hob-cut 66-tooth spur gears with full-depth teeth, 20° pressure angle, and 7/32-in. face were tested in pairs at 785 ft/min with no lubrication.

The results obtained with the 48-pitch teeth are shown in Fig. 4.

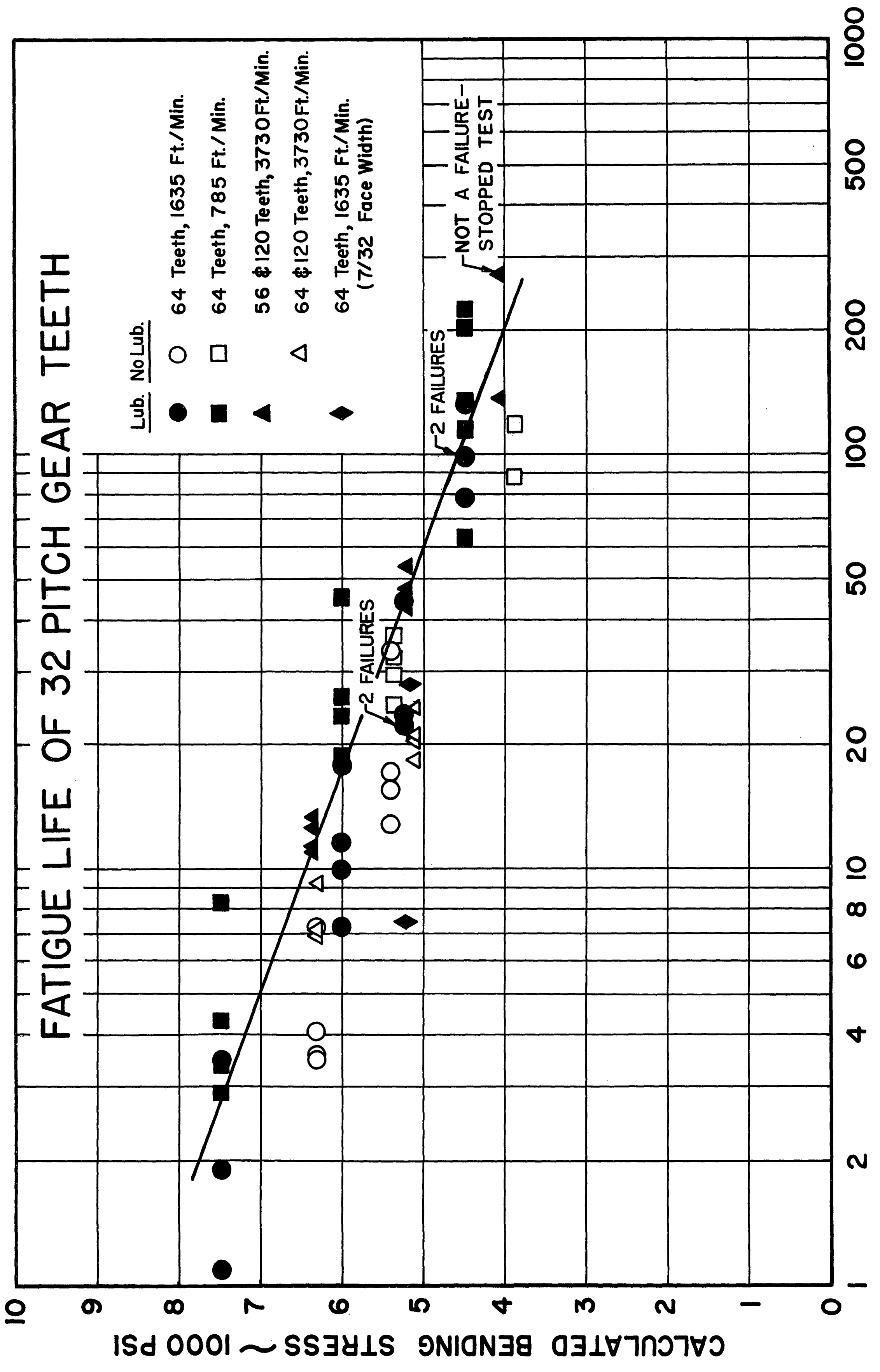


Fig. 3. Fatigue life of 32-pitch gear teeth.

