academicJournals

Vol. 10(17), pp. 549-558, 15 September, 2015
DOI: 10.5897/SRE2015.6264
Article Number: 5C00D7A46891
ISSN 1992-2248

http://www.academicjournals.org/SRE

Scientific Research and Essays

Full Length Research Paper

Identification of the damage model parameters of thrust ball bearings under variable operating regime

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Received 24 May, 2015; Accepted 31 August, 2015

Bearing is an important component of a rotating machine; however, under normal operating conditions, it is subjected to fatigue which results in a defect called spalling. This work presents a monitoring fatigue of a thrust ball bearing on a bearing fatigue test bench. Spalling is artificially initiated on a bearing raceway. Vibration analysis is the method used to characterize the defect. The experimental procedure used is to monitor the operation of the ball bearing to the degradation with an online acquisition of the vibration readings. These data allow plotting the curve of fatigue (mathematical model). In order to obtain more realistic curve, the spalling evolution of several thrust ball bearings is monitored, first, under different loads then at different rotation speeds under a constant load. It found that growth follows a spalling power law and the damage is characterized by the model parameters.

Key words: Predictive maintenance, bearing, vibration analysis, spalling, rolling fatigue.

INTRODUCTION

The increased mechanization and automation of the rotating machines have made that the diagnosis is computerized and prognosis systems are a valuable tool in decision taking by maintenance personnel to intervene timely on equipment. Today, the concept of machinery diagnosis includes automatic detection and classification of defects, while the prognosis is a concept of predictive maintenance based mainly on estimating the likely time remaining before the failure occurs on a machine. Predicting failures promises to significantly reduce maintenance costs such as machine stops, consumption of spare parts, etc. However, it still remains an axis of current research and is a major concern of researchers in the maintenance field. Among approaches about predictive maintenance reported in the literature, it distinguished Pusey and Roemer (1999) who gave an overview of the development of diagnosis and prognosis for high technological performance applicable on turbomachinery until 1999. Jardine et al. (2006) gave an overview and publications on acquisition and processing of data, diagnosis and prognosis of various machines until 2005. Basile (2007) developed a statistical approach to establish a law for equipment reliability; this approach is based on the feedback (time base failures included). Vachtsevanos et al. (2006) identified and described fault diagnosis by artificial intelligence methods and approaches

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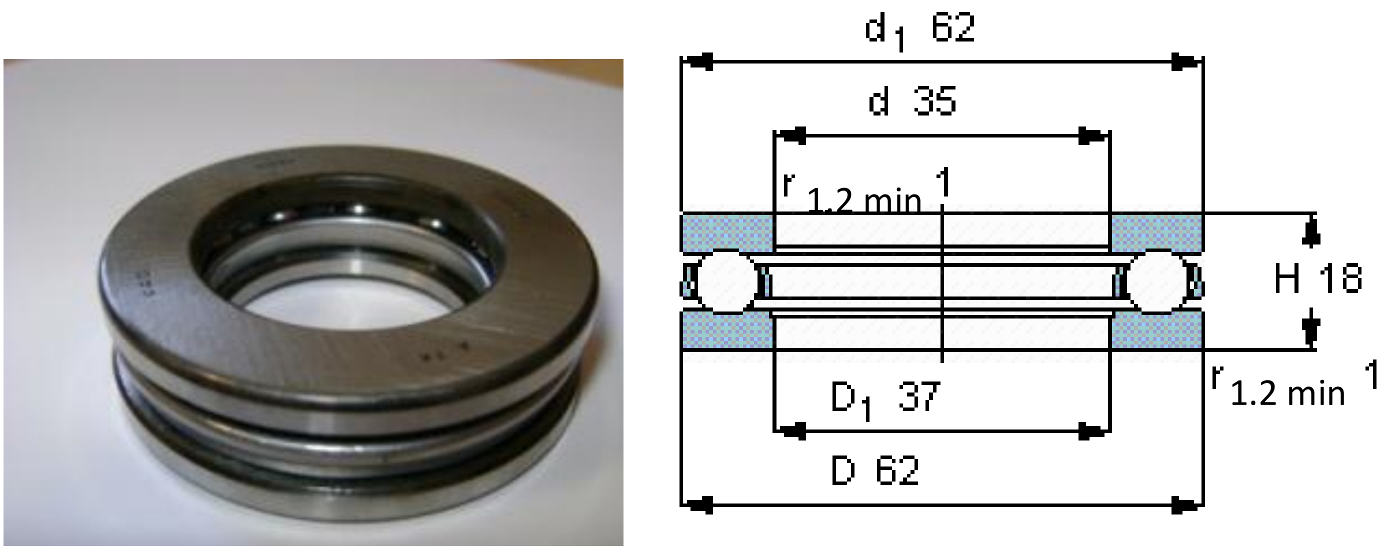


Figure 1. Thrust ball bearing.

to the prediction of failures in engineering systems through examples.

