THE UNIVERSITY OF MICHIGAN
INDUSTRY PROGRAM OF THE COLLEGE OF ENGINEERING

DESIGN OF SPUR GEARS OF "ZYTEL" NYLON RESIN

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ABSTRACT

This paper presents a method of calculating the load carrying capacity of gears made of du Pont "Zytel" nylon resin. A number of design recommendations are given, and the use of gears of "Zytel" in a helical gear speed reducer is described.

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INTRODUCTION

The main objective of this paper is to present a method of calculating the load carrying capacity of spur gears of "Zytel", and to show how this method was established. However, this might well be called a progress report on the work being done at the University of Michigan to establish reliable design methods for gears made of this material. This work is being sponsored by the Polychemicals Dept. of E. I. duPont de Nemours and Company, and is still going on at the time of this writing.

Leading up to the main objective, it is felt desirable to present some information on the "Zytel" nylon resins, to describe the experimental program which has been conducted with gears of "Zytel", and to show some of the test results from which the load carrying capacity has been established. "Zytel" nylon resin, the test equipment, and some preliminary results were described in paper 56-SA-43, presented at the ASME semi-annual meeting at Cleveland, June, 1956. To present a complete picture of what has been done to date, some of the first parts of that paper are repeated here.

MATERIAL

"Zytel" is a nylon resin manufactured by E. I. duPont de Nemours & Company. It is thermo-plastic, and is supplied in the form of granulated powder from which many articles can be molded by conventional molding techniques. The molded material has relatively light weight, is rigid and tough, has a low coefficient of friction, and good resistance to abrasion.

"Zytel" is made in several nylon molding powder compositions, of which "Zytel"101 is recommended for mechanical parts. All of the information in this paper refers to "Zytel" 101, but to avoid needless repetition, the suffix 101 has been omitted.

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

The physical properties are also somewhat affected by the moisture content of the molded material. The physical properties shown in Figure 1 are for material which has been conditioned to have $2\frac{1}{2}\%$ moisture by weight. This represents the amount of moisture absorbed at equilibrium, with air at 50% R. H. and 73°F.

TEST MACHINES

To carry on the test work required to establish the load carrying capacity of gears of "Zytel, five identical test machines were designed and built. These machines operate on the "back-to-back", or "four-square" principle, with two pair of gears in each machine loaded against each other. In this way the driving motor supplies only the power to overcome friction and windage in the machine.

A schematic layout of the test machine is shown in Figure 3. The gears to be tested are mounted on hollow shafts supported in ball bearings. Two steel torsion bars connect the hollow shafts. A

FIGURE 1

"Zytel" Nylon Resin
2.5% Moisture Content

Tensile Strength	73°F	11,200 psi
Yield Strength	73°F	8,500 psi
Elongation	73°F	300 %
Modulus of Elasticity	73°F	175,000 psi
Shear Strength	73°F	8,000 psi
Impact Strength, Izod	73°F	2.0 ft-lb/in.
Hardness, Rockwell	73°F	R 108, M59
Specific Gravity	73°F	1.14

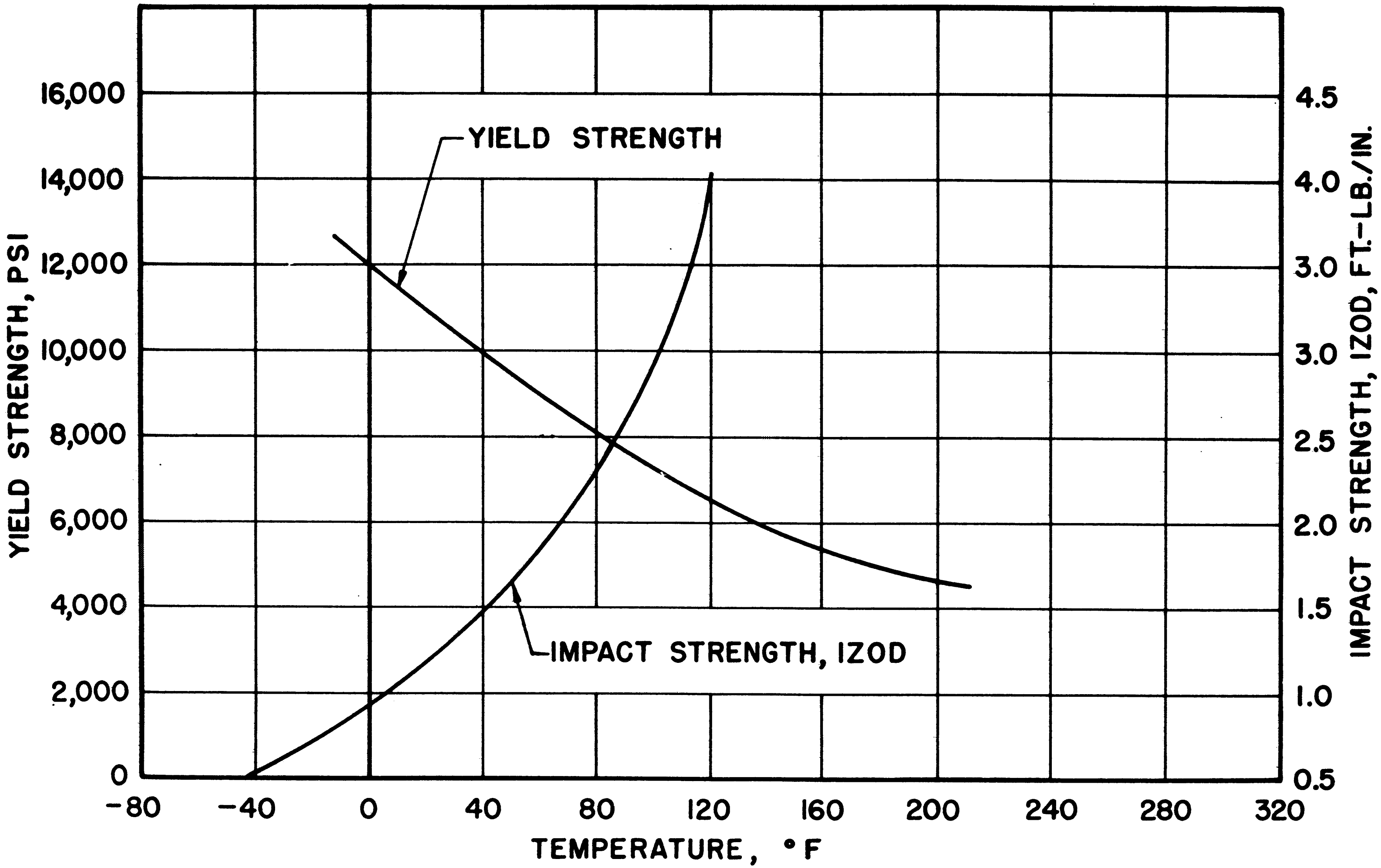


Figure 2. Effects of Temperature on Physical Properties of ZYTEL 101

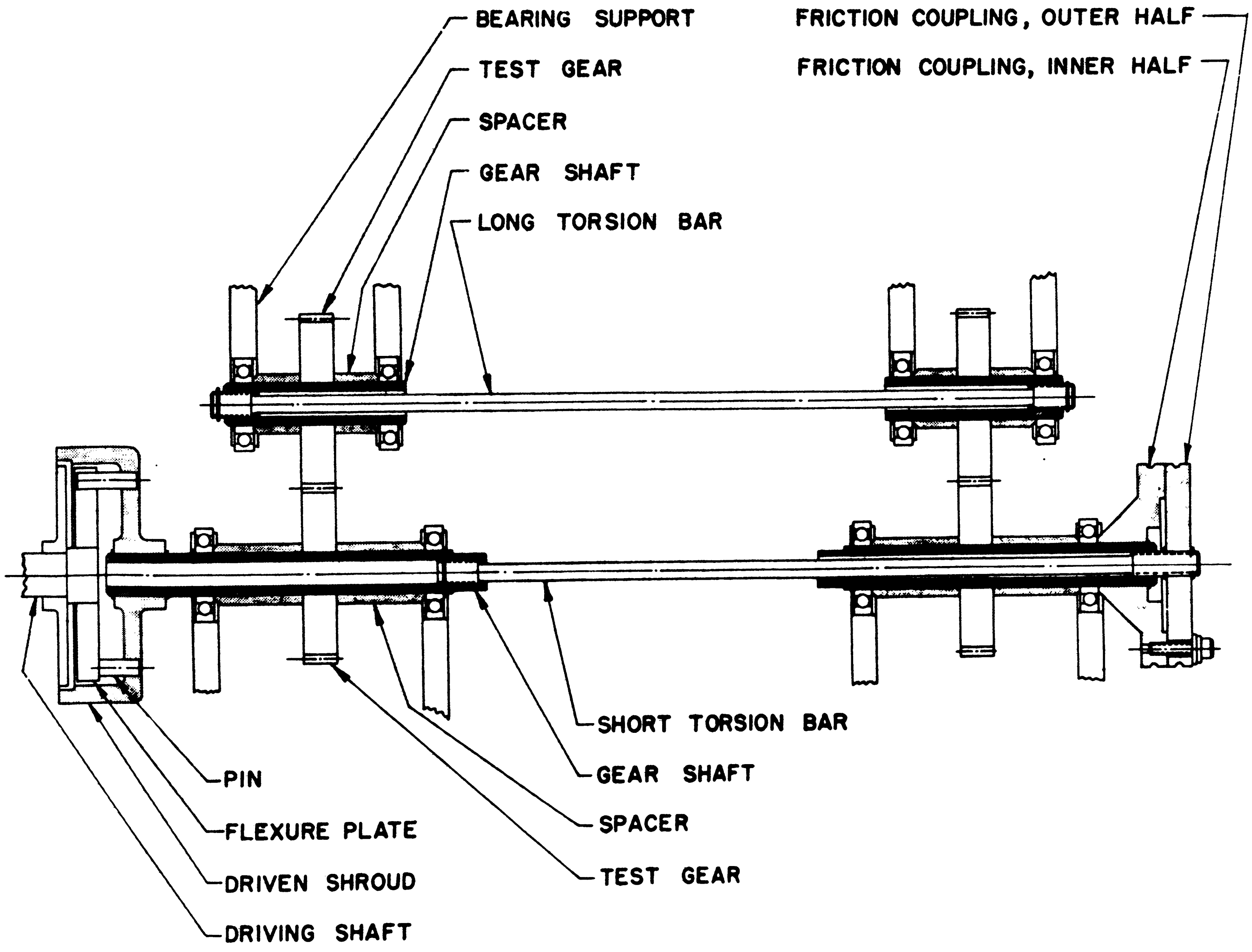


Figure 3. Schematic Layout of Test Machine

friction coupling at the right end is used to twist the torsion bars, thus providing the desired load on the gear teeth.

Figure 4 shows one of the test machines, with the friction coupling in the foreground and the driving motor behind. The bearing supports for the right hand shaft of Figure 4 are movable, thus allowing various sizes of gears to be tested. The gears shown in the machine have $2\frac{1}{2}$" pitch diameters, which might aid in visualizing the actual size of the test machine.

Load is applied to the gear teeth by twisting the friction coupling with a system of small wire ropes and weights operating on the pulleys shown in the foreground of Figure 4. After clamping the friction coupling with the screws provided, the wire ropes are removed to allow the couplings to rotate. Lead shot in buckets is used as the weight. Any desired twisting moment can be applied to the coupling by changing the amount of shot in the buckets.

