LUBRICATION AND CONTAMINATION

effects on bearing life, part 2

Bearing life is influenced by many factors. Two of the most important are lubrication and contamination. Through a better understanding of the mechanisms that lead to reduced bearing life, owing to the effects of such factors, we can improve bearing design and operation. In this second part of our two-part series we focus on contamination.

The use of rolling bearings in machines provides the clear advantage of reducing friction losses and increasing the overall efficiency of the mechanical system. This can be achieved only if the risk of fatigue failure is sufficiently reduced. Particle denting and contamination marks found on bearing raceways can induce stress concentrations and increase the risk of fatigue failure. However, the lubricant film developed at the dent and related local surface stresses are also significant in mitigating the effect of the crack initiation mechanism. In this second part of our two-part article, we present a methodology to link the micro-EHL film and related local stresses due to contamination denting to the fatigue life of rolling bearings. An evaluation of the method applied to rolling bearings dynamic load ratings is also carried out. Comparison between experimentally obtained rolling bearing life and lives predicted using the present theory indicate the ability of the presented model to describe the overall combined effect of the lubrication and contamination condition on the life expectancy of the bearing.

1. THE CONTAMINATION FACTOR

Following Ioannides et al. [1], and the international standard ISO 281:2007 [2], the modified rating life of rolling bearings (at 90 % reliability) is given by the following life equation:

$$L_{10m} = a_{skf}\left(\frac{C}{P}\right)^{p} \qquad (1)$$

In the above equation C represents the basic dynamic load rating of the bearing, P is its equivalent load and p an exponent (3 in case of ball bearings and 10/3 for roller bearings). a_{skf} is the stress-life modification factor described in [1] and [2], using following relation:

$$a_{skf} = \frac{1}{10}\left\langle 1 - \left(\eta\frac{P_u}{P}\right)^{w}\right\rangle^{-c/e} \qquad (2)$$

In equation (2), P_u represents the fatigue load limit of the bearing, w, c and e are constant exponents and η is a penalty factor defined within the range: $0 \leq \eta \leq 1$. This penalty to the load is introduced to give an average account to the actual stress status of the rolling contact that acts in addition to the idealized smooth Hertzian stress. As explained in [3] and in Part 1 of this article [4], this penalty factor can be described as a product of two concurrent quantities, the lubrication factor η_b and the contamination factor η_c; thus it is: $0 \leq \eta_b \cdot \eta_c \leq 1$.

In case of smooth well-lubricated raceways, with lubricant that is free from contaminant particles, we can ideally assume $\eta_b = 1$; $\eta_c = 1$, and no additional penalty is assigned to the life rating of the bearing.

This condition will be denoted with the notation *smooth* and the subscript *s*.

Under similar running conditions, but with the presence of contamination particles in the oil, we can set $\eta_b = 1$ and $\eta_c < 1$ to account for the additional localized stresses originated from the contamination dents on the raceway. We denote this condition the notation *dented* and the subscript *d*.

We can now rewrite the stress-life modification factor for the *smooth* and *dented* conditions as:

$$a_{skf,\,smooth} = \frac{1}{10}\left\langle 1-\left(\frac{P_u}{P}\right)^{w}\right\rangle^{-c/e} \qquad (3)$$

with $\eta_b = 1; \eta_c = 1$

$$a_{skf,\,dented} = \frac{1}{10}\left\langle 1-\left(\eta_c \frac{P_u}{P}\right)^{w}\right\rangle^{-c/e} \qquad (4)$$

with $\eta_b = 1; \eta_c < 1$

The fatigue life reduction, resulting from contamination denting of the raceway, can be quantified by comparing the theoretical fatigue life of a bearing under the *smooth* and *dented* conditions. Thus it can be examined by the following life ratio:

$$\frac{L_{10,\,d}}{L_{10,\,s}} = \frac{a_{skf,\,dented}}{a_{skf,\,smooth}} \qquad (5)$$

The above ratio can be evaluated numerically, reverting to stress and using the Ioannides-Harris [5] fatigue life equation applied to the actual geometry of the rolling contact:

$$\ln\frac{1}{S} \approx A \cdot N^{e}\int_{V_R}\frac{\langle \tau_i - \tau_u\rangle^{c}}{z'^{h}}\,dv \qquad (6)$$