This research consists, firstly to show whether the model of thrust ball bearing fatigue obtained during our previous work is still valid in case operating conditions of the machine change, then to confirm the identification and the significance of the degradation model parameters (Djebili, 2013; Djebili et al., 2013). The occurrence of spalling at a point of contact between the rolling elements and bearing raceway is a random phenomenon whose probability of occurrence is related to the combination:

(i) Between the stress level and the probability of finding a more or less harmful inclusion for chipping, initiated into sub-layers;
(ii) Between the stress level and the probability of finding a surface defect for chipping insiders' surface.

To save time and to control the position of occurrence of spalling, an artificial defect is created using a graphite tool having a piece of rounded shape with a diameter 2 mm at a fixed point of the ball trajectory on the bearing ring. This artificial indentation is formed on an electrical discharge machine and used to initiate a spalling fault by creating a stress concentration. Monitoring the evolution of spalling is then performed in change of machine operating conditions (test bench of bearing fatigue) by successively varying the following parameters:

(i) Axial load applied on the thrust ball bearing,
(ii) Rotation speed of the thrust ball bearing axis.

Then, it will be possible, from the knowledge of loading conditions (corresponding pressure) and operating conditions (rotating speed) to predict the evolution of a thrust ball bearing spalling using the fatigue model with more reliability.

VIBRATION MONITORING OF A THRUST BALL BEARING SUBJECTED TO DIFFERENT LOADS

Description of the thrust bearing

Bearings (Figure 1) mounted on the module are thrust bearings whose characteristics are: FAG 51207 CZECH / ATK, inner diameter di_1= 35 mm, di_2 = 37 mm, outer diameter D = 62 mm, number of balls N_b = 12.

Test procedure

Before initiating a defect on the mobile ring of the thrust ball bearing, it is required to turn the bearing on the fatigue test bench (Figure 2) during an operating lifetime defined by the L_{10} probability (Alfredson and Mathew, 1985). Concerning tests, it used different axial loads on the bearings and L_{10} lifetimes (ISO 281, 2007) are shown in Table 1.

After each L_{10} lifetime to a given axial load, it proceeds to removal of the thrust ball bearing and to initiate a defect on one of the rings (Figures 3 and 4). Then it put back the bearing on the test bench to rotate during the spalling phase. The tests are performed on a test bench on which it sets value of the axial load to transmit at the thrust ball bearing, constant rotation speed of 1800 RPM and flow of cooling liquid. Two piezoelectric accelerometers type DJB3208 and DJB3209 are placed near the bearing in two different directions (axial and radial) to acquire the vibration signals.