Each machine is individually driven by a constant speed motor through a speed variator, providing a 600 to 5000 rpm speed range. An individual oil mist lubricating system is provided on each test machine to lubricate gears and bearings.

TEST GEARS

A fairly large variety of gears have been tested. Many of the gears were machined from molded disks of "Zytel". The teeth of these gears were cut with hobs after the disks had been machined to size. Pitch diameters of these cut gears ranged from 1.375" to 3.75",

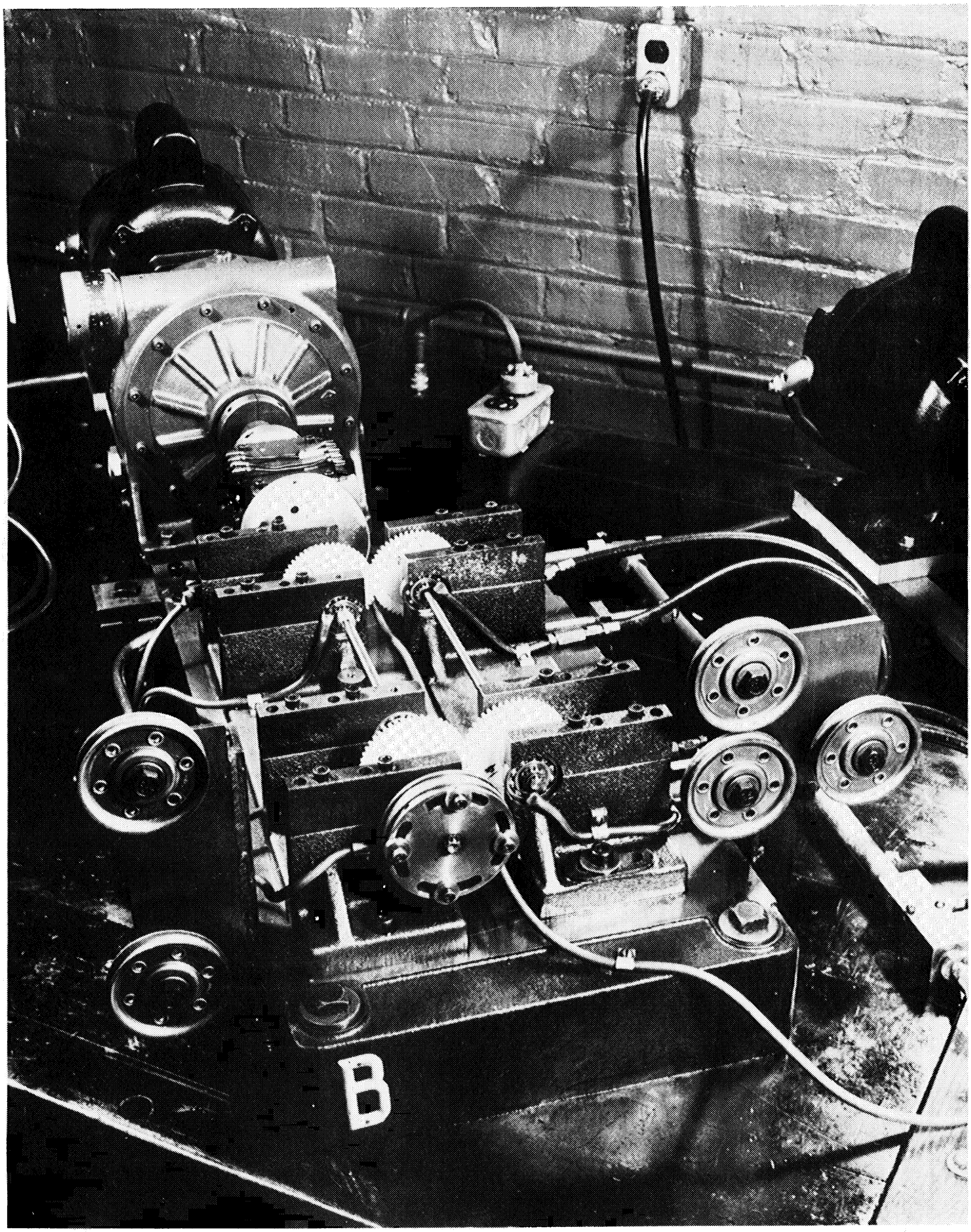


Figure 4. Test Machine

with diametral pitches of 16, 20, 32 and 48. Figure 5 shows an assortment of such gears. All but a few had a 20° pressure angle and full depth teeth, with face widths varying from 7/32" to 7/16". Accuracy limits of the teeth of these gears ranged from AGMA Commercial Class 2 to Precision Class 1, with most of the teeth being within the limits of Commercial Class 2 or 3.

In addition to the cut gears described above, $2\frac{1}{2}$" diameter 20 pitch and 32 pitch gears having molded teeth have been tested. These gears, which are shown in Figure 6, also have a 20° pressure angle, with full depth teeth and 1/2" face width. The web which joins the hub and rim is 1/8" thick.

A single cavity mold, with a ring gate at the hub was used to form these test gears. The action of the ring gate can be fairly well visualized from Figure 7, which shows a gear sectioned through the gate before its removal. The molded 20 and 32 pitch teeth were within the accuracy limits of Commercial Class 1 and 2 respectively, and it is felt that this accuracy would have been difficult to obtain without a ring gated mold.

TESTS AND RESULTS

An extensive life testing program was first conducted with the machined gears having cut teeth, since variations in gear diameter, diametral pitch, face width, pressure angle, etc., can be most easily accomplished with cut teeth. Pitch line velocities ranged from 785 to 3730 ft/min, both with and without lubrication of the teeth. In some cases pairs of identical gears ran together, while in others small pinions were used with larger gears.

