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Study of ball bearing fatigue damage using vibration analysis: application to thrust ball bearings

Toumi M.Yessine*, Bolaers Fabrice[a], Bogard Fabien[b] and Murer Sébastien[c]

GRESPI/MAN, University of Reims Champagne-Ardenne, UFR Sciences Exactes et Naturelles, Campus du Moulin de la Housse -51687 Reims Cedex 2, France

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Abstract. This paper presents a study based on the damage due to the fatigue life of thrust ball bearings using vibratory analysis. The main contribution of this work lies in establishing a relation between modal damping and the rolling contact fatigue damage of the thrust ball bearing. Time domain signals and frequency spectra are extracted from both static and dynamic experiments. The first part of this research consists in measuring the damping of damaged thrust ball bearings using impact hammer characterization tests. In a second part, indented components representing spalled bearings are studied to determine the evolution of damping values in real-time vibration spectra using the random decrement method. Dynamic results, in good agreement with static tests, show that damping varies depending on the component's damage state. Therefore, the method detailed in this work will offer a possible technique to estimate the thrust ball bearing fatigue damage variation in presence of spalling.

Keywords: rolling fatigue contact; modal damping; thrust ball bearing; modal analysis

1. Introduction

Evolutions of modal parameters of a mechanical structure subjected to a fatigue load are known and used in various applications. Over the last decades, research has been done in the area of global vibration-based damage detection (Mazurek 1990, Salawu 1997, Yan *et al*. 2007, Česnik *et al*. 2012), which studies failure by monitoring the changes in the modal parameters of the structure (natural frequencies, damping and mode shapes). The change in the modal parameters can also be utilized for fatigue prediction during the operating life of the structure.

Doebling *et al*. (1996) made a review of damage prediction methods based on the variation of modal parameters. Bedewi *et al*. (1997) proposed an application of failure prediction based on modal parameters which predicted the residual life of composite structures by monitoring the decreasing trend in natural frequency and the increasing trend in damping loss factor as a function of the load cycles during a fatigue test.

*Corresponding author, Ph.D., E-mail: yessine.toumi@univ-reims.fr
[a]Senior Lecturer, E-mail: fabrice.bolaers@univ-reims.fr
[b]Senior Lecturer, E-mail: fabien.bogard@univ-reims.fr
[c]Lecturer, E-mail: sebastien.murer@univ-reims.fr

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Modena *et al*. (1999) showed that damping properties have rarely been used for damage diagnosis, in spite of damping-based crack detection having advantages over detection schemes based on frequencies and modal shapes. They also showed that visually undetectable cracks cause very little change in resonant frequencies and require higher-order mode shapes to be detected, while these same cracks cause larger changes in damping. Curadelli *et al*. (2008) proved that damping is a promising damage indicator in structural health monitoring because it has more sensibility to damage than the natural frequency

Thrust ball bearings are frequently used in rotating machinery in order to allow rotary motion under significant load. Yet even under nominal lubrication conditions, load and installation, it has a limited life because of subsurface initiated contact spall. The spall is due to the phenomenon of rolling contact fatigue which initiates subsurface cracks that propagate to the surface causing a pit and spall on the bearing raceway. Rolling contact fatigue changes mechanical properties because of micro plastic deformation leading to localized structural discontinuities such as inclusions (Voskamp *et al*. 1997). Sadeghi *et al*. (2009) made a review of rolling contact fatigue. They highlighted differences between classical fatigue and rolling contact fatigue. Nevertheless fatigue damage is similar in both cases due to the formation of micro-cracks.

In this paper, the evolution of rolling contact fatigue damage on a thrust ball bearing was studied. The evolution of damping ratio was measured/considered as a function of damage fatigue because it is bonded to rolling contact fatigue (Colakoglu 2003). Two experimental procedures have been developed to extract this parameter. Experimental results show that rolling contact fatigue and the presence of a spall on the raceway of a thrust ball bearing yield an increase in the damping ratio.

2. Experimental procedure

All the experiments in this work were carried out at the laboratories of the Université de Reims Champagne-Ardenne. A ball bearings test bench was designed in 2007 with financial participation of the Région Champagne-Ardenne. It is equipped with a press, a double-action hydraulic cylinder and a pressure-limiting valve (Fig. 1).