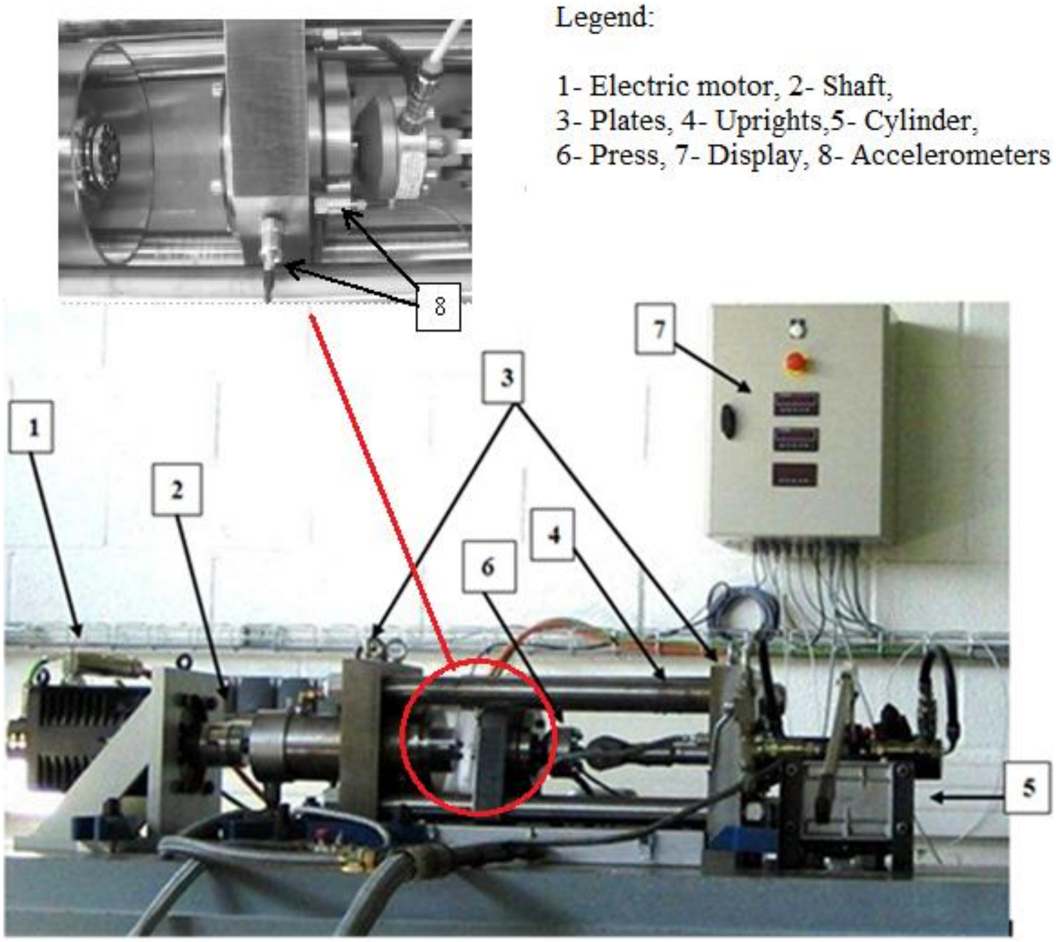


Figure 2. Test bench of the thrust ball bearing fatigue.

Table 1. Bearing lifetimes.

Axial load(N)	Rotation speed (RPM)	Life time L_{10} (h)
30000	1800	$L_{10(1)}$=25
35000	1800	$L_{10(2)}$=13
40000	1800	$L_{10(3)}$=9

These are connected to the installation in order to indicate the vibration level instantly.

This test bench consists of three main parts: (i) base which is consists of two plates (3) linked with four uprights (4); (ii) the training part, including the main shaft (2) which goes through a plate to allow positioning of the thrust bearing to be tested on one of its end and the electric motor (1) which transmits the rotation movement to the shaft by means of a coupling; (iii) the load device support, including the cylinder (5) which loads the press (6), the intensity of the load is indicated by a display (7). The defect size is measured periodically to readjust the model (Figure 5). These measurements are performed using a microscope with a camera and software that quantifies the spalling size.

Evolution of the defect size

Monitoring of the spalling evolution is realized with several thrust ball bearings on the test bench fatigue subjected to variable loads and a constant rotation speed. Figure 6 shows the evolution of the obtained spalling curves.

It is found that spalling growth follows the power law of type $f(x) = a.x^b$ (Richalet, 1998) and depends on the load.

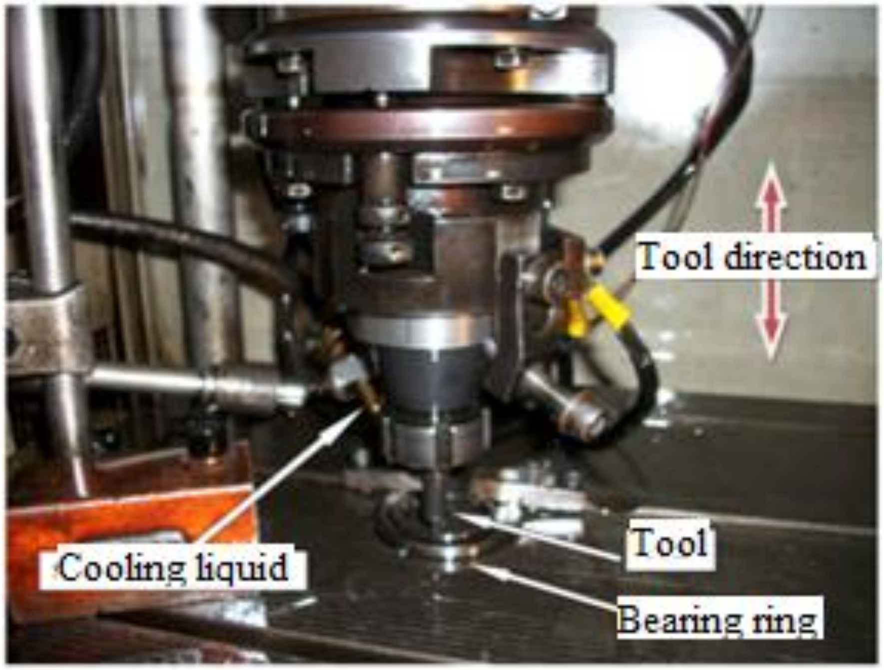


Figure 3. Booting the defect on the bearing ring using the sinking EDM machine.