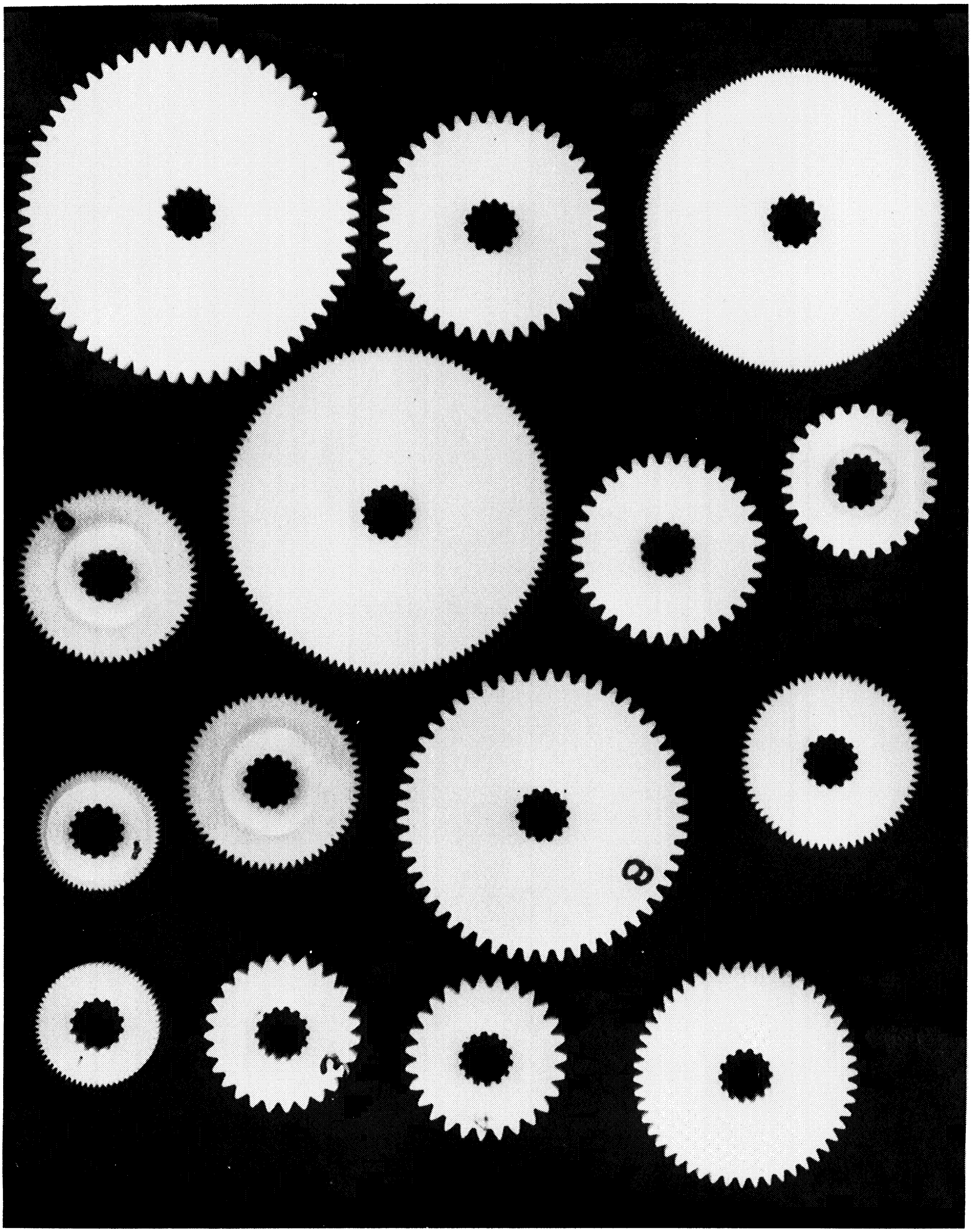


Figure 5. Test Gears with Hob Cut Teeth

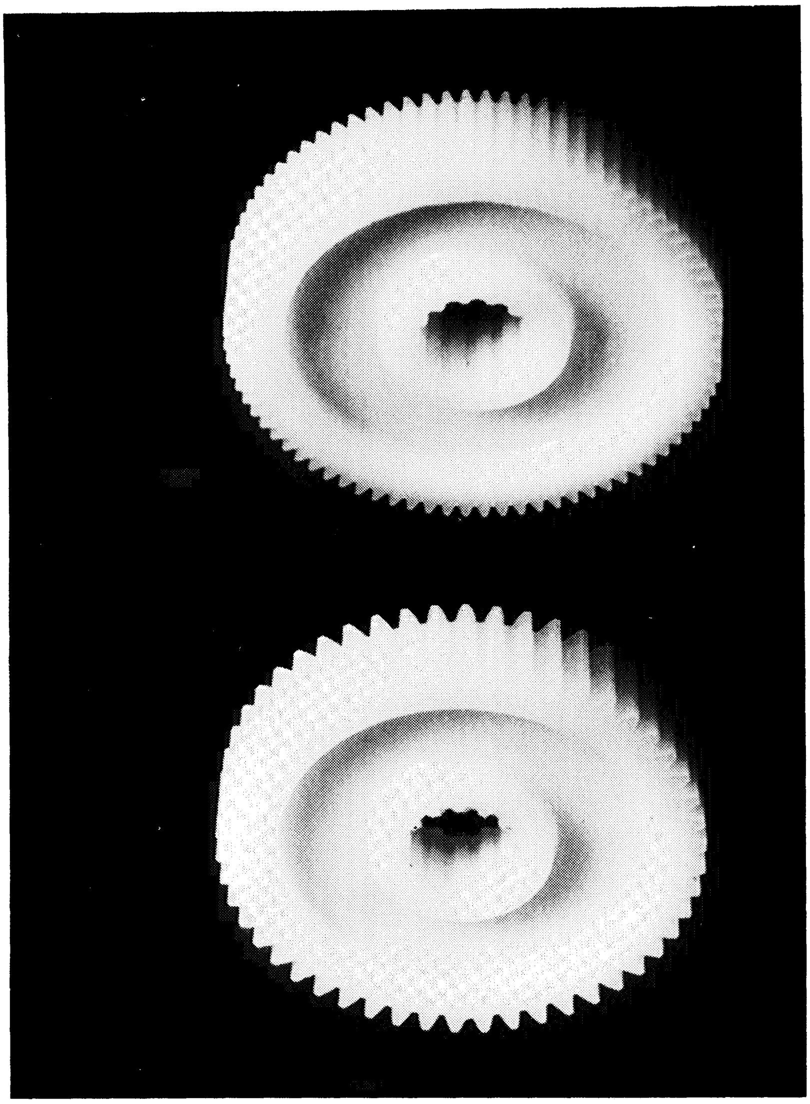


Figure 6. Test Gears with Molded Teeth

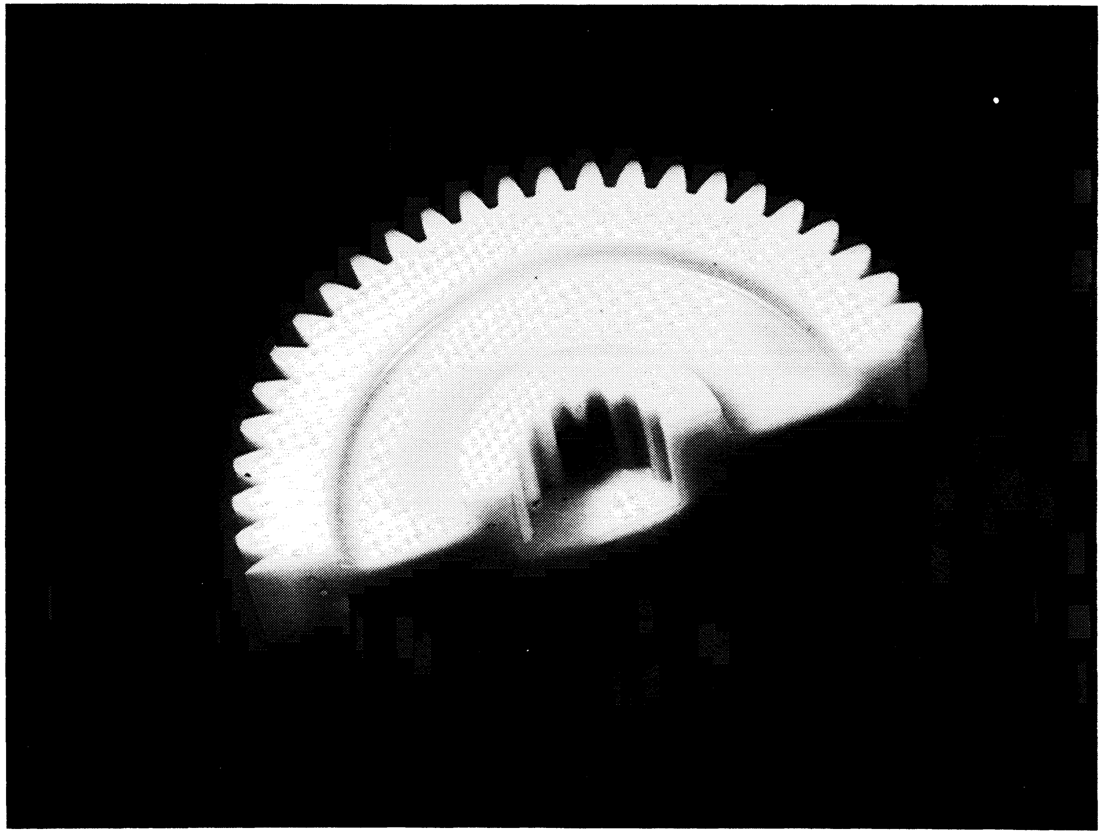


Figure 7. Molded Gear, Sectioned to Show Ring Gate

During this work the machines ran continuously, 24 hours per day, being stopped only occasionally for visual inspection or to replace broken gears. When a gear failed, that gear and the gear with which it meshed were replaced and this was counted as a single failure.

During most of the test work, the teeth were lubricated with oil mist. When no lubrication was desired, the oil mist system was turned off, and the bearings were lubricated with grease. The gears were generously lubricated with oil at the start of the no-lubrication tests, but were not lubricated thereafter.

Ambient temperature and humidity were not controlled at any time. The temperature varied from 68° to 92°F, and the relative humidity varied from 29 to 85% while the test work was in progress.

Some of the results obtained with the gears having cut teeth are shown in Figures 8 and 9. Similar but less extensive data were obtained for 20 and 48 pitch gears having cut teeth.

Figures 10 and 11 show the results obtained with the 20 and 32 pitch gears having molded teeth. Some of these gears had been tested as molded, with no subsequent treatment. Others had been moisture conditioned, and still others had been annealed.

Figure 11 shows that the life of the 32 pitch molded gears was considerably greater at any given load than that of the gears with cut teeth. However, Figure 10 does not show this same increase in life for the 20 pitch molded teeth as compared to those with hob cut teeth. It is believed that the relatively poor showing of the 20 pitch molded teeth, particularly at the higher stresses, was due to the presence of residual stresses in the rim of most of the molded gears tested, and to

