The Modified Life Rating of Rolling Bearings: A Criterion for Gearbox Design and Reliability Optimization

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Engineers typically learn that the bearing L10 life can be estimated using the so called "C/P method" — or the "basic rating life" of the bearing, a method rooted in the 1940s. Major developments have since led to the "modified rating life," released in ISO 281:2007, which includes the a_{iso} life modification factor. In this paper a succession of equations used for bearing life ratings are reviewed, and current bearing life rating practices are discussed in detail. It is shown that — despite the introduction more than 30 years ago of the adjustment factor of the basic rating life, and the standardization in 2007 of the a_{iso} modification factor — use of these improved calculation methods are not practiced by all engineers. Indeed — many continue referring to the old model as a way of seeking compliance with existing, established practices. The result is the potential for many gearbox manufacturers to continue making design decisions based on the old ISO 281:1977 "basic rating life" standard. This paper addresses these issues in the specific context of industrial gearbox bearing design, using as an example the design analysis of a helical gearbox application. The implication of not adopting modern rating life as described in ISO 281:2007 is equivalent to disregarding 30 years of bearing technology development.

Introduction

The concepts of rolling bearing rating life and basic load rating (load carrying capacity) were introduced by A. Palmgren in 1937 (Ref. 1). At that time and until the 1950s, most bearing manufacturers listed in their catalogues the load admissible on the bearing for thousands hours of operation at five different speeds. In those days the selection of a bearing size for a given application was rather a simple matter.

The concept of a single rating factor to characterize the dynamic capacity of the bearing was new and it was initially used only within the bearing company that developed this new technology. This rating method was backed by the theory of Lundberg and Palmgren (L-P) (Ref. 2) and by the Weibull statistics (Ref. 3). It was found that it could provide a correct interpretation of the many series of endurance tests available at the time, (Refs. 2, 4 and 5). This calculation method prevailed on all the others methods used at the time and it was adopted by ISO in 1962.

Before ISO acceptance the L-P model for life ratings was independently validated by Lieblein and Zelen in 1956 (Ref. 4) of the U.S. National Bureau of Standard, using endurance test data provided from different bearing manufacturers. In total, 213 test series were analyzed amounting to a total of 4,948 endurance-tested bearings. Furthermore, the statistical setting of the bearing life dispersion was also assessed by Tallian of the Philadelphia testing laboratories in 1962 (Ref. 5). In the Tallian investigation, a composite sample totaling more than 2,500 endurance-tested bearings was analyzed. The original L-P model constituted the foundation and it is remains today the nucleus of all national and international standards for fatigue life rating of rolling bearings — including subsequent theories and developments. Basically, the L-P theory (Ref. 2) developed the basis for the calculation of the dynamic load rating and equivalent dynamic load of rolling bearings as it is applied today in the ISO 281 (Ref. 8) basic rating life equation:

$$L_{10} = \left(\frac{C}{P}\right)^p \tag{1}$$

Where
- L_{10} is rated fatigue life, at 90% reliability, in million revolutions
- C is basic dynamic load rating of the bearing for a rated fatigue life of one million revolutions
- P is standardized dynamic equivalent load of the bearing
- p is life equation exponent

The availability of a standard method for the dynamic rating of rolling bearings is useful to the mechanical industry, as it allows streamlining product specifications for large-scale manufacturing and worldwide compatibility and exchangeability of rolling bearings.

The dynamic load rating allows bearing users to compare similar bearing types made by different manufacturers. Manufacturers, on the other hand, can profit from the ISO standards to rate their products, of any size and type, using just the internal nominal geometry of the bearing. Apparently the ISO standard for bearing load ratings provides a win-win situation for all parties, and this explains the widespread use of this standard in the mechanical industry.

Mechanical designers, however, need to be well informed in order to take full advantage of the opportunities offered by standardized bearing load ratings. In particular, they must be aware of the many aspects and changes that have taken place in this field through the years and how these changes have impacted gearbox performance and design practices.

In this paper we will first examine the evolution of standardized bearing life rating that has taken place after ISO 281 was first instituted in 1962. The technical justifications behind each different change will be explained, showing also the impact that variation of bearing life ratings had on gearbox design and product performance over time.

Present use of the standard will also be discussed, showing that there are different interpretations—and some misuses—of the present standard in the marketplace. This introduces distortions and uncertainty to what should be the rather straightforward task of selecting the proper bearing size for a given application.

Methods to avoid possible misleading situations and risks are suggested and explained, using an example of a bearing design analysis of a helical gearbox application. The limitations implicit in the definition of standard load rating are also considered and discussed in detail. Finally, the concept of robust design based on the latest rating rules and the modified life, i.e., ISO 281 (Ref. 8), is introduced with reference to the performance and reliability optimization of industrial gearboxes.