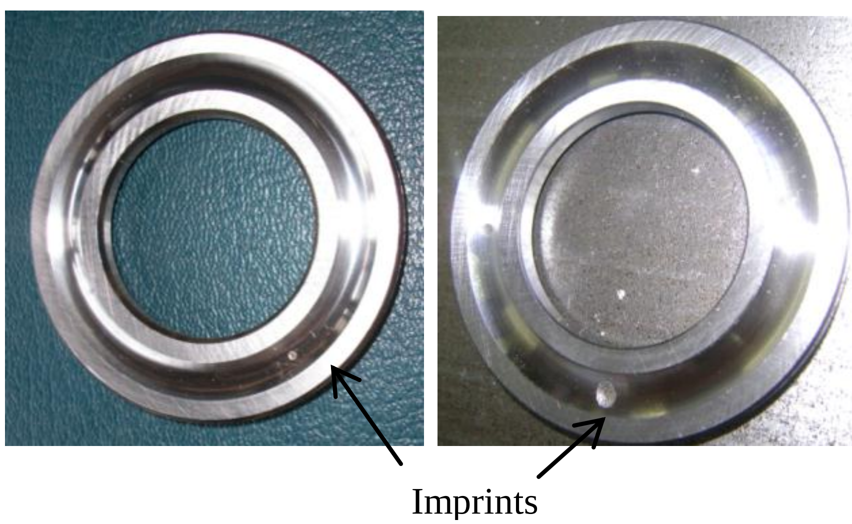


Figure 4. Spalling defect on a thrust ball bearing.

Calculation of Hertz pressure

It uses the formula concerning the Hertz contact between two solids to calculate the Hertz pressure generated between the ball and the ring of the thrust ball bearing.

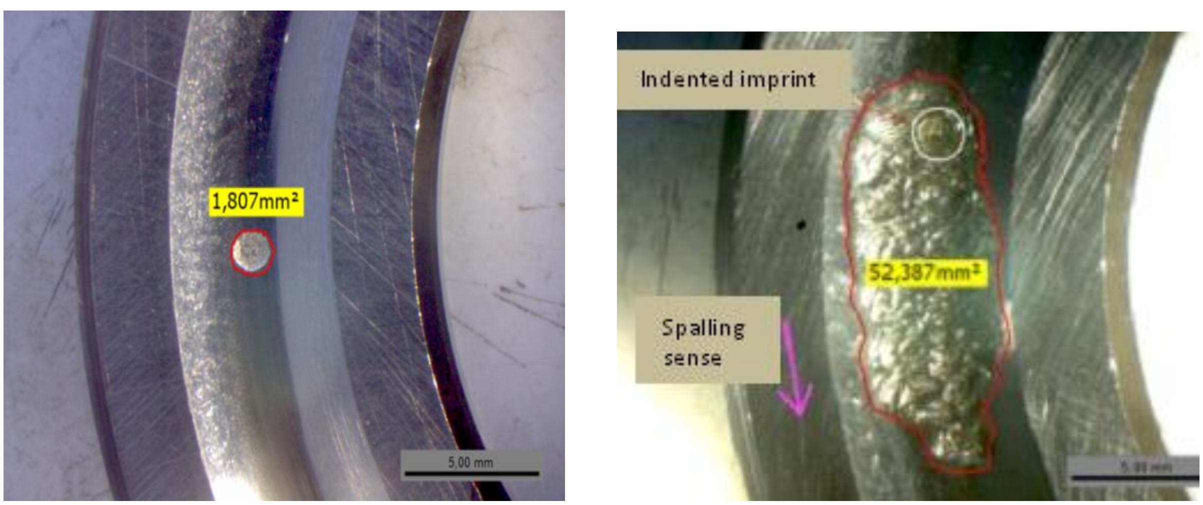
$$P_{max} = \frac{3N}{2\pi kl} \quad (1)$$

The materials of the ball and the ring are identical. The contact surface generated is elliptical shape with large semi -axis k and small semi-axis l. N = Q/12: axial load applied to each bearing ball. For full load Q = 30000 N, the maximum pressure is: P = 3.3 GPa (Table 2).

Experimental results

The power law coefficient 'a' grows with increasing load and the spalling time decreases with increasing load

Figure 5. Measurement of the spalling fault size.



(a) (b)

Figure 6. Evolution of spalling on a thrust ball bearing ring, (a): Initiated fault, (b): Spalling fault.

during the spalling phase (until fault area threshold 48 to 52 mm^2) (Figure 7).

Evolution of the vibration indicator

We proceed in the same way to monitor the vibration indicator over time. Monitoring of the spalling evolution is realized on several thrust ball bearings under continuous operation and subjected to varying loads. Measurements are taken on the test bench in a wide range of acquisition frequency (0 - 20 kHz). It is to define the band of frequencies (f1, f2) in which a vibration energy is maximum (Li et al., 2000). This energy corresponds to

Table 2. Cycles number of spalling phase.

