9-1-2017

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Run-out Based Crossed Roller Bearing Life Prediction by Utilization of Accelerated Testing Approach and FE Numerical Models

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Highlights

- Linear and non-linear models for wear phenomena description in bearings.
- Limited number of material characterization tests required.
- Numerical estimation of the geometry modification due to wear (DEFORM SW).
- Numerical estimation of the wear-out in the full-scale bearing system (ABAQUS SW).
- Precise bearing life-time prediction based on a chosen wear-out failure criterion.

Abstract

Surface wear and spalling phenomena are the most important factors affecting roller bearing performance and operational lifetime, in terms of number of cycles the bearing can operate. Although bearing life has been traditionally defined by the onset of spalling, as specified by international standard ISO 281, this definition does not always reflect the real operational conditions, since bearing wear can also represent a failure criterion when excessive run-out occurs. In such cases, bearing life needs to be predicted according to the amount of bearing run-out resulting from wear due to load and number of cycles. In the current study, the wear phenomenon has been modeled using the Archard wear and Lemaitre damage models, both included in a numerical simulation of a roller bearing assembly. Wear and damage models, along with the results of accelerated life experiments, have been used to develop a reliable method enabling bearing life prediction with a minimal number of material characterization tests. Accelerated life testing experiments on real bearings have been performed to confirm the reliability of the methodology, showing good correlation with numerical simulation results and proving that the developed model can be utilized for prediction of the lifetime of bearings.

Graphical abstract

Keywords

Bearing life prediction, Run-out, Abrasive wear, Archard wear model, Lemaitre damage model, Wear update

1. Introduction

Generally, roller bearing life is calculated according to ISO 281 standard, based on the idea that spalling initiation determines the failure of the bearing, thus its actual lifetime. However, wear is also an important factor that affects roller bearing performance and can influence the time the bearing can be actually used before it is replaced. Concerning the spalling effect, several contributions are available in the literature, as follows. Tallian and Mccool [\[1\]](#) studied the spalling caused by the fatigue in the lower part of the material surface, according to the

stress concentration. Laird et al. [2] introduced crack patterns according to residual and Hertz stresses generated during rolling contact. Ding and Gear [3] developed a spalling depth prediction model based on the ligament collapse of the spalling mechanism in the gears. Slack and Sadeghi [4] derived an FE explicit model for the investigation of the crack initiation and spall formation in machine elements subjected to rolling contact fatigue, considering spalling initiation and friction direction. Warhadpande et al. [5] inferred the influence of stress and strain in the spalling area from the results of fatigue tests. Weinzapfel and Sadeghi [6] verified the results of FE simulations for the analysis of the spalling effect in the material sub-surface by developing a fatigue damage model.

Bearing research has also greatly progressed for the determination of the bearing precision and for the study of the vibration effect, but only a few studies have measured quantitative data or developed real lifetime predictions. Lynagh et al. [7] studied bearing precision according to non-linear spring contact by applying a numerical model able to simulate the effect of bearing working frequency. Maru et al. [8] studied the influence of the contaminations in the lubricant oil on the wear in the bearing systems. Karacay and Akturk [9] introduced vibration spectrum analysis in order to predict bearing defect initiation time and position. Singh et al. [10] developed a bearing non-linear model for vibration prediction due to defects caused by impulsive force.

In addition to that, various studies have been also conducted in order to develop accelerated life test methods able to give satisfactory results and predictions. Mettas and Vassiliou [11], by analyzing the stress factors, studied the influence of the failure of a part in a multi-component system. Srivastava et al. validated the adequacy of the ramp stress testing method after applying different types of stress conditions in accelerated life testing [12]. Rajkumar et al. [13] developed a power law-Weibull-based model for the prediction of the life-stress relationship by utilizing accelerated wear tests. Zhang et al. [14] applied the accelerated life testing in numerical analysis in order to predict the component life considering its defects [14]. Cheon et al. developed a method to predict refrigerator door hinge life, based on allowable hinge wear, by using accelerated life testing coupled with FE analysis [15]. Hence, based on the above-mentioned recent studies [11], [12], [13], [14], [15], effectiveness and reliability of the accelerated life testing and FE simulations, aiming to predict the life of the component, have been proved.