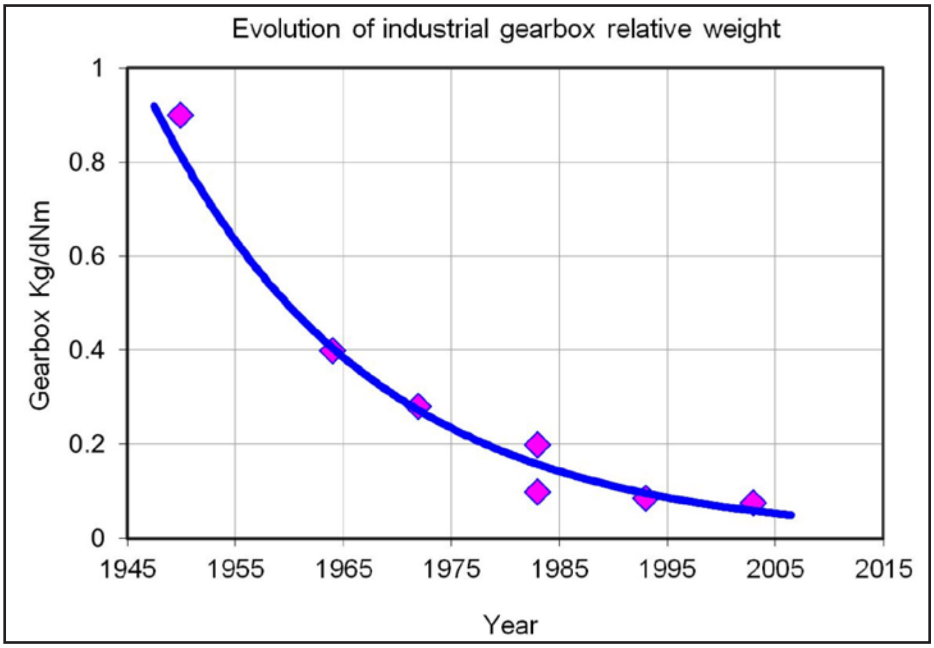

Figure 1 Typical relative weight evolution in industrial gearboxes during the last 50 years (Ref. 6).

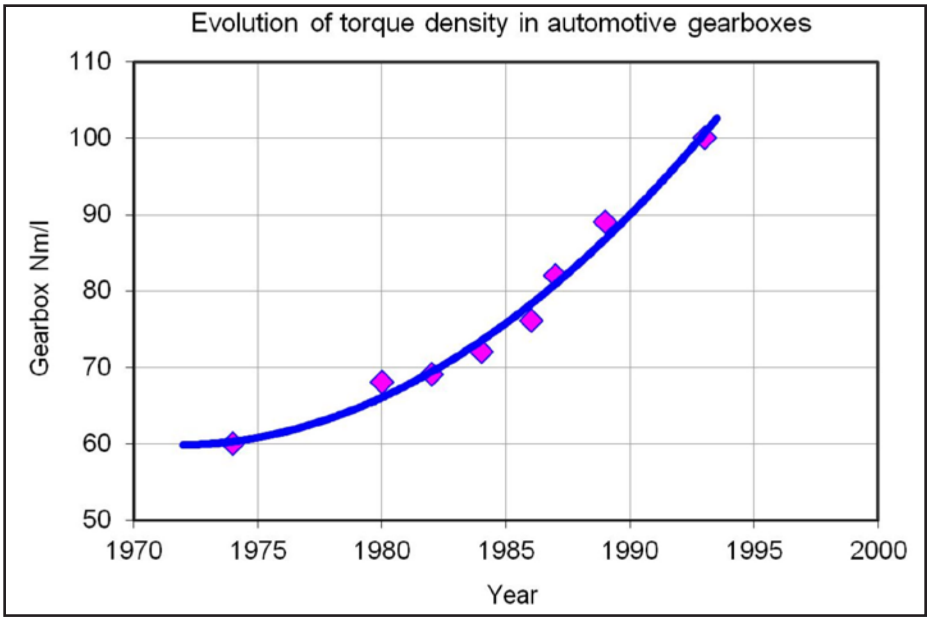

Figure 2 Progress in automotive gearbox design in terms of increased torque density (Ref. 37).

Standardization and Evolution of Bearing Life

The increase of transmitted torque, the decrease in overall dimensions and weight, together with the increased reliability and service life, are undoubtedly the technical aspects that have dominated the rapid progress in the design of mass-produced gearboxes and mechanical transmissions over the last 50 years.

Previous analysis has shown that the ratio between the transmitted torque and the mass of a typical industrial gearbox has increased up to a factor of 12 since the 1950s (Ref. 6; Fig. 1).

This progress can also be assessed by looking at the power density of the gearbox that is particularly relevant in the matter of automotive transmissions. An analysis of automotive gearboxes (Ref. 7) shows that this parameter approximately doubled during the last 30 years. During the same time period the reliability of rolling bearings for gearboxes also increased by a factor of three (Ref. 7), allowing for a 70% improvement of the torque density of automotive transmissions (Ref. 37; Fig. 2).

This real progress would not have taken place had gearbox designers not benefitted from the simultaneous, continuous progress in rolling bearing technology that characterized the same time period. Indeed, due to the stress concentration of the rolling contact and the number of rolling elements, rolling bearings are in general the heaviest-stressed and the highest-fatigue-cycled component of a mechanical system.

The fact that the life expectancy of an entire system depends on its weakest link makes the reliability of a few critical bearing components vital for the reliability of the whole transmission, and it pushes for the development of bearings with an extended life. The progress achieved in increased rolling bearing reliability can be visualized in the development of the ISO 281 rated life, relative to the original ISO 281:1962 level (Fig. 3).

Figure 3 shows normalized rated lives to the initial ISO 281:1962 rating. As discussed in the introduction, the ISO 281:1962 was the direct result of a draft proposed by the Swedish delegate Palmgren to the ISO Technical Committee in 1952. This draft basically contained the bearing rating rules developed by SKF during the previous two decades of research.

In the following period, thanks to the newly discovered elastohydrodynamic lubrication (EHL) mechanism, the effect of the lubrication quality on the expected bearing life could be addressed and an intensive research program was initiated at SKF. This research work was carried out with the prominent contribution of T.E. Tallian in Philadelphia (Refs. 9-11) and by S. Andréason in Gothenburg (Refs. 12-14). The results of this work were also made available via SKF catalogs (Refs. 15-16) and to the ISO Technical Committee for further standardization (Ref. 17). This led to ISO 281:1977 (Ref. 17). In this new version of the ISO standard, adjustment factors for the lubrication condition of the bearing, i.e., the viscosity ratio (*The viscosity ratio, κ, is defined as the ratio of the actual viscosity, ν, to the rated viscosity, ν_1, for adequate lubrication, when the lubricant is at normal operating temperature. To separate the bearing contact surfaces, a minimum viscosity ratio $\kappa = 1$ is required. Full-film conditions exist when $\kappa \geq 4$, i.e., a sufficient hydrodynamic film is formed for adequate lubrication.* $K = \nu/\nu_1$ Ref. 16), and material quality were introduced into the life rating equation. Although extensive guidelines were given, the adjustment factors were not directly provided in the standard but they needed to be specified by the bearing manufacturer.