In equation (6), S represents the survival probability of the rolling contact, N is the number of fatigue stress cycles, A is a scaling constant, τ_i is the fatigue criterion, τ_u is the fatigue limit shear stress, z'^{h} is the stress-weighted average depth, and V_R is the stress volume at risk of the Hertzian contact. Note, however, that the relevant quantity of equation (6) affecting the life ratio (5) is the volume-related stress integral, which is:

$$I = \int_{V_R}\frac{\langle \tau_i - \tau_u\rangle^{c}}{z'^{h}}\,dv \qquad (7)$$

Using the above notation, the fatigue life of a rolling contact, (with u the number of overrolling per revolution), can be expressed as:

$$L_{10} = \frac{N}{10^{6}\,u} \approx \frac{1}{u}\cdot\left(\frac{\ln(1/S)}{A \cdot I}\right)^{1/e} \qquad (8)$$

In this equation, the stress integral *(I)* can be computed for a dented contact and for an ideally smooth contact. Thus, it can be used for the estimation of the expected effect on the bearing life, as also given by the life ratio of equation (5). In other words, the following applies:

$$\frac{L_{10,\,d}}{L_{10,\,s}} = \frac{a_{skf,\,dented}}{a_{skf,smooth}} = \left(\frac{I_{smooth}}{I_{dented}}\right)^{1/e} \qquad (9)$$

Inserting equations (3) and (4) into equation (9) enables deriving the theoretical equation, describing the contamination factor η_c in terms of the actual stress condition of a particle dented contact:

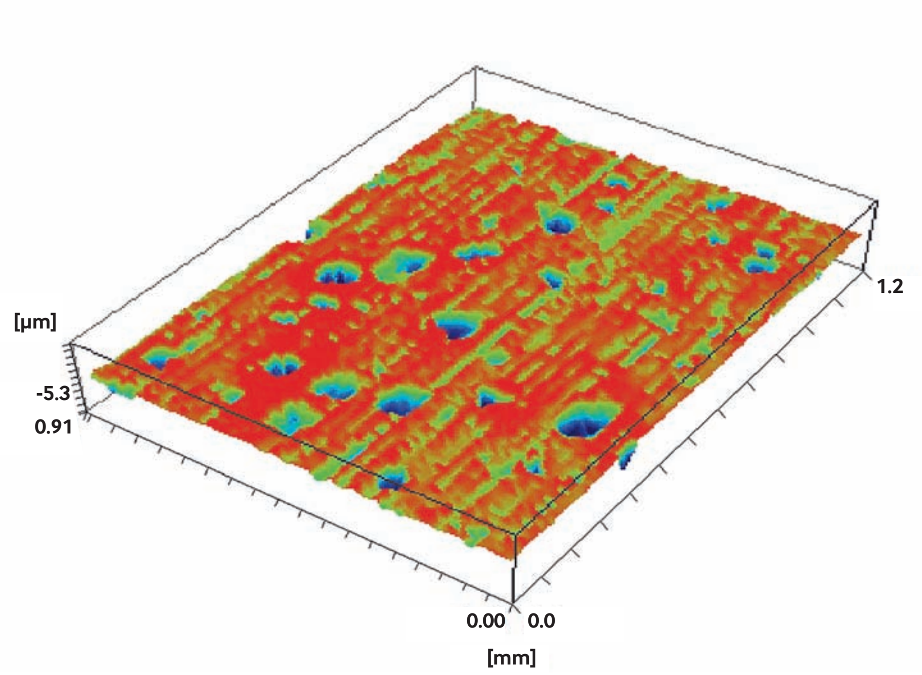

Fig. 1: Example of a typical 3D sample map of a contamination-dented rolling bearing raceway.