Axial load (N)	Hertz pressure (Gpa)	Spalling cycles number
30000	3.3	N_1=468000
35000	3.48	N_2=288000
40000	3.61	N_3=180000

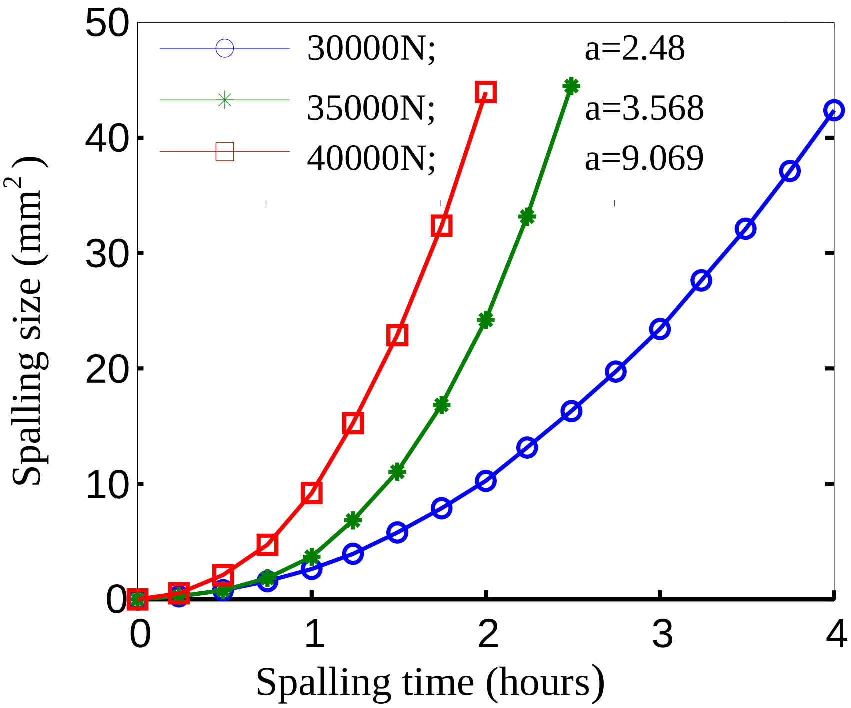


Figure 7. Evolution curves of spalling size at different loadings.

the structure resonances generated by a bearing fault. These resonances are typically between 10000Hz and 15000Hz in case of rotating machines (Djebili et al., 2013). Figure 8 shows spalling evolution curves of three thrust bearings at a constant speed. It finds that spalling growth follows the power law of type f (x) = $a.x^b$ and depends on the load. The spalling of bearings grows with different cycle numbers N1, N2 and N3 according to the applied pressure. These results are presented in Table 3.

In Tables 2 and 3, the cycle number define for both cases, the duration of the spalling phase just shows increase with decreasing load. The evolution of the thrust ball bearing spalling follows a power law model, therefore confirms the results of previous tests and the coefficient ‘a’ of the power law increases with increasing of the load according to Table 4.

Discussion of results

According to the obtained results from the fatigue tests on thrust ball bearings, the statistical indicator of spalling fault evolves with applied axial load on the bearings. Figure 9 shows three curves representing the variation of the RMS indicator values according to the pressure on the bearing. Through these curves, the RMS values of the statistical indicator grow rapidly with increasing of the axial load on the thrust ball bearing and the points of curves diverge increasingly further. In fact, the differences between the points of the curves relating to load 40000 N are greater than the differences between the points of the curves relating to load 35000 N and further larger than those between the points of the curves relating to load 30000 N. It can conclude and confirm that more load is important during the thrust ball bearing operation, the spalling increase is relatively fast.

