Gearbox Bearing Service Life: A Matter of Mastering Many Design Parameters

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Management Summary

The availability of high-strength shaft materials, in combination with bearings with high carrying capacity, allows use of slimmer shafts. However, the modulus of elasticity remains the same, so seat design for bearings and gears must be given close attention.

Introduction

The service life of a gearbox is determined by many factors. Bearings, for example, play a major role since they contribute an important function while also interacting with the shafts, casing and oil. Without a doubt, the sizing of the bearings is of great importance in gearbox reliability. For over 50 years, bearing dynamic carrying capacity has been used to determine a suitable size needed to deliver a sufficient fatigue life. But despite the existence today of advanced calculation methods, they do not fully predict service life. Producers of high-quality bearings have introduced better ways to express (quantify) improved performance, but only in terms of increased dynamic carrying capacity.

This article will cover the following:

- Sizing of bearings based on dynamic carrying capacity and how this relates to service life
- How the design of the interface between bearing and shafts should be adapted to modern shaft materials
- How the design of the interface between bearing and gearbox casing influences service life of the gearbox
- Influence of modern electric motor speed controls on bearing-type selection

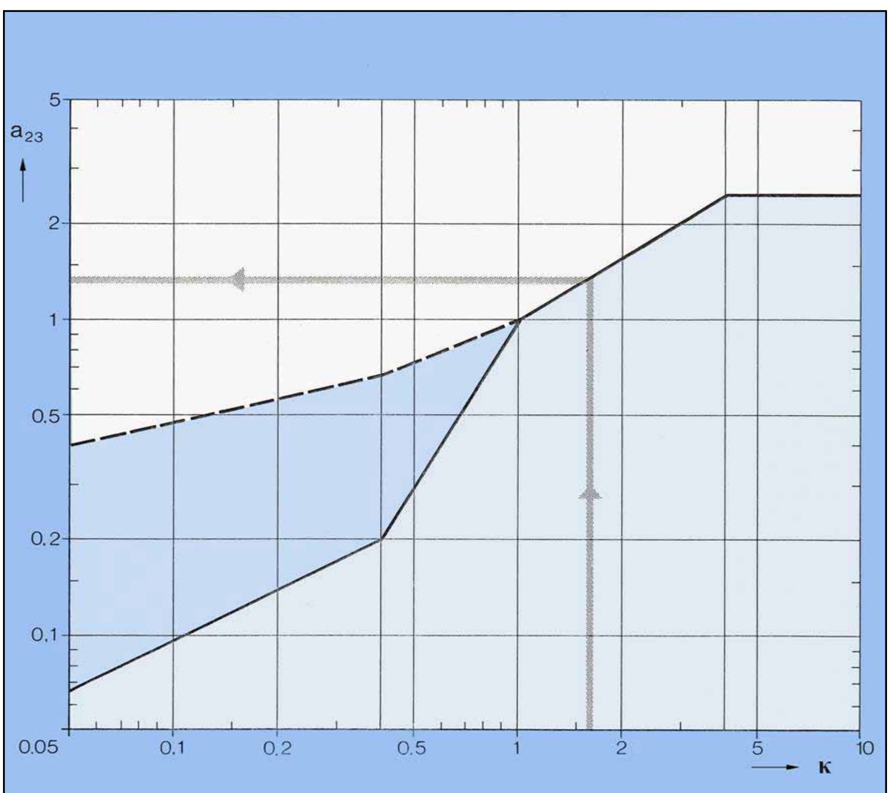

Figure 1—Life adjustment factor a_{23} for oil film influence.

Sizing of rolling element bearings based on dynamic carrying capacity. For modern high-quality bearings, the classic basic rating life can deviate significantly from the actual service life in a given application. Generally speaking, service life in a particular application depends not only on load in relation to bearing size, but also on a variety of influencing factors, including lubrication, the degree of contamination, misalignment, proper installation and environmental conditions.

The first method accepted by ISO for determining a suitable bearing size is the classic Lundberg and Palmgren equation $L_{10} = (C/P)^p$, making it possible to determine a suitable dynamic C-value (which in turn defines the bearing size) needed to satisfy a need for a fatigue life L_{10}.

The influence of material properties and lubricant film thickness was introduced in the 1960s, represented by the a_{23} factor (Fig. 1).

(1)

$$L_{10} = a_{23}\,(C/P)^{P}\ [a_{23} \text{ is a function of } \kappa \text{ and the material}]$$

As an attempt to take some of those factors into account when determining a suitable bearing size, the DIN ISO281:1990/AMD2:2000 contains a modification factor a_{SLF} to the basic rating life $L_{10} = (C/P)^{P}$. The method makes provision for bearing manufacturers to recommend a calculation methodology for this life modification factor to be applied to a bearing based on operating conditions. Some life modification factors apply the concept of a fatigue load limit P_u analogous to that used when dimensioning other machine components. Furthermore, the life modification factor makes use of the lubrication conditions and a factor η_c for contamination level to reflect the application's operating conditions.

Basic Conditions

The life modification factor considers bearing load level, oil film thickness and the stress-inducing influence of indentations in raceways and rollers from oil contaminants.

The influence of the oil film thickness is strong, and is represented by the κ–value. κ is the ratio of the actual operating viscosity to the rated viscosity for adequate lubrication (Ref. 1).

The influence of indentations from oil contaminants is very important, and complex to model. The η_c–value (contamination factor) acts as an inverted stress concentration factor defined as a value between 0 and 1, where 0 represents "severe contamination" and 1 represents "extreme cleanliness." The DIN ISO 281 Addendum 4:2003 describes a method to obtain an η_c–factor for a given application. SKF has developed a standardized method to estimate the η_c–value using load level; oil film thickness; size and type of contamination particles; mounting practice; filter effectiveness; and seal effectiveness into account (Refs. 2–3).

(2)

$$L_{10m} = a_{SKF}\,(C/P)^{P}\ [a_{SKF} \sim \kappa, \eta_c, P_u, P]$$

SKF and other bearing producers over the last decade have introduced high-performance-class bearings that are given higher dynamic carrying capacity figures and, in SKF's case, another scale for the life modification factor a_{SKF} to adapt L_{10m} calculations to the new technological developments in material and manufacturing (Fig. 2).

Therefore, the development from the purely sub-surface fatigue theory—"Acta Polytechnica" from Lundberg & Palmgren (Ref. 4) equation $L_{10} = (C/P)^{P}$—to a more realistic bearing application model, has introduced some adjustment factors to account for other, influencing phenomena in the sizing of rolling bearings. Observe that by choosing how to define the different factors in relation to the operating conditions—such as lubrication quality and oil cleanliness—it is possible to consider more effects than load, exclusively; this makes it trickier to evaluate bearings from different manufacturers based only on published dynamic carrying capacity figures.