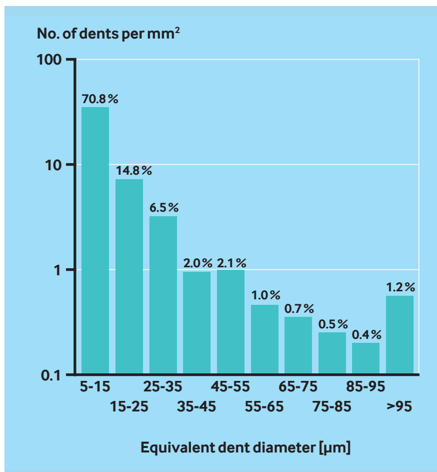

Fig. 2: Example of dent population statistics obtained from 3D sample maps of a rolling bearing raceway. The bearing had operated under severe contamination conditions, comparable to an ISO 4406 cleanliness code classification within the range -/19/16 to -/21/17.

$$\eta_c^{'} = \frac{P}{P_u}\left\langle 1 - \left\langle 1 - \left(\frac{P_u}{P}\right)^{w} \right\rangle \cdot \left(\frac{I_{smooth}}{I_{dented}}\right)^{-1/c} \right\rangle^{1/w} \tag{10}$$

From equation (10), numerical values of η_c can be computed starting from the calculation of the volume-related fatigue-stress integral, estimated from the different amount of contamination denting. Basically, the life ratio of equation (9) is evaluated to represent bearings exposed to lubricants with a different amount of contaminant particles, using the micro-EHL methodology as described in Part 1 of this article [4]. In order to perform this calculation, it is necessary to have a measure of the population of dents that are found on typical raceways of bearings exposed to lubricants with various degree of particle contamination. Statistical measurement of the dent population found on the bearing raceway (fig. 2) can provide a direct representation of the effect of the cleanliness of a given oil and related operating conditions.

The evaluation of the stress conditions resulting from several types of dent distributions can be performed in different ways: i) Using an explicit direct method, starting from the 3D mapping of actual dented regions of bearing raceways (fig. 1) and proceeding as explained in Part 1 [4] in the case of the lubrication factor stress integral calculation; or: ii) Implicitly, by computing the stress integral for different dent geometries (reference dents) and related lubrication conditions. This basic data can then be used to compute actual dented surfaces by properly adding the effect of the volume-related stress integral, describing each specific type of dent population found on the dented region of the bearing. The application of this method requires the automatic counting and categorization of dent population from 3D surface samples of the bearing raceway. A specific type of this implicit method was applied using a dent counting and classification system developed in house. This is required to characterize different dent patterns found in bearing applications, which