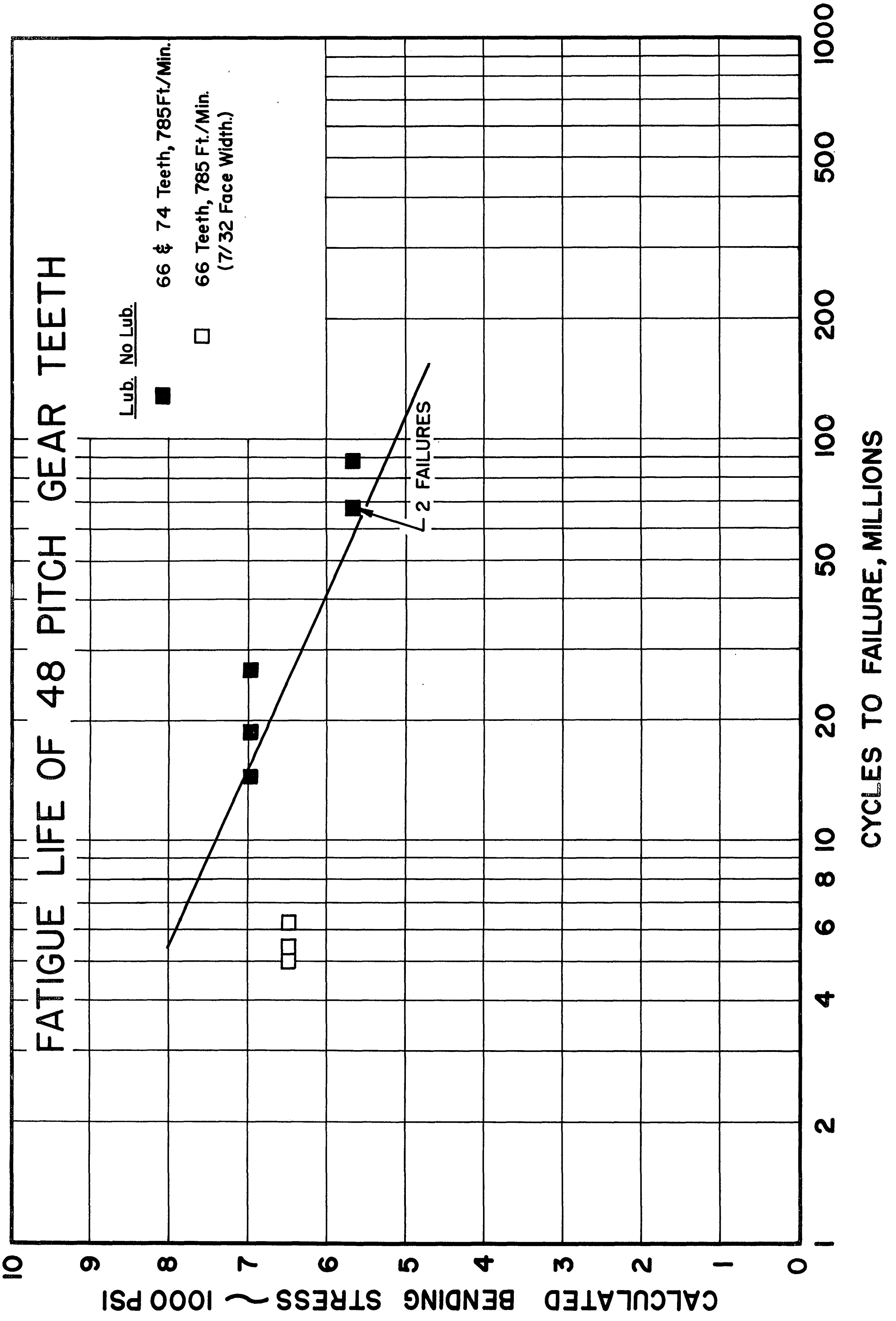


Fig. 4. Fatigue life of 48-pitch gear teeth.

7. DISCUSSION OF RESULTS WITH 48-PITCH TEETH

a. Hob-Cut Teeth, Lubricated.—The test data obtained with the 48-pitch teeth were not as extensive as those of the 16- and 32-pitch teeth. Nevertheless, a reasonable line can be obtained as shown in Fig. 4.

b. Hob-Cut Teeth, Not Lubricated.—Only one test was run with no lubrication, and the gears used for this test had 7/32-in. face width rather than the common 7/16-in. face width of other test gears. The results indicate that the lack of lubrication might reduce the load-carrying capacity of the 48-pitch teeth, but the data are not sufficient to establish a line on Fig. 4.

It was felt that the 66-tooth gears with 7/32-in. face were perhaps too small to be tested effectively in the test machines. The rotating parts other than the gears were very large compared to the small test gears, hence vibration, inertia, etc., could adversely affect the test results.

8. RESULTS OF ALL FATIGUE TESTS

Previous to the period covered by this report, some fatigue testing had been done with 16- and 20-pitch gears. The results of these tests were shown in Progress Report No. 2.

The results previously obtained with the 16-pitch gears are included in the results shown in Fig. 1. To provide an easy reference, the results obtained with the 20-pitch gears are shown in Fig. 5.

The 20-pitch data are not sufficient to establish a line by themselves. However, by including the 16- and 32-pitch lines as shown in Fig. 5, a reasonable line can be established for the 20-pitch teeth.

Figure 6 shows the results of all fatigue tests run to date. A total of 175 failures are plotted, to indicate the magnitude of the entire testing program.

9. METHODS OF CALCULATING BENDING STRESS

a. Hob-Cut Teeth.—Bending stresses in the hob-cut teeth were calculated by means of the basic Lewis equation as described in Progress Report No. 2.