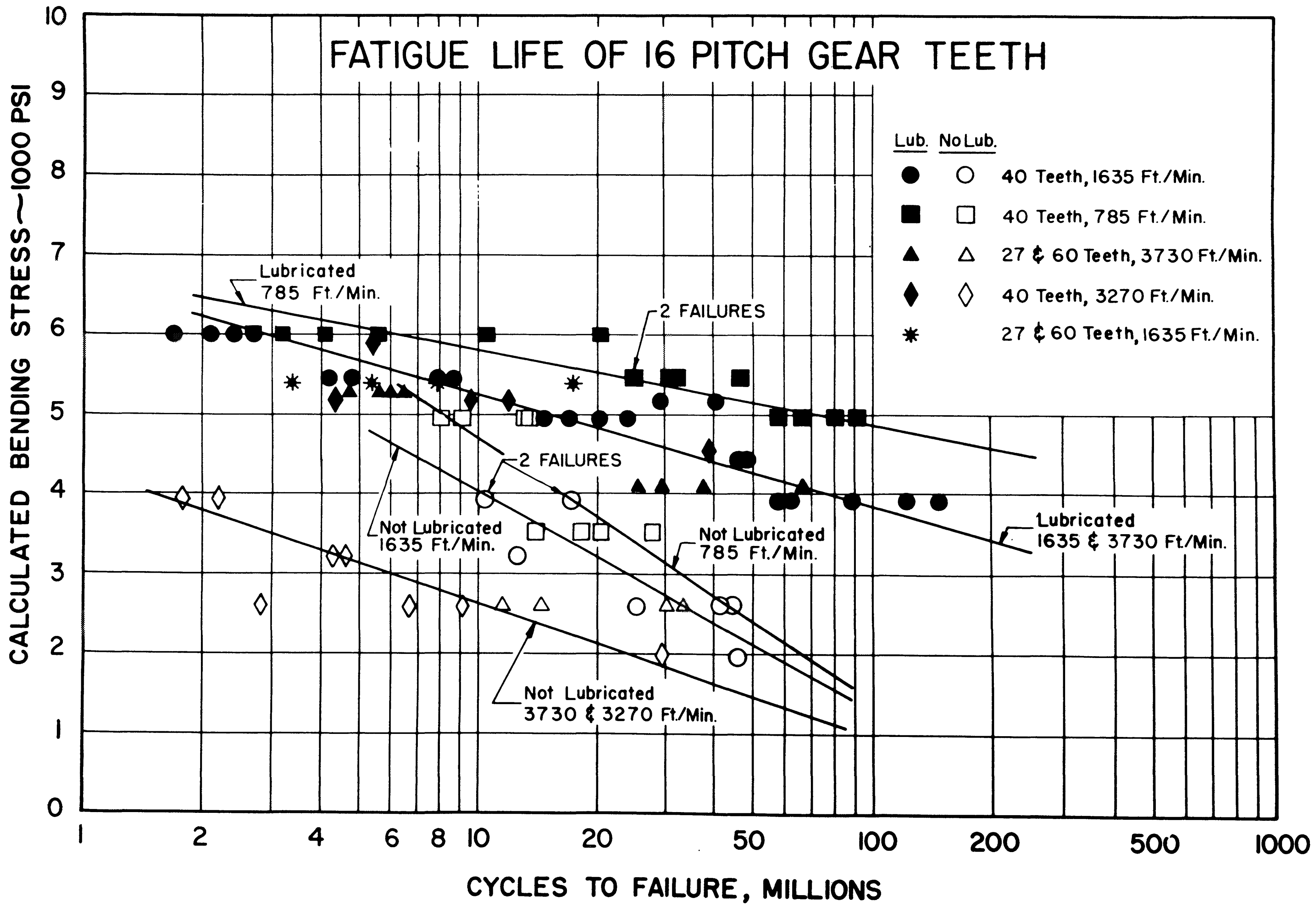


Figure 8. Fatigue Life of 16 Pitch Hob Cut Gear Teeth

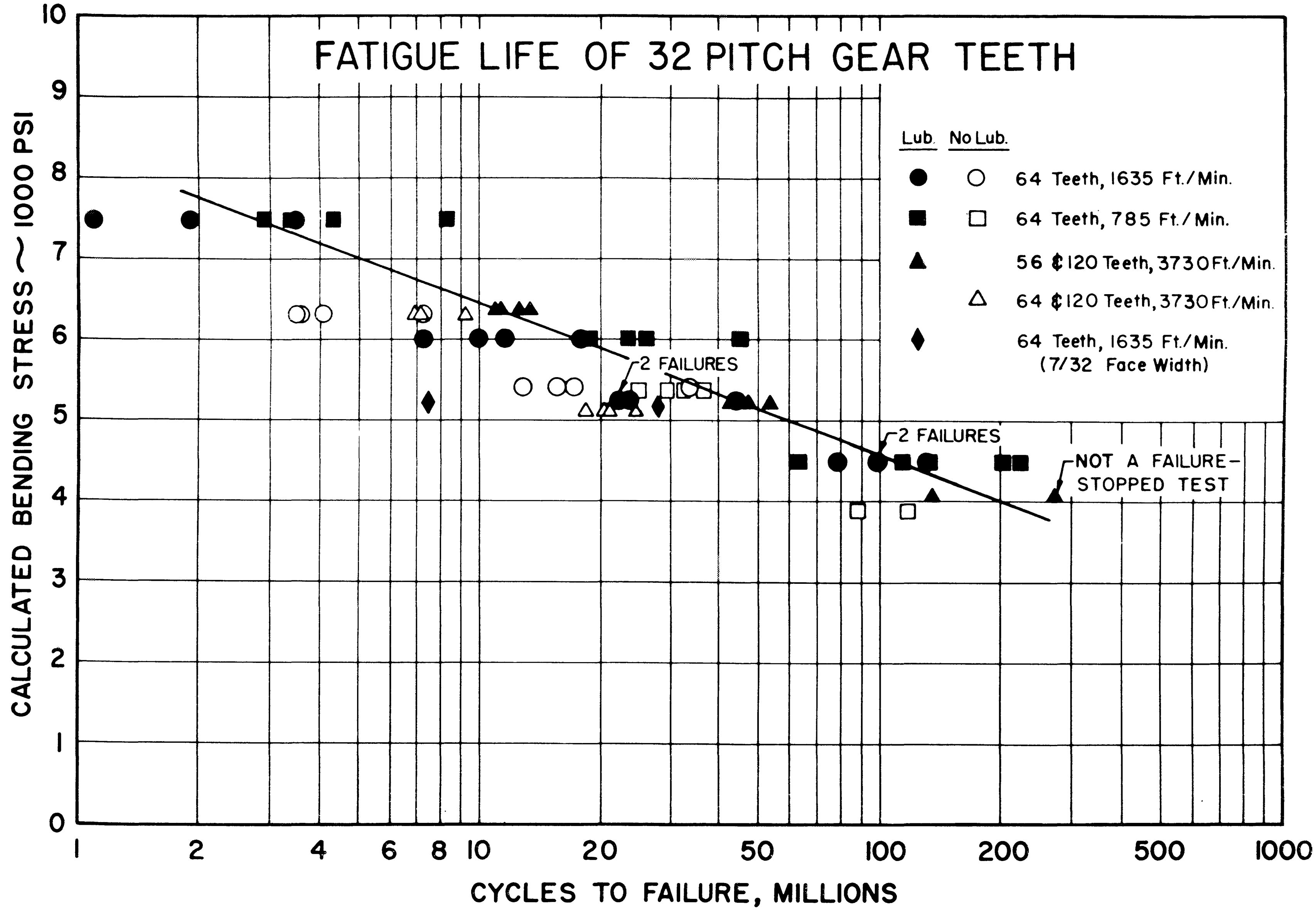


Figure 9. Fatigue Life of 32 Pitch Hob Cut Gear Teeth

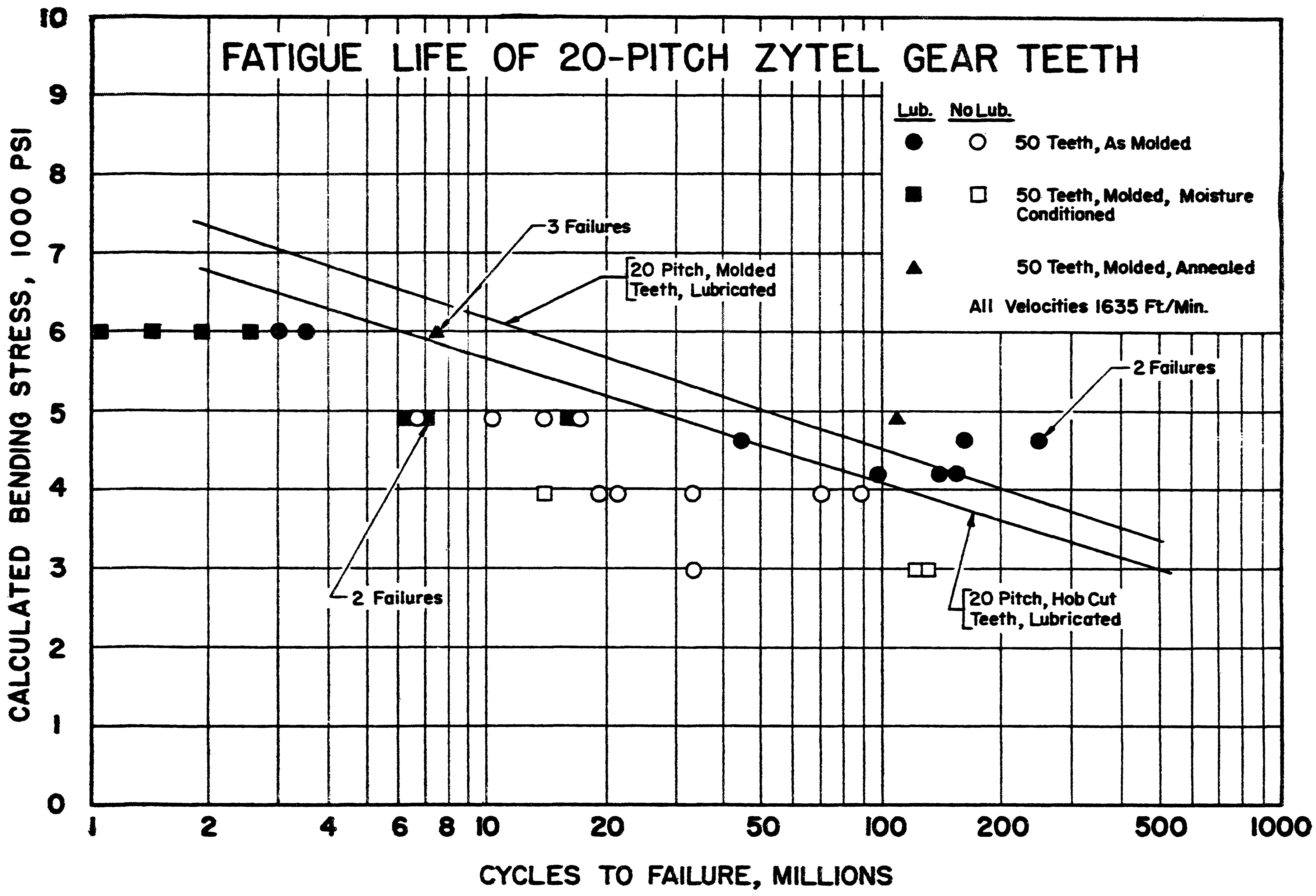


Figure 10. Fatigue Life of 20 Pitch Molded Gear Teeth

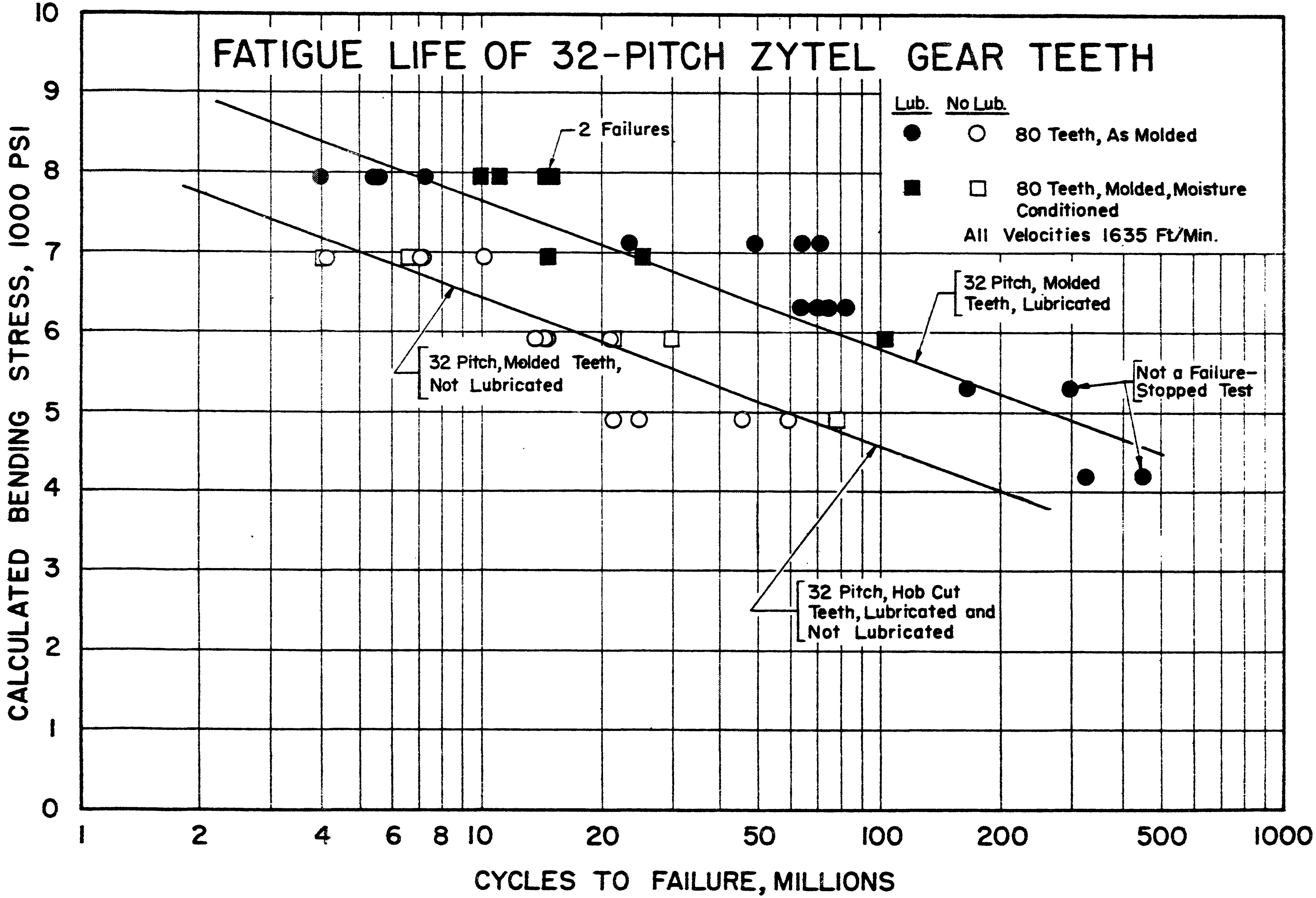


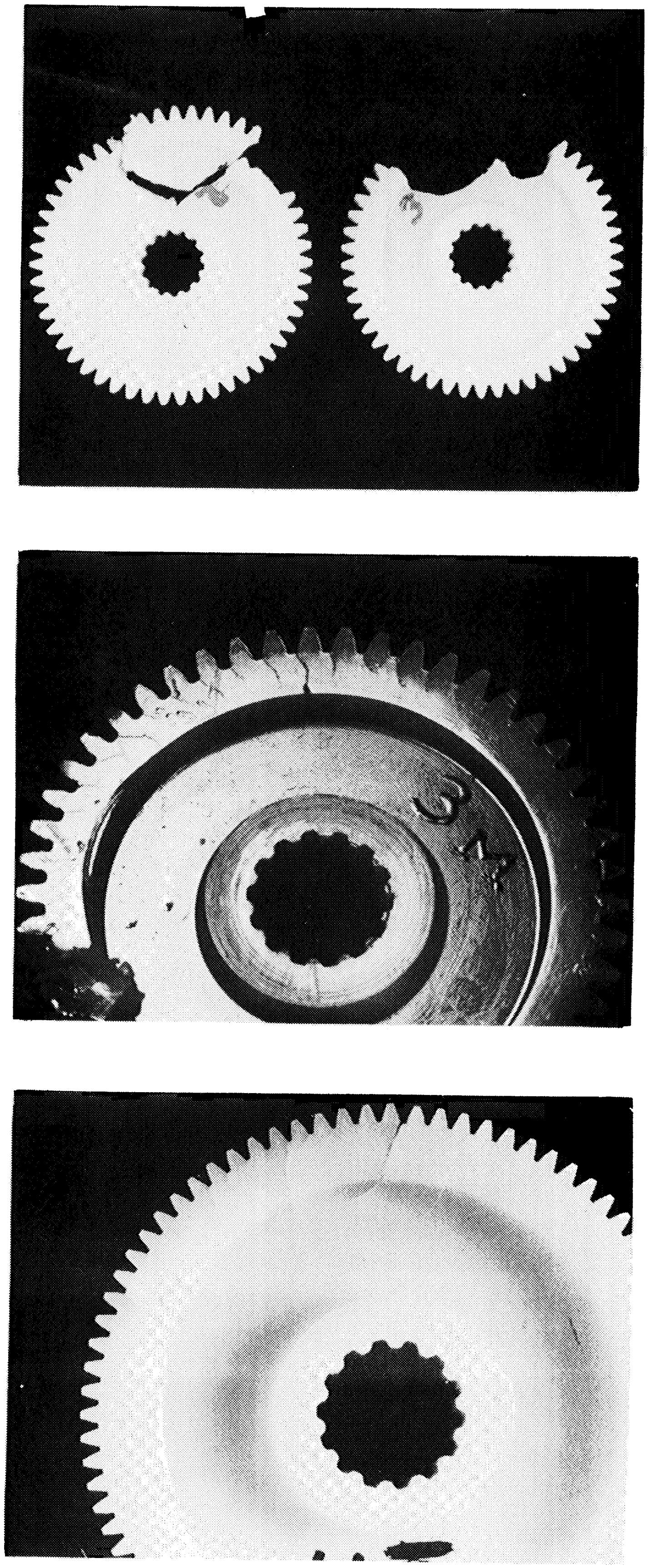
Figure 11. Fatigue Life of 32 Pitch Molded Gear Teeth

the thin web which did not provide enough support for the rim and teeth at high loads.