Limitations of the classical and modified sizing models. The different issues of the L_{10} models for determining bearing

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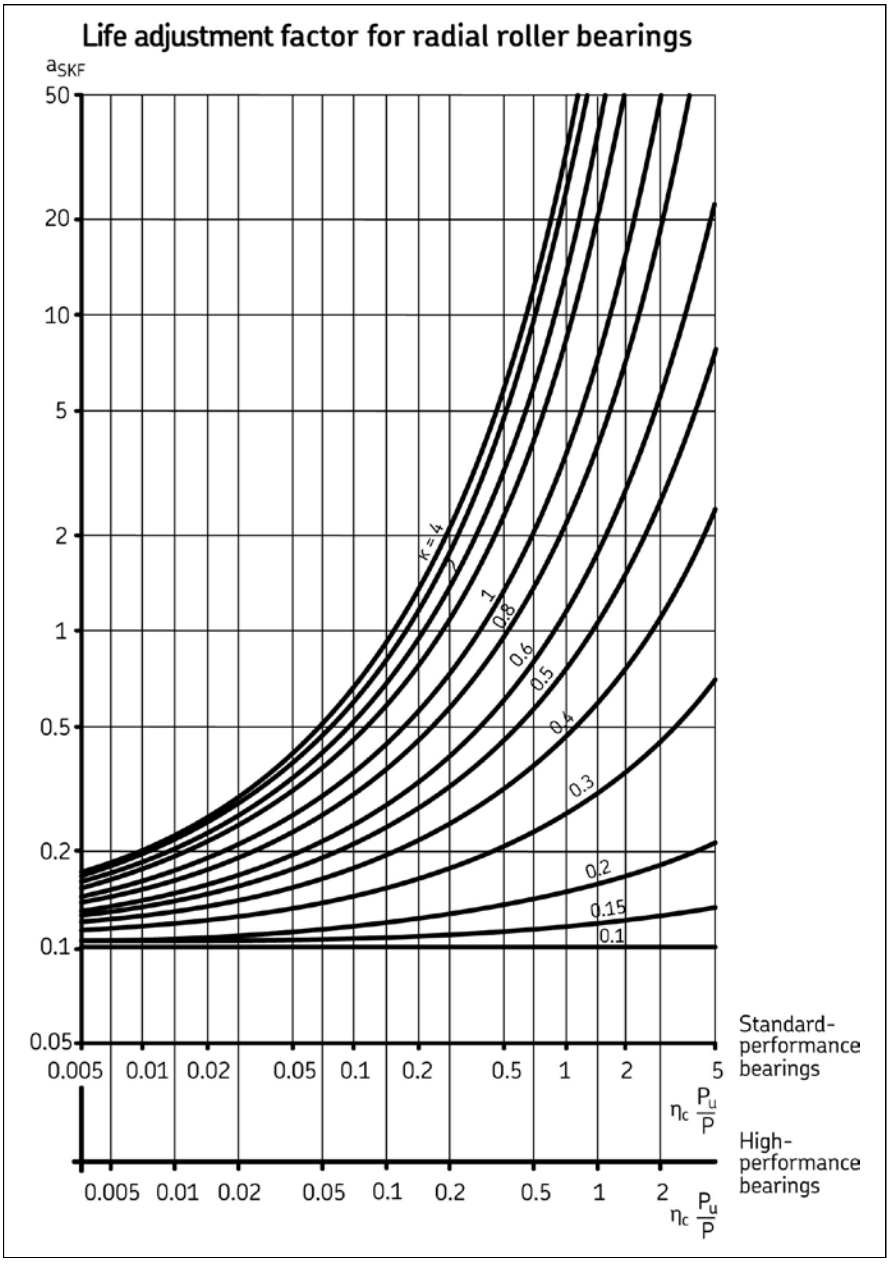

Figure 2—Life adjustment factor a_{SKF} for oil film quality, load level and indentation influence; separate "high-performance scale."

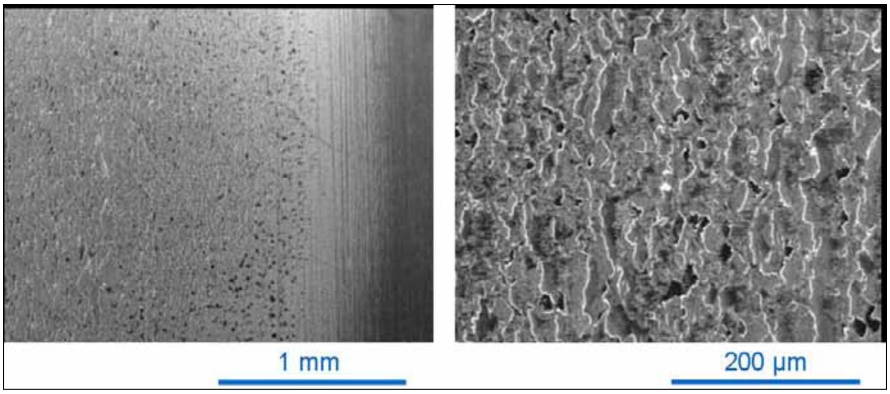

Figure 3—Surface distress.

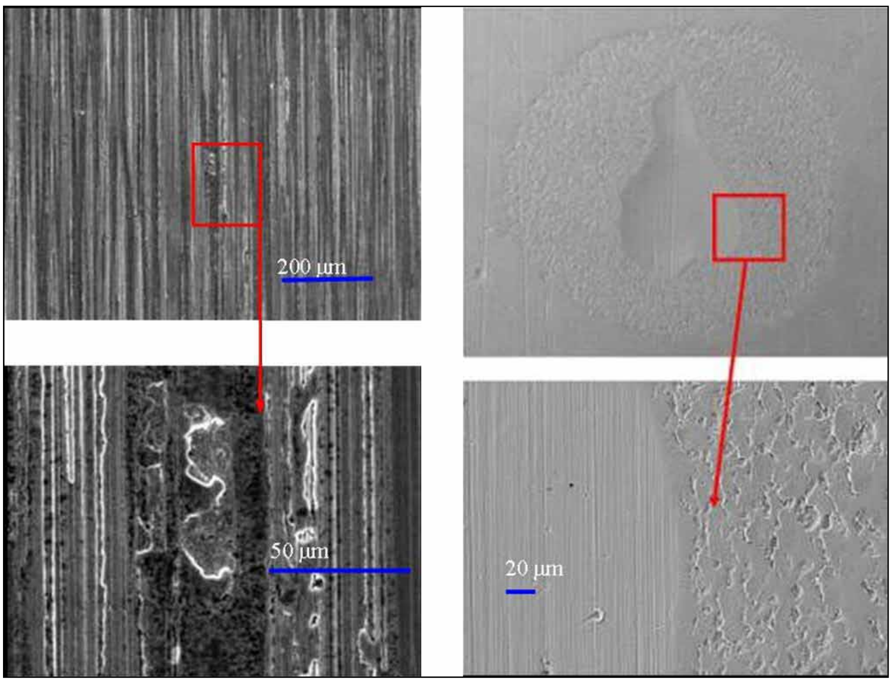

Figure 4—Surface distress on asperity summits (left) and in the area around an indent (right).