In the 1970s material manufacturing technology related to cleanliness made substantial progress, thanks to vacuum degassing and other techniques, to prevent or reduce the formation of micro-inclusion and defects in the steel matrix. Research to quantify the effect of material-increased cleanliness on bearing fatigue life was conducted, primarily at the two main laboratories in Gothenburg and Philadelphia, and also in the new corporate SKF Engineering & Research Center (ERC) located in the Netherlands.

This intensive effort provided hundreds of test results and a very robust experimental base to justify an upgrade of the dynamic load ratings of rolling bearings. This upgrade was introduced in the SKF catalogue of 1981 (Ref. 18)

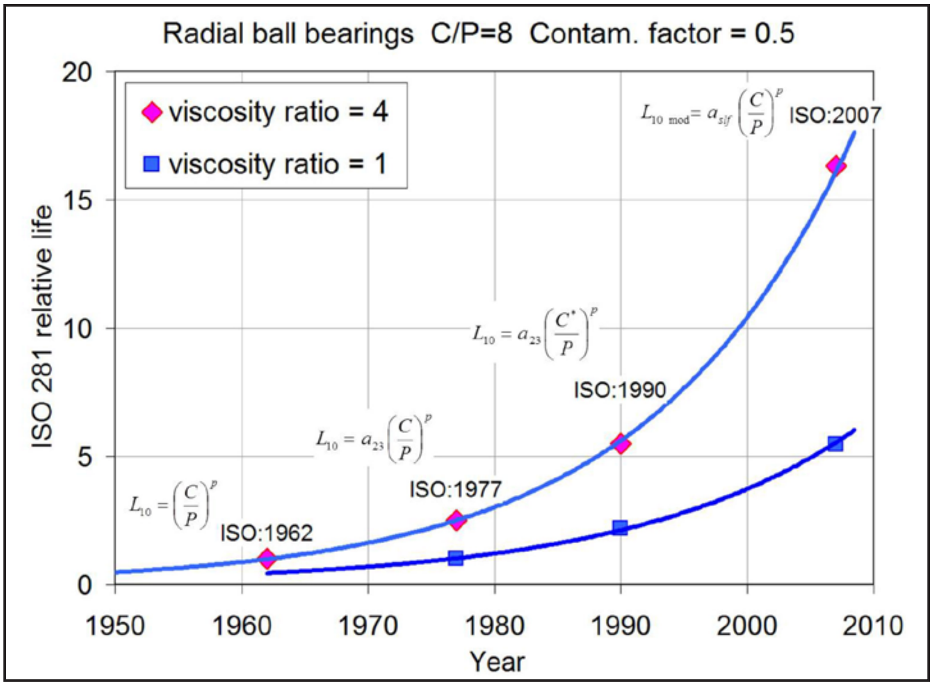

Figure 3 Typical progression of the ISO rated life of a radial ball bearing, loaded at $C/P = 8$; contamination factor = 0.5; viscosity ratio of lubrication = 1 and 4.

and integrated into the ISO dynamic load rating standard about 9 years later, with ISO 281:1990 (Ref. 19).

Introduction of Modified Bearing Rating Life

In the 1980s, SKF research (Ref. 20) found that the performance of rolling bearings could no longer be accommodated with just a linear adjustment of the basic load ratings, as was done until then. Substantial modifications were required; i.e., a significant non-linear modification of the basic load rating of the bearing was developed (Refs. 20-21).

It was found that the new calculation method could introduce, under certain conditions, a change to the rated life for a factor up to 50 times (Eq. 2). Despite the groundbreaking modifications to the old calculation routines, the new methodology was introduced in the SKF catalogue 4000 in 1989 (Ref. 22) with the addition of a stress life modification factor, a_{slf}, that combines the effect of the bearing fatigue load limit and the additional stress system related to the contamination level and lubricant quality of the contact.

$$L_{10\,mod} = a_{slf}\left(\frac{C}{P}\right)^p \tag{2}$$

Where

$L_{10\,mod}$ is modified rating life (at 90% reliability), million revolutions

a_{slf} is life modification factor.

In 2003 the calculation method was in use already for many years with good results, thus on initiative of the German standardization organization DIN, the SKF life rating method was adopted as DIN 281 Addendum 1:2003. Further discussions for the standardization of the new methodology were also initiated by the ISO Technical Committee.

To support this process, disclosure of the SKF theory and related experimental bases of the new method was also undertaken (Ref. 23). More than 260 test series (approx. 8,000 bearings) were tested to support the development and validation of the new method. This and other results were published (Refs. 23-24) to further sustain ISO standardization of the modified life rating calculation model. This process was concluded in 2007 and is the basis of the present ISO 281 rating standard (Refs. 25-26).

From the analysis of the evolution of ISO 281 it is evident that rolling bearing technology has made gigantic strides during the last 50 years, and this progress is an important aspect of the substantial improvement in the total efficiency and reliability of mechanical systems such as gearboxes and transmissions. This progress, however, does require the availability of a significant amount of endurance test data for the statistical validation of the improved rating rules in the dynamic loading of bearings.

Given the high costs involved for the endurance testing of large numbers of bearings, only some large bearing manufacturers are able to financially support the investment to finance and conduct such large test campaigns. In time, this also leads to dynamic load rating standards that reflect the performance and quality of the bearing products of the main bearing manufacturers, rather than the average or lower-quality present in the market.

