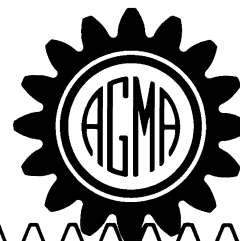


An Analytical Approach to the Prediction of Micropitting on Case Carburised Gears

by: D. Barnett, Renold Gears, J.P. Elderkin, MTM Precision
and W. Bennett, MoD (Navy)

06FTM06

American Gear Manufacturers Association



TECHNICAL PAPER

An Analytical Approach to the Prediction of Micropitting on Case Carburised Gears

**Dave Barnett, Renold Gears, John P. Elderkin, MTM Precision
and William Bennett, MoD (Navy)**

[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

Micropitting is one area of gear failure that has become more predominant over recent years, mainly because of its effect on gear noise and transmission error. This paper will outline an approach to analysing micropitting by looking at the critical factors for a given gear design. A practical calculation procedure, which incorporates a three-dimensional spring model, was used to predict the micropitting wear rate and the position that wear would take place on test gear pairs. Case studies have been included that directly compare the predicted levels of micropitting with those actually measured. A simplified formulation suitable for manual calculations will also be discussed.

Copyright © 2006

American Gear Manufacturers Association
500 Montgomery Street, Suite 350
Alexandria, Virginia, 22314

October, 2006

ISBN: 1-55589-888-2

An Analytical Approach to the Prediction of Micropitting on Case Carburised Gears

Dave Barnett, Renold Gears, John P. Elderkin, MTM Precision
and Lt. William Bennett, MoD (Navy)

Introduction

Micropitting is one area of gear failure that has become more predominant over recent years, mainly because of its effect on generated noise and transmission error. It is also possible that micropitting could be considered as the precursor to other failure mechanisms. At the present time, practical testing is being undertaken by many organisations across the world including: – the British Gear Association (BGA), the American Gear Manufacturers Association (AGMA), FZG and others. This work has primarily been targeted at the determination of the mechanics of micropitting and how to reduce or prevent it.

Previously, the use of sophisticated lubricants, with their appropriate additive packages, has been the main method of attempting to control the generation and propagation of micropitting. However, the advent of superior manufacturing methods and improvements in the quality and cleanliness of steels, has allowed the design engineer to significantly increase the gear tooth loading. This factor has indirectly led to the fact that micropitting is now currently seen as a major problem in commercial gears. The primary author has been involved with gear testing over a number of years and this has enabled him to compile a mathematical model that predicts the likelihood of micropitting occurring. The model is flexible enough to allow calculation of the various parameters, under specified load conditions, for a given design with or without involute and lead modifications. At present, the procedures include calculations for lubricant viscosity and temperature, but not those for oil chemistry. Once the actual effects of the additive packages are known these can be incorporated into the model.

Calculations

The practical calculation procedure utilises a three-dimension model that is based on spring theory. It was originally developed for the specific purpose of analysing bending strength and noise in high con-

tact-ratio spur gears. It was later modified to include helical gears. Specifics of the initial work were first published in 1986 and updated in 1994. The model itself functions by mathematically dividing the surface of the tooth into nodes, based upon the base pitch of the gear pair to be considered. Contact is assumed to take place along the line of action (LOA), although in practice this is incorrect. However, measurement of the true stresses on actual gears with tooth modifications has shown that the errors are insignificant. The analysis also checks the approach and recession of the gear mesh for any premature engagement or delayed disengagement. These factors have been shown to change the load distribution between meshing teeth due to deflection under load. The model calculates the normal tooth force exerted on each tooth as it moves through mesh. This is influenced by the gear design, involute and lead modification, tooth stiffness and applied load.

The calculated example (Figure 1) illustrates the magnitude of load sharing on a standard FZG 'C' type gear (which has no involute or lead modifications) as it rolls through mesh. The resultant loads shown to the left and right of the "length of theoretical engagement" are due to premature engagement and delayed disengagement. These factors effectively increase the profile contact ratio.

