b-12 CHARACTERISTICS OF REGIMES OF GEAR LUBRICATION

Darle W. Dudley
DUDLEY ENGINEERING COMPANY, INC.
17777 Camino Murrillo
San Diego, CA 92128
United States of America

Gear teeth operate in three relatively different regimes of lubrication. At very slow pitch line speeds there is boundary lubrication only. Scoring and pitting are particularly troublesome. Very thick oils with strong additives can help improve Regime I conditions.

At medium pitch line speeds there is mixed lubrication. Pitting life is improved. Lubricant requirements are still quite critical in Regime II.

At faster pitch line speeds and high speed, a full EHD oil film is obtained. In Regime III, lubricants can be thinner and mild additives are usually sufficient. Scoring becomes a hazard due to the flash temperature developed in the oil film.

INTRODUCTION

All gear operation can be divided into three kinds of gear lubrication.

Regime I - Almost no useful film due to pitch line velocity.

Regime II - Partial oil film from pitch line velocity.

Regime III - Full oil film developed by pitch line velocity.

In the early 1950's attempts were made to calculate oil film thickness between contacting gear tooth surfaces and to explain why a high speed gear unit could run for years and still have finish marks visible over about 90% of the contacting tooth surface.

In a book (1) published in 1954 I reported this data on page 296:

"In one recent test made by the author a set of precision spur gears were operated for 20 million cycles at a pitch line speed of 6,000 fpm. Involute measurements showed wear to be less than 0.0001". Then the set was run at 1,000 fpm with the same torque. After 5 million cycles at this speed the set was stopped. Wear was measured at 0.0005"."

In the 1960's the problem of oil film thickness between gears and bearings was largely solved. The discovery that oil viscosity increases greatly when oil is trapped in the Hertzian contact zone between gear teeth made it possible to calculate the elastohydrodynamic (EHD) oil film thickness and get values that seemed reasonable in comparison with surface finish values and observations of wear in gears. The work of Dowson and Higginson is a classic (2) in this field. Some observations of mine on gear wear in this time period were given (3) at a world conference on elastohydrodynamic wear in Leeds, England in 1965.

In the 1970's there was increasing realization that the tendency of gears to pit or score was considerably influenced by the adequacy or inadequacy of the EHD oil film. The published and unpublished work of the late Charles Bowen was particularly effective in pointing out that the rating of gears from the pitting standpoint was greatly influenced by the regime of lubrication. His 1977 ASME paper (4) is particularly valuable in this regard.

In 1980 ASME published the WEAR CONTROL HANDBOOK. In developing the material for a chapter on gear wear (5) I became convinced that pitting was one of the most serious causes of gear wear. This led me to sort out the relation of pitting to regime of lubrication in more depth than had been done previously. It seems obvious that serious pitting has to be avoided to prevent serious (or premature) wear. Granting this, the gear designer or user must have a clear headed understanding of pitting characteristics in each of three regimes of lubrication if he is to avoid or control the pitting hazard so that the wear in gears is under control.

In this paper I will discuss some general characteristics of the regimes of lubrication. Much gear development work is needed in the 1980's to fully understand these regimes and to develop gear rating formulas for pitting and scoring that are properly matched to each regime. I hope that this paper will be helpful in pointing out some of the considerations involved.

GENERALITIES OF REGIMES

Regimes of lubrication need to be thought of like world climate zones. For instance, we all have a concept of an arctic climate, a temperate climate and a tropical climate. The way of life is quite different in each climate zone.

When we get specific about climate it gets difficult. Alaska has much arctic climate, but some places like Anchorage, Alaska are as temperate as some cities in Japan or Germany. Japan and the United States are normally thought of as temperate, but some locations in both southern Japan and the southern part of the United States are definitely tropical.