This implies some uncertainties for the bearing user and gearbox designer, as the ISO ratings are universally employed. In principle, the same dynamic load rating is obtained from bearings with the same internal geometry, but quite different surface microgeometry; waviness; raceway; rolling element profilometry; shape; internal precision and tolerances; material fatigue strength; and type of heat treatment. Indeed, there are many other detailed aspects of the bearing design, such as cage and seals, which are not included in the ISO 281 rating system but are known to affect the performance of the bearing in a very significant way.

To cope with this situation, main bearing manufacturers have developed in-house, advanced computer software for the detailed modeling and simulations of rolling bearings, surrounding parts, and complete mechanical system. Advanced simulation tools include the static, quasi-static and dynamic analysis of the shaft-bearing housing system. The bearing internal geometry, mounting interface as well as shaft and housing behavior are taken into account when analyzing a bearing solution (Refs. 28-29). These computer tools are maintained and updated with the most recent results of bearing performance from new product development and endurance testing.

On the other end of the spectrum, it is also found that other bearing companies that lack specific knowledge and testing facilities can simply exploit the ISO 281 simplicity of rating rules to their own advantage. Dynamic ratings of bearings based on the simple application of ISO 281 without testing and validation procedures of the product may lead to bearing dynamic load ratings that appear equal or even superior to its competitive offerings on the surface, only to fall short to a close examination or when they are in use in the actual application.

Different Practices behind Catalogue Values

As just described, it is important to verify that catalogue values are backed up with sufficient test data and that are in line with ISO 281, by reading carefully a supplier's catalogue and technical material. This section will provide examples of misleading practices sometimes in use.

To provide an overview of the different dynamic load rating practices presently in use, roller bearings of different manufacturers were investigated and their ISO 281 dynamic load ratings were calculated and compared to the their catalogue values. The result of this survey is given in Figure 4 for five samplings. This will be discussed in terms of generic rating strategies found in today's marketplace that are in current use.

For Case (A), Figure 4, bearing products of standard (std.) quality are rated according to ISO 281; the basic rating life therefore remained unchanged. For a given assumed load, lubrication and speed conditions, it is possible to have a typical ISO 281 modification factor equal to 2.5. Thus the application of such modification factor provides an increase of the rated life of 2.5 times as expected. Case (A) has also introduced a new bearing class different from the standard for extended (ext.) performance. After exhaustive testing, Case (A) found that the load rating of the new product needed to be adjusted to accommodate for the increased performance of the product. The fatigue performance of the new (ext.) product indicated a 15% increase of the dynamic load rating with the required high level of experimental confidence.

This moderate change of the dynamic load rating is then amplified by the load-life exponent, leading to a 60% increase of the basic rated life (Fig. 4). This life increase will be further magnified by the application of the life modification factor (i.e., 2.5), leading to a modified life that is four times the original ISO basic rated life of a standard product.

Case (B), following the increased rating introduced by case (A), after some product and process development, is able to match the new performance class and, with the help of validation tests, it releases a product that is also rated with a 15% increase of the dynamic load rating. In this way, Case (B) is able to match the technical challenge of the competing product to the benefit

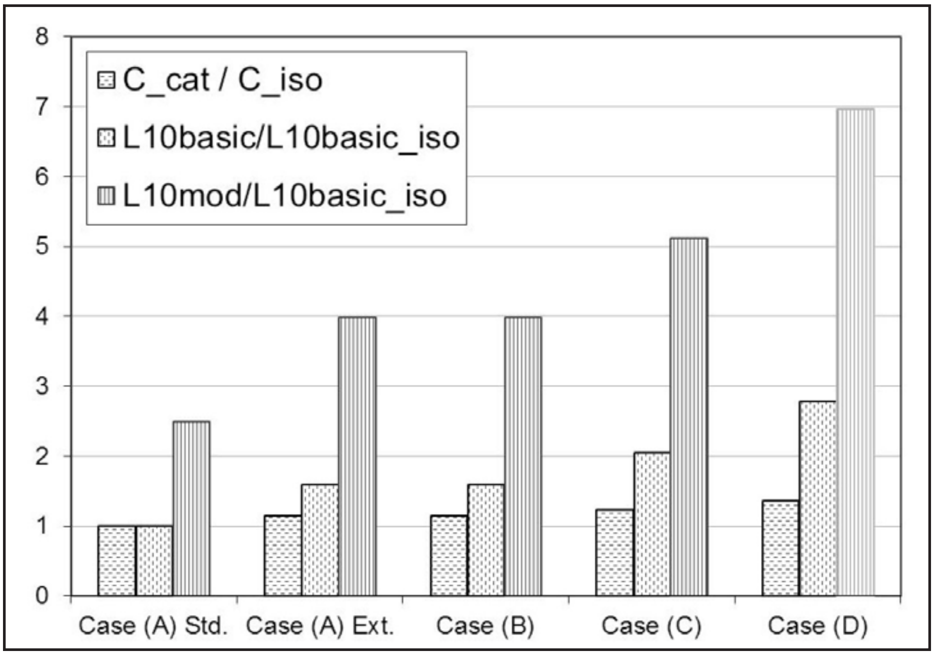

Figure 4 Overview of different dynamic rating rules found in the market place; to simplify the discussion, the life modification factor — a_{ISO} — is taken for all cases equal to 2.5.

of widening consumer choice and market competition.

However, there is also Case (C), which has the goal to present the most favorable impression possible of the strength of its products, thus publishing a 24% increase above the ISO 281 ratings for some of its products — without known statistical test back-up data.

This 24% increase happens to correspond exactly to the doubling of the basic rated life of the bearing, which is quite fortunate as this can facilitate the communication with costumers and the marketing of the bearings. Note that this increase of the basic rated life is further amplified by the use of the modification factor leading to a modified rated life that is five times the original ISO basic rating.