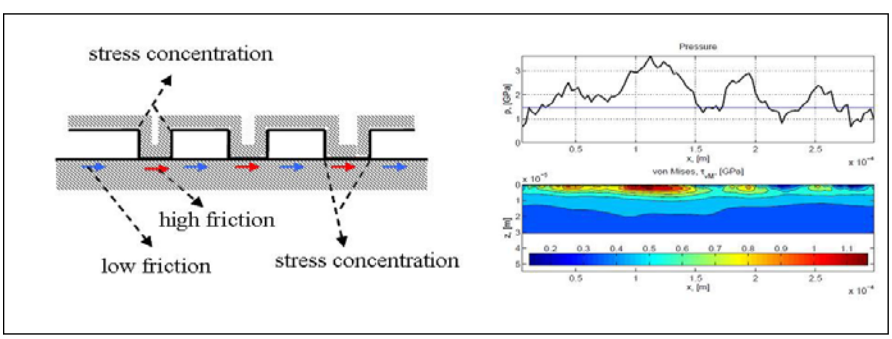

Figure 5—Schematic representation of the traction discontinuities and stress concentration areas in boundary or mixed-lubrication regime of rough surfaces.

size are well-grounded in mathematical models and practical testing, but limited in the use of only one failure mechanism—i.e., fatigue of the raceways or the rolling elements to deliver a suitable service life when determining a suitable bearing size. In the a_{SKF} diagram, very thin oil film (e.g., κ equal to 0.1) is associated with a life modification factor of 0.1—independent of load level—which is to be understood as predominant, surface-induced fatigue. In the bearing (and in the adjustment factor model), surface-induced fatigue gains importance at low values of κ and η_c.

Wear from abrasive particles in the oil film is a complex phenomenon. It will, for instance, surely change the contact geometry—which in turn will change the wear progress—so its progress is difficult to predict. Work on predictive models is in progress but will not be discussed here in detail. Recent findings in the field of surface distress in the thin-film regime—applicable to bearings in the lower-speed part of gearboxes and on gears—are, however, bringing new insights that are finding practical applications in bearing design and manufacturing (Ref. 1).

Surface Distress

In many industrial applications having lubricated rolling/sliding contacts (rolling bearings, gears, cam-followers) the power density has increased accordingly, due to the need for higher efficiency, reduction of weight and costs (i.e., downsizing). However, with the increasing severity of the working condition—that is, by heavier loads in combination with higher temperatures, thinner oil films and/or boundary lubrication conditions—machine components can sometimes suffer from surface distress (Ref. 5). This phenomenon manifests itself initially with a change of coloration/dull appearance of the surface, and grows as the damage progresses. Under the microscope, the affected surface areas show the presence of tiny microspalls, microcracks or micropits (Fig. 3).

Today it is recognized that surface distress is a surface damage phenomenon associated with poor lubrication conditions, thus high, local friction and pressures at asperity level. This phenomenon has been the subject of many recent experimental and numerical studies (Refs. 7–12).

Surface Friction

Real contacts, even when running under "nominal pure rolling" conditions, always have a small amount of slip; this results in some sliding friction and, consequently, in the possibility of surface distress risk. Testing has shown that nominal, pure rolling conditions can also exhibit surface distress. Under equal conditions and number of cycles, it has also been found that as the boundary friction coefficient is increased, surface distress is more severe. One easily concludes that boundary friction is a very important factor in promoting surface microcracks when the contact operates under boundary or mixed lubrication.

Importance of Lubrication and Roughness

Lubrication plays a major role in the life performance of rolling bearings, which is why life models account for the effect of the lubrication parameter κ (Ref. 1). The importance of lubrication and roughness in surface damage is very much related to the effect of local friction forces and stress concentrations (at asperity level). In boundary or mixed-lubrication, having irregularities (roughness or indentations, Fig. 4) on the surface will influence the way the dry and lubricated spots are distributed within the contact. Furthermore, discontinuities on surface traction and possible stress concentrations (Fig. 5) must also be considered. High roughness or high roughness slopes might promote local film collapse, high contact pressures and tractions. This will enhance stress concentrations in the critical areas of traction discontinuities.

Indeed, from the test results and theoretical modeling, surface distress appears first in areas of pressure discontinuities (high-pressure gradients) associated with increased roughness. It becomes apparent on the borders of grooves or in the summits of asperities or surface rises from indentations (Fig. 4). Rather unexpectedly, it is, in general, the smoother of the two mating surfaces that will first start the distress process. Some observations that may help to explain this discussed in the next section.

The Contact of Two Rough Surfaces

In industrial applications the contact always takes place between two real surfaces having a certain roughness. This is also the case in tests carried out using a surface distress test rig (SDTR), which has a rotating rod in contact with three discs (hardened bearing steel). As observed, when the test rod was rougher than the load-applying discs, surface distress did not appear for a reasonable time—even under the harshest conditions (Fig. 6a). However, when the discs were rougher than the rod (Fig. 6b) the surface distress easily appeared on the rod surface. This is also a common observation elsewhere (Ref. 11). As discussed (Ref. 13), the most likely explanation for this is the load history from the fatigue microcycles imposed by the roughness.

When the conditions in the contact are, in general, more towards boundary or mixed-lubrication, the stress history is then imposed by the dominant, rougher surface upon the smoother one, as long as there is some sliding. In real contacts, both surfaces will be rough and in movement (with some sliding); but if they have different roughness, the rougher surface will prevail over the smoother when imposing load microcycles. Therefore, the smoother surface is more susceptible to surface distress in the presence of some sliding, provided that the mechanical properties of both surfaces are the same.