Concerning the bearing, and the consequent estimation of their lifetime, a key aspect lies in the estimation of both friction and lubrication conditions, especially when experiments are meant to be carried out for the determination of the wear behavior in terms, for instance, of the wear model constants. In the literature, several authors spent effort in studying the friction, from analytical, experimental and numerical points of view. Among them, Halme and Andersson [16] reviewed the state of the art about the rolling contact fatigue in roll bearing and established a correlation between tribological mechanisms and bearing diagnosis, in terms of vibrations, for the estimation of the remaining lifetime of roller bearing systems. Olofsson et al. [17], [18] underlined how to, under boundary lubrication conditions, the sliding between elements can make mild wear conditions to arise, leading to a considerable change in the surface profile of the roller. To simulate this phenomenon, the Archard model has been utilized, obtaining a good agreement between simulations as experiments results. In another work of Olofsson [19], an analytical model was developed for the estimation of the geometrical change of the contact surface, due to wear, in spherical roller thrust bearings, showing that, i) by utilizing their

proposed model the fatigue life is reduced by 80% after less than 500.000 cycles; ii) the initial wear has little influence on the fatigue life; and that the highest wear is observed on the roller rather than on the housing. In addition, Andersson et al. [\[20\]](#) overviewed several different friction models to be applied under different lubrication conditions, namely sliding dry, boundary and mixed lubricated, discussing them from both the numerical and tribological point of view. In the conclusion of their work, a combined friction approach is proposed to better represent, from the numerical point of view, the various friction conditions those can arise during the operational components lifetime.

In many applications, such as machine tools, bearing wear and run-out are considered as the failure criterion, rather than spalling initiation, due to the low level of precision in the prediction of this second one. Currently, most bearing users must measure the amount of run-out in loco, in order to determine when bearing failure occurs. For instance, in the case of large roller bearings used in CNC machines or wind power plants, bearing wear must be periodically measured to ensure proper functioning and to determine whether replacement of the component is needed. However, this is a reactive approach and it seems that, so far, not predictive approaches, able to predict the bearing lifetime based on the wear, have been developed neither in literature nor as industrial standards.

In the research presented in this paper, a combined application of linear and non-linear wear models is utilized to update the geometry of the roller in the bearing system and, accordingly, estimate the increase of run-out, for increasing number of cycles. The proposed procedure has been implemented in a numerical simulation, allowing determining the amount of surface abrasion, and eventual pitting, in terms of material removal, for increasing number of cycles. The update of the surface allows accounting for the increase of clearance between roller and race, which results in an increasing displacement along radial and vertical direction and which is utilized, in this paper, as a parameter to be compared with the chosen failure criterion. The bearing is considered to have reached the end of its operation lifetime when the displacement along the radial, or vertical, direction reaches the chosen failure criterion, in terms of maximum allowed displacement. In order to determine the constants to be utilized in the linear and non-linear wear models, namely, Archard wear and Lemaitre damage models respectively, tensile test, pin-on-disc test and accelerated life tests on real bearings have been carried out. In the pin-on-disc tests, accelerated test conditions have been applied in order to reduce the burden, in terms of time, for the laboratory experiments. The material properties and model constants determined from the above-mentioned experiments have been implemented in the developed numerical model for the estimation of the radial and vertical displacement for increasing number of cycles and for different applied loads. In order to validate the proposed model, the results of the numerical simulations have been compared with those of laboratory experiments carried out on real bearings, with 50 mm, 70 mm and 100 mm outer diameter crossed roller bearings and for a 70 mm outer diameter tapered bearing. The comparison between numerical predictions saw a maximum error below 15% in the estimation of the number of cycles that the bearing can operate before it has to be substituted due to the out-of-specification displacement along one, or both, radial and vertical directions, allowing to prove the reliability of the proposed approach.

2. Wear phenomena in crossed roller bearing and utilized wear models

During the working life of bearing systems, the clearance between roller and race increases due to the rising of surface defects such as wear, spalling and pitting, [Fig. 1](#). These defects cause vibrations and imprecise rotation, ending to compromising the good functioning of the gear. The bearing run-out is usually defined as axial- and radial-direction displacement of the inner race and usually measured using dial gauges.