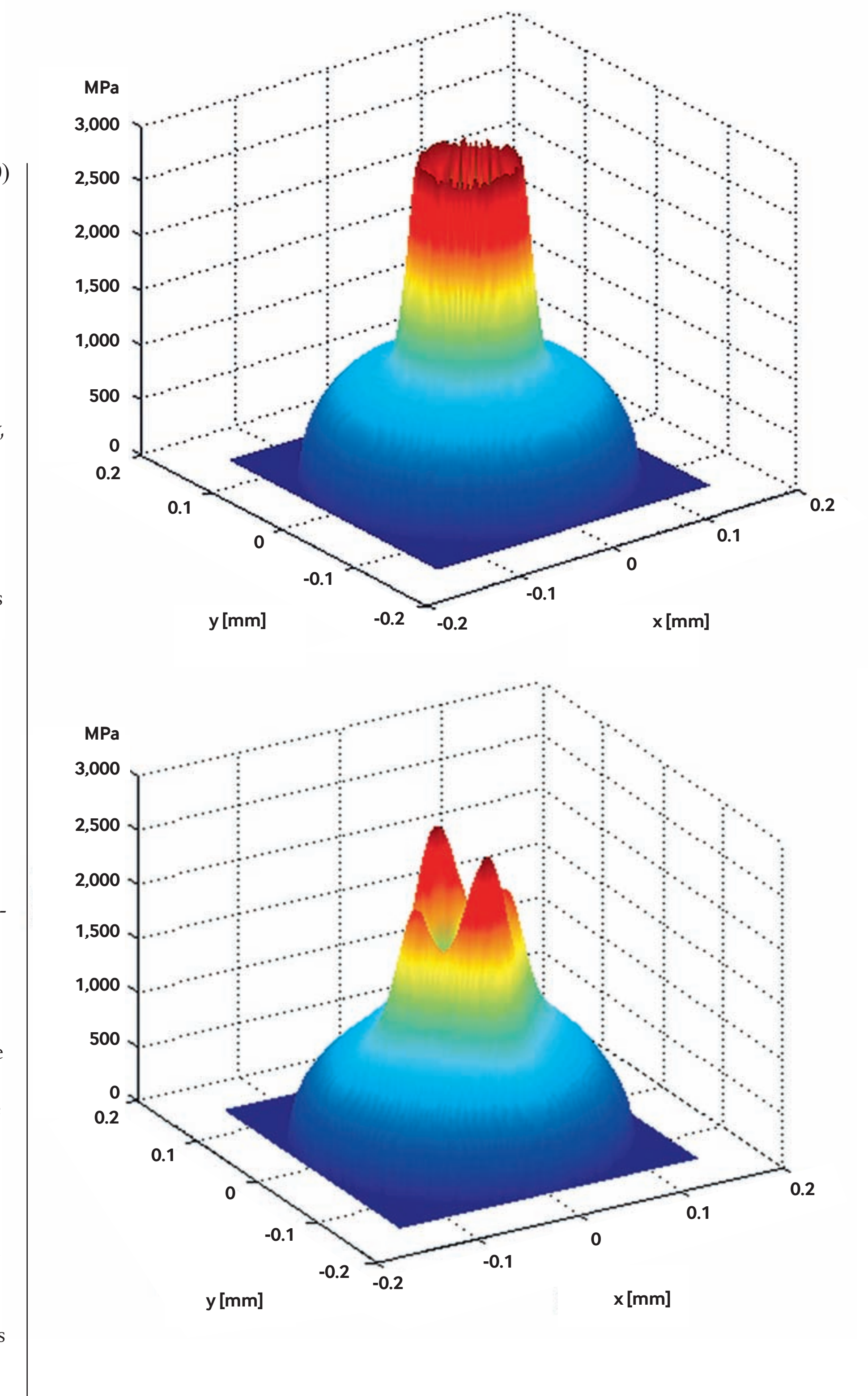

Fig. 3: Example of contact pressure calculation of a typical dent (150-micron diameter and 5-micron depth) under dry and lubricated conditions.
Top: contact stress under dry condition (no lubricant film).
Bottom: same dent showing the contact stress attenuation induced by a 0.3-micron oil film present in the rolling contact.
In both cases the nominal Hertzian pressure is $P_o = 1.255$ *GPa*.

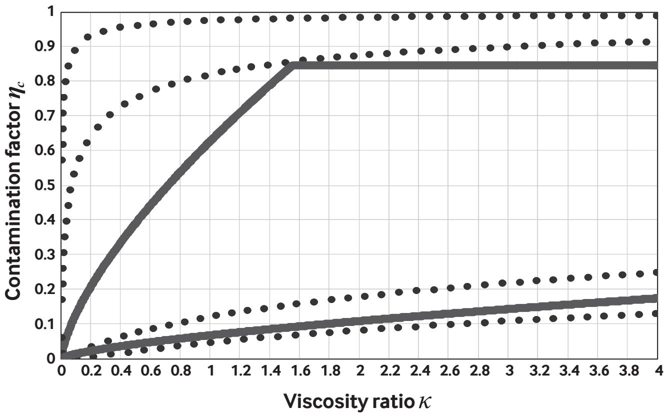

Fig. 4a: Comparison of the numerically derived contamination factor (calculated value-range indicated by dots) and the contamination factor obtained from equation 11 (solid line). Bearing mean diameter d_m = 50 mm and lubricated with high cleanliness oil (ISO 4406 code -/13/10; upper curves) and with severe contamination conditions (ISO 4406 code -/19/16; lower curves).

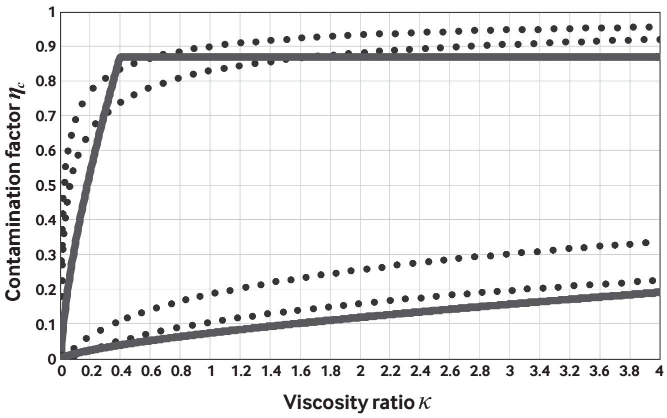

Fig. 4b: Comparison of the numerically derived contamination factor (calculated value-range indicated by dots) and the contamination factor obtained from equation 11 (solid line). Bearing with mean diameter d_m = 2,000 mm and oil cleanliness level ISO 4406 -/15/12 (upper curves) and mean diameter d_m = 25 mm and oil cleanliness level ISO 4406 -/17/14 (lower curves).

dent populations can be qualified in relation to the operating condition of the bearing (fig. 2).