Evidence of residual stresses being present can be seen in Figures 12a, b and c, which show failures of some of the 20 pitch molded gears. Rather than failing at the root of a tooth in a typical bending fatigue failure, the gears of Figure 12a and b failed by breaking through the rim. This type of failure was preceded by the formation of cracks at the roots of the teeth, as shown in Figure 12c. It was not uncommon to have these cracks form at the root of virtually every tooth on the gear after a relatively short period of operation. The gears would then continue to run for a long period of time, during which time one or more of the cracks would progress through the rim to the web. When the crack reached the web, a piece generally broke out to end the test.

When the 20 pitch gears were annealed to remove the residual stresses, the life of the gears was considerably increased, and the line of Figure 10 is drawn to represent the test results obtained with the annealed gears. The annealed gears failed at the root in a typical fatigue failure as shown in Figure 13.

At higher loads there was evidence of distortion of the 1/8" thick web of the 20 pitch gears, and it was believed that this adversely affected the life of the 20 pitch molded gears. Nevertheless, the molded teeth showed a definite improvement over the cut teeth in the sizes tested. Although no 16 or 48 pitch molded gears were tested, it



A

B

C

Figure 12. Fatigue Failure of 20 Pitch Molded Gear Teeth, Not Annealed

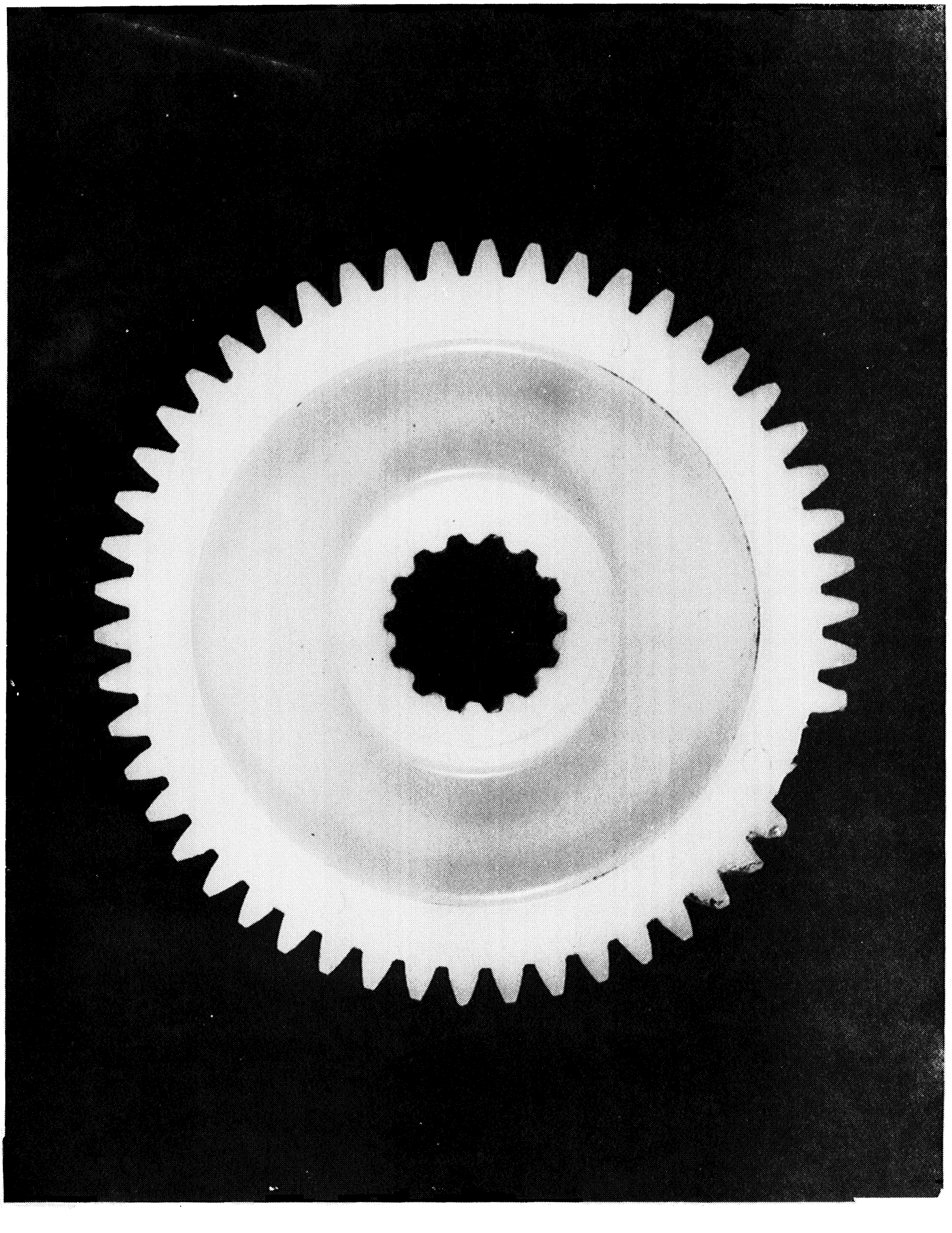


Figure 13. Fatigue Failure of 20 Pitch Molded Gear Teeth, Annealed

was concluded that molded gears of those pitches should have somewhat greater life at any given load than the cut teeth tested. This line of reasoning was followed to establish allowable stress curves of Figure 15.

The stresses plotted in Figures 8 through 11 were calculated by the Lewis Equation for bending stress in a gear tooth:

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

where S = bending stress, psi

F = tangential force on tooth, lbs

f = face width, inches

Y = form factor, load near pitch point

The above equation considers all the load to be carried by one tooth when in contact near the pitch point. A photographic analysis of the teeth under load, both stationary and in motion, showed this is essentially the condition which exists with the 16 pitch teeth. With smaller teeth, however, some of the load is carried by the adjacent teeth. Nevertheless, the above equation is simple to use, and calculating the stress in this manner results in the points falling along a straight line within the range of random error deviation.

The form factor Y was calculated from the tabulated figures on page 477 of Buckingham's "Analytical Mechanics of Gears", McGraw-Hill. Buckingham's values have been multiplied by 3.1416, to obtain the values shown in Figure 14.

FIGURE 14

Tooth Form Factor Y for Load Near Pitch Point

Number of Teeth	20° Full-Depth Form	20° Stub-Depth Form
15	-----	0.556
16	-----	0.578
17	-----	0.587
18	-----	0.603
19	-----	0.616
20	0.544	0.628
22	0.559	0.648
24	0.572	0.664
26	0.588	0.678
28	0.597	0.688
30	0.606	0.698
34	0.628	0.714
38	0.651	0.729
43	0.672	0.739
50	0.694	0.758
60	0.713	0.774
75	0.735	0.792
100	0.757	0.808
150	0.779	0.830
300	0.801	0.855
Rack	0.823	0.881

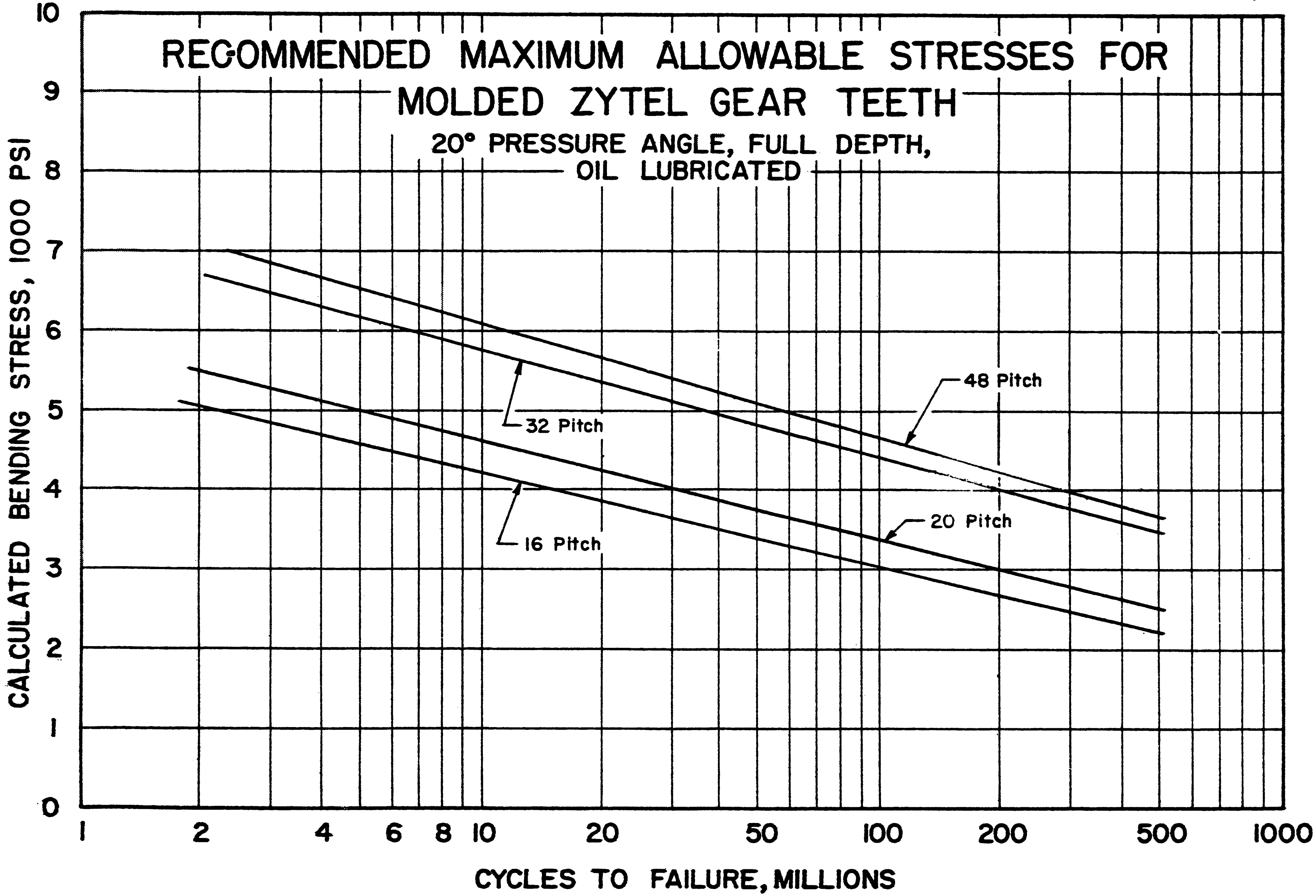


Figure 15. Recommended Max. Allowable Stresses for Molded Gear Teeth

LOAD CARRYING CAPACITY OF GEARS OF "ZYTEL"