Fig. 1. Bearing run-out mechanism of crossed roller bearing.

Most of the focus of this paper is put on crossed roller bearing, shown in [Figs. 2a](#) and [b](#) in terms of real product and model with the partial section. In [Fig. 2c](#), the cross-section of the bearing race, Section A shall be interested by linear wear due to sliding contact with the bearing race. Section B, instead, is normally characterized by rolling contact but, under certain heavy friction conditions or due to elastic deformation of the roller, also by sliding contact. In this study, wear-based bearing surface defects are classified into two different categories: linear and non-linear. Linear wear gradually erodes the roller surface, mainly because of the friction between roller and race surfaces. Non-linear wear is evidenced by volume loss, such as when spalling and pitting occur, as a consequence of the fatigue cracks.

Fig. 2. (a) 50 mm crossed roller bearing, (b) relevant model section and (c) race-roller cross-section.

2.1. Linear wear model (Archard)

To simulate the linear wear, and in order to take into account the influence of different surface roughness on the wear, the modified Archard wear model proposed by Hwang et al. [\[21\]](#), [Eq. \(1\)](#), has been considered.

$$W = \int k(1 + C_w W^{i_w}) \frac{P^a V^b}{H^c} dt \quad (1)$$

In [Eq. \(1\)](#), W and k are wear depth and wear coefficient, respectively. P , V and H are normal pressure, sliding velocity, and hardness, respectively. a , b and c are material constants. C_w and i_w are surface roughness constants. The material properties for the linear wear model have been determined from inverse calibration of the results of the pin-on-disc tests, detailed in following [Section 4](#).

2.2. Non-linear wear model (Lemaitre)

In order to simulate the non-linear wear using the FE analysis, the coupled elastic model proposed by Lemaitre [\[22\]](#) has been utilized in order to take into account the reduction of Young's modulus as a consequence of the damage, for increasing number of cycles. The term "elastic damage" is related to micro-plasticity effects happening in the material in case of fatigue loading, as in the case of the considered roller bearing, where the pressure applied by the balls is not constant but follows a $R = 0$ cycle. The definition of the coupled equations for the elastic damage formulation is similar to those of the Lemaitre damage model, defined for the failure under plastic conditions [\[23\]](#), except for the definition of the damage state, which is defined, in this case, based on the variation of the Young's modulus of the damaged material, in comparison to the original one, as shown in [Eq. \(2\)](#). $\tilde{\sigma}$ and σ are the stress of the original and damaged material, respectively, E is Young's modulus and ϵ_e the elastic strain, respectively.

$$\epsilon_e = \frac{\tilde{\sigma}}{E} = \frac{\sigma}{E(1-D)} \quad (2)$$

The coupling between elasticity, considered as isotropic, and damage, results in a definition of the elastic strain energy as shown in [Eq. \(3\)](#), where $\bar{\sigma}_e$, $\sigma_{H,e}$, ν are the equivalent stress, the hydrostatic stress, and the Poisson ratio, respectively. Based on the estimation of the elastic strain energy, the increment of damage is calculated according to [Eq. \(4\)](#), where represents the increment of the Lemaitre damage factor, $\dot{\epsilon}_e$ the elastic strain rate whereas the remaining model constants, S_0 and b_0 , are material based constants, which are determined based on the results of the accelerated life tests, presented in following [Section 4](#).

$$Y_e = \frac{\bar{\sigma}_e^2}{2E(1-D)^2} \left[\frac{2}{3}(1 + \nu) + 3(1 - 2\nu) \left(\frac{\sigma_{H,e}}{\bar{\sigma}_e} \right)^2 \right] \quad (3)$$

$$\dot{D} = \dot{\epsilon}_e \left(\frac{Y_e}{S_0} \right)^{b_0} \quad (4)$$

Based on the variation of the stress and strain components, calculated in the numerical simulation, a coupling with the damage state is established, allowing to account for the reduction of the mechanical performance of the material as the damage increases. If a portion of the material reaches the critical damage value, D_{cr} , it is considered to have reached its

fatigue load-carrying limit and, in the case of a numerical model, it stands for the moment when the relevant elements are removed from the mesh. This last statement partially anticipates the surface update criterion utilized in the developed numerical model, for the estimation of the increasing clearance between roller and race and which is explained more in detail in [Section 6](#) of the paper.