ROTATION SPEED VARIATION OF THE THRUST BALL BEARING

It acts on the variation of a second parameter of the

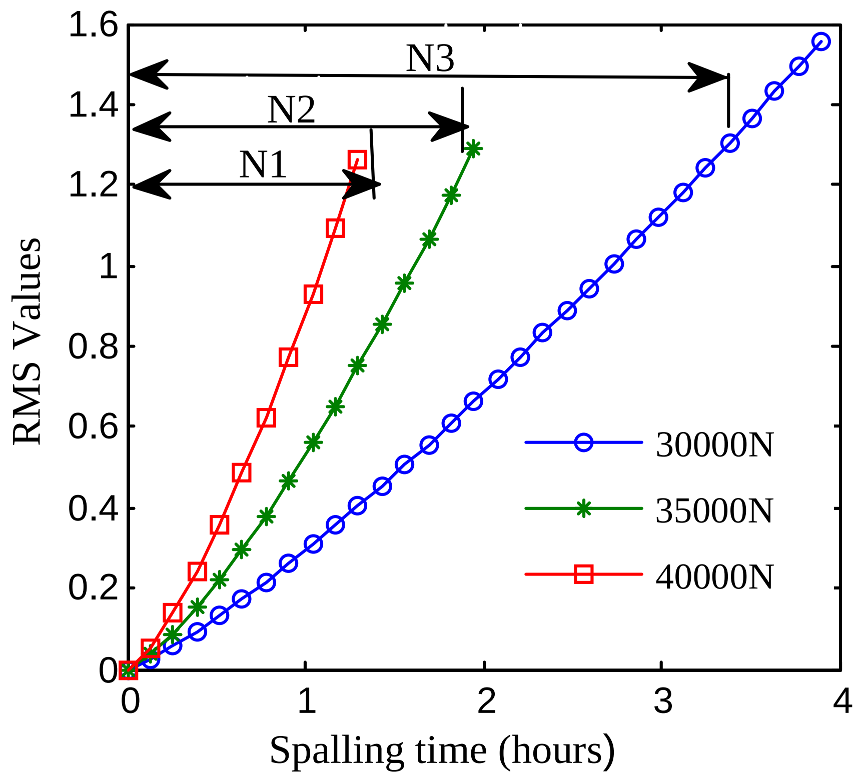


Figure 8. Evolution curves of the root mean square (RMS) indicator (Hoeprich, 1992) at different loads.

Table 3. Cycle numbers of spalling phase.

Axial load (N)	Hertz pressure (Gpa)	Spalling cycles number
30000	3.3	N1=369000
35000	3.48	N2=225000
40000	3.61	N3=108000

Table 4. Values of the power law coefficients.

Axial load (N)	Coefficient (a)	Coefficient (b)
30000	0.292	1.23
35000	0.53	1.33
40000	0.885	1.38

operating conditions of the machine which is the rotation speed of the thrust ball bearing to see its effect on the fatigue model. For this purpose, other fatigue tests are made again on thrust ball bearings. They consist of:

(i) Initiating the fault on one of the bearing rings,
(ii) Applying the same constant load on bearings,
(iii) Varying the rotation speed of bearings,
(iv) Operating the bearings continuously on the test bench until degradation.

Damage curves evolution

Monitoring the evolution of damage is then performed under a constant load of 30000 N. T1, T2 and T3

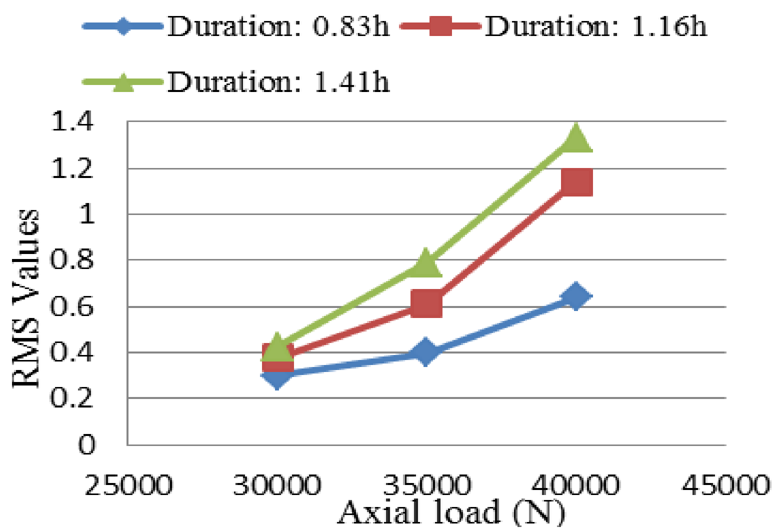


Figure 9. Vibration amplitude according to the axial load during the spalling phase.

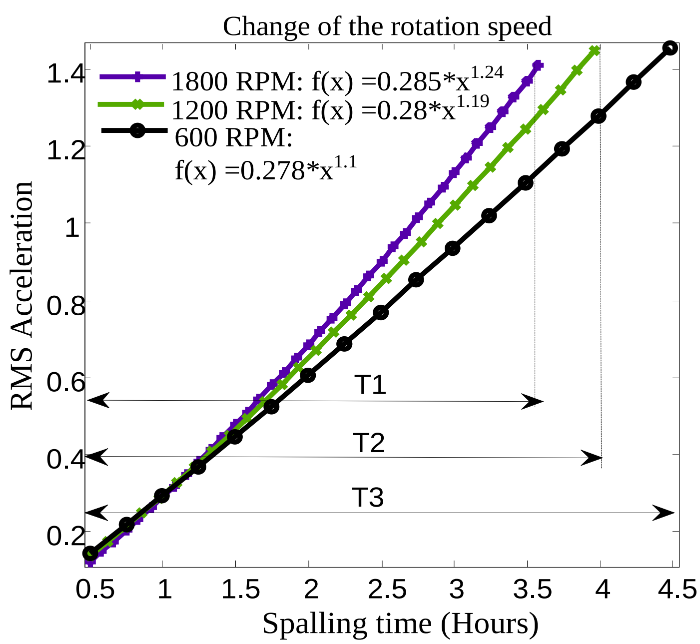


Figure 10. Comparison of evolution curves according to different rotation speed.