The investigation of bearings from Case (D) revealed a rather interesting rating strategy. Similar to Case (C), this case does not seem to have any catalogue data back-up with test results. Nevertheless, Case (D) introduces a 36% increase of the dynamic load rating in his bearing catalogue.

The basic life rating associated to such an increase is 2.8 times the ISO basic rated life. Thus it seems that this case of manufacturers has simply transformed a life modification factor equal to 2.8 into a dynamic load rating of the bearing. A further investigation about the way the life is calculated by Case (D) confirms this. Indeed, contrary to ISO 281, it appears that Case (D) advises customers to apply a life modification factor that must be, at the best, lower than one. This clearly indicates that Case (D) applies a strategy to obtain load ratings that are artificially inflated just to create the perception of benefit into the buyer, and not at all in line with ISO 281 definition.

This practice might create several problems to a not-well-informed designer. Indeed, such designer, working on the assumption that the dynamic load rating declared by this manufacturer is ISO 281-compatible, will most likely introduce the declared catalogue rating in calculation routines that make use of the ISO 281 modification factor, falsely leading to a predicted life that can be up to seven times the original ISO 281 basic life (for a 2.5 modification factor, as used for the other cases).

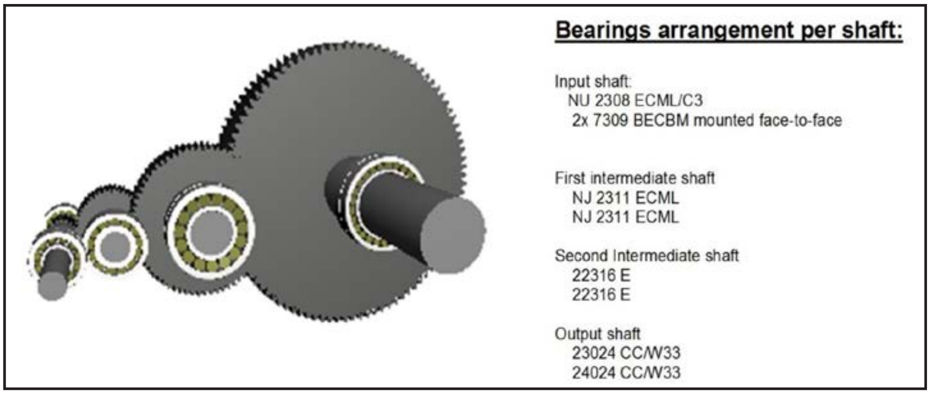

Figure 5 Schematic of a three stages helical gear unit and bearing arrangements.

This significant effect on the calculated life is shown with the light gray column of Case (D) (Fig. 4), to indicate that this rated life is not compatible with the ISO 281 modification factor. Clearly, this is a misleading way of rating bearing performance, which may lead to products that, in real applications, may not be able to reach their reliability target and fail their design expectations.

The examples depicted in Figure 4 show that engineers should be skeptical of bearing manufacturers catalogue rating information, and that they should understand what is being offered in detail — even if the information is labeled as ISO 281.

A Safe Criterion for the Selection of Rolling Bearings

At the present time, making a decision based only on the dynamic load rating, (the C value) of the bearing taken from manufacturers catalogues can be a treacherous affair.

Reliance only on the declared catalogue figure of C can in some cases lead to the wrong choices. Some suppliers will always look for ways to prove the benefits of their products to the customers, and inflating the dynamic load rating C is a simple and effective way to do this. To make the most informed decision possible, users must pay attention to the details, ask hard questions of their suppliers, and always read the fine print.

Once the C value is understood and verified, the next step to do a proper design selection is to also look at the modified life rating of the particular bearing application, rather than only the published value of C. As it will be discussed in the following gearbox calculation example, and, as general rule, it is indeed the modified rated life of the bearing, L_{10m}, rather than the dynamic load rating C, that provides the most valuable information regarding the performance of a bearing product with regard to the particular application.

The modified rated life combines the basic rating life with the stress-life modification factor. The comparison of L_{10m} will avoid hidden aspects in the particular definition of either factor. As will be shown hereafter, by using a bearing selection criterion based on the modified rated life L_{10m}, users can avoid pitfalls and arrive at informed decisions for the selection of their bearings and the optimization of their products.

Example of Newest Method for Gearbox Bearing Design

The following bearing application example was selected to illustrate that the dynamic load rating of the bearing C is less crucial as generally believed, than is the actual expected performance of a bearing application. The purpose of this example is to highlight that using modern life calculation methods provides more understanding to the designer of what issues may be faced by bearings, and consequently more chance to the designer to make adequate design choices. Comparisons of different life methods have been done by Uherek (Ref. 36) for several gearboxes. In these analyses, it is observed that contamination and lubrication were already key factors in the design.

The application example is a three-stage helical gear unit, with the following general characteristic and design:

Table 1 Calculation data and resulting basic and modified rated life										
	C, kN	P, kN	C/P	Speed, rpm	Temp, °C	κ	η_c	Basic rating life, L_{10}, hrs	Life modification factor a_{SLF}	Modified rating life, $L_{10\,mod}$, hrs
Input shaft - A bearing 1 7309 BEP	56	1.2	47	1,400	80	4	0.5	> 1,000,000	50	> 1,000,000
Input shaft - A bearing 2 7309 BEP	56	3.9	14	1,400	80	4	0.5	34,000	30	> 1,000,000
Input shaft - B bearing NU 2308 ECP	129	12.3	10.5	1,400	80	3.9	0.43	30,500	14.4	440,000
Intermediate shaft 1 - A bearing, NJ 2311 ECP	232	31.5	7.4	443	80	1.9	0.33	29,000	2.6	76,000
Intermediate shaft 1 - B bearing, NJ 2311 ECP	232	28.1	8.3	443	80	1.9	0.33	43,000	3.1	134,000
Intermediate shaft 2 - A bearing, 22316 E	490	83.9	5.9	116	80	0.7	0.22	52,000	0.4	20,500
Intermediate shaft 2 - B bearing, 22316 E	490	105.1	4.7	116	80	0.7	0.22	24,000	0.35	8,500
Output shaft - A bearing 24024 CC	430	71.8	6	35	80	0.3	0.13	183,000	0.15	29,000
Output shaft - B bearing 23024 CC	355	82.7	4.3	35	80	0.3	0.13	60,500	0.14	8,500