In addition to that, an important consideration related to the non-linear wear model is related to the intrinsic probabilistic behavior of this phenomenon. Occurrence region of non-linear wear is caused according to the initial flaws and void in the material, which are not uniform in the bearing rollers. In other words, this wear phenomenon can be said to occur probabilistically.

3. Bearing life prediction procedure

In this study, bearing life is predicted by utilizing the accelerated life testing procedure, coupled with the numerical analysis. Compared with conventional method based only on experimental testing, the proposed allows reducing the time related to the mechanical test, obtaining the results in a more efficient way. The followed procedure is composed of four steps, as shown in [Fig. 3](#) and here below summarized.

Fig. 3. Bearing life prediction method procedure using FE analysis and accelerated life testing.

As first, the linear and non-linear models constants have to be determined by proper material characterization, which include simple tension test and pin-on-disc test. After, the set-up data for the accelerated life test, namely load and number of cycles, are chosen and, after the tests, the run-out vs cycles and load charts are derived. Linear and non-linear wear models are implemented in the numerical simulations, run under accelerated life testing conditions. Based on the results of both accelerated life testing in experiments and numerical analysis, the life prediction curve can be calculated. Based on the prediction curve, the lifetime of different

bearings can be predicted, in a short amount of time, based on the results of the numerical simulation.

4. Material characterization and testing

In order to determine the relevant material properties of the JIS SUJ2 (AISI 52100) material to be utilized in the wear and damage models, three different material tests have been carried out: i) Tensile test, to determine both elastic and plastic properties; ii) Pin-on-disc test experiments, performed utilizing the procedure detailed in literature [\[24\]](#), for the determination of the linear wear model constants; and iii) accelerated life tests on real bearings, in order to determine the constants for the non-linear wear model.

The simple tension tests have been carried out specimen made of JIS SUJ2 according to the ASTM E8 standard, specimen type 1. From the results of the simple tension test, Young's module has been calculated as $E = 218.000 \text{ MPa}$ whereas the Poisson ratio has been set as 0.28 according to the material data sheet. In [Fig. 4](#), both the plastic region of the true stress–true strain curve and the relevant interpolation, utilizing the Swift power law, are shown. The power law and the relevant model constants are reported in [Eq. \(5\)](#) and have been obtained by inversely calibrating the results of the tensile test with those of the numerical simulation until a maximum deviation below 5% has been obtained.

$$\sigma_y = 690(0.0015 + \varepsilon)^{0.2107} [\text{MPa}] \quad (5)$$

Fig. 4. JIS SUJ2 material Swift power law interpolation.

4.1. Pin-on-disc test

Aiming to determine the material constants to utilize in the linear wear model, pin-on-disc tests have been performed and, in this section, both testing procedure and results are shown. A pin-on-disc test machine, [Fig. 5](#), has been designed, manufactured, and utilized to conduct tests at different velocities and applying different weights on the pin.

Fig. 5. Pin-on-disc machine (a) model and (b) functioning concept.

Both pin and disc have been realized with the same material, JIS SUJ2, as in the case of the real crossed roller bearing system. As previously anticipated, in the present research work, accelerated testing conditions have been utilized and, for this reason, no lubrication has been utilized during the test. This choice has been made taking into account that, during the operational lifetime, the lubricant initially placed between rollers and race tends to disappear or to be contaminated by metal particles those detaches from both the race or the rollers, making the friction to increase. By adopting the choice of having a metal-on-metal surface contact with no lubrication, the calculation of the surface abrasion due to wear will be in favor of safety.