The same thing goes for regimes of lubrication in gears. Normally a spur or helical gear at a pitch line velocity of 0.5 m/s (100 fpm) and a finish of 1 µm (40 micro-in.) would be in Regime I. If there is an extra heavy oil and good oil additives, the teeth may polish up to a finish around 0.3 µm and the gear set may operate up to 80° C with essentially Regime II conditions prevailing. (The EHD oil film in this case would be around .1 µm.) This is a special case where favorable conditions permit a gear set to operate in a better regime than would normally be expected.

As an opposite situation, we might think of a high speed gear pair running at a pitch line speed of 50 m/s. Normally such gears would have a finish around 0.5 µm and would use a medium weight turbine oil at temperatures around 60° C. The operation would be well into Regime III.

If this part had a finish of only 0.7 µm, was operating at 100° C and was using a light turbine oil, the EHD film might be only around 0.4 µm. At the moderate tooth loads used for high speed gears, there would be very little finish improvement in running. Such a situation can (and does) often lead to operation in Regime II even though the "climate zone" should have been Regime III operation.

Pitch Line Velocity

The most important variable in setting regimes of lubrication is pitch line velocity. Figure 1 shows what might be taken as nominal positions of the three regimes of lubrication with respect to pitch line velocity.

Figure 1 is drawn for carburized gears at 700 HV (60 HRC). It is assumed that the accuracy and finish are normal for the pitch line speed.

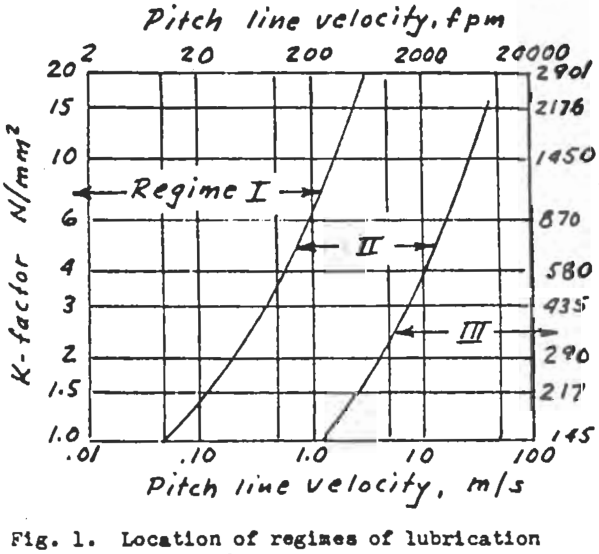

Fig. 1. Location of regimes of lubrication for average gears

(High speed gears are made more accurately than low speed gears.) Another assumption is that the fast running gears use a typical turbine oil, while slow running gears use the heavier oils typical of vehicle or industrial practice.

The boundary lines between Regime I and II and between Regime II and III are no more certain than the line on a geographic map that defines arctic, temperate and tropical climate regions. Obviously, there will be change over situations where the gear operation is midway between Regime I and II or midway between Regime II and III.

Oil Film Thickness

Figure 2 shows a typical curve sheet for a turbine oil used with high speed gears. This sheet was drawn for 2 to 1 ratio helical gears on a center distance of 260 mm. The sheet would not change much if the gears had been 5 times larger or if the load was made twice as great or half as great. (The assumed load is normal for good quality turbine gears designed for thousands of hours of life.)

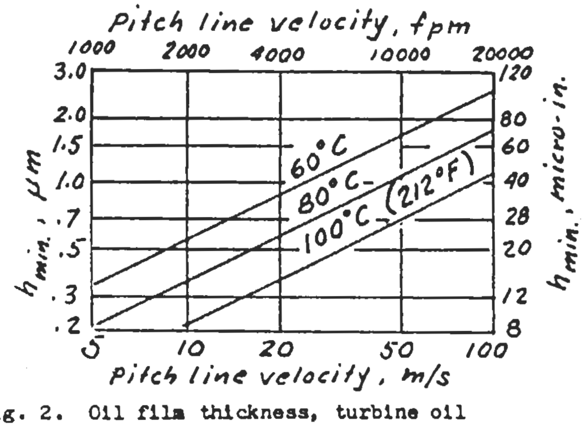

Fig. 2. Oil film thickness, turbine oil

The prime variables in gear oil film thickness are pitch line velocity, oil temperature and kind of oil.