This equation is given as follows:

$$S = \frac{F\ P}{f\ Y} , \qquad (1)$$

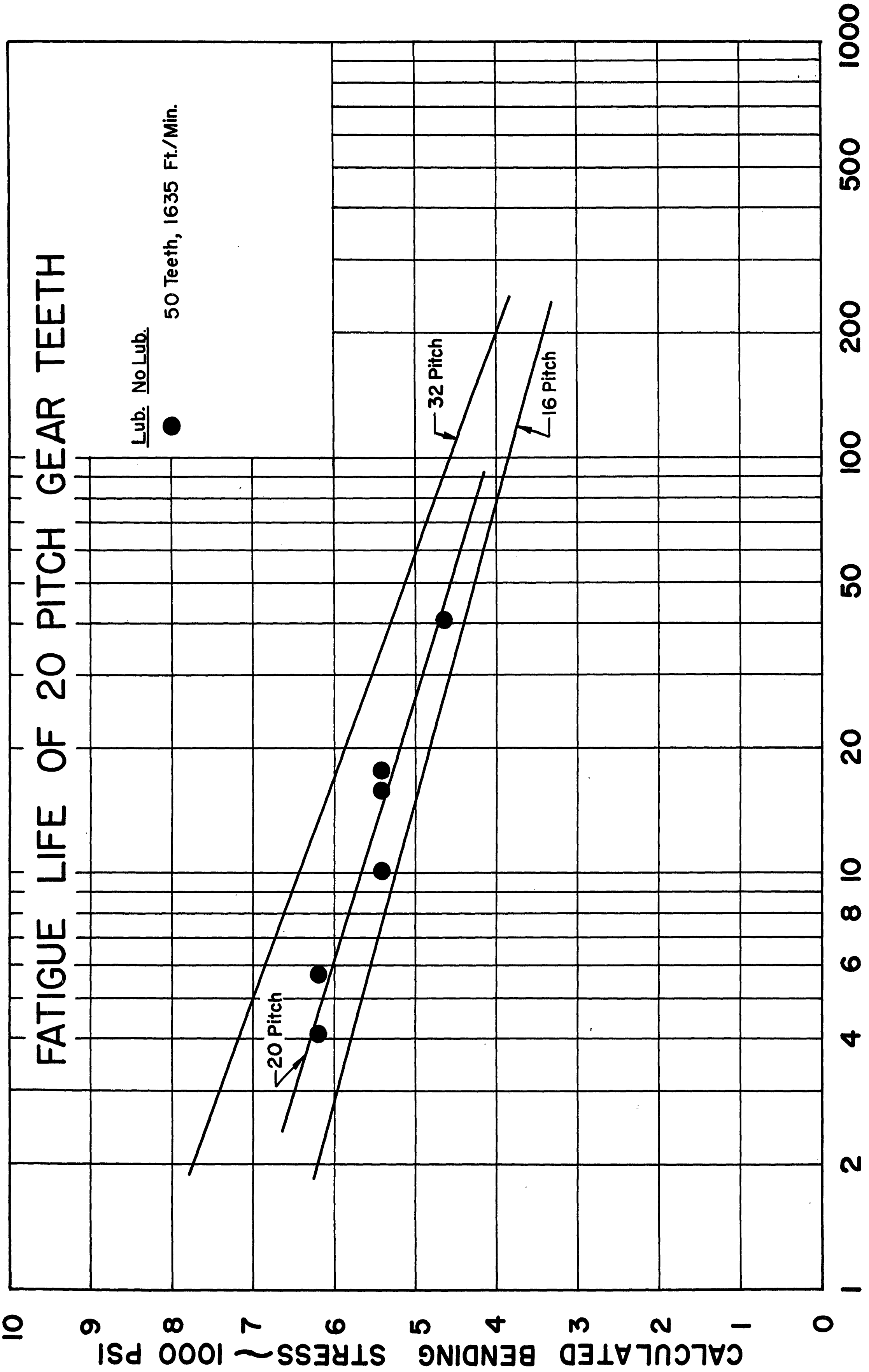


Fig. 5. Fatigue life of 20-pitch gear teeth.