Pin-on-disc tests have been carried out utilizing three different loads, namely 100 kgf, 150 kgf and 200 kgf, and two different rotational velocities for the disc, namely 250 rpm and 500 rpm. Both pin and disc have been realized with the same bearing steel material JIS SUJ2. The amount of pin wear was determined through measurements made by utilizing a micrometer gauge (accuracy ± 0.002 mm). The surface of the pin, after 5 min and 15 min from the beginning of the test, for the experiment conducted at 250 rpm and with the load on the pin of 150 kgf, are shown in following [Fig. 6a](#) and [b](#) respectively.

Fig. 6. Wear on the pin in the pin-on-disc test after (a) 5 min and (b) 15 min for 150 kgf and 250 rpm experimental conditions.

By measuring the amount of wear on the pin, in terms of reduction of its weight, for increasing time, the curve shown in Fig. 7a, for 250 rpm velocity, and Fig. 7b, for 500 rpm, parametrized on different weight, have been obtained. The constants of the model Eq. (1) have been obtained by minimizing the difference between the results of the pin-on-disc test and those of relevant FEM simulations and have been derived considering the wear amount only on the pin while neglecting the one on the disc. Archard wear model constants are summarized in Table 1.

Fig. 7. Pin-on-disc results for (a) 250 rpm and (b) 500 rpm rotational velocities.

Table 1. Linear wear models constants.

Parameter	Description	Value / Measurement unit
k	Wear coefficient	$3.57 \cdot 10^{-5}$
a	Degree of freedom of parameter	1.38
b	Degree of freedom of parameter	1.48
c	Degree of freedom of parameter	2.86
C_w	Surface roughness constant	1.00
i_w	Surface roughness constant	0.15

4.2. Accelerated life experiments

In order to determine the constant to utilize in the non-linear wear model, and to carry out the experiments with which to compare the results of the developed numerical model, a bearing wear test stand has been designed and manufactured, [Fig. 8a](#) and [b](#), where also the detail of the coupling between test machine and bearing is shown. In the test machine, the load is supplied by a hydraulic cylinder and the force is measured by three piezoelectric load cells meanwhile the bearing is put into rotation by an electric motor, controlled by Sigma Win software. Since the constants, to be utilized in the non-linear wear model, are material properties, they have been calculated considering the study cases of the 50 mm outer diameter crossed roller bearing whereas the remaining experiment, 75 mm and 100 mm outer diameter crossed roller bearing and 35 mm outer diameter tapered roller bearing, have been utilized for the validation of the proposed numerical model. The comparison between numerical and experimental results is presented in following [Section 6](#). As previously anticipated in [Section 2](#), in the present research work, the run-out is measured as displacement, along vertical and radial directions thus, for increasing number of cycles. For this reason, two dial gauges have been placed along these two directions in order to record the amount of wear per rotation of the table, as shown in [Fig. 8c](#).

Fig. 8. (a) Bearing wear test machine, (b) stand dimensions and (c) measurement system schematic diagram.

Concerning the processing conditions for the 50 mm crossed roller bearing, experiments have been carried out by applying the following forces on the bearings: 1020 kgf, 893 kgf, 765 kgf and 638 kgf, those correspond to 200%, 175%, 150% and 125% of the dynamic load applied to the bearing. In order to avoid misinterpretations of the results, in each experiment, a new bearing has been utilized. Moreover, in all the four cases, the rotational velocity of the bearing has been set to 400 rpm. The results, in terms of run-out vs cycles for increasing number of cycles, along with both radial and vertical directions, are shown in following [Fig. 9a](#) and b. In the case of 638 kgf, the bearing system does not any relevant increase in the run-out, along with neither the radial nor the vertical directions, until 900,000 cycles. However, starting from that amount of cycles, the run-out starts to increase along both directions, following a steep path.

Fig. 9. Run-out along (a) radial and (b) vertical direction for the 50 mm crossed roller bearing.

In the considered crossed roller bearing, during their manufacturing process, a layer of NLGI grade 2 grease is placed in the race, in order to reduce the friction, thus the wear, of both rollers and race. However, the effect of the grease tends to disappear during the operation lifetime of the bearing, resulting in the abrupt increase of the run-out after a certain number of cycles. The higher the load applied on the bearing the lower the number of cycles at which the run-out starts to abruptly increase, as testified by the results shown in [Fig. 9a](#) and b.