- This gear unit has a reduction ratio of 40, with 3 stages
- The input speed is 1,400 rpm
- The nominal power is 66 kW, which correspond to an output torque of 18 kNm
- Lubricated with circulating oil with filters, mineral oil of 320 mm^2/s @ 40°C
- The contamination level (for the contamination factor η_c) is ranked ISO -/17/14
- It is assumed that the operating temperature is 80°C
- Examining the results of Table 1 and Figure 6, the following observations can be made:
- There is no direct correlation between the dynamic load rating, i.e., basic rated life, and the actual modified rating life of the application.
- The basic rating life doesn't show the bearings that are at risk, i.e., shorter lives. Indeed the bearings that are most at risk are: the Intermediate shaft 2B and the output shaft B, both bearings were indicated as quite safe by the basic rated life.
- The modified rating life clearly indicates that two bearings that have shorter lives, and also the technical reasons behind the reduced endurance, i.e., lubrication (low κ) and contamination issues (low contamination factor η_c). Thus surface fatigue problem can be expected rather than subsurface fatigue.

A usual procedure that is utilized when life values are found too short is to select a bigger bearing, due to its higher capacity. However, the designer must be aware that the selection of bearings with an increase of the dynamic load rating *C* would not solve the lubrication and contamination issue. Even if an increased capacity would show a slightly longer life (a 10% higher capacity leads to 33-37% higher life), it does not change the phenomena: the problem is surface induced stresses, originated by the surface micro-geometry and the effect of solid particles. Also in today's gearbox, due to their high power density, the space is often limited, and it can be impossible to consider a bigger bearing size, as a solution.

These stresses can't be reduced by an increase of the dynamic load rating of the bearing. Corrective actions to improve the reliability of the intermediate (2B) and output shaft (B) bearings require not an increased dynamic load rating of the bearing but actions to improve the tribology of the surfaces in contact, (Refs. 30-31). Therefore, proven EP additives in the lubricant and preventive measurements to reduce the presence of particles and debris in the oil can provide good results.

According to the analysis of the modified life rating calculation, it is possible to propose the following improvements to increase the life of the weakest two bearings:

- Use oil containing effective EP additives

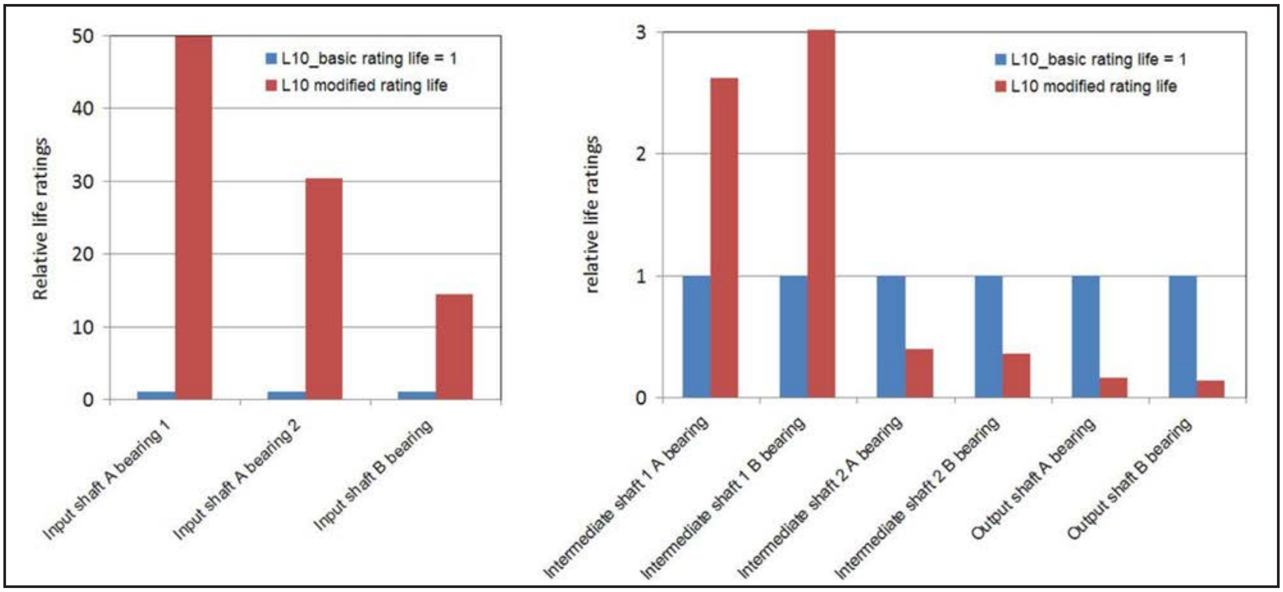

Figure 6 Overview of the modified life rating of the gearbox bearings normalized to the basic rating.

- Improve the cleanliness to ISO -/15/12 level
- Both corrective actions at the same time

Figure 7 shows the relative improvements that would be possible to achieve in relation to the original modified life rating shown in Figure 6.

- Examining the results of Figure 7, the following can be found:
- The use of oil containing proven EP additives would provide the most significant improvement—especially in the two low-speed output shaft bearings.

For the intermediate shaft bearing (2B), an improvement of 40% is obtained by improvement in the oil cleanliness. Combining this improvement with an optimized EP-additive of the oil can provide a further 80% life increase, thus enabling attainment of the required rated life for this class of applications.