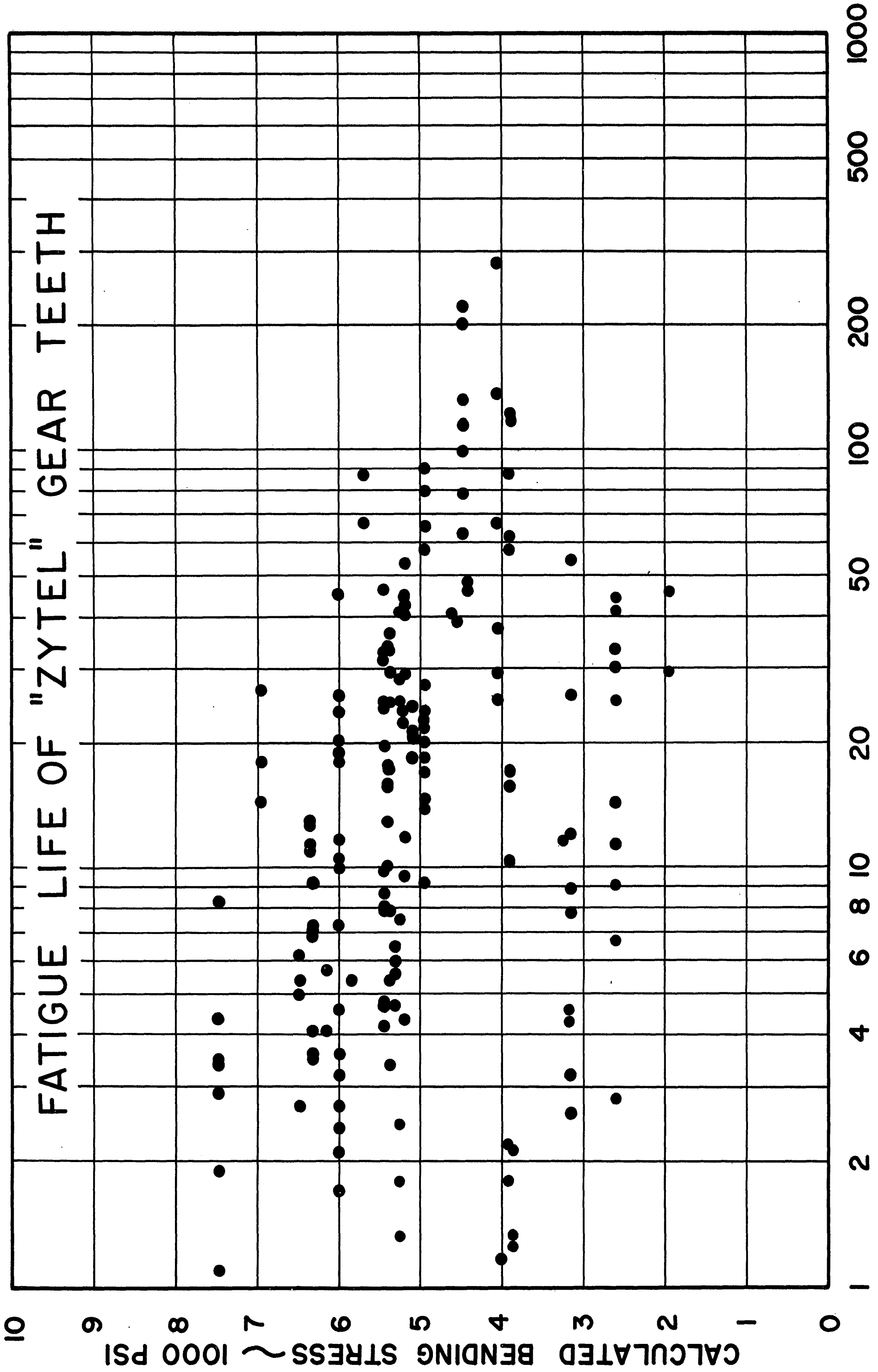


Fig. 6. Fatigue life of "Zytel" gear teeth.

where S = bending stress, psi
F = tangential force on the tooth lb, due to power transmitted
P = diametral pitch of the tooth
Y = tooth form factor, for load at pitch point.

The value of force F can be calculated by the following equation:

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

where T = torque, lb-in., being transmitted, and
D = pitch diameter, in.

The value of the form factor Y depends upon the number of teeth in the gear, the tooth height and pressure angle, and the point on the tooth where the load is applied.

Values of this form factor, for teeth of standard proportions, have been worked out and published, for the load applied near the end of the tooth and for the load applied at the pitch point. As described in Progress Report No. 2, photographic studies had shown that one tooth carries the entire load only when it is in contact near the pitch point. Hence the form factor for the load applied at the pitch point was used with the Lewis equation.

For reference, a table of form factors for 20° full depth and stub teeth, with the load applied at the pitch point, is shown in Table I.

These values do not apply to the 27-tooth gears having 16-pitch 20° teeth but with an oversize outside diameter, nor to the 27-tooth gears having 16-pitch 30° stub teeth. The values of the form factors for those two gears were worked out individually, and found to be 0.678 and 0.923 for the 20° and 30° teeth, respectively.

b. <u>Molded Teeth</u>.—Because the exact shape of the molded teeth was not known, and was difficult to measure, and because of the sharp fillet at the root of the tooth, it was felt that any form factor determined for these teeth would be of questionable accuracy. Hence for the molded teeth the stress was calculated by using the conventional bending stress for a cantilever beam. Applied to a gear tooth, with the load at the pitch point, this equation is written:

$$S = \frac{6\,F\,d}{f\,t^2} \quad , \tag{3}$$

where S = bending stress, psi,
F = tangential force on the tooth, lb,
d = dedendum of tooth, in.,

TABLE I

TOOTH FORM FACTOR "Y" FOR LOAD AT 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	0.481	0.628
21	0.490	0.638
22	0.496	0.722
24	0.509	0.732
26	0.522	0.741
27	0.528	0.746
28	0.534	0.751
30	0.540	0.760
34	0.553	0.773
38	0.565	0.785
40	0.660	0.791
43	0.672	0.795
50	0.694	0.804
53	0.700	0.808
56	0.706	0.812
57	0.707	0.813
60	0.713	0.817
64	0.719	0.820
66	0.721	0.822
74	0.734	0.828
75	0.735	0.829
100	0.757	0.842
120	0.766	0.847
150	0.779	0.854
164	0.781	0.856
300	0.801	0.867
Rack	0.823	-----

f = face width, in.
t = tooth thickness at the root, in.