Based on the results of the accelerated life testing conducted on the 50 mm outer diameter crossed roller bearing, and considering only the part of the run-out vs cycles curve where the run-out abruptly increases, the constants for the non-linear wear model have been determined by inverse calibration of the results of the experiments with those of the numerical model. The quality of the inverse calibration is testified by the comparison between experimental and numerical results shown in [Fig. 9a](#) and b whereas the model constants, relevant for the elastic damage model, are reported in following [Table 2](#).

Table 2. Non-linear wear model constants.

S_0	Damage resistance parameter	5.06 MPa
b_0	True plastic strain at UTS	0.57
D_{cr}	Critical damage	0.68

The material properties and model constants determined in this paragraph of the paper have been utilized in the numerical models presented in the following section of the paper. In addition to that, the procedure detailed in [Section 4.2](#) concerning the accelerated life-testing test has been adopted also for the experiments utilized for the validation of the developed numerical models, whose results are presented in following [Section 6](#).

5. Numerical model implementation

The numerical simulations have been run utilizing two different commercial software, DEFORM and ABAQUS, and a self-developed program used to update the geometry of the roller after each predetermined number of cycles intervals. The sequential implementation of the numerical models is shown in following [Fig. 10](#). By utilizing the numerical simulation implemented in DEFORM, the amount of wear for each cycle interval is estimated based on linear and non-linear wear models, where the model constants determined in [Section 4](#) have been used. Afterward, utilizing a self-developed routine, the geometry of the roller is updated after each cycle interval. The number of cycles, as well as the size of the cycle interval, can be chosen by the user. The geometry information, output of the “wear update” block in [Fig. 10](#), are stored for each cycle interval and are afterward inputted in the ABAQUS simulation for the estimation of the clearance in radial and vertical direction.

Fig. 10. Numerical simulation flow chart.

Concerning the implementation of the wear models in the numerical simulation in DEFORM, additional details about their influence on the mesh update are here given. By utilizing the linear model, the surface is progressively updated in order to take into account the wear. This implies that the position of the nodes in the mesh is updated based on the wear caused by the relative movement between roller and race, as shown in [Fig. 11a](#). However, the utilization of the non-linear wear model implies that, when an element reaches the critical damage, it is removed from the mesh and the element under it starts to accumulate damage, as shown in [Fig. 11b](#). By applying both linear and non-linear wear model and then linearly combining the results, it is possible to precisely update the mesh in the numerical simulation, hence to estimate the variation in the geometry of the roller caused by the wear. Afterward, based on

the updated geometries and by utilizing the ABAQUS numerical model, the increasing clearance can be measured and, based on a chosen failure criterion, the number of cycles the bearing is able to operate can be estimated. The results of the life predictions are shown in [Section 7](#).

Fig. 11. Schematic diagram for the geometry-updating-algorithm (a) linear wear (b) non-linear wear.

In the DEFORM simulations, the outer and the inner race of the numerical model have been meshed using tetrahedral elements with 1.25 mm element side size whereas the rollers have been meshed with the same tetrahedral elements, but with a finer mesh of 0.6 mm side size. For all the simulations, an implicit sparse solver has been used, with a convergence error limit of 0.01 for the velocity and of 0.1 for the force. Three different numerical simulations for crossed roller bearing with an inner diameter of 50 mm, 70 mm and 100 mm and one for a 35 mm tapered roller bearing have been set-up and run. An example of the numerical implementation in DEFORM is given in [Fig. 12](#).

Fig. 12. Full cross roller bearing numerical model in DEFORM.

6. Results and discussion

Following the rationale explained in the end of the previous paragraph, an example of the DEFORM numerical simulation output is shown in following [Fig. 13](#), where the different influence on the surface of the roller after the application of the linear wear model, [Fig. 13a](#), and of the non-linear wear model, [Fig. 13b](#), are presented.

Fig. 13. Simulated distributions of (a) wear depth and (b) surface damage in the DEFORM FEM model.