Figure 3 shows a similar curve sheet for slower speed industrial or vehicle gears. An SAE 90 gear oil was assumed. A 2 to 1 ratio set of spur gears on a 260 mm center distance was used for the calculations.

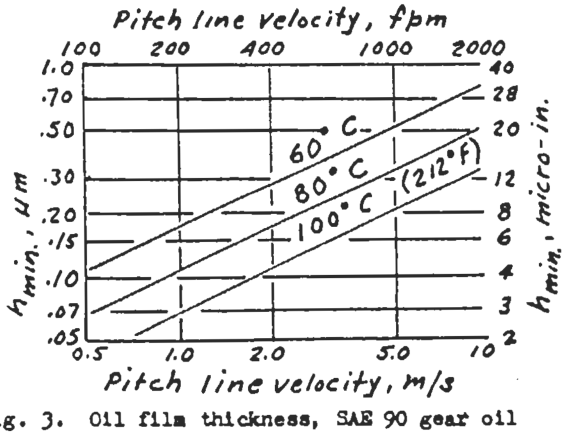

Fig. 3. Oil film thickness, SAE 90 gear oil

Pitting Tendency

Figure 4 shows the stress vs. cycles tendency for each of the three regimes of lubrication.

The general equation for life of gears (in contact cycles) in respect to the tangential tooth load is

$$\frac{L_b}{L_a} = \left(\frac{W_{ta}}{W_{tb}}\right)^x \qquad \text{Eq. 1}$$

where:

W_{ta} = tooth tangential load in Newtons for L_a cycles before pitting

W_{tb} = tooth tangential load in Newtons for L_b cycles before pitting

x = slope exponent for load vs. cycles curve

Gear design practice uses the compressive stress, s_c. Figure 4 is therefore plotted as s_c vs. cycles. The compressive stress is proportional to tooth load to the 0.5 power.

The K-factor index of tooth load intensity can be converted to compressive stress by using a tooth geometry factor, C_k to evaluate the tooth shape and a derating factor to evaluate misfit and dynamic load effects that make some parts of some teeth carry more load than they would have carried when the load was distributed evenly over all the tooth surfaces. The simplified equations[1] to cover this for external spur or helical gears are:

$$K = \frac{W_t}{d \times b}\left(\frac{u+1}{u}\right) \qquad \text{Eq. 2}$$

where:

d = pinion pitch diameter, mm

b = face width, mm

u = gear ratio

= no. gear teeth $\div$ no. pinion teeth

then,

$$s_c = C_k\left(K \cdot C_d\right)^{0.50} \qquad \text{Eq. 3}$$

where:

C_k = geometry factor

K = load intensity factor, (Eq. 2)

C_d = overall derating factor

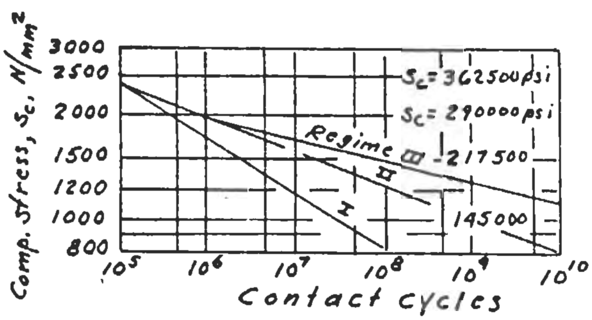

Fig. 4 Trend of contact stress for Regimes I, II and III. Carburized gears.

[1] References 6, 7, 8 each show calculation procedures for calculating compressive stresses in gear teeth.