This gearbox bearing calculation example shows how to use the modified rated life of bearings to improve the performance of a few critical bearings. Yet doing so will provide significant increased performance to the complete system. This transparent and simple optimization process would not be possible using the basic rating life or selecting the bearing based only on the value of the dynamic load rating *C*.

Indeed, for the input shafts A and B the dynamic load rating is quite unimportant as the size of the bearing is dictated by the required shaft size. On the other hand, for the intermediate and output shafts, the use of the basic load rating C would give no useful information for the improvement of the long term reliability of this gearbox. Selection of the input shafts A and B bearings based on inflated values of the dynamic load ratings C to reach an increased reliability for the system would fail. This is because it would not address the elimination of the surface stress system related to the low speed of the output shaft with a viscosity ratio (κ) of 0.3.

Of course the final design decision must take other parameters and constraints into account, but applying the modified life rating method allows the designer to make more informed decisions on alternative solutions with an enhanced understanding of bearing operating conditions and the expected consequences of the application performance.

Further Discussion on Selection Criteria of Rolling Bearings

The above gearbox bearing calculation example shows that the design optimization process can only be applied if the life calculation informs the designer on potential problems and hidden risks. This should be done by looking at the modified life rating, as the basic rated life would be not able to provide such critical information.

The basic rated life and the dynamic load rating *C* basically represent the subsurface fatigue performance of the bearing. In other words, the fatigue performance at extremely high load C/P~2, and as such it cannot contribute much to applications normally operating at much lower loads for reliable operation and extended periods of time.

Gearbox designers that rely solely on the C/P parameter as a selection criterion for the bearing can also easily fall into believing that higher *C* values will ensure a higher reliability for the application. In such a case the attraction for the selection of bearings with inflated dynamic load ratings *C* will be irresistible, with possibly unfortunate consequences for the actual field performance of the application.

To explain this important issue in some detail we refer to the contact pressure and related subsurface von Misses stress field in the case of two rolling bearing contacts — one (a) with an idealized nominal smooth geometry, the other (b) with its real surface micro-geometry in contact (Ref. 27) (Fig. 8).

In the case of reduced lubrication or a presence of contamination particles, the contact (b) will exhibit a severe stress system right at the surface of the rolling contact. This stress system is the direct result of the asperities and micro-profile geometry that are not part of the ISO specification of dynamic load rating *C* (which refers to the nominal geometry of the bearing and good lubrication conditions, i.e., good separation of surfaces by a clean lubricant film).

Therefore these stresses can't be reduced by adopting a bearing with increased *C*. Note that if a bearing rated with an inflated *C,* and used under the same C/P conditions, will lead to the actual increase of the contact pressure and a further intensification of the surface stresses of the contact.

The problem of reducing high surface stress in the rolling bearing is therefore not by acting on the dynamic load rating, but by acting on the tribology of the contact for the restoration of the protec-

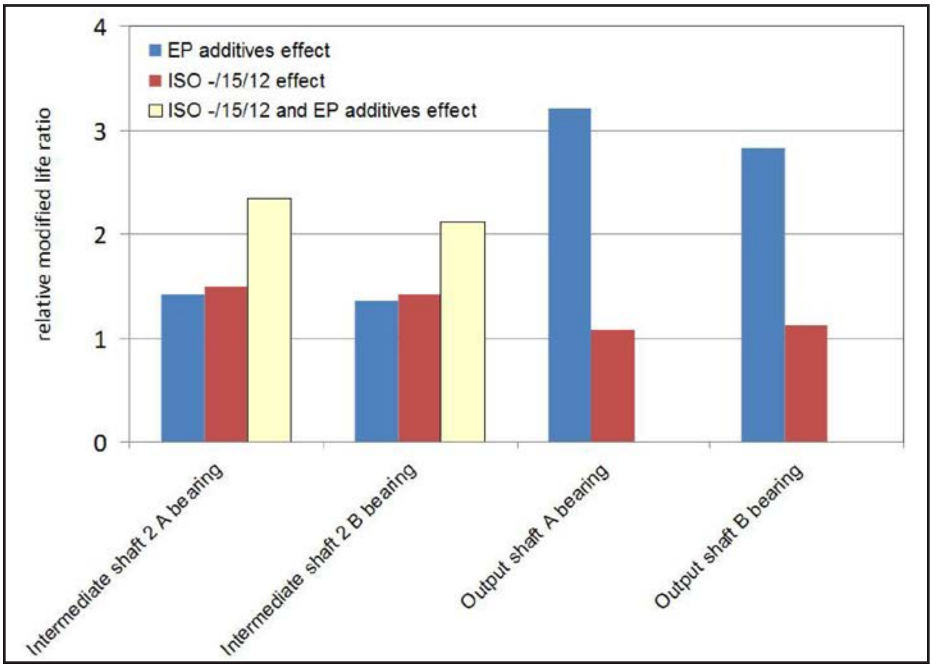

Figure 7 Relative increase of bearing life as result of corrective actions about lubricant EP additives and improved cleanliness of the oil.

tive lubricant film or reduction of geometrical imperfections. As discussed earlier in this paper, this means working with the chemistry of the lubricant and preserving the geometrical quality of the rolling surfaces by improvement of the lubricant cleanliness.

Failure Mode from the Field and Implications for Bearing Selection

There are several studies about the type of damage found in bearings that are replaced during routine maintenance work. Engel and Winter (Ref. 32) in 1979 reviewed the results of damaged bearings originated from an estimated total installed bearing population of several million bearings and arrived at the conclusion that although actual failed bearings are very small in proportion of the original population (0.05%), the predominant failure mode in the field is either lubrication- or contamination-related.