It is known that bearings that run in similar conditions may have significant variations in their denting pattern and resulting dent population. This may be due to a difference in the local oil flow or the geometrical configuration of the bearing itself. However, it is in general apparent if a bearing has operated under heavy contamination or, conversely, under high cleanliness conditions.

Intermediate cases can also be recognized and the corresponding cleanliness characterized according to a simple scaling. Therefore, this method is suited to rate the contamination dent population found on an actual bearing raceway, as for instance for the dent population shown in fig. 2, and relate this to a corresponding cleanliness rating of the lubricant, expressed within a given range.

For the use of the above methodology, the increased volume-related stress integral of a specific set of reference dent geometries was carried out. This calculation was extended to include the effect of the oil film thickness (lubrication conditions) on the resulting stress risers at the dent.

This is an important feature of the present model, enabling local oil film effects to be explicitly accounted for within the assessment of the contamination factor. The stress reduction effect induced by the lubricant film, as shown in the calculation example of fig. 3, can be significant and must be included in the analysis. In this way the effect related to the lubricant film can be decoupled from the global overall surface roughness effect, thus allowing setting $\eta_b = 1$ within the assessment of the contamination factor of equation (10).

As illustrated in fig. 3, the magnitude

Table 1: Summary of test conditions used for bearing life testing.

Bearing type	Designations	Load – C/P	Lubrication – κ
Deep groove ball bearing	6305, 6205, 6206, 6207, 6309, 6220	1, 2.8, 2.4, 2.1, 3.1, 3.5, 4, 6	4, 3.4, 2.1, 2, 1
Cylindrical roller bearing	NU 207 E, NU 309 E	2.5, 2.77, 2.82	4, 1, 0.8
Spherical roller bearing	22220 E, 22220 CC	2.2, 2.3, 2.5, 2.3, 2.7, 3, 4.7	4, 3.6, 1.8, 0.37, 0.28
Tapered roller bearing	331274, K-LM11749/10, K-HM89449/10, K-580/572	1, 1.1, 1.3, 2.5, 3.5	4, 2.9, 0.9

and distribution of the stress rise at the dent from a given dent geometry is strongly affected by the lubricant film present in the rolling contact. Thicker lubricant films will result in a reduction (damping) and redistribution of contact stress developed at the dent, while a negligible film thickness will sharpen the stress and raise the stress concentration to its maximum.

To include this effect, stress damping related to the lubricant film was applied in the parametrical evaluation of the dent contact pressure distribution in accordance to the numerically calculated results [6], [7], [8] and [9]. In this approach, the stress attenuation is related to the viscosity ratio κ of the bearing, under the assumption that the expected averaged lubricant film of the rolling contact will be proportional to this lubrication parameter. Note that bearing size variation will also affect the life ratio of equation (9) and the corresponding η_c of equation (10). Large bearings will have a large smooth stress integral, which will have a dominant effect over the dent-related stress integral. Furthermore, the maximum dent-related stress concentration has a natural upper limit due to the maximum particle size that may be transported in a lubricant stream; thus it will not be dependent on the bearing size but rather will be only affected by the cleanliness grade of the lubricant. Therefore, large diameter bearings have an advantage in terms of sensitivity to the effect of contamination, compared to bearings of smaller diameter.

2. CONTAMINATION FACTOR IN PRACTICE

As discussed earlier, the numerical solution of equation (10) provides a theoretical base for the estimation of the contamination factor η_c, enabling the parametric assessment of this factor for various degrees of particle contamination rating and lubrication conditions of bearings of various sizes.