The dedendum was found by taking one half the difference of the pitch diameter and the root diameter. To do this, the root diameter was measured, and the pitch diameter was calculated from the diametral pitch and the number of teeth.

The tooth thickness at the root was determined by measuring.

10. STATISTICAL ANALYSIS OF TEST RESULTS

A statistical analysis was made of the test results obtained with lubricated teeth to see if the test data were reasonably consistent, and to establish the slopes and intercepts of the lines best representing the data.

The analysis showed that the data were satisfactorily consistent. The standard conditions were well within the range of random error deviation, and were of the same order of magnitude for all combinations of pitch and velocity tested, with the single exception of the 32-pitch gears operating at 3730 ft/min. In this one case, the standard deviation was much less than for any other gears tested, and hence was considered entirely satisfactory.

The lines determined from the statistical analysis of the results obtained with lubrication are shown with the test results in Figs. 1 through 6. Because of the limited data available, the statistical analysis did not include the 20-pitch teeth, nor the 16-pitch stub, oversize, and molded teeth.

No statistical analysis was made of the test results obtained with no lubrication. The lines representing the no lubrication data in Figs. 1 and 3 were drawn by judgment only.

C. Investigation of Gear-Tooth Temperatures

Gear-tooth temperatures were measured with a thermocouple imbedded in a cork holder contoured to fit the space between teeth. The thermocouple wire junction contacted the tooth ot the pitch point at approximately the center of the tooth face.

The temperature data were obtained by operating the gears under load for a period of time to allow the temperature to become stabilized, then shutting down and quickly placing the thermocouple against the teeth to measure the temperature rise above ambient temperature.

The results obtained with 16- and 32-pitch teeth are shown in Fig. 7. The temperatures and stresses shown are for the smaller of the two gears meshing together. The ambient temperature was approximately 80°F during this test work.

At any particular bending stress, the temperature increase was considerably greater for the lubricated 16-pitch teeth than for the 32-pitch teeth running at the same velocity. This is to be expected, since for the same stress, the 16-pitch teeth carry a much greater load than do the 32-pitch teeth. Furthermore, the sliding velocity was somewhat greater for the 16-pitch teeth than for the 32-pitch teeth. This combination of greater load and sliding velocity could logically account for the higher temperature of the 16-pitch deeth.

The 16-pitch, 27-tooth gears rotated at a higher speed than the 32-pitch, 64-tooth gears at the same velocity. Thus the load application occurred more frequently, and the time between load applications was less with the 16-pitch teeth than with the 32-pitch teeth. Furthermore, there was more mass in the blank of the 32-pitch gears than in the 16-pitch gears. All these could help account for the higher temperatures of the 16-pitch teeth.

The higher temperatures of the 16 pitch teeth could partially account for the tendency of the 16-pitch teeth to fail at lower stresses than the 32-pitch teeth, especially at high stresses and high velocities. At lower stresses and velocities, the difference in temperature between the 16- and 32-pitch teeth is not very large.

When operated without lubrication, the temperatures of the 16-pitch teeth were higher than when lubrication was provided. However, these differences in temperature do not seem to be enough to account for the lower failure stresses of the 16-pitch teeth without lubrication. As described in Progress Report No. 2, it is felt that failure without lubrication is often caused by local heating of the surface at the point of contact starting a shear type of failure. This local surface temperature could instantaneously be much higher than that measured with the thermocouple after stopping the gears.

D. Investigation of Energy Required to Cause Tooth Failure

Trying to reduce the tremendous amount of time required for fatigue testing, an attempt was made to establish a correlation between the energy lost at the gear teeth during operation, and the life of the teeth. Some energy is lost at the teeth due to friction, but it was felt that perhaps most of the energy lost at the gear teeth was absorbed by hysterisis losses due to flexing the gear teeth. If such was the case, a correlation between the energy absorbed and gear tooth failure might possibly be found. Such a correlation could