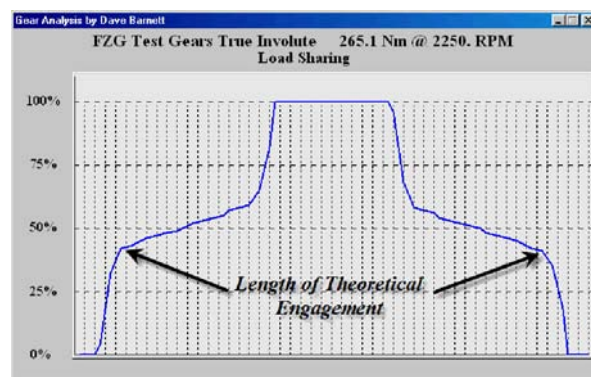


Figure 1. Load Sharing - Standard FZG Test Gear

Using the stiffness, the selected increments along the LOA and any tooth modifications, the load intensity per mm of face width is calculated over the total tooth surface for each node point.

Figure 2 depicts the load intensity over the entire contact area. The analysis shows that premature engagement and delayed disengagement exist on this gear (as shown previously). The black zone depicts zero load intensity. The load intensity gradually increases, as indicated by the various shades of grey until it reaches a maximum (depicted by the white band). The yellow lines mark the limits of theoretical contact and the grey areas above and below these lines show the zones of potential conflict.



Figure 2. Load Intensity - Standard FZG Test Gear

The calculation procedure considers each individual node point to be equivalent to a 'roller bearing' with its own individual radius of curvature, sliding velocity, entrainment velocity and slide roll ratio. Hence, knowing the applied torque, it is possible to calculate the contact stress and the oil film thickness that is generated. The empirically derived analysis has been corroborated against actual gear tests and it can be concluded that micropitting is primarily dependent on the following factors:

1. Contact stress
2. Sliding velocity
3. Oil film thickness
4. Oil temperature
5. Slide roll ratio (SRR)
6. Surface finish (the RMS parameter 'Ra' is used in the calculations at present, but this may require modification)

7. Surface hardness
8. Number of contact cycles
9. Direction of sliding and the time that each surface is in contact with the corresponding gear in the pair.
10. Oil additive package (at present this is not incorporated in the calculations as more work is required to define these parameters)

Description of the Test

Testing was commissioned by the MoD (Navy) and conducted at QinetiQ, as part of the BGA's Project 6 program of work into the "understanding of micropitting". This segment of the work consisted of trials on standard FZG 'C' type spur gears, as well as modified 'C' type spur gears. The modified gear sets were calculated to precisely defined project specifications and were manufactured by FZG. Two oils were considered for the work program, one with a micropitting additive and the second without. Similar trials were also conducted (in parallel) using a PCS (3-roller type) disc-testing machine.

Analysis of the Standard FZG Gears

As stated previously, the analysis does not consider the effects of oil chemistry at present. Hence the following examples have been derived from the gear test results utilising a lubricant without micropitting additives. The trials were conducted with an oil sump temperature of 50°C.

Figure 3 graphically represents the predicted erosion as the gear progressed through the load stages (LS) 5 to 10 and onto the endurance levels. Up to LS 10, erosion of the tooth surface due to micropitting can be seen in the dedendum portion of the tooth and is concentrated at the start of active profile (SAP). During the endurance phases of the test, the maximum erosion changed to a point close to the lowest point of single tooth contact. These observations suggest that this is due to premature engagement and can be directly attributed to the lack of tip and root relief on the standard FZG 'C' type gears. The tip of the mating gear would appear to contact the flank of the test gear above the SAP. This premature engagement resulted in high contact stress, even for small loads, which could be as much as four (4) times higher than that normally anticipated. As the micropitting erosion progressed, the erosion

reduced the level of premature engagement. This, in turn, reduced the contact stress and could slow down the rate of micropitting erosion, or transfer it to a different point on the tooth.