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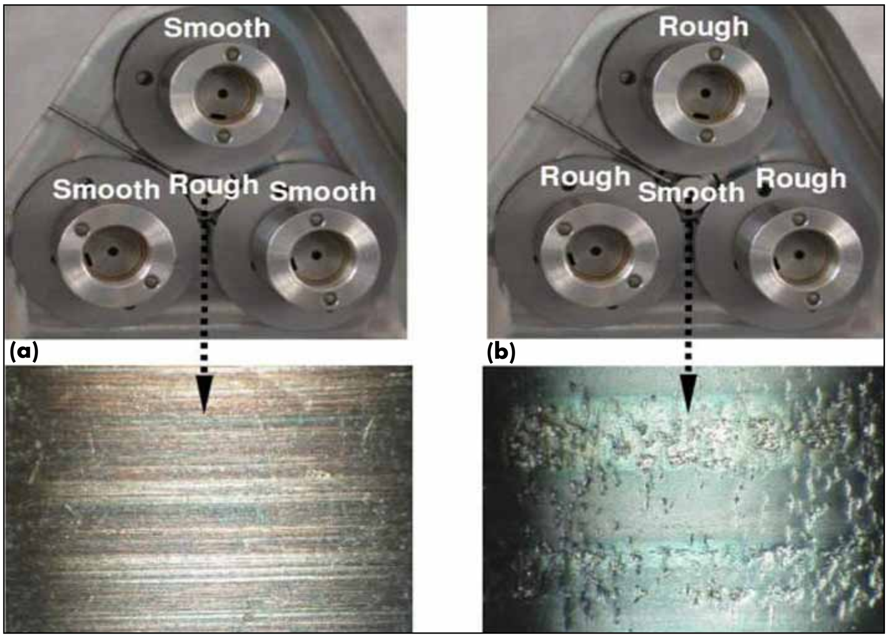

Figure 6—Effect of roughness location: (a) smooth discs on rough rod, and (b) rough discs on smooth rod.

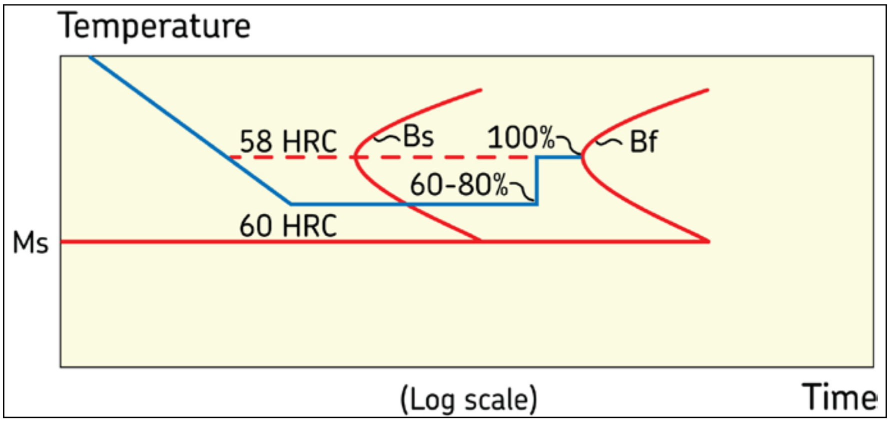

Figure 7—Bainite transformations with conventional and new process.

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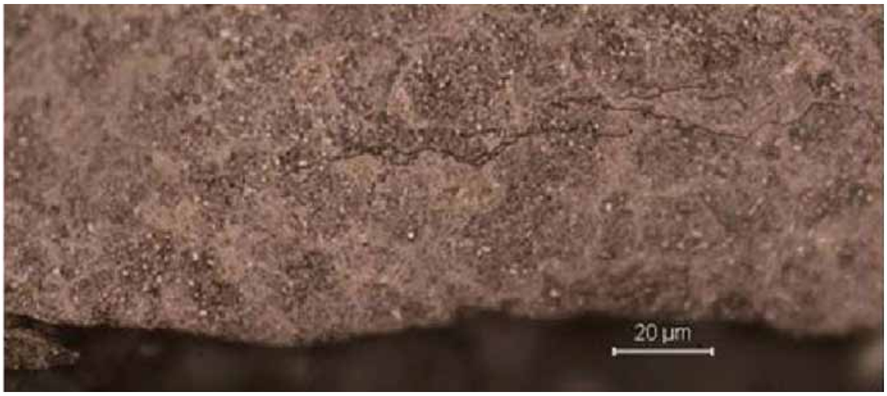

Figure 8—Severe wear damage phase under contaminated conditions.

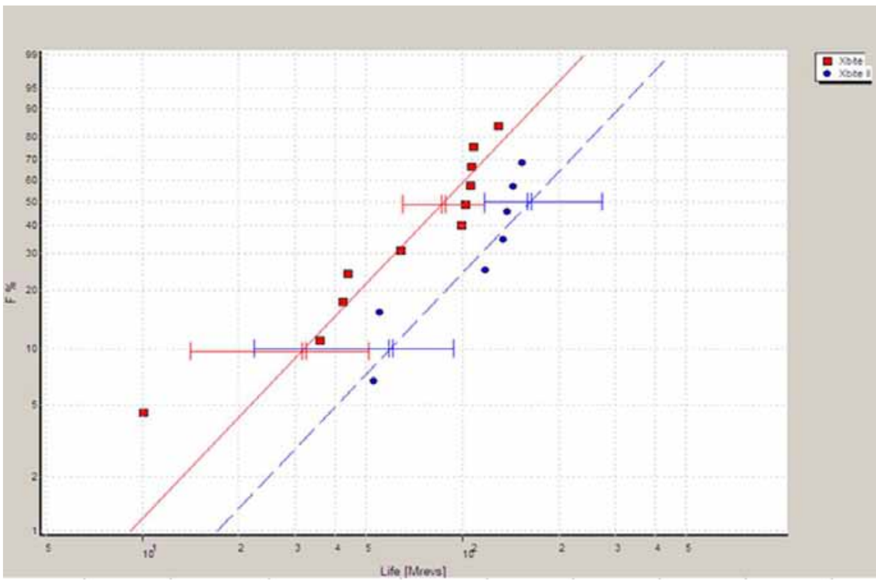

Figure 9—Weibull estimates for bearings tested under severe contaminated conditions.

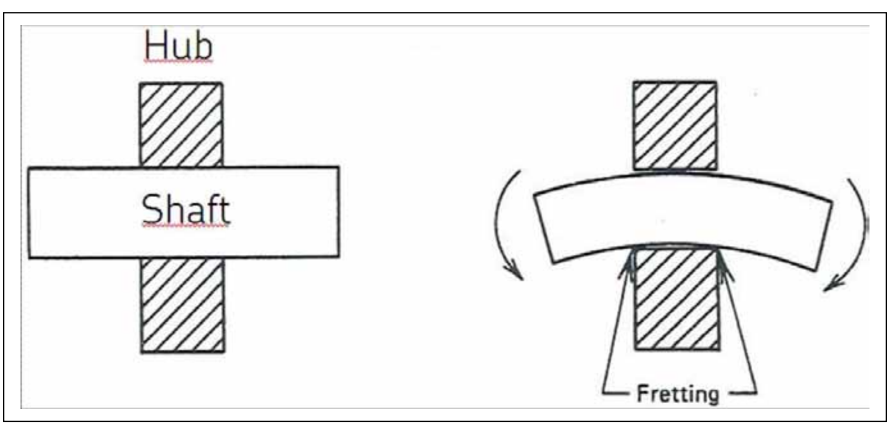

Figure 10—Straight shaft seat critical fretting points (Ref. 14).