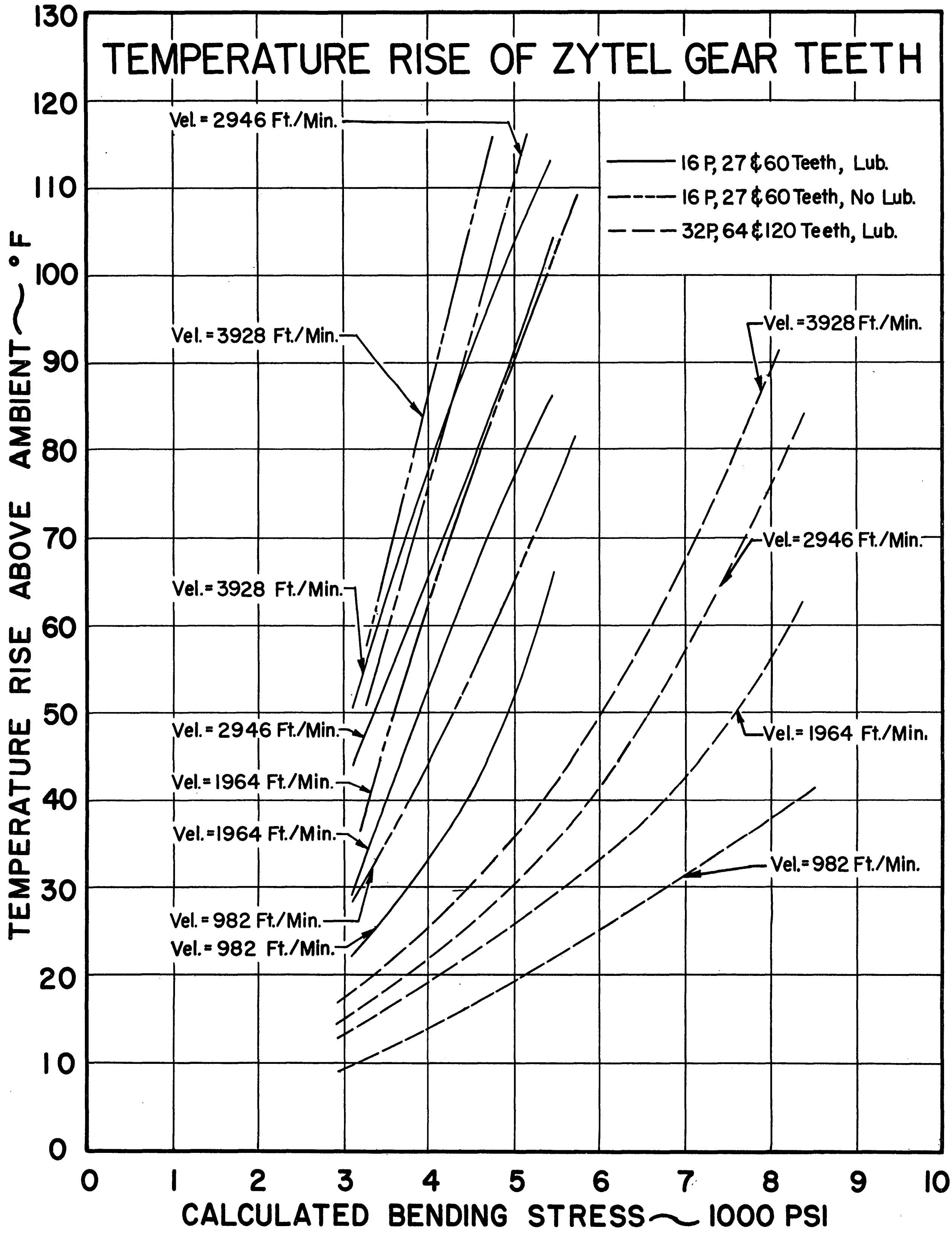


Fig 7. Temperature rise of "Zytel" gear teeth.

allow the life of the gears to be predicted by measuring the torque, and hence the energy, lost in the teeth during operation.

The method of measuring the torque lost in the gear teeth was described in Progress Report No. 2. During the fatigue testing, this lost torque was measured, and the lost energy was calculated from this measured torque, the speed of the gears, and the hours of operation to failure.

Some results of this investigation are shown in Fig. 8. The energy lost in the teeth of 4 gears is plotted against the torque which the great transmitted.

If a line such as that shown could be firmly established, it would be possible to predict the life of the gears by measuring the torque lost in the teeth while the gears were transmitting a known torque. From a line such as that of Fig. 8, the energy lost in the teeth to cause failure could be found. Having measured the torque lost in the teeth, and knowing the speed of rotation and having determined the energy to cause failure, the number of cycles to cause failure could be calculated.

The results of the first 5 tests run are shown in Fig. 8. These 5 tests seemed to indicate that such a line could be established. However, the results of subsequent testing were so widely divergent that it was felt that no further effort should be put into this investigation, and it was abandoned.

IV. CONCLUSIONS

A. Method of Calculating Torque and Horsepower Capacity of Hob-Cut Teeth

The Lewis equation presented in Section III-9 of this report can be rewritten as follows:

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

To calculate the torque capacity of the gear teeth, the equation can be further rearranged to:

$$T = \frac{S\,D\,f\,Y}{2\,P}\ , \tag{5}$$

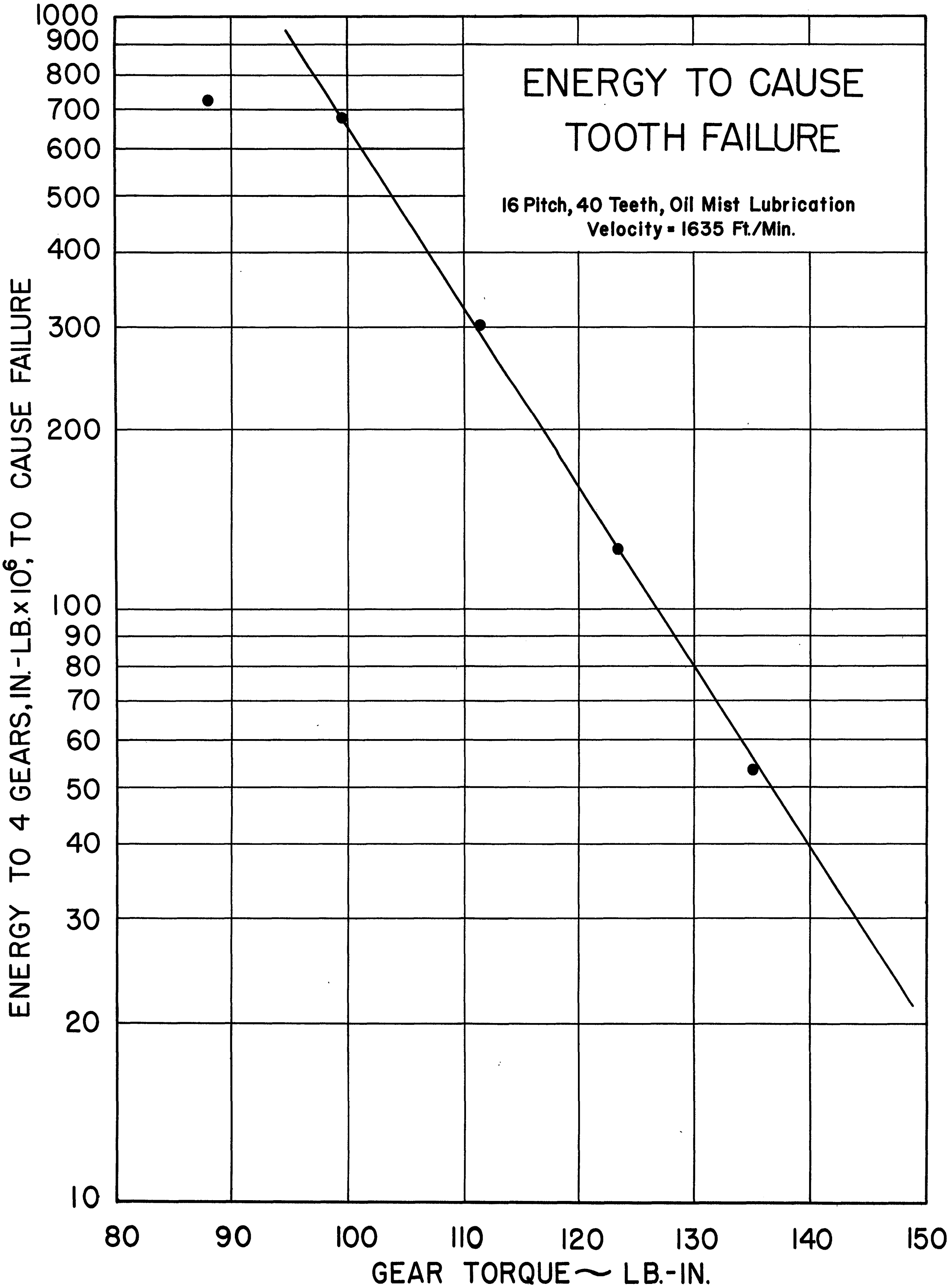


Fig. 8. Energy to cause tooth failure.

where T = torque teeth can carry, lb-in.,
S = allowable bending stress, psi,
D = pitch diameter, in.,
f = face width, in.,
Y = form factor, load at pitch point,
P = diametral pitch.

The torque which can be safely carried by a gear of known pitch, diameter, and face width can be found by using the above equation, and by selecting the allowable stress from Fig. 9 for the life desired. It should be understood that the lines of Fig. 9 show the maximum stress which should be used. Adverse operating conditions, or conservative design, would dictate allowable stresses lower than those shown.

The form factor can be taken from Table I of this report.

To calculate the horsepower capacity of the gear teeth, Eq. (4) can be written in this form:

$$Hp = \frac{S\,D\,f\,Y\,N}{126{,}000\,P}, \tag{6}$$

where Hp = horsepower teeth can transmit,
N = gear speed, rpm.

It should be kept in mind that many machines operate with varying values of torque and horsepower. When calculating the load-carrying capacity of gears for such an application, the torque or horsepower calculated by Eqs. (5) or (6) should at least equal the maximum instantaneous torque or horsepower which the machine will have to transmit. If this maximum instantaneous torque or horsepower is not known, it should be approximated by the use of a suitable service factor applied to the average torque or horsepower.

The allowable stress for Eq. (6) is taken from Fig. 9, and the form factor from Table I.

No correction need be applied for velocity, if the velocity is no more than 4000 ft/min when the teeth are lubricated. At greater velocities, it might be well to use lower stresses than indicated by the lines of Fig. 9.

To provide a reasonable safety margin, the lines shown in Fig. 9 are 25% lower than the lines which represented the test data in Figs. 1 through 6. Hence Fig. 9, and not the preceding figures, should be used when selecting allowable stresses.