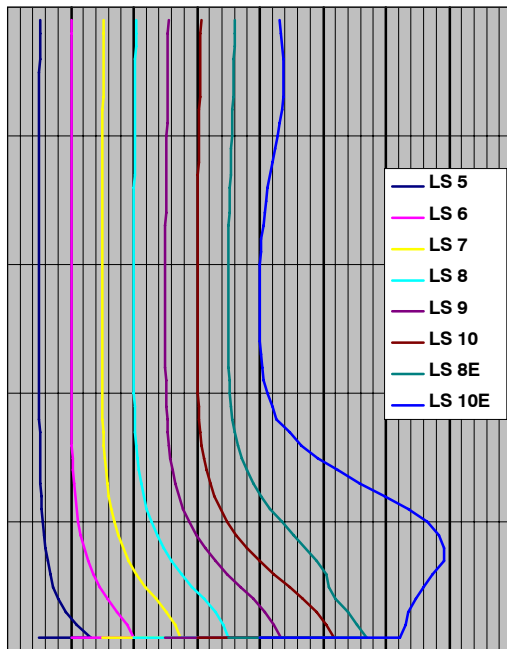


Figure 3. LS 5 to LS 10 and Endurance Standard FZG Test Gear

Analysis of the Modified FZG 'C' type gears

The modified 'C' type gears were re-designed standard FZG gears with a 0.038 mm tip relief, starting just above the highest point of single tooth contact. The amount of tip relief was chosen such that under a torque level of LS10, premature engagement was just avoided.

As can be seen in Figure 4, the prediction indicated that, with tip relief, the maximum erosion due to micropitting would occur just below the lowest point of single tooth contact. In the case of the standard gear, this would have been at the SAP. Similarly, on the modified gear, the actual predicted depth of the erosion due to micropitting was also reduced when compared to the standard gear. The measured results, for both depth and position, on the actual test gears gave a good correlation to the predicted values.

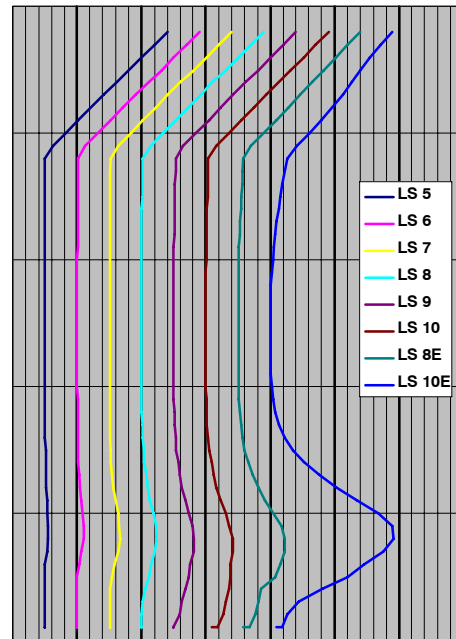


Figure 4. LS 5 to LS 10 and Endurance Modified FZG Test Gear

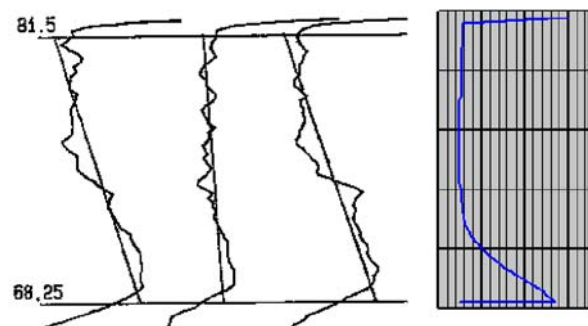


Figure 5. Standard FZG at end of Load Stage 10 – Actual / Predicted.

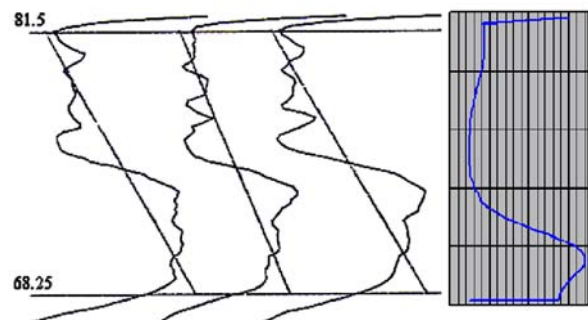


Figure 6. Standard FZG at end of the Endurance Stages – Actual / Predicted.