LOW CYCLE BEHAVIOUR

From 1 cycle to 10^4 contact cycles pitting is not apt to occur. If the loads are too high there will be tooth surface damage, but the damage will usually be in the form of scoring (also called "scuffing"), cold flow, rippling or swadging.

If the load is heavy enough to cause pitting at 10^5 cycles the look of the teeth at 10^4 or even 3×10^4 cycles may be essentially a burnished appearance. Figure 5 shows a medium hard gear at 2×10^4 cycles of such high load that pitting was severe at 10^5 cycles. (This was known because several units were in service, loaded the same, and they all would pit badly when the cycles got high enough.) The Figure 5 gear is burnished and pitting had just started. Some teeth had no pits and some teeth had one or two large pits.

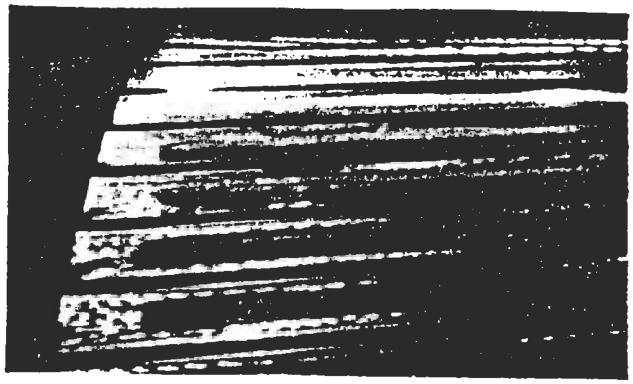

Fig. 5 A Regime I gear in service that will be pitted by 10^5 cycles

Up to 10^5 cycles there seems to be no particular difference between the three regimes so far as pitting goes. This suggests that either high speed or slow speed gears might be designed for the same high load at 10^4 or 10^5 cycles. From a practical standpoint, it is not reasonable to do this. A high speed gear that survives severely high loads for 10^5 cycles without pitting may have micro cracks or other forms of surface damage that will prove fatal at 10^9 or 10^{10} cycles even when the loading at 10^9 or 10^{10} cycles is low enough to be well within proven acceptable gear rating practice. (More on this point later.)

The scoring hazard is greatest when new gears first see maximum torque and temperature conditions. In Regime I the scoring hazard is primarily due to the lack of separation of the tooth surfaces. High viscosity oils are used, but this alone is often not enough. Special extreme pressure additives (normally called "EP" additives) are used. The EP additives tend to substitute a chemical film for an EHD film. The chemical film provides some type of low shear chemical reaction product that keeps replacing itself as the gear tooth contact action tends to rub it off.

In Regime III the low cycle scoring hazard seems to be caused by a breakdown of the EHD oil film due to excessive temperature in the film. The normal design procedure (9) calculates a flash temperature of the oil. Flash temperature design limits have been in common use now since the 1950's. The pioneering work of H. Blok (10) led to the development of the "flash temperature" concept of scoring.

Regime III gears have a kind of hot scoring due to the flash temperature behaviour. Regime I gears score, but their scoring is a sort of cold scoring due to lack of EHD oil film. Regime II gears may have troubles due to either kind of scoring. In the author's opinion, much more research is needed on tooth scoring -- and it is most important to correlate scoring test data with the regime of lubrication.

REGIME I

Final drive gears in mills, lifting devices or winches may have to run very slowly under very heavy load. Regime I gears are not too frequent in the gear trade, but they are often necessary.

This regime has a steep slope for the load vs. cycles curve of pitting capacity. When $x = 3.2$, the load capacity at a 10-fold increase in cycles is only 49%. The compressive stress capacity is 71% at a 10-fold increase in cycles. (Stress is proportional to load to the 0.50 power -- as was mentioned earlier.)

Figure 6 shows a plot of percentage load vs. cycles for the three regimes of lubrication. The basing point for 100% load was taken quite arbitrarily at 10^5 cycles. The reason for doing this is that all three regimes seem to have about the same capacity to resist pitting at this relatively low number of cycles.