Fig. 1 (a) Experimental test bench

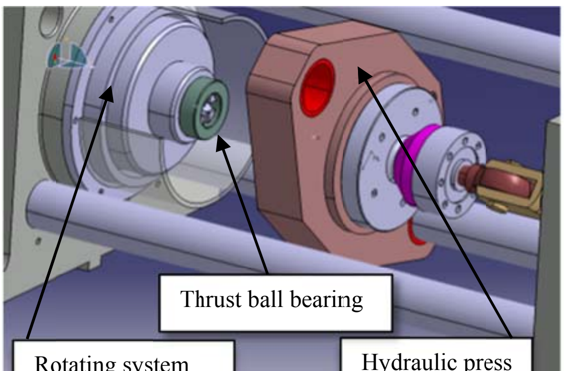


Fig. 1 (b) Graphic presentation of the hydraulic press and the rotating system

Fig. 2 (a) Thrust ball bearing components

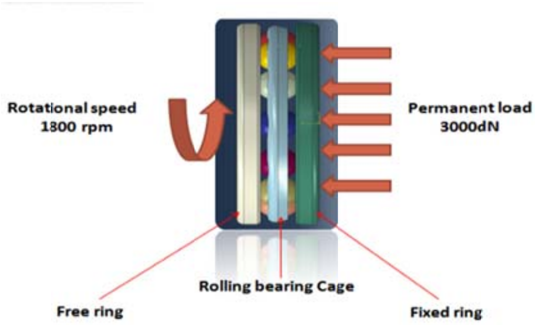


Fig. 2 (b) Description of the experimental procedure

For handling reasons, 51207 stainless steel thrust ball bearings (100Cr6) were chosen (Fig. 2(a)). One-way thrust ball bearings can withstand axial load in one direction but not radial loads. These bearings feature a fixed ring, a free ring, a cage and 12 balls. Young's modulus is 206 GPa, Poisson's ratio is 0.29 and density is 7800 kg/m³. The dynamic load capacity of the studied components is about 3900 daN while the test bench can deliver a pressure up to 5000 daN. The free ring is rotated by a 10 kW electric motor going up to 3000 rpm. To keep suitable industrial conditions for the experiments, the free ring rotation was set to1800 rpm. A permanent load of 3000 daN was applied on the fixed ring of the rolling bearing. A hydraulic pump lubricates the whole system with D68 MAGNAGLIDE synthetic oil (viscosity 71.4 mm²/s at 40°C). The operating temperature of the machine is 60°C, while that of the lubricating oil temperature reaches up to 50°C.

The test bench can withstand 24-hour-per-day operation, being monitored by a controller which stops the machine in case of overheating or lubrication fault.

To detect bearing defect, an accelerometer was installed in the axial direction. Vibration signals were captured by OROS data acquisition system. In the test conditions detailed above, 90% of 51270 thrust ball bearings reach 24-hour life time. However, fatigue damage and micro cracks can appear during the first few working hours. This paper primarily focuses on damping values variation along the fatigue life-time of bearing components. Thus, damaged thrust ball bearings were studied at different levels of rolling contact fatigue. This work will be divided into two parts: a static and a dynamic experimentations, which will be detailed in sections 3 and 4 respectively.

3. Static mode

Modal damping is a phenomenon deeply bonded to the material properties. It varies with different environmental parameters, such as temperature, number of fatigue cycles, applied stress, etc (Colakoglu 2003). In static mode, both natural frequencies and damping ratios can be firstly determined by an impact hammer test for instance.

3.1 Experiments

In this section, the damping factor of damaged thrust ball bearings was determined using shock hammer characterization tests. This procedure consists in exciting the component independently