Improving Wear and Metallic-Surface Contact Resistance

One way to enhance wear resistance is to increase the hardness of the components in contact. But this has consequences. If hardness increases, toughness is reduced, as well as a safe failure mode; spalling occurs instead of cracking, leading to catastrophic failure.

Development work was performed to find a way to improve the resistance to surface-initiated damage without losing the fundamental advantages of current processes. The bainite hardening used today can be manipulated to retain toughness, compressive residual stresses and—at the same time—increase wear and debris-contaminated condition life without loss of productivity. The process (patent-protected; Fig. 7) provides products with prolonged life under harsh running conditions while retaining all the benefits of existing long-term, successful hardening processes.

Wear tests conducted using significant amounts of debris in the form of chilled cast iron particles significantly delay the onset of heavy wear (Fig. 8).

Tests run under marginal lubrication conditions confirm the advantages of the new heat treatment method—about twice the life is attained under very low kappa conditions (Fig. 9).

This enhanced heat treatment will become the future standard and enhance the ability of bearing components to withstand environmental threats posed by application conditions encompassing debris contamination and marginal lubrication.

How the design of the interface between bearing and shafts should be adapted to modern shaft materials. Another surface failure-related mechanism that may limit the performance of a gearbox is fretting. Modern, high-strength steels offer great possibilities for transmitting high torque and load in gearbox shafts and gears, despite limited dimensions. However, modern high-strength steel still has the same modulus of elasticity as older, lesser steel material. Hence the deflection for a given condition is the same; conversely, if the increased stress resistance of a better shaft material is utilized, the shaft will deflect and/or twist more than if a less-stress-resistant material (resulting in a larger diameter shaft) was used.

To avoid damage to the joints between shafts, rings, hubs or gears, any increased shaft deflection/twisting due to downsizing requires close attention to the design of the shaft seats.

On Fretting Developing to Shaft Cracking

In a mechanical context, fretting is commonly used to describe wear and/or corrosion caused by small movements in interfacing, metallic surfaces. Also, surfaces that bind to each other via friction, e.g., interference fit joints, will, when the frictional binding force is overcome, move in relation to each other. This motion will also cause exposure of the metallic surfaces with oxidation as result. Typically, the oxidation results in discoloring. If the relative motion is large and the contact pressure high, the relative motion may, however, develop fretting. And if the fretting occurs in a critical place on a shaft surface, it may initiate surface damage that propagates to shaft breakage (Fig. 10.)

In a rolling bearing context, bearing suppliers work with fitting practices aimed at eliminating fretting caused by bearing rings subjected to rolling element loads. In the vast majority of rolling bearing applications, this is sufficient because the bearings are mounted in shaft positions where the bending/torsional stress is very low. In this situation, contact pressure between ring and shaft is only needed to eliminate motion/fretting—mainly from the shear stress between seat and ring bore resulting from the rolling elements.

In positions where shaft bending/twisting is more pronounced, the seats for shrunk components must be designed in a way that permits local motion and development of a limited surface damage without risk that the surface damage propagates. Using uncomplicated calculations, it is easy to show that interference between an inner ring and a shaft cannot eliminate motion caused by bending and or torsion. A solution to limit and "disarm" fretting due to shaft bending/torsion has to be sought outside the interference-dependent contact pressure domain.

Example: More Interference is Not Always a Solution

The contact pressure between a rectangular cross-section ring and a massive shaft when the ring is shrunk onto the shaft is expressed as a function of the ring hoop stress:

$$p = \frac{(d_e - d_i)}{d_i} \sigma_{hoop} \qquad \textbf{(3)}$$

where:
d_e is ring inner diameter, mm
d_i is ring outer diameter, mm

For bainitic-hardened rings, a maximum hoop stress of 100 MPa is allowed from a fatigue life influence point of view; maximum contact pressure expected between a bainitic-hardened inner ring and a shaft is estimated at $100 \times (d_e - d_i)/d_i$ (MPa).

The inner ring dimensions of a bearing depend on the cross-section. For a typical ISO diameter series 0 roller bearing, the radial thickness of the inner ring is roughly 5.5–7.5 percent of the bore diameter. Smaller bearings have relatively thicker rings, and vice versa. For a 500 mm bore ISO series 0 bearing, the above gives that the maximum practical contact pressure in the bore at approximately $100 \times (1.055 - 1)/1 = 5.5$ MPa.

A heavier ring cross-section would result in a larger contact pressure, but within the ISO dimension system it is virtually impossible to find large inner rings with a radial thickness larger than 15 percent of the bore diameter. And so a ring-to-shaft contact pressure of 15 MPa is considered a practical, maximum pressure to achieve.

If such a ring is sitting on a shaft seat subjected to bending and/or torsion, the bending and/or torsional strain in the surface of the shaft results in a motion between shaft and ring—unless the ring is forced to develop the same strain as the shaft. If there is motion in the mating surface, fretting is likely to follow. The only way to eliminate all motion between ring and shaft is to ensure that the surfaces "bind" to each other. In normal machines, the only binding mechanism available is friction. If friction somehow results in a ring binding sufficiently to a shaft surface subjected to bending and/or torsion, the contact pressure X coefficient of friction must be larger than the shaft surface bending and/or torsion stress.

A commonly used maximum coefficient of friction given (for degreased and heated steel components) is 0.2. Using the above-estimated 15 MPa maximum level of contact pressure and the 0.2 coefficient of friction, the maximum shaft surface bending and/or torsion stress allowable to avoid motion between shaft and ring is estimated as $15 \times 0.2 = 3$ MPa.

Since maximum-allowable shaft bending and/or torsion stress values in modern shaft materials can amount to 100 MPa, it is obvious that interference between inner rings and shafts cannot

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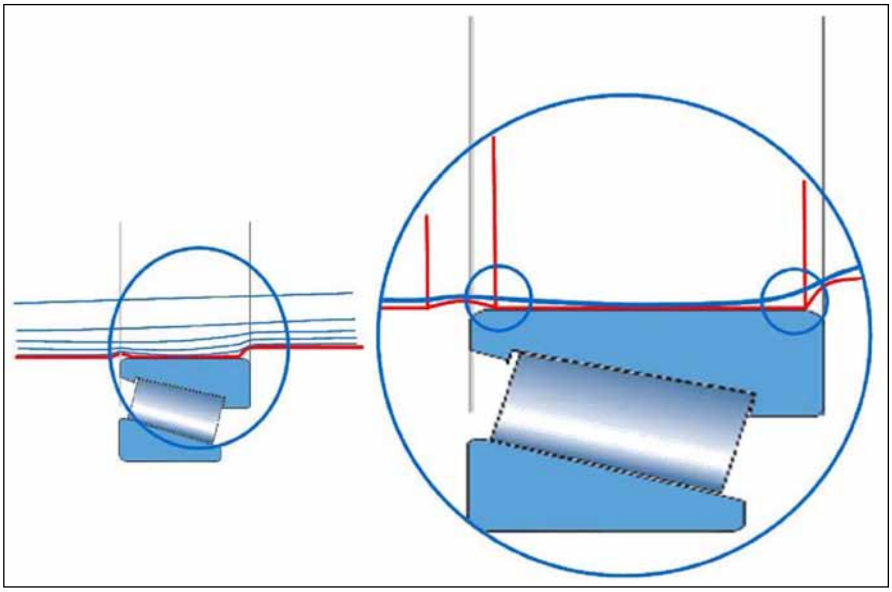

Figure 11—Shaft seats designed to create low stress areas at the borders of interference joints.