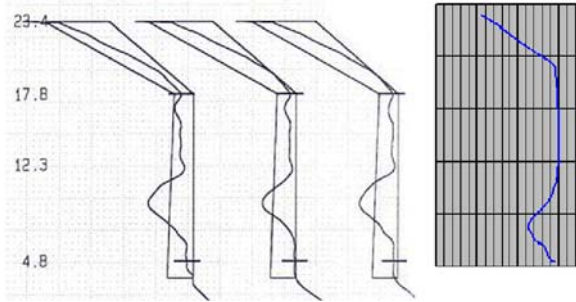


Figure 7. Modified FZG at end of Load Stage 10 – Actual / Predicted.

Comparing the results of the Standard and Modified FZG gears: It can be noted that, during LS 5 to 10, the level of micropitting erosion developed a magnitude of root relief of a similar size to the tip relief on the modified gears. Once the level reached was approximately 0.04mm, the position of the micropitting erosion moved to approximately the same point on the tooth as seen on the modified gears. The results of these tests indicated that if the correct level of involute modification had been applied at the gear design stage, it could have reduced and in some cases eliminated micropitting altogether.

Analysis has taken place on the gears produced for AGMA, which were tested for pitting but also exhibited micropitting. These gears not only have profile modification, but also lead crowning. The results are contained in Appendix 1.

Similarly, an analysis has been performed on a helical gear set. Again, an example is shown in Appendix 1.

Derivation of the Calculation Procedures

The following micropitting equation was empirically derived based on the observations from actual gear and roller testing. It was found that the level of micropitting erosion was proportional to the localised contact stress, number of cycles and sliding velocity (at the point of consideration). It was also found to be inversely proportional to the surface hardness and lambda ratio (oil film thickness/surface roughness). The actual tests on gears and rollers indicated that the slide roll-ratio had a significant effect and this was also included in the calculation procedure. Significantly, it was found that it was difficult to obtain micropitting with a slide roll-ratio of less than approximately 40%. It was further noted that the erosion due to micropitting was more dominant in

the dedendum portion of the tooth of both the pinion and gear. It was concluded that this was possibly due to the motion of the interacting tip of the pinion as it moved across the flank of its mate. As a result, a retardation factor was also included such that:

$$\text{Retard} = \frac{\rho_1 \times Z_1}{\rho_2 \times Z_2} \quad (1)$$

Where:

- ρ_1 = Radius of curvature of flank 1 (gear 1)
- Z_1 = Number of teeth in gear 1
- ρ_2 = Radius of curvature of flank 2 (gear 2)
- Z_2 = Number of teeth in gear 2

A factor for the *rate* of micropitting was also included to allow for the possible use with gear sets made from differing materials.

The results of the early tests appeared to indicate that there was a contact stress threshold below which micropitting would not occur.

It is also believed that micropitting will not normally occur once the lambda ratio (λ) exceeds a factor of 3. A switch has been included to prevent calculation once these conditions are met.

The resulting equation for micropitting erosion is as follows:

$$M_p := X \cdot MP_{\text{rate}} \frac{C_y}{V_{pn}} \sigma_{H1} \cdot U_s \left(\frac{Ra}{h_{i?}} \right) \left(\frac{U_s}{U_{e1}} \right)^2 \text{Retard} \quad (2)$$

Where:

- MP_{rate} = Rate of micropitting erosion per cycle
- C_y = No of cycles
- V_{pn} = Vickers hardness
- σ_{H1} = Local contact stress at point 1 (start of engagement)
- σ_{Hp1} = Local contact stress at point 1 (for pre-engagement)
- U_{s1} = Sliding velocity at point 1 (m/sec)
- Ra = Mean surface roughness of the mating surfaces
- $h_{i?}$ = Oil film thickness
- U_{e1} = Entrainment velocity
- Retard = The ratio of sliding of one surface over the other

As indicated above, X is a switch (at present) and is dependent on the λ ratio. For the purposes of this paper, X can be either 1 or 0 and is currently set to 0 if the formulation $\lambda > 3$ is satisfied. It is envisaged that, in the future, this could be utilised as a