

Comparing Surface Failure Modes in Bearings and Gears: Appearances versus Mechanisms

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American Gear Manufacturers Association



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ABSTRACT:

For gear systems and rolling element bearings, there are similar designations for the failures that are identified within their respective contact surfaces. By concentrating on the contact tribo-system, it is possible to go beyond the appearance and determine the mechanisms that promote or prevent contact fatigue damage.

It is also important to identify the lubricant regime within the contact. This can be done with the modified lambda ratio. This is a ratio of the central film thickness occurring within a contact divided by the composite roughness of the two surfaces forming the contact, the roughness being modified to match the contact width occurring in the direction of motion

The contact fatigue modes are identified over a range of modified lambda values. An attempt is made to describe the failure modes and interpret the wear, fracture and/or fatigue mechanisms that lead to the failure initiation. Considering the similarity in appearance of the gear and bearing failure modes and yet the considerably different relative surface motions and tractions of the two contacts, this comparison helps provide insight into the basic causes of the failures and perhaps suggests methods to avoid them.

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INTRODUCTION

In the 1960's and early 1970's, considerable work was done to identify the various modes of damage that ended the lives of rolling element bearings (1)(2)(3)(4)(5). A simple summary of all the damage modes that could lead to failure is given in Table 1. In bearing applications that have insufficient or improper lubricant, or have contaminants (water, solid particles) or poor sealing, failures such as excessive wear, corrosion or excessive vibration can occur rather than contact fatigue. Usually other components in the overall system besides bearings also suffer. Over the years, builders of transmissions, axles and gear boxes that comprise such systems have understood the need to improve the operating environment within such units so that some system life improvements have taken place.

Those of us who manufacture bearings realized that identifying the damage modes was not enough but that an understanding of the causes and underlying mechanisms at play was important to improve the ability to predict bearing life in these variety of systems. However, whenever adverse operating conditions prevailed and actual life was below expected life, it was recognized that the understanding of modes of failure within bearings could be used to determine what improvements were needed to allow bearings to have extended life even under adverse conditions.

Considerable work has been done to accomplish this, and more remains to be done, but by being able to understand expected failure modes based on known operating conditions and applying this to the design cycle, significant improvements in bearing performance have been achieved. For line contact bearings, the prime factors are (1) significantly cleaner steel, (2) much improved surface finishes, which includes roughness, waviness and other factors beyond R_a (the arithmetic center-line average of roughness), and (3) geometry that optimizes internal stress profiles so that even under high load or nonaligned conditions much more uniform stresses can be maintained along the contact line.

It would seem the same development path has been occurring for gear contacts. Thus, it may be we can review the failure modes that occur in gears and, based on the understanding of what mechanisms underlie the various failure modes that have been identified, see what design changes may be appropriate that would also extend gear performance life beyond present limits.

One way to do this is to compare the modes of damage identified in line contact bearings and gears to determine what is happening within their respective contacts that cause the final failures. Since both bearings and gears function primarily with fluid lubrication, a tribological model of line contact, as shown in Fig. 1, can be the starting point to discuss the

TABLE 1
DAMAGE CLASSIFICATION FOR BEARINGS

I. FATIGUE

Contact Fatigue - Subsurface Origin

- Inclusion - within macroshear classical shear stress zone below contact surface
- Inclusion - near surface zone of macroshear greatly influenced by surface roughness (asperities)
- Subcase Fatigue - origin near case-core interface if yield strength is exceeded by applied stress

Contact Fatigue - Surface Origin

- Origin at the end of contact aggravated by edge geometry
- Multiple origins of micropitting (peeling or frosting)
- Point Surface Origin - at localized stress risers (dents, grooves, surface inclusions)

II. PLASTIC FLOW

- Brinelling or debris denting
- Load excursions above the material yield point
- Yielding aggravated by high temperature excursions

III. WEAR

- A. Adhesive
 - Normal - mild or "controlled" - usually identified as "run-in"
 - Severe - irreversible scuffing, scoring, smearing or seizure
- B. Abrasive
 - Normal - usually 3 body system, medium to fine particles that are also associated with "run-in"
 - Severe - grooving, gouging, denting with ridges that cause serious surface stress risers
- C. Corrosive
 - Water or acidic constituents from lubricant breakdown or temperature chemically aggressive additives in lubricant
- D. Fretting
 - Microscale adhesive and abrasive wear
 - Corrosion involvement depending on environment and contacts

mechanisms that contribute to the various failure modes reported.

THE BASIC LINE CONTACT MODEL

Whether gears or roller bearings, the contacting surfaces can be represented as two cylinders. It is possible to include transverse profiles or radii on these cylinders and determine the appropriate elliptical or truncated elliptical contact formed; but for this discussion two simple cylinders of the same or different radii are sufficient. Under running conditions, each cylinder has surface velocities either close to the same or considerably different that results in tangential or frictional forces within the contact that can range from less than one percent to well over twenty percent of the normal force that forms a rectangular contact area between the two cylinders.

The two cylinders under load determine the Hertzian contact area on the X-Y plane shown on Fig. 1. The cylinder length along the Y axis and $2b$, the width of contact along the X axis, form the Hertzian contact shown by the straight dashed lines. The Hertzian contact pressure reaches a maximum P as indicated by the dotted lines above the contact rectangle. P directly relates to the sub-surface principal and shear stresses to be discussed later.

If there is sufficient lubricant in the entering meniscus to the contact, an elastohydrodynamic lubricant (EHL) film(h) will be formed. Recognizing that the cylindrical surfaces have some amount of roughness, the lubricant film(h) and roughness (σ) can be compared to give an indication of the quality of the lubricant operating regime within the contact. That is, the EHL film(h) divided by the composite roughness of the two surfaces (σ) provide h/σ or λ (λ). Documents