As more data becomes available it may turn out that the three regimes are somewhat different even at 10^5 cycles -- much will probably depend on the steel hardness and the definition of whether or not a slight amount of pitting or something like 10% of the tooth surface pitted constitutes pitting failure.

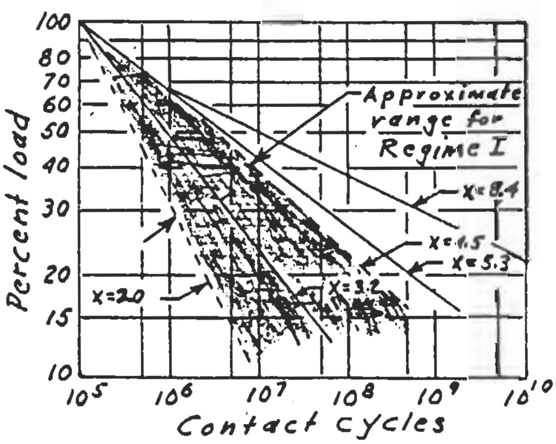

Fig. 6 Variation in gear load capacity that may be expected in Regime I. Note typical slope curves for Regimes II and III

When the lubricant is particularly thin, or the lubricant has little or no additives, Regime I may have a slope up to $x = 2.0$. Conversely, a reasonably heavy oil and strong EP additives may make Regime I do much better and get over as far as $x = 4.5$.

Figure 6 shows the general area of Regime I operations. The gear designer of course is faced with a critical problem in Regime I. Data available in 1981 is still quite inadequate. The best solution is to design conservatively and then test the gears and the lubricant. If premature pitting occurs, improve on things like the lubricant and its additive package, the gear surface finish, or lower the maximum operating temperature. (Also design changes like higher hardness, better profile and load modifications, or slightly larger gears may be needed to do the job. Figure 7 shows a medium hard Regime I gear.

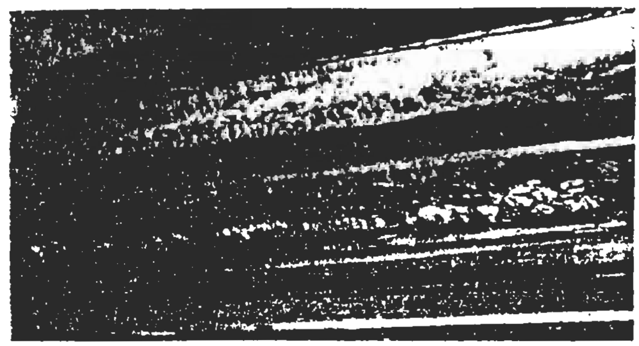

Fig. 7 Slow speed gear in Regime I with severe pitting. Note lack of polish in gear addendum

REGIME II

This regime is typical of vehicle gears. Relatively heavy oils are used. Strong EP additive packages are used -- particularly if the pitch line speed is as slow as 0.5 m/s. The gears are generally not made to a very fine finish, but they tend to wear in quickly to good finish. In appearance, they are often very bright and polished after some hours of full torque operation.

Regime II has a moderate slope. The load capacity for a 10-fold increase in cycles is 65% at a value of $x = 5.3$. The compressive stress capacity is 80% at a 10-fold increase in capacity.

Figure 7 is another plot of percentage load vs. cycles for the regimes of lubrication. The three basic curves are the same as Figure 6. Figure 7 shows the general area to be expected for Regime II. This area is bounded by slope curves at $x = 4.5$ and 6.0.

The variation in slope for Regime II is usually not too great. The AGMA vehicle standard (8) shows design curves for compressive stress vs. cycles that are in the middle of the Regime II region just described. In general, gears for Regime II wear into rather good finish and fit. Relatively heavy oils are used with appropriate additive packages.