The Lewis equation which was used to calculate the stresses in the gear teeth can also be used to calculate the load carrying capacity of the teeth by using the test data to establish allowable stresses. However, this equation is generally more useful when re-written in either of the following forms:

$$T = \frac{S\ D\ f\ Y\ K}{2\ P} \qquad (2)$$

where T = torque gear can transmit, lb-in

S = allowable stress, psi, from Figure 15

D = pitch diameter of pinion, inches

f = face width, inches

Y = form factor for pinion, from Figure 14

K = design factor, from Figure 16

P = diametral pitch

or:

$$HP = \frac{S\ D\ f\ Y\ N\ K}{126{,}000\ P} \qquad (3)$$

where HP = horsepower gear can transmit

N = pinion speed, RPM

What are considered to be maximum recommended stresses for molded and lubricated teeth are shown in Figure 15. To provide a reasonable margin of safety, the lines shown in Figure 15 have been reduced 25% from the lines which represented failure of the gears on test.

Figure 15 shows higher allowable stresses for small teeth than for large teeth. This is in part due to the fact that the load is distributed among more teeth when the teeth are small, even though the method of calculation considers the entire load to be carried by one tooth. Furthermore, the larger teeth tend to run at a higher temperature than the smaller teeth. This has the effect of weakening the larger teeth.

It should be kept in mind that Figure 15 shows the maximum stresses which should be used when the load is reasonably steady with virtually no shock or impact loading. These should be used with reasonable judgment by the designer, and should be further reduced to compensate for overloading, impact or shock loading, or other such conditions of operation.

The design factor K, shown in Figure 16, compensates for those cases where cut teeth may be used instead of molded teeth, for lack of lubrication, and for velocities in excess of 4000 ft/min. Other than this, no correction need be made for velocity.

It would seem that the use of Equation 2 or 3 to determine either the torque or horsepower capacity of gears of "Zytel" is fairly straight forward, hence no examples are given. Within reasonable limits, it should be possible to approximate allowable stresses for pitches other than those shown in Figure 15. It would seem unwise, however, to carry this to extremes and use the data of Figure 15 to establish allowable stresses for teeth which are much larger than those tested. The combination of large loads and high sliding velocities encountered in large teeth might exceed some limiting value not reached with the teeth tested, beyond which the teeth

FIGURE 16

Values of Design Factor K for Use with

Lewis Equation and Design Charts

Teeth	Lubri-cation	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

may fail by surface deterioration rather than by bending fatigue. In such a case, the load carrying capacity would be limited by resistance to wear rather than by bending fatigue strength.

It would also seem unwise to apply the stresses of Figure 15 to teeth having pitches finer than 48, unless the dimensions of the teeth are very carefully controlled. Small errors could greatly reduce the carrying capacity of such small teeth.

To facilitate the design of gear teeth of "Zytel", Equation 2 and the stresses of Figure 15 have been combined to form the design charts of Figures 17, 18, 19 and 20. These design charts can be used to establish the torque capacity of any given gear, or can be used to establish the size of gear required to transmit a given torque. A design factor K, from Figure 16, is used with the charts to increase the torque which must be transmitted, or to decrease the torque which the gear is capable of transmitting.

A few brief examples will illustrate how the charts are used.

Example 1: A 32 pitch pinion having a 1/2" face width is to transmit a torque of 54 lb inches at 1760 RPM.

The teeth will be formed by molding, and will be lubricated. A life of 100 million revolutions is desired. What should be the pitch diameter of this pinion?

Figure 16 shows a design factor of 1.0 for molded and lubricated teeth with a pitch line velocity below 4000 ft/min, hence the design torque will be the same as the actual torque, 54 lb-inches.