(Figure 10) are spalling phase's durations of thrust ball bearings respectively to rotation speed 1800 RPM, 1200 RPM and 600 RPM.

Discussion of results

Through the damage curves evolution obtained in Figure 10, it is noted that the coefficient value b of the power law on Table 5 increases with increasing of the rotation speed of the thrust ball bearing.

The damage curves analysis concluded that more rotation speed is high, the spalling phase duration is reduced and the coefficient value b is large. These results show that the thrust ball bearing damage is characterized by the coefficient b. Three curves are plotted in Figure 10 to represent the evolution of the RMS indicator values with respect to the rotation speed.

Regarding curves on Figure 11, the RMS values of the statistical indicator grow with rotation speed increase of

Table 5. Coefficients values of the power law.

Rotation Speed (RPM)	Coefficient (a)	Coefficient (b)
600	0.278	1.1
1200	0.28	1.19
1800	0.285	1.24

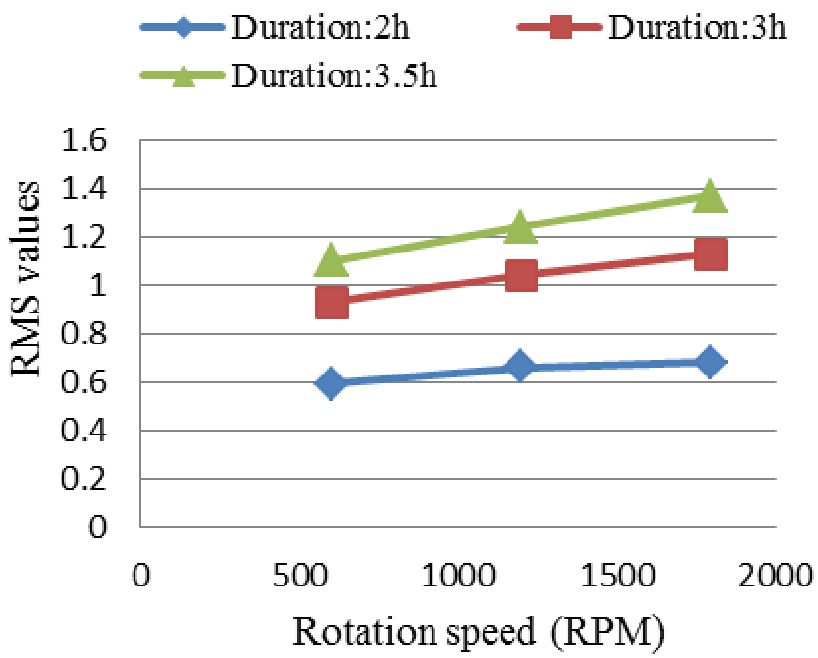


Figure 11. Vibration amplitude according to the rotation speed during a spalling phase.

the thrust ball bearing and the points of curves diverge increasingly from the small to the large speed but not with the same importance as those of curves representing the indicator evolution according to the load.

It can also confirm that more rotation speed is important, the increased spalling is relatively significant. In fact, the differences between the curves points relating to load 40000 N are greater than the differences between the curves points relating to load 35000 N and further larger than those between the curves points relating to load 30000 N. Fatigue tests on the thrust ball bearings subjected to a change in the operating conditions of the test bench have shown that the increased spalling under increasing axial load is more important than increased spalling under increasing rotation speed.

CHARACTERIZATION OF THE DAMAGE LAW PARAMETERS

The experimental results have identified the damage law parameters concerning a spalling phase. This law is a power function (Li et al., 1999):

$$D = aN^b \tag{2}$$

Where D: damage at time t_i to the pressure P and the rotation speed n, N: operating time or number of cycles at time t_i, a: empirical parameter depending on the load, b: empirical parameter depending on the rotation speed of the thrust ball bearing.

Therefore, monitoring the evolution in time of a vibration indicator allows establishing a trend curve (Williams et al., 2001; Li et al., 2000). From the above results, a and b parameters (coefficients) values are determined by the least squares method applied to each vibration reading using a Matlab program (Richalet, 1998; Adrian and Moshe, Version 6 and 7). The results of various tests have also shown that law coefficients of the spalling growth vary with change of the applied axial load and the rotation speed of the thrust ball bearing. The type of power damage model seems to characterize the thrust ball bearings degradation. This spalling growth model is a phenomenological model obtained by a physical approach with experimental identified parameters using vibration analysis method (Alfredson and Mathew, 1985; Alfredson and Mathew, 1985; Djebili et al., 2013). Therefore the spalling growth is subjected by determination of **a** and **b** coefficients which characterize respectively the Hertz pressure and damage (Djebili, 2013).