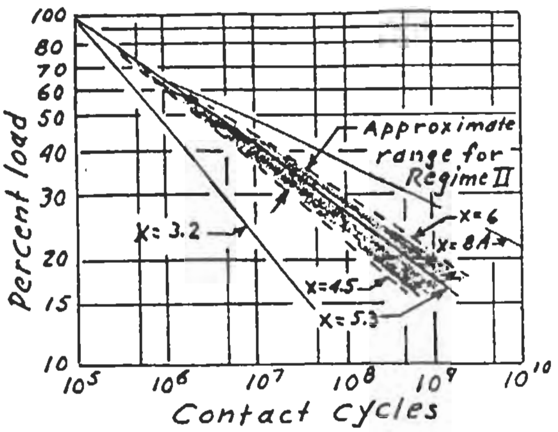

Fig. 8 Approximate variation in load capacity in Regime II. Typical slope curves shown for Regimes I and III.

The gear designer can work with much better confidence in Regime II than Regime I. Testing of new designs is still often necessary. Test results may require small changes in the lubricants, some adjustment in tooth modifications and/or some improvement in surface finish of the gear teeth. Figure 9 shows a Regime II gear.

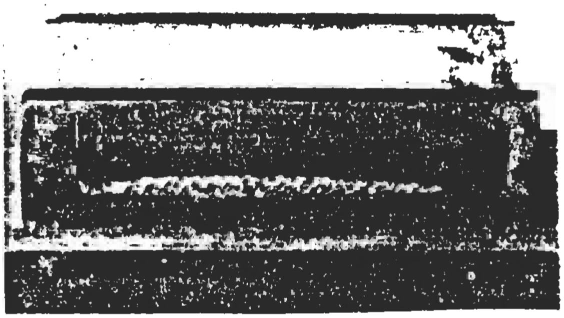

Fig. 9 Pitted Regime II gear. Note polish in contact area.

REGIME III

This regime is typical of high speed gears used with turbines. Usually the first reduction of an electric motor drive and some times the second reduction is fast enough to be in Regime III.

Turbines use fairly thin oils, but the pitch line speeds are so fast that it is usually possible to develop an EHD oil from 0.6 µm to as much as 2.0 µm. Turbine first stage gears often run from 50 to 150 m/s pitch line velocity. Even second stage turbine gearing will usually run from 20 to 50 m/s pitch line speed.

High speed turbine gears are generally finished by grinding or shaving to a finish of 0.5 µm to 0.8 µm. After initial break-in the finish usually improves so that final running finish is apt to be in the 0.4 to 0.6 µm range. It is not difficult in most high speed gears to develop the full EHD oil film needed for essentially complete separation (by oil) of the meshing gear tooth surfaces.

Regime III has a rather gentle slope. The load capacity for a 10-fold increase in cycles is 76% at a value of x = 8.4. The compressive stress is 87% at a 10-fold increase in capacity.

Figure 8 shows a plot of percentage load vs. cycles with the same three basic curves shown. The general area of design for Regime III is now shown.

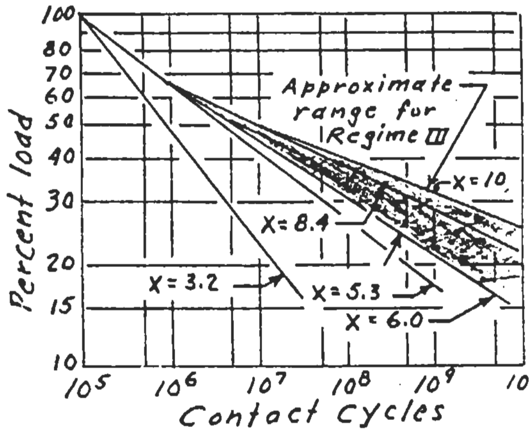

Fig. 10 Approximate variation in load capacity in Regime III. Slope curves, Regimes I and II shown.