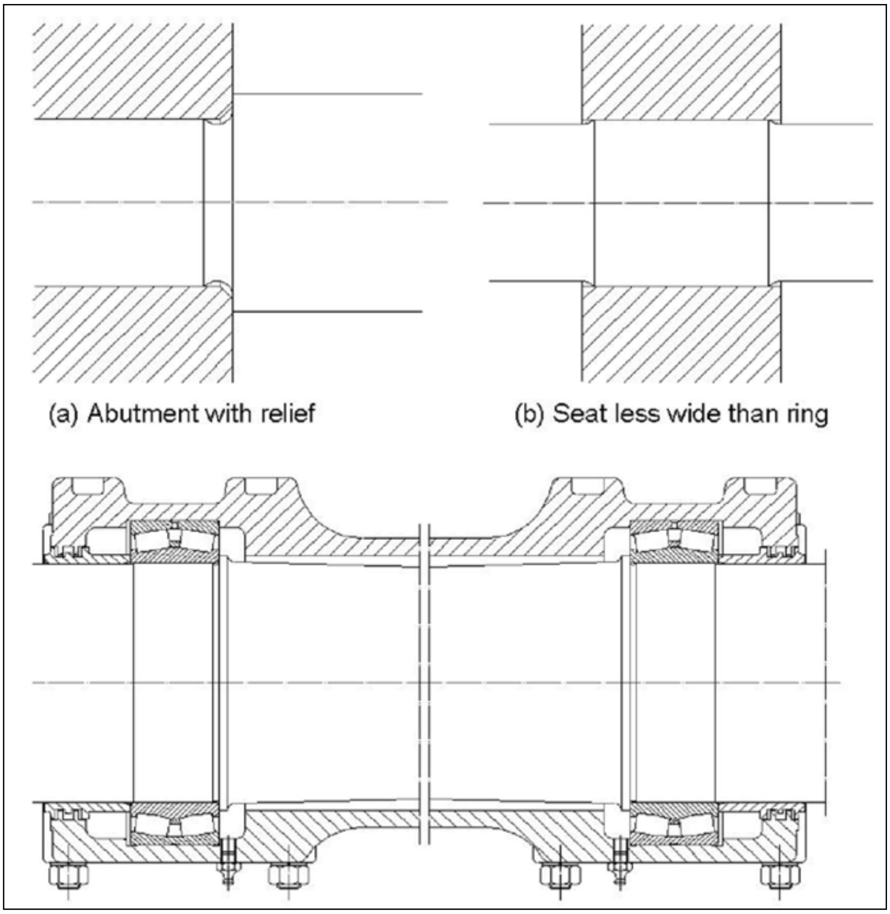

Figure 12—Design examples of components shrunk on shafts subjected to rotating, bending and/or torsion.

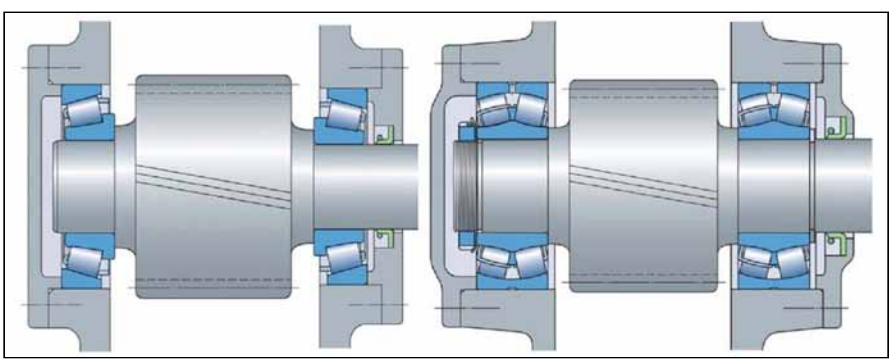

Figure 13—"Cross-locating" arrangements.

eliminate bending/torsion motion and, consequently, the fretting that may escalate into damage. Since the contact pressure available from a rolling bearing inner ring interference is not even close to eliminating the relative motion between inner ring bore and shaft seat resulting from shaft bending or shaft torsion, the task of eliminating, minimizing or disarming such motion/fretting is left to seat design.

One commonly used solution is to make the shaft stiffer under the ring, gear or hub to minimize strain (and consequently stress) in the interface surface, and to work with relieves or undercuts and make the hub part of the interference joint wider than the shaft part to relocate the fretting to the least-dangerous place (Fig. 11) in the interference joint. In Figure 11 examples "a" and "b" the softer shaft will rub against the hard inner ring or gear bore, which is far less dangerous from a shaft fatigue perspective than the opposite situation where (on a straight shaft) the inner ring or gear would rub into the softer shaft and then create a stress-raiser.

Small, sharp surface damages are stress concentrations that can develop into global fatigue cracks. In the shown examples, the rubbing part of the shaft is not subjected to bending stress and, as a consequence, not sensitive to the stress-raising effect of rubbing leading to fretting corrosion.

Figure 12 shows some design examples from machines where components (e.g., bearing inner rings) are shrunk on shafts subjected to rotation, bending and/or torsion.

Notice in the shown examples that the shaft seating is narrower than the cylindrical part of the inner ring bore; i.e., the transition between the ground bore and the machined chamfer is not contacting the shaft. This design eliminates the effect that the hard transition area between bore and chamfer of the inner ring is fretting or "coining" a groove into the soft shaft. Rather, the soft shaft is mildly deformed (crowned) at the ends of the seating.

How the design of the interface between bearing and gearbox casing influences gearbox service life. From the examples, it follows that the interface between shafts and components shrunk on shafts deserves attention—particularly if large deflections present. The interface between bearing outer rings and gearbox casing may in this respect seem less demanding, since there is no rotational load that generates motion. That perhaps is too optimistic, considering the potential effects of variable frequency drives.