It will be noticed that a single line represents the allowable stress for lubricated 16-pitch teeth in Fig. 9. The line which represented the 785 ft/min data in Fig. 1 has been omitted. It is felt that this is desirable to simplify the use of Fig. 9. Any error thus introduced is on the safe or con-

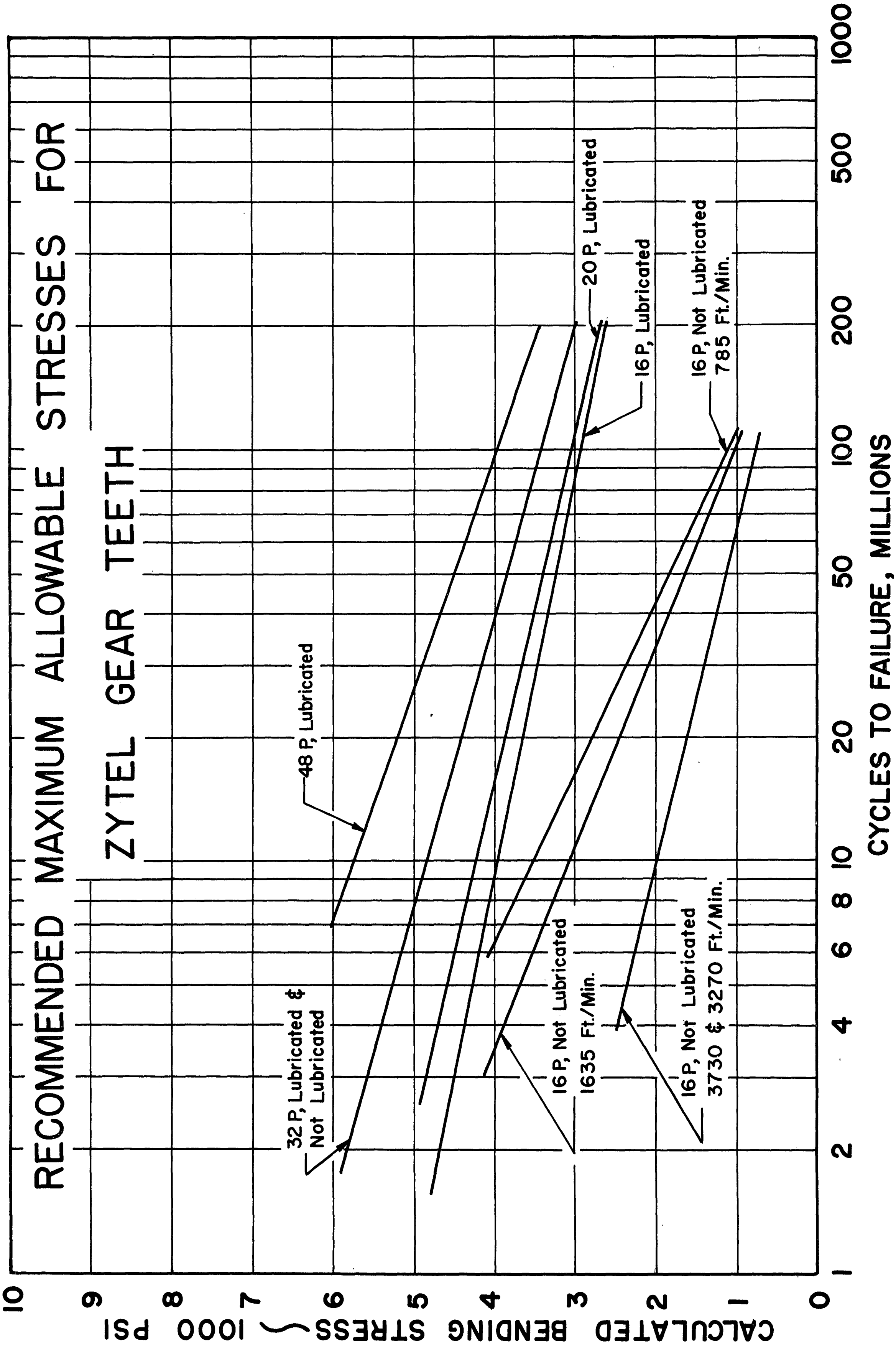


Fig. 9. Recommended maximum allowable stresses for "Zytel" gear teeth.

servative side, and hence should cause no difficulty.

B. Design Recommendations

A number of design recommendations can be made from the results of the test program. For easy reference these are listed as follows:

1. Size of Teeth.—As with metal gears, it is desirable to use the smallest teeth reasonably possible that are strong enough to carry the load.

2. 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.

3. 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.

4. Face Width.—Almost any reasonable face widths should be satisfactory, since the performance does not seem to be affected by the face width.

5. Accuracy of Manufacture.—It appears to be desirable to have the teeth as accurately made as is reasonably possible. The life of the gears seem to be shortened when the teeth are not accurately formed and spaced.

6. Backlash.—Backlash must be provided, but performance does not seem to be affected by reasonable variations in backlash. For high-speed or heavy-load operation, which may heat the teeth, the backlash should be somewhat more liberal than that used for metal gears.

7. 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.

V. PROPOSED FUTURE WORK

The following proposals are made for future work:

1. Fatigue tests will be conducted on hob-cut "Delrin" gears to obtain a comparison of "Zytel" and "Delrin" as a gear material.

These fatigue tests are in progress at the time of this writing.

2. A fatigue testing program, similar to the one just completed with hob-cut teeth, will be carried out with "Zytel" and "Delrin" gears having molded teeth. Thus design data and the torque and horsepower capacity of molded teeth will be established.

3. An industrial speed reduction unit, in which the steel helical gears are replaced with "Zytel" helical gears, will be tested to verify further the results obtained on the test machines. An a-c induction motor and a positive displacement oil pump will supply and absorb the required power. The equipment is now being set up for this work.