They found that lubrication and oil contamination account for 75% of all bearings failure. This same conclusion is also reported by Nierlich and Volkmuth (Ref. 35), who reviewed detailed previous data, correlating this to detailed observations and measurements of damaged bearing surfaces.

In a separate and extensive investigation supported by the German research council for drivelines technology (Ref. 33), conducted among the member companies of the FVA association, it was found that the most predominant bearing failure modes experienced in the field were inadequate lubrication and contamination. This investigation also reported and discussed in detail with other results by Gläntz (Ref. 34).

From the observation of the failure modes of field bearings, one can conclude that subsurface fatigue-initiated failure is in fact very rare. This is to be expected, as most bearing applications are selected based on a *C/P* design criterion, which is a good method to avoid classical subsurface-initiated fatigue failure—particularly under good lubrication conditions (Fig. 9).

However, the presence in the marketplace of artificially inflated dynamic load rating *C* may put in jeopardy this good record. Indeed, as observed earlier, today there is the risk that the unusually increased *C* figures may correspond in reality to a lower expected dynamic carrying capacity of the bearing, leading to a reduction, rather than an increase, of the actual reliability of the application.

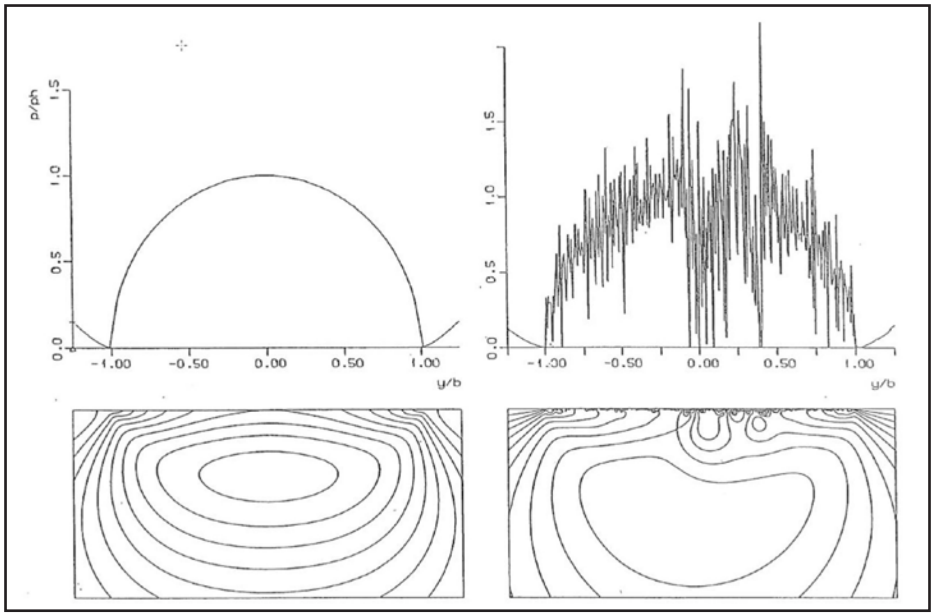

Figure 8 Example of stress field a) nominal smooth contact geometry, b) actual rough geometry (Ref. 27).

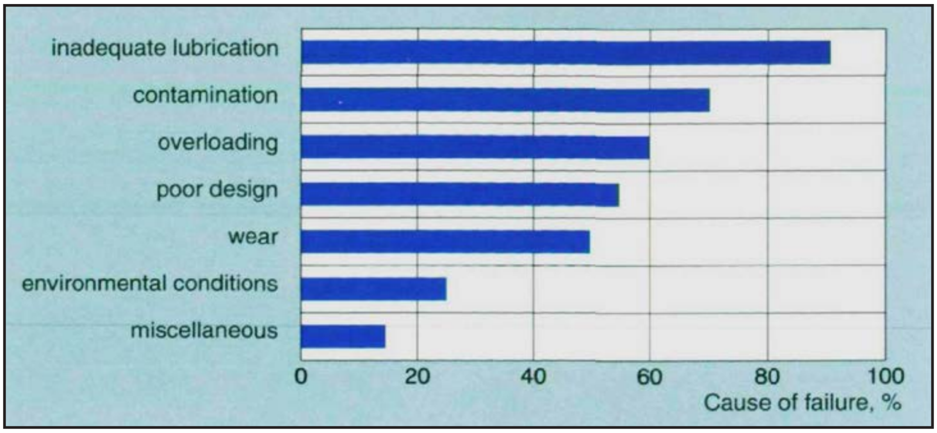

Figure 9 Causes of rolling bearing failure as percentage of responses from an enquiry among the companies of the FVA associations (Ref. 34).

Conclusions

Today's gearboxes require high reliability and extended life. The trend to increased performance will continue in an effort to develop more energy-efficient mechanical systems. At present the most commonly used method of selecting a bearing for a given application is based on the dynamic load rating *C*. This paper has shown very clearly that this methodology has several drawbacks and may not lead to an increased reliability for the system. To avoid risk of reduced or unexpected performance, well-informed designers should base their decision on these simple and practical rules:

- Engineers should look with critical eyes the printed *C* values in bearing catalogues, looking for consistency with either ISO 281 or bearing manufacturer testing practices.
- Today, the use of the modified rated life is straightforward and transparent. It is fully documented in the ISO 281 and based on physical operating characteristics of the bearing, as viscosity ratio of the oil, pitch diameter, lubricant cleanliness class, rotary speed of the bearing etc. The use of this calculation should replace current simple practices based only on the use of the C value.
- As shown in the gearbox calculation example:
  - ¤ Bearing lives based on the modified rated life can provide expected lives that can be larger

than the basic rating life. These are the bearings that are normally scrapped as not failed at the end of the gearbox life.

¤ There are also some bearings that are critical for the reliability of the system. The modified rating life "ISO 281" is the only public tool available to detect bearing criticality and evaluate possible corrective design modifications for performance optimization.

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