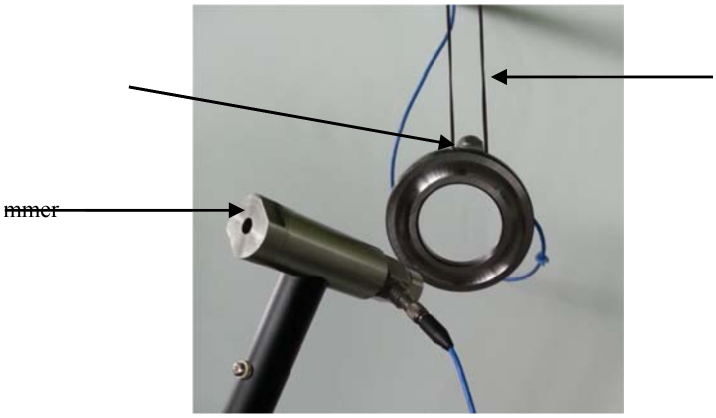


Fig. 3 Impact hammer test

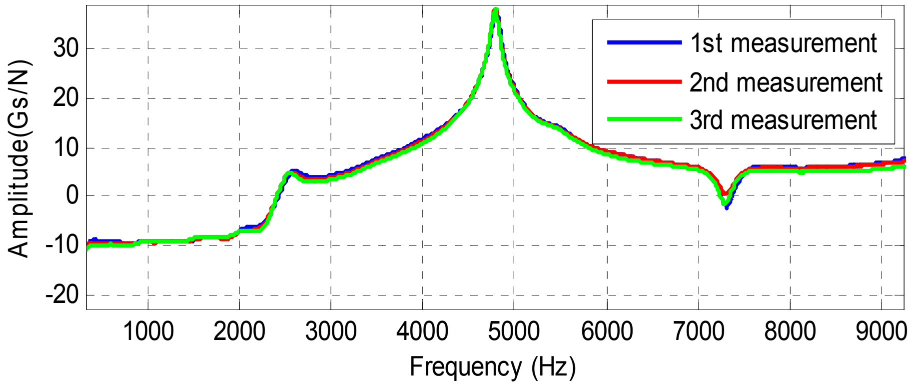


Fig. 4 Frequency spectra of 3 different measurements of the same component

from the housing with free-free boundary conditions. Vibratory acquisitions were performed using SIGLAB vibratory monitoring system. Force sensor set on the impact hammer is a PCB/IMI with a sensitivity of 112.41 mV/N. B&K accelerometer (sensitivity 10 mV/m/s²) was installed on the upper backside of the free raceway of the studied thrust ball bearing. The ring was hung up using a very thin rubber band and hit on the lower front side of the component as shown in Fig. 3.

Excitation was applied at different points of the ring. Each measurement represents an average of 5 impacting tests and three measurements were carried out for each step, as shown on Fig. 4. A 10 kHz sampling frequency was set. As sample temperature drastically affects damping estimation, all tests were performed in almost identical ambient conditions. All results in this part were processed by SIGLAB monitoring system yielding normal modes and corresponding natural frequencies.

Fig. 4 shows that frequency spectra are quite similar. In some cases, they slightly vary from one measurement to another. A particular vigilance was given in order to ensure identical experimental conditions. Therefore, no effects were noticed on natural frequencies due to the impact tests. For accurate results, the most appropriate spectrum was selected.

Two different test campaigns have been carried out in static mode. The first one consists in studying damping of several thrust ball bearings damaged respectively at 24.5 Million cycles,

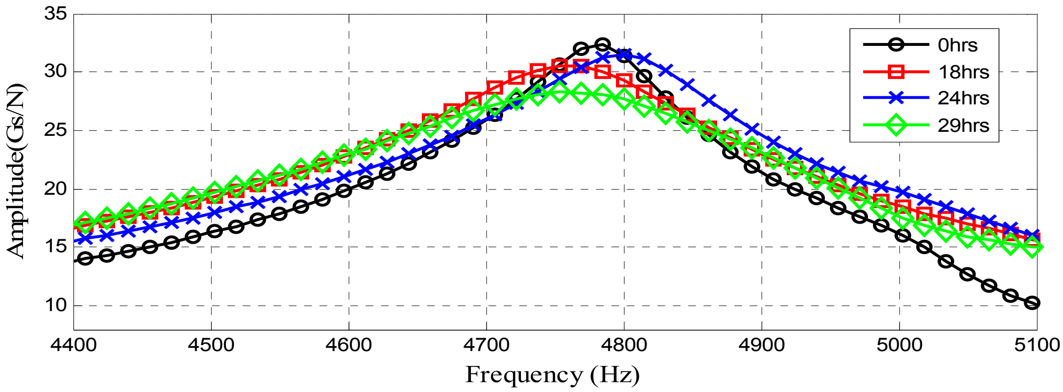
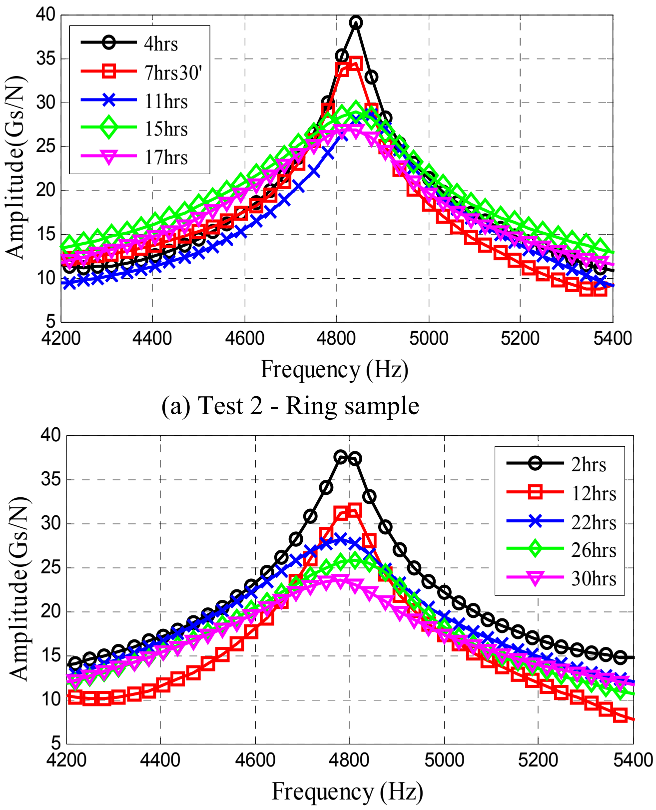