Results of this analysis can be compared to the engineering model of η_c obtained from diagrams and tables available in [1] and [2]. In [2] the contamination factor η_c is provided as a function of κ for different values of the bearing mean diameter d_m. Plots are given for a few basic ranges of the cleanliness classification rating of the lubricant. Basically, the engineering model of η_c can be described using the following basic equation [2]:

$$\eta_c\,(\kappa, d)_{\beta\,cc} = \min\left(c_1(\beta_{cc})\,\kappa^{\,0.68}\,d_m^{\;0.55}, 1\right) \cdot \left[1 - \left(c_2(\beta_{cc})\,d_m^{\;-1/3}\right)\right] \tag{11}$$

where c_1 and c_2 are constants assigned according the oil cleanliness classification rating. This classification is based on the ISO 4406 cleanliness scale (or to a corresponding filtration quality grading, i.e., ISO 16899) [2]. Unlike the η_b model, η_c depends on three parameters, thus the comparison between the numerically estimated values of η_c derived from equation (10) and the η_c values obtained from equation (11) is complex. In order to simplify the comparison we consider two separate cases related to oil circulation systems with filtration. The results of this comparison are shown in figs. 4a and 4b.

- Case i) in which the bearing diameter is maintained constant and two extreme cleanliness classes related to oil circulation with filtration system are evaluated, (fig. 4a).
- Case ii) in which the bearing diameter is varied between two different extreme sizes and an intermediate levels of contamination related to oil circulation with filtration is examined, (fig. 4b).

Regarding the functional dependency of η_c to the lubrication parameter κ, the following can be noticed:

i) for high κ values, the engineering model indicated with a solid line in figs. 4a and 4b displays a good correlation with the η_c value from the theory (value-range indicated by dots), while,
ii) for low values of κ the engineering model (11) is in some cases more conservative.

In relation to this, it must be noticed that it is indeed in the low κ region that the theoretical model has greater uncertainty, as it is based on a simple nominal film thickness while the failure mechanism is mainly a local event. The conservative approach that is adopted by the engineering model (11) seems therefore justified. It can be concluded that equation (11) is a reasonable engineering model for the assessment of the contamination factor used in bearing life ratings.

3. EXPERIMENTAL RESULTS

Endurance testing of bearings subjected to predefined contamination conditions is not a simple undertaking [10]. There are many difficulties to simulate in a test environment the type of over-rolling dent patterns and dent damage that is expected in a standard industrial application, e.g., gearbox, characterized by a given ISO 4406 cleanliness classification of the oil. For instance, in a test environment the lubricant reservoir can be much larger (a factor >100) than in a normal bearing application. Furthermore, the way the oil is flushed through the bearing may significantly differ from what is in general occurring in an actual bearing application. Therefore, in setting up the test conditions, the actual total number of particles that will reach the test bearing and will be over-rolled must be considered as the contamination reference. This is required to avoid excessive dent damage that will misrepresent the conventional conditions of typical rolling bearing applications. Furthermore, the contamination level will be the result

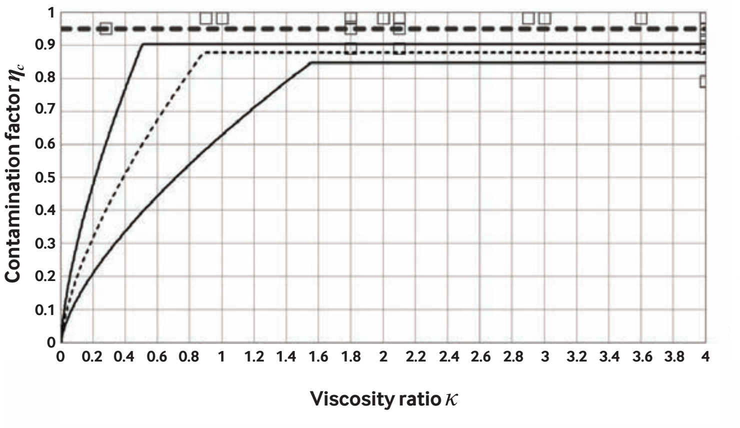

Fig. 5a: Comparison of the contamination factor η_c obtained from bearing life testing (box symbols) and the corresponding η_c curves (solid lines) obtained from equation (11), bearings with mean diameter d_m = 50-200 mm (lower and upper curve). Live testing performed under clean conditions (comparable to an ISO 4406 code -/13/10; -/14/11). Thick dash line is the trend line (curve fit of the experimental data points).