CONCLUSION AND PERSPECTIVES

This research is focused on vibratory monitoring of the spalling initiated on thrust ball bearings with operating conditions change. The vibration analysis method is used to determine the growth model parameters of the spalling phase using vibration measurements and identify those parameters through the obtained results. Fatigue tests have shown that coefficients of the bearing fatigue model grow with increasing load and rotation speed values concerning the test bench operating regime. Through these results, change in parameters related to the operating conditions of the machine showed clearly the interpretation of the coefficients **a** and **b** of the fatigue model. They respectively characterize the Hertz pressure and damage. As the final objective of the work is to determine the remaining life defined as the fatigue cycles number to achieve a spalling area (in the case of thrust ball bearings tested ≈ 50 mm^2), so it is possible to estimate from modeling the spalling growth. Use of the damage model whose parameters are determined for the actual operating conditions data may be more interesting to predict failure. Therefore, from loading and operation conditions of thrust ball bearings, it is possible to further estimate the spalling growth law and predicts the evolution of damage and duration corresponding remaining lifetime knowing the threshold fault (must not be exceeded).

The work done until now still needs more efforts to improve the growth model of thrust ball bearing fatigue. The improved model is to consider other parameters that can affect the bearing fatigue during running time such as temperature. Our wish is to build abacuses for fatigue model to then be used for:

(i) All operating conditions of the machine,
(ii) State of the environment (local, outside, etc.) of the machine.

The exploitation of these abacuses will be more convenient by maintenance users to ensure quality and optimal maintenance.

Conflict of Interest

The authors have not declared any conflict of interest.

REFERENCES

Adrian B, Moshe B (Version 6 and 7). Matlab pour l'ingénieur.

Alfredson RJ, Mathew J (1985). Frequency domain methods for monitoring the condition of rolling element bearings. Mech. Eng. Trans,--Inst. Eng. Aust. 2:102-107.

Alfredson RJ, Mathew J (1985). Time domain methods for monitoring the condition of rolling element bearings. Mech. Eng. Trans,--Inst. Eng. Aust. 2:108-112.

Basile O (2007). Prise en compte de l'incertitude dans les modèles fiabilistes industriels, Extensions aux sollicitations variables. Doctoral thesis, Faculté Polytechnique de Mons.

Djebili O (2013). Contribution à la maintenance prédictive par analyse vibratoire des composants mécaniques tournants, application aux butées à billes soumises à la fatigue de contact de roulement. Doctoral thesis.

Djebili O, Bolaers F, Laggoun A, Dron JP (2013). Follow up of the growth of a rolling fatigue spalling within a predictive maintenance. J. Mech. Ind. 14:85-93.

Djebili O, Laggoun A, Bolaers F, Dron JP (2013). Methodological approach of selecting a vibration indicator in monitoring bearings. Int. J. Phys. Sci. 8 (12):451-458.

Hoeprich MR (1992). Rolling Element Bearing Fatigue Damage Propagation. J. Tribol. 114(2):328-333.

Jardine AK, Lin D, Banjevic D (2006). A review on machinery diagnostics and prognostics implementing condition-based maintenance. J. Mech. Syst. Signal Process. 20:1483-1510.

Li Y, Billington S, Zhang C, Kurfess T, Danyluk S, Liang S (1999). « Adaptive prognostics for rolling element bearing condition ». J. Mech. Syst. Signal Process. 13(1):103-113.

Li Y, Kurfess TR, Liang SY (2000). Stochastic prognostics for rolling elements bearings. J. Mech. Syst. Signal Process. 14(5):747-762.

Norme ISO 281 (2007). Roulements - Charges dynamiques de base et durée nominale.

Pusey HC, Roemer MJ (1999). An assessment of turbomachinery condition monitoring and failure prognosis technology. The Shock and Vibration Digest 31:365-371.

Richalet J (1998). Pratique de l'identification. Hermes Sciences Publications.

Vachtsevanos G, Lewis FL, Roemer M, Hess A, Wu B (2006). Intelligent fault diagnosis and prognosis for engineering systems. John Wiley & Sons, Inc. Hoboken. ISBN: 978-0-471-72999-0.

Williams T, Ribadeneira X, Billington S, Kurfess T (2001). Rolling element bearing diagnostics in run-to-failure lifetime testing. J. Mech. Syst. Signal Process. 15(5):979-93.