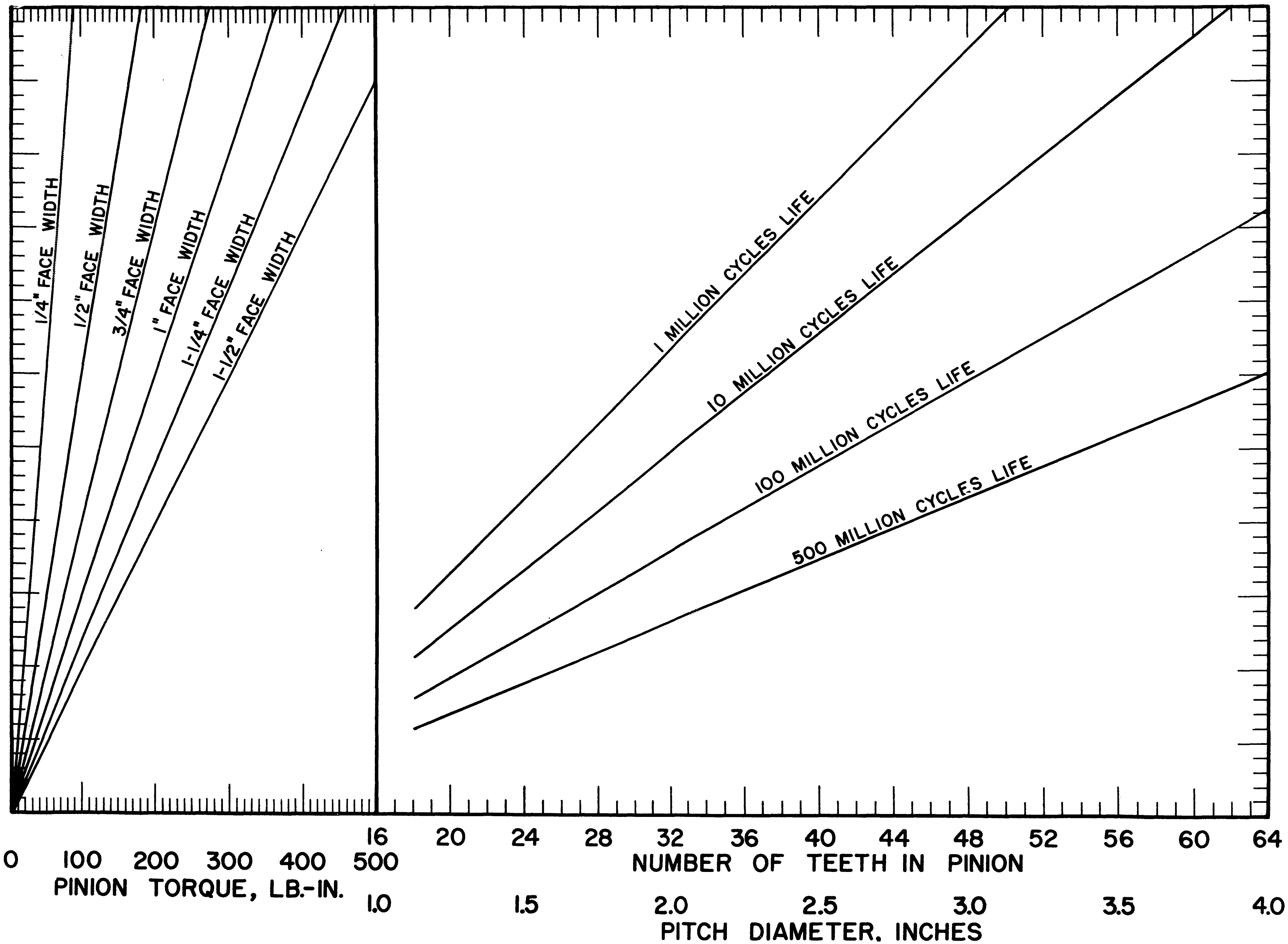


Figure 17. Design Chart for 16 Pitch Gears

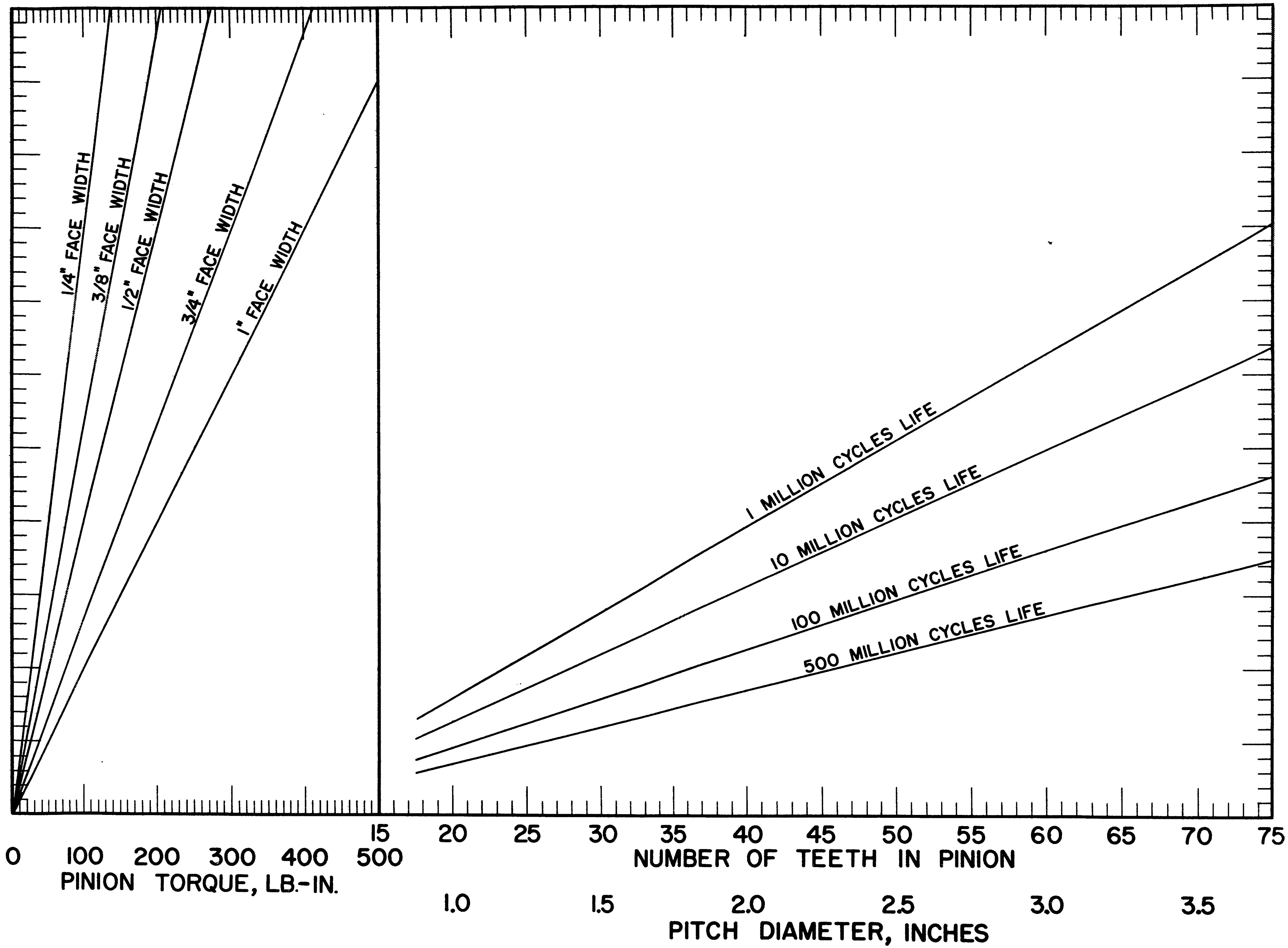


Figure 18. Design Chart for 20 Pitch Gears

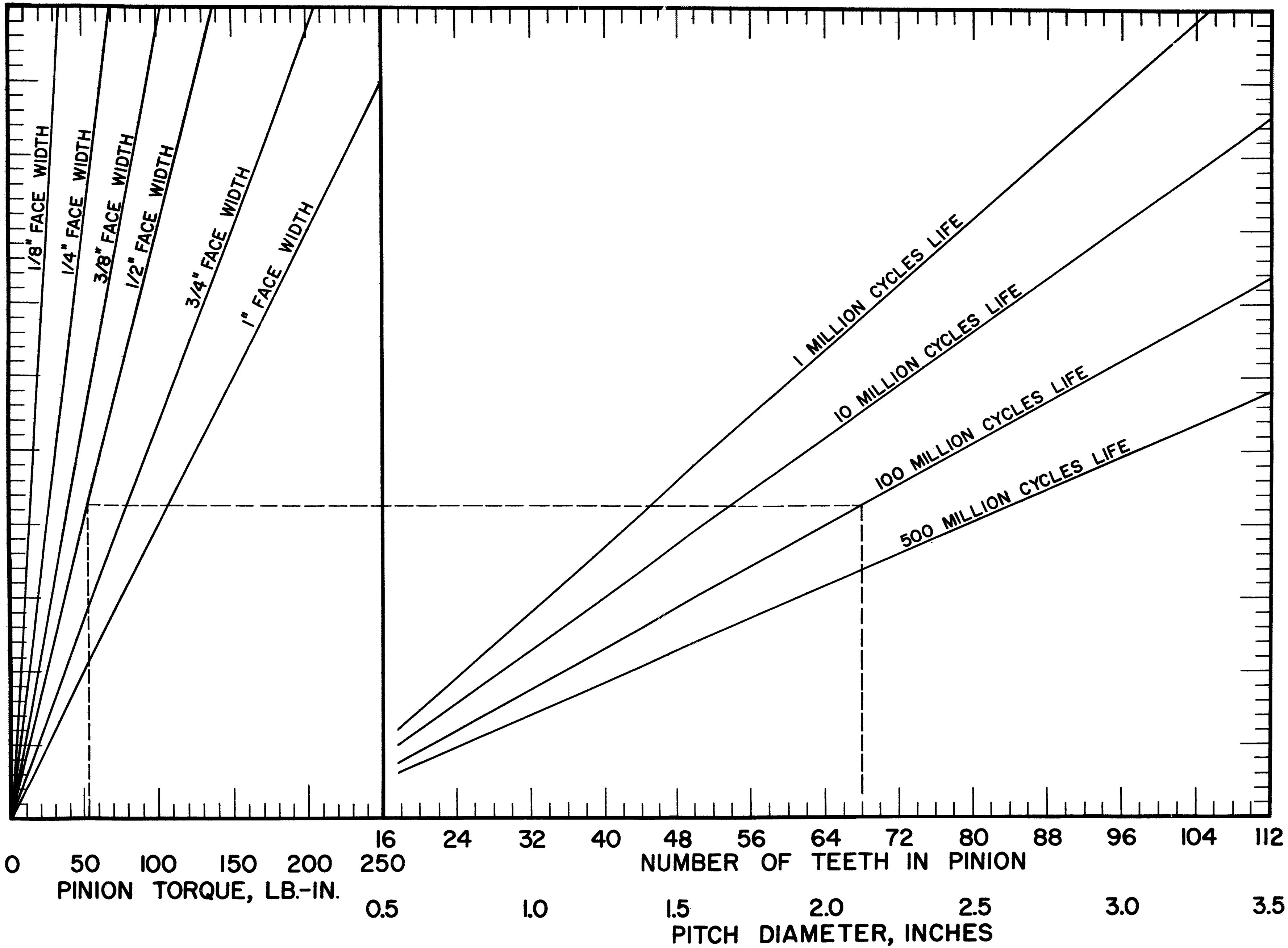


Figure 19. Design Chart for 32 Pitch Gears

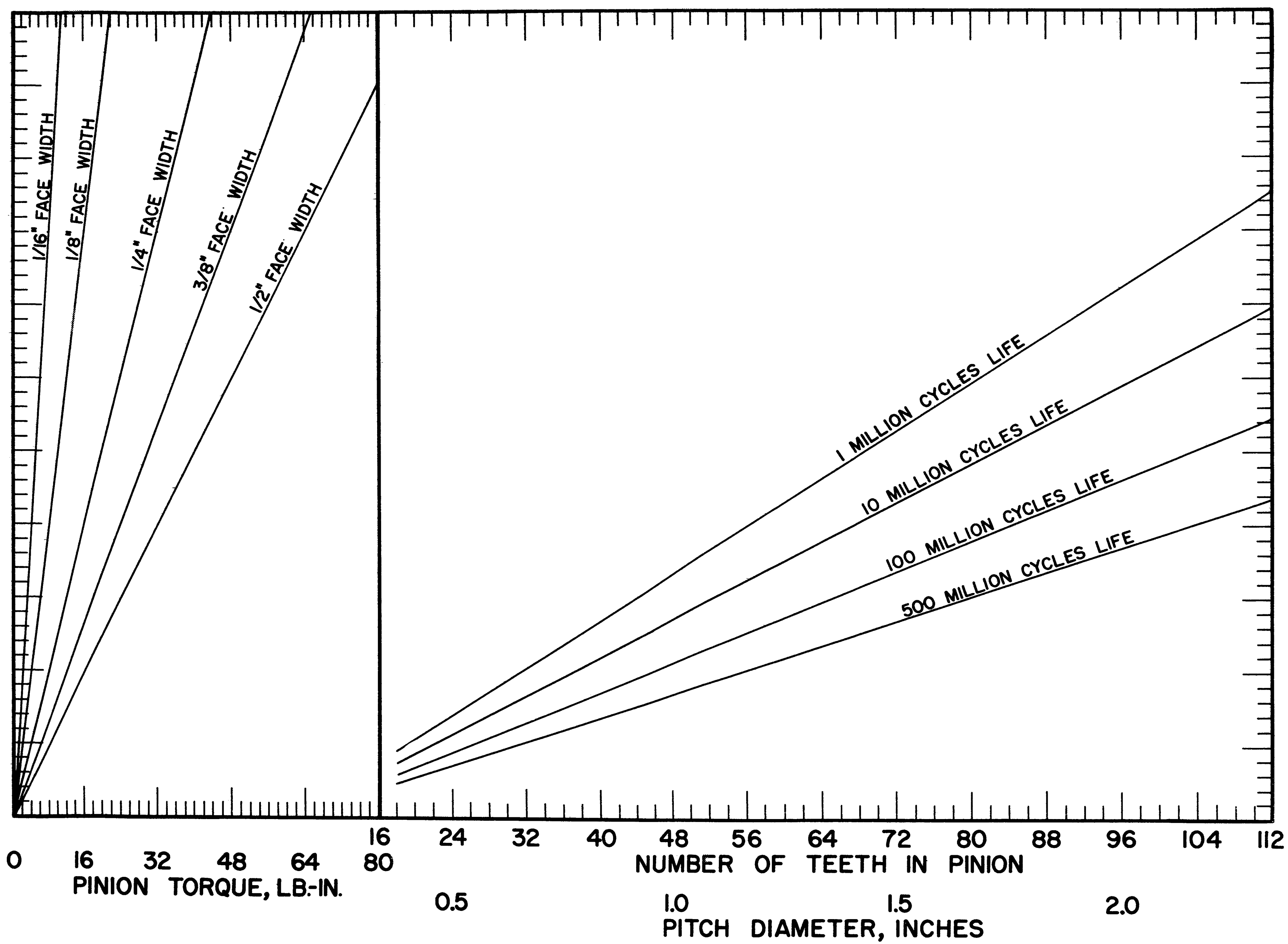


Figure 20. Design Chart for 48 Pitch Gears

The solution is shown by the dashed lines of Figure 19, the Design Chart for 32 pitch teeth. The dashed line is started at 54 lb-inch on the torque scale, is extended upward to intersect the 1/2" face line, then is drawn horizontally to the right to intersect the 100 million cycle line, then downward to find the number of teeth to be 68, and the pitch diameter 2.125 inches.

Example 2: What diameter would be required for the pinion of the preceding example if the teeth had been formed by hobbing, and were to operate with no lubrication?

Figure 16 shows a design factor of 0.70 for 32 pitch cut teeth without lubrication. Then the design torque will be:

$$\text{Design torque} = \frac{54}{0.70} = 77.1 \text{ lb-in}$$

The solution is similar to that of Example 1, except that the line is started at 77.1 lb-in on the torque scale, thus resulting in a pitch diameter of 3.0 inches, with 96 teeth.

Example 3: What torque could the pinion of Example 1 transmit if the teeth were not lubricated?

Figure 16 shows a design factor of 0.80 for 32 pitch molded gears without lubrication when the velocity is below 4000 ft/min. Then the torque which the pinion can transmit will be:

$$\text{torque} = 54 \text{ lb-in} \times 0.80 = 43.2 \text{ lb-in.}$$

As when calculating the load capacity of the gears by means of Equations 2 or 3, the results obtained from the design charts should be modified by the judgment of the designer to allow for impact or overloading which may be anticipated.

OTHER DESIGN RECOMMENDATIONS

The following design recommendations are based on information obtained from various tests and investigations with gears of "Zytel":

1. Size of Teeth: As with metal gears, it is desirable to use the smallest teeth which are strong enough to transmit the necessary power. The limitations of the load carrying calculations should be considered when determining the size of teeth to be used.

2. Pressure Angle and Tooth Form: Full depth teeth with a 20° pressure angle appear to be quite satisfactory. The full depth teeth may be strengthened by making the blank oversize, but this must not be carried to extremes, since the strength gained by making the tooth thicker at its base may be more than offset by the weakening effect of higher temperature caused by the increased normal force and sliding velocity.

   The 16 pitch, 27 tooth, 20° pressure angle gears made with blank diameters 1/16" oversize had greater life than these same gears made with standard full depth teeth. Similar gears with 20° stub teeth, and with 30° stub teeth, had less life than the 20° full depth teeth.

3. Accuracy of Manufacture: It is desirable to have the teeth as accurately made as is reasonably possible. The life of the gears are shortened when the teeth are not accurately formed and spaced.

4. 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 given below:

Diametral Pitch	Backlash, inches
16	0.004 - 0.006
20	0.003 - 0.005
32 and finer	0.002 - 0.004

For high speed or very heavy load operation, which may heat the teeth, the backlash should be somewhat more liberal than the values listed.