The AGMA design standard for aircraft engine gears (7) follows a slope close to x = 8.4. Data available on successful turbine gearing in marine, land and turbo prop aircraft shows that high speed gears (usually in Regime III) fit into the general area shown on Figure 8 rather well.

The designer in Regime III does not have to worry much about the slope of the load vs. cycles curve. Things of greater concern are quality of material, accuracy and fit of the gears.

Figure 11 shows a Regime III pinion just starting to pit at 2 x 10^7 cycles.

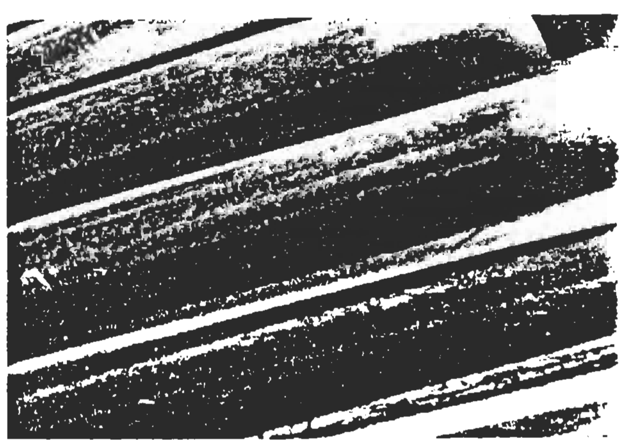

Fig. 11 Initiation of pitch line pitting on case hardened helical pinion.

In Regime III the oil film thickness is great enough to separate the contacting surfaces. Finish marks are not worn off. This is shown rather clearly in Fig. 12.

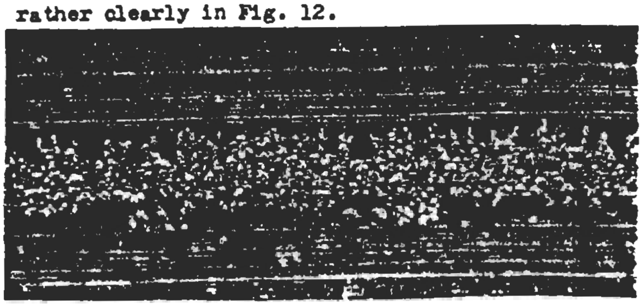

Fig. 12 Scanning electron microscope view of pitting seen in Fig. 11. Note finish marks.

CLOSING COMMENTS

The material just presented shows the general characteristics of the three regimes of lubrication. The tendency of gears to score (scuff) or to pit is strongly influenced by the regime of lubrication.

The concept of regimes of lubrication explains things gear designers noticed in earlier years, but could not explain. For instance, why did slow speed gears score when scoring was thought of as a speed related problem? In regard to pitting, some design data (11) showed a rather steep slope of stress vs. cycles, while other data showed a rather gentle slope. Variations in slope seem to be mostly due to the regime of lubrication.

The material presented in this paper comes in considerable part from field experience with gear units that have been in service for 5 to 10 years.

In the oil and gas industry, large numbers of turbine driven units have been in service for over 30,000 hours. At 10,000 RPM a gear part will have 1.8×10^{10} cycles. When a 10,000 RPM pinion meshes with three star gears in an epicyclic drive, there will be 5.4×10^{10} cycles in 30,000 hours.

Most laboratory bench tests of gears are only run for 10^7 to 10^8 cycles. Gear data for operation in the region of 10^7 cycles to 10^{10} cycles must come primarily from units in service.

Figure 13 shows an example of a high speed gear operating in Regime III that failed by pitting at 5×10^9 cycles.

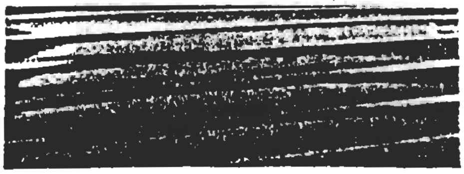

Fig. 13 Gear eroded by pitting, 5×10^9 cycles. Profile wear about .025 mm (.001").