For industrial transmissions, a very common bearing-to-casing design is to use a loose fit. The loose fit is often used—and needed—to make axial motion of the bearing outer rings possible; e.g., when adjusting clearance or preload in taper roller bearings, or to make two spherical roller bearings axially floating in a "cross-locating" arrangement (Fig. 13). The loose outer ring fit design is robust as long as the fit is loose enough to allow axial motion when needed, yet tight enough to distribute bearing load favorably—a sometimes-challenging compromise when involving, for example, large temperature gradients and/or varying torque.

If the casing bore tolerance is chosen correctly, the bearing outer ring is free to adjust axially; and as long as the load direction and magnitude on the bearing are constant, there will be no motion and thus no wear between outer ring and casing. Should for any reason the load direction frequently change, the outer ring contact will move in the casing bore, and wear may develop. Wear generates particles and increases looseness—typically, a self-generating mechanism.

Influence of modern electric motor speed controls in bearing type selection. In the previous section we dwelled on the outer ring to casing contact and the influence of the fit chosen.

Constant gearbox torque gives constant bearing load magnitude and direction; the loose outer ring fit works perfectly well. Should the torque change in magnitude and/or direction, bearing load magnitude and direction will change.

Variable frequency drives (VFDs) allow conventional, alternating current motors to be run at any speed, not just the nominal 900, 1,800 or 3,600 rpm associated with 60 Hz frequency.

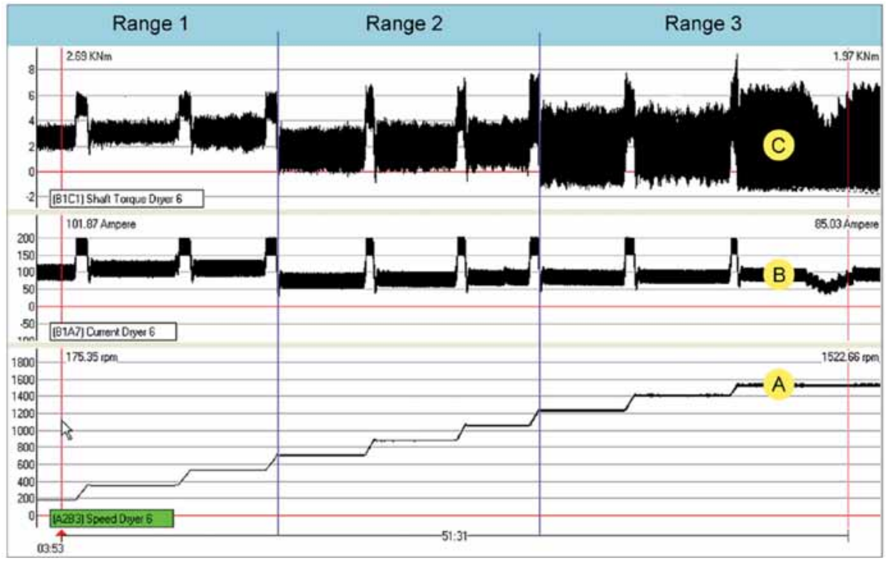

Figure 14—Measurements from paper machine drivetrain (courtesy Contec Control Technology AB, Sweden).

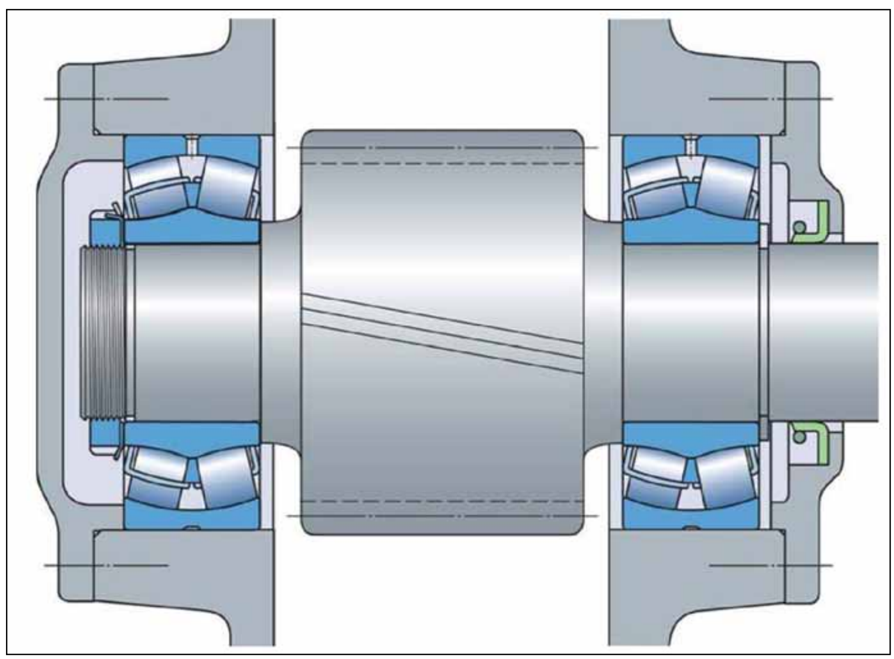

Figure 15—Two spherical roller bearings in "cross-locating" arrangement.

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Many reports on subjects like pulsating torque, torque ripple and short-time torque peaks resulting from VFDs have been written, with the main focus on electric effects like bearing common-mode voltage, currents, voltage spikes, etc. (Ref. 15). To some extent concerns such as broken couplings and twisted shafts have also been reported (Ref. 16). Such extreme examples are fortunately rare. From a transmission bearing perspective, however, torque variations far less than what are needed to destroy couplings or shafts may generate motion and, consequently, wear between the bearing outer ring and casing.

VFDs are used to drive a machine with a constant and variable speed, typically including a feedback loop to keep the machine speed constant even if the load changes. VFDs are also normally programmable to smoothly ramp the speed of the drive motor up from, say, standstill to the desired operating machine speed. Depending on how the complete drivetrain—motor/coupling/gearbox/drive/shaft/driven machine—behaves in the torsional resonance domain with relation to how the VFD speed control or speed ramp-up behaves, the drivetrain may or may not excite any of its natural (torsional) frequencies. If the speed control loop or the speed ramp-up parameters are set with, e.g., unfortunate gain, a torsional resonance excitation may be sustained. Typically, one can easily monitor a drivetrain's performance; on the other hand, the gearbox in the middle may incur damage that goes unnoticed. A pulsating torque may (in a system with low damping) develop into reversing torque, in turn reversing shaft, gear and bearing loads.

In such a situation a gearbox designed with loose outer-ring fit seat tolerances will suffer from wear. The wear will generate iron oxide—a mild but powerful abrasive agent also contributing to wear—as well as harder components such as gears and bearings. Eventually the gearbox ceases to function due to bad gear mesh and/or worn out bearings.