5. 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 3 times the thickness of the teeth at the pitch circle, the web thickness should be about equal to the rim thickness, the hub diameter at least $1\frac{1}{2}$ times the shaft diameter, and the hub length at least equal to the shaft diameter.

If the gear is large and the teeth are to be heavily loaded, it is better to mount the gear of "Zytel" on a metal flange rather than key or spline the gear directly to the shaft.

Molded gears should be designed to reduce or eliminate the formation of residual stresses, and the mold should preferably have a ring gate to maintain concentricity.

6. 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 prevent the formation of these stresses, 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.

ADDITIONAL INVESTIGATIONS AND RESULTS

A considerable amount of experimental work has been done in addition to the life testing previously described. Some of the most interesting is work including here.

From a rather extensive wear testing program carried on with cut gears, it was concluded that tooth wear is insignficant in the sizes tested, whether the teeth are or are not lubricated. Inspection of a number of the molded gears after long periods of operation showed that this was also true for the molded teeth.

To try out the load carrying calculations on something other than the test machines, a Boston helical gear speed reducer was equipped with the helical pinion and gear of "Zytel" shown in Figure 21. These

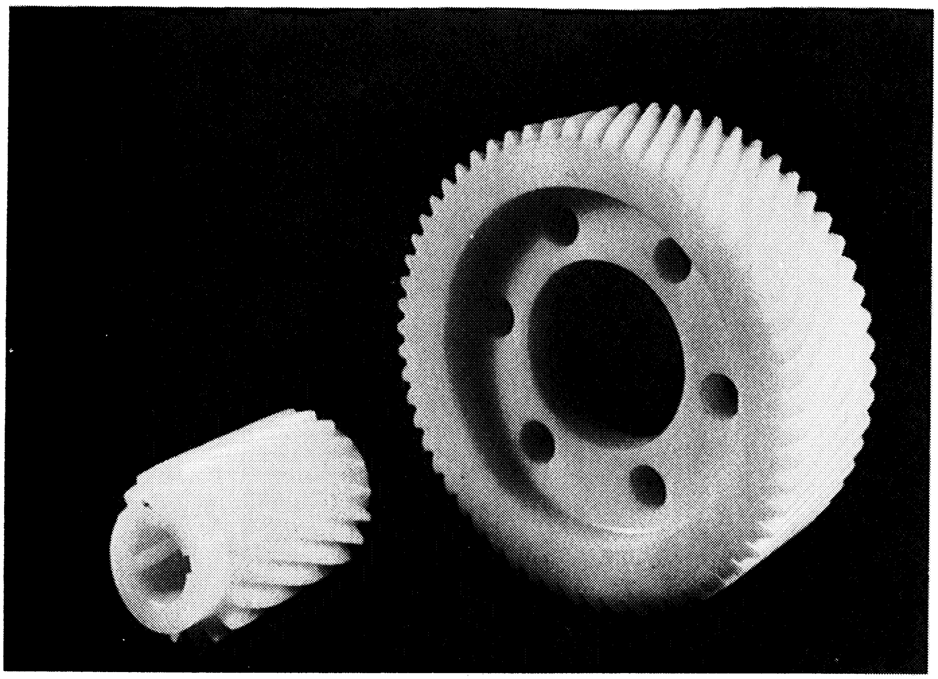


Figure 21. Helical Pinion and Gear, Hob Cut

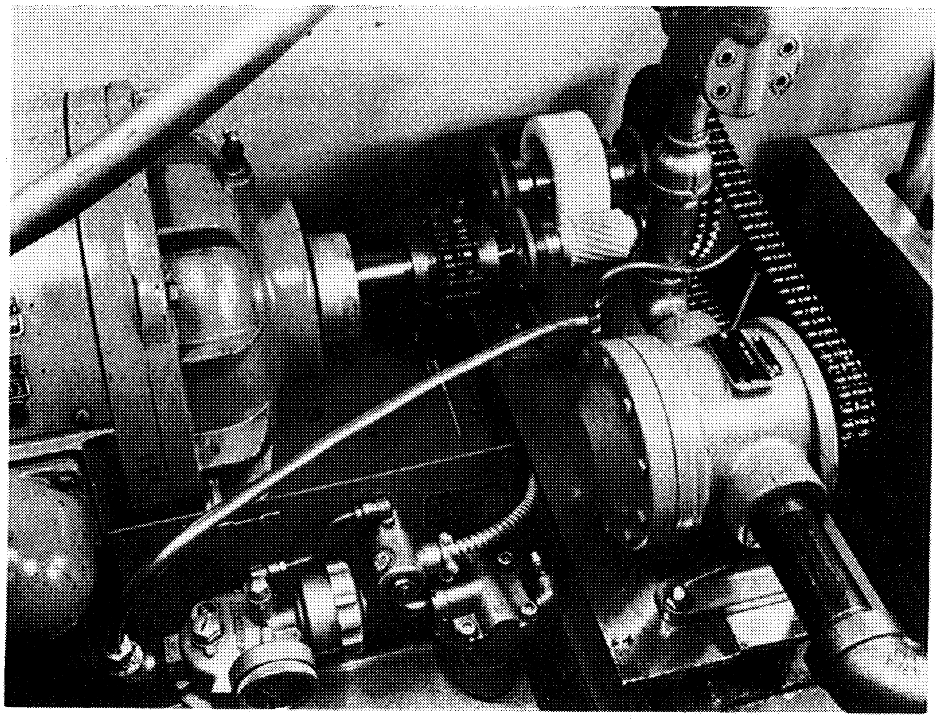


Figure 22. Helical Gear Speed Reducer Test Stand

have a normal pitch of 12, with 19 and 57 teeth, and a 23.56° helix angle. The pinion is 0.030" oversize on the outside diameter, while the gear is this same amount undersize. The face widths for the pinion and gear are 2.25" and 2.0" respectively.

The gears of "Zytel" replaced 10 pitch steel gears with face widths of 2.25" and 2.50". With the steel gears the reducer had a rating of 10 HP at 1760 RPM of the pinion. The gears of "Zytel" transmitted $7\frac{1}{2}$ HP at 1760 RPM for 2091 hours before failure. This represents 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 predicted life based on our test machine data.

Figure 22 shows the gears installed in the speed reducer. A vane pump, driven by a roller chain is used to load the speed reducer. The gears of "Zytel" survived the shearing off of all the teeth in the sprocket mounted on the reducer, and an oil leak in the gear housing which allowed the teeth to run without lubrication for an extended period of time.

Figure 23 shows some results of an investigation of gear tooth temperature. The temperature shown were obtained by operating the gears under load for a period of time to allow the temperature to become stabilized, then stopping the machine quickly and measuring the tooth temperature with a thermocouple imbedded in a cork holder contoured to fit the space between the teeth. This work was done with gears having cut teeth, and the ambient temperature was approximately 80°F.

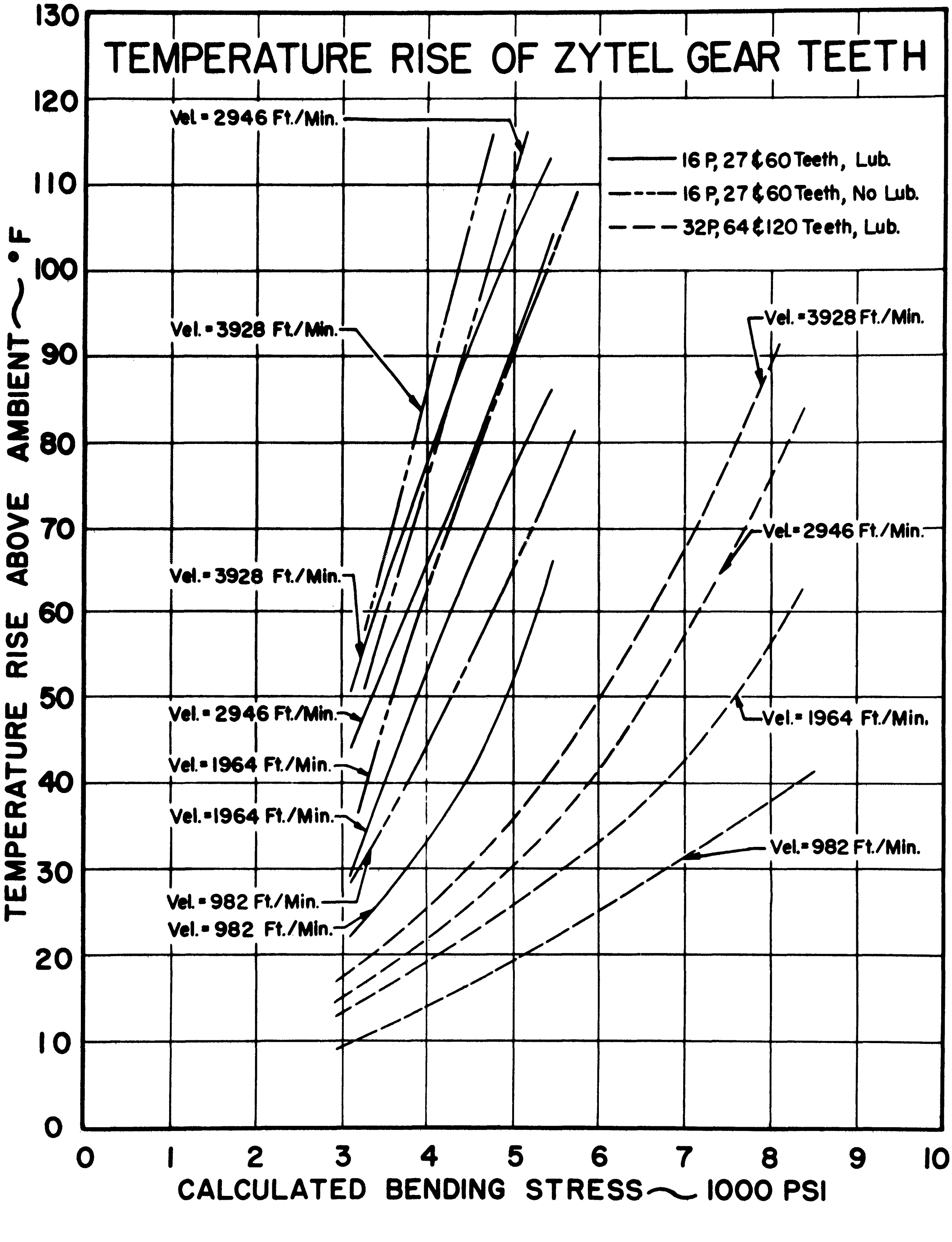


Figure 23. Temperature Rise of Gear Teeth

At any particular bending stress, the temperature increase was considerably greater for the lubricated 16 pitch teeth than for the lubricated 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 is somewhat greater for the 16 pitch teeth. This combination of greater load and sliding velocity could logically account for the higher temperature of the 16 pitch teeth.

It is felt that the higher temperature of the larger teeth accounts in part for the lower stresses at which these larger teeth fail. However, no simple method of predicting the temperature rise has been established.

Some experimental work concerning the application of "Zytel" to gearing is continuing at the present time, although the life testing program is now being concentrated on a new material. Gears of "Zytel" meshing with steel gears are under test, and an investigation of the relationship of sliding velocity, contact pressure, friction coefficient, and wear is in progress. Results will be published when available.

In the meantime, it is hoped that the information in this paper will be useful to those who may design gears of "Zytel".