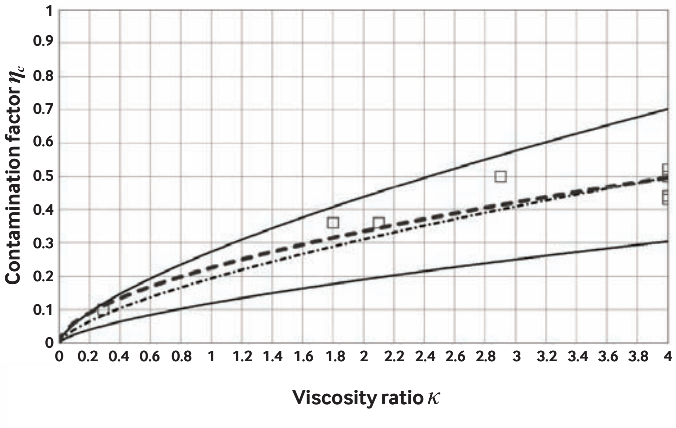

Fig. 5b: Comparison of the contamination factor η_c obtained from bearing life testing (box symbols) and the corresponding η_c curves (solid lines) obtained using equation (11), bearings with mean diameter d_m = 30-100 mm (lower and upper curve). Life testing of bearings with filtering seals (slight contamination conditions: ISO 4406 -15/12; -/16/13). Thick dash line is the trend line (curve-fit of the experimental data points, imposing the origin point). Dash dot line is the corresponding engineering model with d_m = 60 mm.

of the balance between the contaminant originally present in the system and the particles that are generated and removed in the circulating oil. These difficulties, among others, hindered past attempts to adopt purely experimental methods in the development of a contamination factor for bearing life ratings.

Nevertheless, endurance tests under different lubrication and contamination conditions were performed in the past, and a significant number of test results have become available in time [10]. Thus, it is conceivable to check the response of the contamination factor model (11) with respect to these life tests. The test results reported here are related to 172 bearing population samples tested during the past few years. Considering that each bearing sample is normally formed with a group of 30 bearings, several thousand bearings were endurance tested for this set of experimental results. A summary of some relevant information regarding the bearing types and testing conditions is given in table 1. As shown in the table, the test bearings were mainly of small to medium size and lubricated with turbo oil of basically three ISO viscosity grades: VG 9, VG 32 and VG 68. The tests were conducted at various rotational speeds, ranging from 1,000 to 6,000 r/min and selected so that the outer ring temperature was maintained within the required limits of the test.

The evaluation of the contamination factor is based on the direct comparison of the factor derived from the endurance test median L_{10} life using a back calculation procedure. The experimentally derived contamination factors have then been compared with the engineering model of η_c as provided by equation (11). This comparison is shown in figs. 5a, 5b and 5c.

Basically, the cleanliness conditions used in the bearing life testing can be categorized in three classes:

- Standard cleanliness tests, fig. 5a. These tests are carried out with good oil filtration provided by a multi-pass high-efficiency filtration system $\beta_{x(c)} = 3$ (or better). With this filtration, cleanliness codes ISO 4406: -/13/10 to -/14/11 can be expected. Considering the mean diameter range of the tested bearings, the expected η_c factor that results from this type of testing with full film lubrication should be between 0.8 and 1.
- Filtering seals tests, fig. 5b. Life tests in which the oil is pre-contaminated with a fixed quantity of hard (750 HV) metallic particles. The contamination particles are normally distributed in the size range between 25 to 50 microns. The oil is let to flow around the test bearing that is equipped with rubber seals. The bearing seals provide a filtering action, and only a limited amount of particles of small sizes will be able to penetrate the seals and, by that, contaminate the bearing. These types of tests can be rated as slight contamination (oil bath, ISO 4406 codes: -/15/12 to -/16/13). Under the given test conditions the expected η_c factor of this type of testing can be between 0.3 and 0.5.
- Pre-contaminated tests, fig. 5c. The test starts with a 30-minute run-in with an oil circulation system that is contaminated with a fixed quantity of hard (~750 HV) metallic particles (size range 25 to 50 microns). After this run-in time under contamination conditions, the bearing is tested in standard clean conditions. This procedure has shown to be very effective in producing a repeatable dent pattern, i.e., predefined denting on the bearing raceways. Under the given test conditions, this type of test is rated as typical to severe contamination (oil bath, ISO 4406 codes: -/17/14 to -/19/15). The expected η_c value for this type of endurance test can be between 0.01 and 0.3.