The measurements in Figure 14 made on a drivetrain for a dryer section of a paper machine illustrate the above behavior. The measurements were made in an attempt to find the reasons for gearbox noise and limited drive shaft card and joint service life. Three physical parameters were monitored: driving motor speed (1), driving motor current (2) and driven machine torque (3).

The speed was increased following a programmed ramp-up; the motor current appeared as a range around 100 amps with the expected peaks, when the speed made an increment (i.e., when the machine was accelerated, the current was higher) and the torque delivered to the driven machine behaved similarly to the current—at least within Range 1. After that the torque showed an increasing variation (Range 2) and after that (Range 3) the torque progressed from varying to reversing. Naturally, the complete drivetrain rotated continuously; it was only the torque that was reversing—not the rotational direction.

Unless the gear and shaft masses are very large, operation under conditions such as those in Range 3 will, for most bearing positions in the gearbox, result in large changes in load magnitude and load direction. Naturally, bearing fatigue life is reduced due to increased mean loads. But the wear associated with a loose outer ring to casing design subjected to frequently changing load direction will very likely be noticed much sooner than fatigue of a bearing raceway.

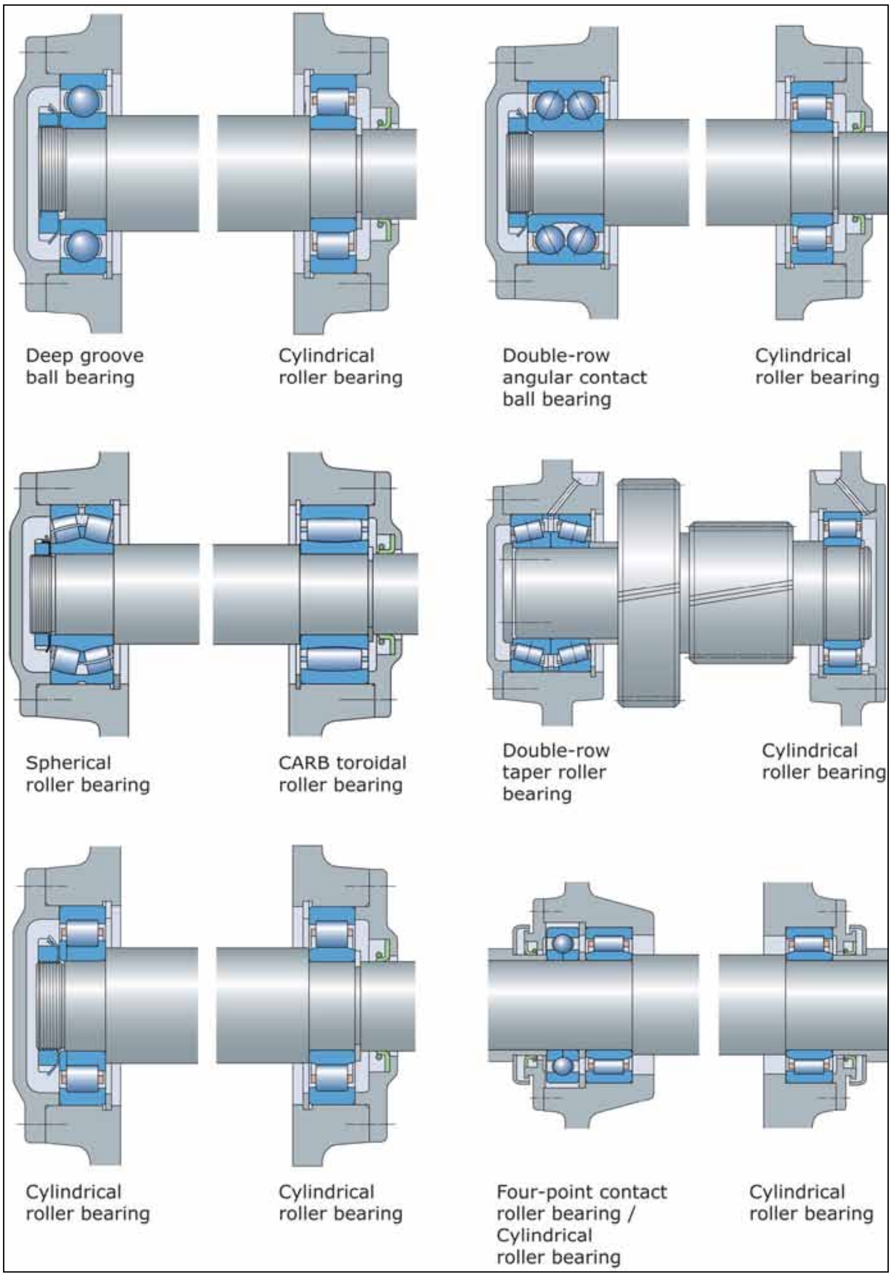

Figure 16—Different fixed-floating bearing arrangements.

The solution to gearbox service life limitations such as worn bearing seats in the casing and the resulting consequences is to *design for changing load direction*—i.e., leave the classic loose outer ring to casing fit "cross-location" designs (Fig. 15) in favor of tighter outer ring to casing fitting tolerances. *This means that axial expansion of shafts in relation to casing no longer can be managed via axial movement of one or both outer rings, so other solutions are necessary*.

Bearing arrangements such as two cylindrical roller bearings (for taper roller bearings, a tight outer ring fit makes axial clearance/preload adjustment more challenging) or arrangements of the fixed–floating type are then the right choice.

Fixed–floating arrangements (Fig. 16) can be made in a number of ways. For the examples shown the common factor is that the axial shaft expansion occurs inside one of the radial bearings, not between the outer ring and the housing. In the past, fixed–floating bearing arrangements were limited to so-called "stiff" bearings (i.e., not self-aligning). But with the introduction of the toroidal roller bearing design, self-aligning bearing arrangements are suited to tight outer ring to casing fit fixed–floating bearing arrangements.

Discussion and Conclusions

Despite the existence of advanced fatigue models for the sizing of bearings, the service life of bearings in gearboxes (and the service life of gearboxes themselves) is a more extensive matter than simply finding a sufficiently large bearing dynamic carrying capacity. Phenomena such as poor lubrication and contaminant-initiated surface fatigue are at least as important. New developments in the understanding of surface-initiated damage and how to treat material and surfaces to better-resist motion under high contact pressure offer significant improvement, but only as they relate to a larger dynamic carrying capacity in the bearings.

Finally, modern material and modern electric drive systems offer new opportunities for industrial gearboxes. But in order to benefit—not suffer—from some of the effects of strong shaft materials and multi-megawatt VFDs, bearing interface designs, tolerances and bearing arrangements must be adapted to the new operating conditions.

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