Once the wear contributions have been calculated, they are fed to the “wear update” block, [Fig. 10](#), which operates the average of the two estimated geometries. For increasing number of cycles, the surface of the roller will experience a progressive modification due to wear, as testified by the SEM microscope pictures, [Fig. 14a](#) and b, taken to a new roller and to one after 400.000 cycles. Concerning the developed numerical model, the update of the mesh due to wear, for increasing number of cycles, is shown in [Fig. 14c](#), as results of the “wear update” block.

Fig. 14. SEM picture of a (a) new roller and (b) after 400.000 cycles; (c) numerical roller surface update due to linear and non-linear wear models application.

After the geometry update, the roller model is imported in the ABAQUS simulation and positioned in the race groove, as shown in [Fig. 15](#).

Fig. 15. ABAQUS numerical model for the crossed roller bearing (inner race and roller).

The full model of the ABAQUS simulation, at the first time step, is shown in [Fig. 16a](#). In [Fig. 16a](#) the grey cylindrical element is coupled with the bearing model and a rotational boundary condition, around the y-axis is applied on it, as in the real application. As a consequence of the wear, the roll will move inside the race, causing an increasing run-out, both in the radial and vertical directions, those will increase as the damage increases, hence for increasing number of cycles.

Fig. 16. (a) Initial and (b) final step of the run-out simulation in ABAQUS.

By applying the described numerical procedure for all the cycles intervals, it is possible to measure the evolution of the displacements along radial and vertical direction and, based on a chosen failure criterion, to understand when one or both displacements exceeds the chosen limit. According to the chosen limit, and based on the results of the ABAQUS simulation, the lifetime of the bearing can be predicted.

The developed combination of numerical models have been applied to three different crossed roller bearings with outer the diameter of 50 mm, 70 mm and 100 mm and to a tapered bearing with the outer diameter of 35 mm. The results of the numerical simulations have been compared with those of accelerated life testing experiments carried out by utilizing the procedure described in previous [Section 4.2](#). For all the four bearings, different load cases have been taken into account and tested both from the experimental and numerical point of view, allowing determining the bearing life prediction curves. For each bearing, based on a chosen failure criterion, the number of cycles the bearing can operate before it has to be substituted can be calculated based on the relevant life prediction curve. In all the considered cases, the rotational speed of the bearing, both in the simulation and in the experiment, has been set to 400 rpm.

Concerning the 50 mm crossed roller bearing, the results have been already presented in previous [Fig. 9a](#) and [b](#) so are no repeated in this section. For this case, the maximum error, considered as the deviation of the numerical simulation in comparison to the relevant experiment, in terms of radial and vertical displacement, has been measured in 8.08%. As expectable, the higher the load the faster the run-out occurs and this phenomenon is properly simulated by the numerical model. Moreover, the axial run-out has shown to be more severe than the radial one. Based on the results shown in [Fig. 9a](#) and [b](#), if the failure criterion for the

considered roller bearing is set to 2.5 μm both in radial and vertical directions, the life prediction curve can be derived, as shown in [Fig. 17](#).

Fig. 17. Bearing life prediction curve for 50 mm crossed roller bearing.

The life prediction curve shown in [Fig. 17](#) was derived from the results of the numerical model for the chosen amount of radial and axial maximal allowable clearance (failure criterion). The comparison between the results of the proposed numerical model and the relevant experiment have shown to match in predicting the lifetime of the considered bearing in approximately 2530,000 rotations but, if the same estimation is made according to ISO 281, for a load of 420 kgf, the lifetime is estimated in 191 million cycles.

In order to study also the influence of the size effect on the accuracy of the prediction of the run-out, different crossed rollers with the inner diameter of 70 mm and 100 mm have been analyzed both with experiments and with the developed numerical model. In addition, also a tapered roller bearing with the inner diameter of 35 mm has been tested. All the analyzed bearings are realized with the same JIS SUJ2 steel and, being both material and heat treatment same, the material properties derived in [Section 4](#) have been utilized also in the numerical simulations of these additional three cases. For the crossed roller bearings (70 mm and 100 mm), the applied load have been set to 150%, 125% and 100% of the real dynamic load and the inputted data are summarized in [Table 3](#). The comparison between numerical estimation and relevant experiment is shown in [Figs. 18](#) and [19](#) for the 70 mm crossed roller bearing and in [Figs. 20](#) and [21](#) for the 100 mm crossed roller bearing. [Figs. 18](#) and [20](#) show the results for the radial run-out whereas [Figs. 19](#) and [21](#) those for the vertical run-out, respectively. In these two analyzed cases, the maximum error between experimental and simulated results have been calculated in 14.22% for the axial run-out and in 11.05% for radial run-out, showing that, if the wear models are properly formulated the maximum error is below 15%.