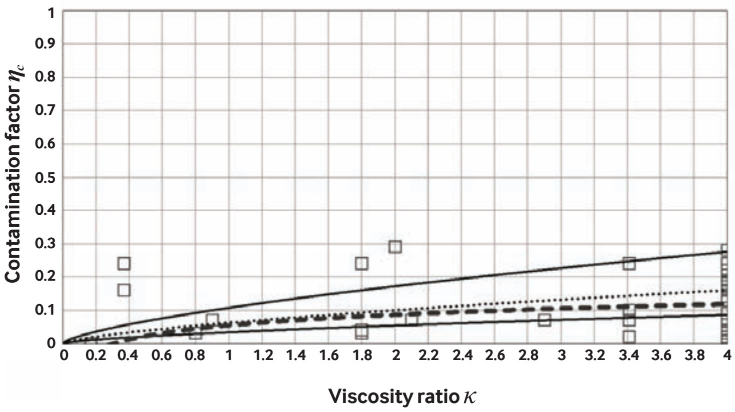

Fig. 5c: Comparison of the contamination factor η_c obtained from bearing life testing (box symbol) and the corresponding η_c curves (solid lines) obtained using equation (11), bearings with mean diameter d_m = 25-100 mm (lower and upper curve). Life tests results of pre-contaminated run-in bearings (condition related to a typical to severe contamination comparable to ISO 4406 -/17/14 to -/19/15). Thick dash line is the trend line (curve fit of the experimental data points, imposing the origin point). Light dot line (just above the trend line) is the corresponding curve from equation (11).

In figs. 5a, 5b and 5c it can be noticed that the experimental data points are limited in number, thus they are unable to show a clear trend to compare to the engineering model η_c of equation (11). Nevertheless, a fairly good match can be appreciated between the average values of the points related to the three different cleanliness classification levels used in bearing life testing and the contamination factor η_c obtained, using the simple engineering model of equation (11). Indeed, the trend line fitted from the experimental data points shows to be well aligned with the corresponding η_c curve for all three cases that were examined. It is also clear that a detailed evaluation of the model response based only on experimental data is difficult, due to the inherent dispersion shown by fatigue life results. The theoretical support provided by resolving equation (10) is thus essential to the development of the simplified engineering mode η_c of equation (11).

4. DISCUSSION AND CONCLUSIONS

The basic methodology for the derivation of the lubrication factor η_b and contamination factor η_c used in bearing life rating is presented. It is shown that a simple basic theory of micro-EHL can be applied for the evaluation of both factors. The theory is based on the application of micro-EHL pressures and the parametrical evaluation of the Ioannides-Harris volume stress integral related to smooth/dented conditions of real bearing surfaces. The estimation of the volume-related fatigue stress integral is based on a FFT numerical calculation scheme of the surface and subsurface stresses of the rolling contact.

This method is particularly suited to deal with the special kind of stress fields generated during the over-rolling of bearing dents. An important feature of the rolling contact stress calculation is the inclusion of the effect of the lubricant film into the elastic response of the surface asperities during over-rolling. This leads to a more realistic prediction of the subsurface stress field in case of variation of the lubrication condition of the rolling contact. Using the above described calculation scheme, an evaluation of the equations used by bearing standards for the estimation of the lubrication and contamination factor is carried out. The following conclusions can be drawn:

1. As shown by equations (4) and (10), the predicted effect on bearing life from contamination is load-dependent. This is different from the models available in the open literature but is well aligned with the common experience. Comparison between the present model and full-bearing experiments of figs. 5a, 5b and 5c validates this effect.

2. The basic theory of the lubrication factor η_b and the contamination factor η_c clearly shows that the quality of oil film and the cleanliness condition of the lubricant are important operating conditions of the bearing. The proper characterization of these factors is essential for a realistic prediction of the life expectancy of rolling bearings.

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