It is not possible to list all the items of published and unpublished data that have formed a background for this paper. However, for those interested in further study, several good additional references are given besides the ones specifically mentioned in the text.

REFERENCES

(1) Dudley, Darle W., PRACTICAL GEAR DESIGN, McGraw-Hill, New York, 1954; Kōgakusha Company, Ltd., Tokyo.

(2) Dowson, D. and Higgenson, G.R., ELASTOHYDRODYNAMIC LUBRICATION, Pergamon Press Ltd., 1966.

(3) Dudley, Darle W., "Elastohydrodynamic Behaviour Observed in Gear Tooth Action," presented to the Institution of Mechanical Engineers, in Sept. 1965 at Leeds, England.

(4) Bowen, C.W., "The Practical Significance of Designing to Gear Pitting Fatigue Life Criteria," ASME Paper 77-DET-122 presented at the Sept. 26-30, 1977 meeting in Chicago, Illinois.

(5) Peterson, M.B. and Winer, W.O., WEAR CONTROL HANDBOOK, ASME, New York, 1980. (See chapter "Gear Wear" by Darle W. Dudley.)

(6) Dudley, Darle W., GEAR HANDBOOK, McGraw-Hill, New York, 1964.

(7) American Gear Manufacturers Assoc., "Design Procedure for Aircraft Engine and Power Take-off Spur and Helical Gears," AGMA 411.02, 1966.

(8) American Gear Manufacturers Assoc., "Design Guide for Vehicle Spur and Helical Gears," AGMA 170.01, 1976.

(9) American Gear Manufacturers Assoc., "Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears," AGMA 217.01, 1965.

(10) Blok, H., "Les Temperatures de Surface Dans Les Conditions de Graissage Sous Pression Extreme," 2nd World Petroleum Congress, Paris, 1937.

(11) Barish, Thomas, "How Sliding Affects Life of Rolling Surfaces," Machine Design, Oct. 13, 1960.

(12) Winter, H., and Weiss, T., "Some Factors Influencing the Pitting, Micro-Pitting (Frosted Areas) and Slow Speed Wear of Surface Hardened Gears," ASME Paper 80-C2/DET-89, presented at the Aug. 18-21, 1980 meeting in San Francisco, Calif.

(13) Young, I. T., "A Wider Scope for Nitrided Gears," ASME Paper 80-C2/DET-46, presented at the Aug. 18-21, 1980 meeting in San Francisco, Calif.

(14) Ueno, Taku, Ariura, Yasutsune, and Nakanishi, Tsutomu, "Surface Durability of Case Carburized Gears—On a Phenomenon of 'Grey-Staining' on Tooth Surfaces," ASME Paper 80-C2/DET-27, presented at the Aug. 18-21, 1980 meeting in San Francisco, California.

(15) Fujita, K., and Yoshida, A., "Effects of Case Depth and Relative Radius of Curvature on Surface Durability of Case-Hardened Chromium Molybdenum Steel Roller," ASME Paper 80-C2/DET-37, presented at the Aug. 18-21, 1980 meeting in San Francisco, California.

(16) Ishibashi, A., and Tanaka, S., "Effects of Hunting Gear Ratio Upon Surface Durability of Gear Teeth," ASME Paper 80-C2/DET-35, presented at the Aug. 18-21, 1980 meeting in San Francisco, California.

(17) Ichimaru, K., Nakajima, A., and Hirano, F., "Effect of Asperity Interaction on Pitting in Rollers and Gears," ASME Paper 80-C2/DET-36, presented at the Aug. 18-21, 1980 meeting in San Francisco, California.

(18) Wellauer, E.J., "Application of EHD Oil Film Theory to Industrial Gear Drives," ASLE/ASME Paper 75-PTG-1, presented at the Oct. 20-23, 1975 meeting in Miami Beach, Florida.