Table 3. Accelerated life test conditions for 70 mm and 100 mm crossed roller bearings.

Case No.	70 mm diameter crossed roller bearing			100 mm diameter crossed roller bearings		
	1	2	3	1	2	3
Load [kgf]	897	747	598	1073	894	715

Fig. 18. Experiment and simulation run-out results of 70 mm crossed roller bearing (radial run-out).

Fig. 19. Experiment and simulation run-out results of 70 mm crossed roller bearing (vertical run-out).

Fig. 20. Experiment and simulation run-out results of 100 mm crossed roller bearing (radial run-out).

Fig. 21. Experiment and simulation run-out results of 100 mm crossed roller bearing (vertical run-out).

As previously derived for the 50 mm crossed roller bearing, also for the case of the 70 mm and 100 mm crossed roller bearing, if the same failure criterion of 2.5 μm for both in radial and vertical directions is set, the life predictions curves can be calculated and are shown in [Figs. 22](#) and [23](#), respectively. Based on the prediction and considering a load of 550 kgf, the 70 mm crossed roller bearing will operate approximately 2050,000 cycles whereas the 100 mm crossed roller bearing 1320,000, respectively.

Fig. 22. Bearing life prediction curve for 70 mm crossed roller bearing.

Fig. 23. Bearing life prediction curve for 100 mm crossed roller bearing.

Considering now the last case, the 35 mm tapered roller bearing, the accelerated testing conditions are reported in following [Table 4](#) and are related to 150%, 125% and 100% of the real dynamic load. The results of the radial and vertical run-out are shown in [Fig. 24](#) and [Fig. 25](#), respectively. The maximum error between numerical simulation and relevant experiment has been calculated in 12.37% so, also for the analyzed 35 mm tapered roller bearing, the run-out can be predicted with an error lower than 15%, showing the reliability of the proposed approach for the prediction of the lifetime of bearings. Finally, by considering a 2,200 kgf and a failure criterion of 2.5 μm both for radial and vertical directions, the estimated lifetime is of 2470,000 rotations ([Fig. 26](#)) whereas, if the ISO 281 for spalling failure criterion is used, the life is estimated in about 221 million cycles.

Table 4. Accelerated life test conditions.

Fig. 24. Experiment and simulation run-out results of 35 mm tapered roller bearing (radial run-out).

Fig. 25. Experiment and simulation run-out results of 35 mm tapered roller bearing (vertical run-out).

Fig. 26. Bearing life prediction curve for 35 mm tapered roller bearing.

7. Conclusions

In this study, by using accelerated life testing and finite element analysis, a quantitative life prediction method, based on run-out, was developed for crossed and tapered roller bearings. In particular, the difference between the results of experiments and simulation has always shown to be lower than 15%, showing that the application of the two wear models is reliable in the estimation of the run-out, hence of the bearing lifetime. In order to properly predict the run-out, wear models constants must be accurately derived from the material characterization and, for this purpose, standard material characterization, such as the tensile test, as well as pin-on-disc and accelerated life testing should be carried out. The proposed approach well fits the nowadays industry demand for quick and precise estimation of the real lifetime for bearing, especially in the case of large bearings utilization, such as CNC machines and wind power plants. Moreover, by utilizing the proposed approach, the run-out and the relevant bearing lifetime can be predicted according to the wear, and not only to the spalling effect, allowing a more realistic estimation of the actual lifetime of the bearings.

Acknowledgement

This work was supported by the National Research Foundation of Korea (NRF) grant funded by the Korean Government ([NRF-2015R1D1A1A09059763](#)) and the Research Grant of Sogang University No. [201210031](#). The support is gratefully acknowledged.

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