Fig. 5 Comparison of frequency spectra from the first test campaign (Test 1)



(b) Test 2 - Ring sample

Fig. 6 Comparison of frequency spectra from the second test campaign

31.1 M cycles and 37.5 M cycles respectively corresponding to 18, 24 and 29 operating hours. These results were compared to those of an undamaged component. Free vibration structural responses are presented in Fig. 5. It displays a slight attenuation of natural frequency curves depending on the sample fatigue life.

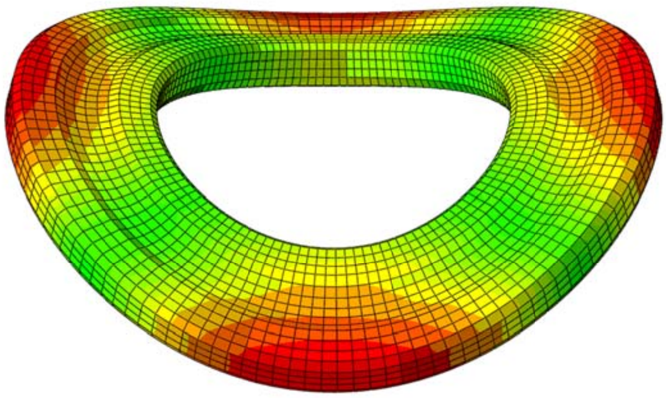


Fig. 7 Numerical simulation of free ring natural mode using FEM

Since every bearing has its own fatigue behavior, a second test campaign was carried out. Its purpose consists in estimating damping evolution of a single thrust ball bearing damaged along fatigue cycles. Experiments were performed almost in the same boundary conditions on two samples. To satisfy the identical temperature condition, comparison with undamaged components will be avoided. For these experiments, test bench machine was preheated to its optimal work temperature as described in section 2. Impact hammer tests were performed immediately after sample extraction, yet a temperature decrease from 2 to 4°C should be taken into account. Tests were stopped as soon as natural spalling was detected on free rings.

According to Figs. 5 and 6, it is noticed that natural mode curves shift randomly to the right or to the left around theoretical natural mode as ring fatigue level increases. This could be due to the position of the structure micro cracks during impact hammer tests. In fact, subsurface cracks and micro cracks may affect the sample mode shape. In addition, considering that vibratory response and free ring hitting were made approximately at the same point for each test due to the complex shape component, curves could shift to the right or to the left. A possible improvement would consist in placing two more accelerators on both left and right sides of the free ring, but this method was discarded since accelerators' weight could alter vibratory response.

3.2 Numerical simulation

Natural modes of the free ring were numerically estimated using the finite element method (FEM). The ring model consists in 9384 linear hexahedral elements (type C3D8R).

The mode shape corresponds to a free-free vibration with zero initial displacement. As shown on Fig. 7, the first mode (4707 Hz) perfectly matches that which was extracted from experiments. In the second test campaign, natural modes of the two and four hours working free ring are respectively about 4800 and 4840 Hz. With an error below 2%, the experiments can be considered as relevant for such use.

3.3 Results

In order to estimate the damping ratio of all studied components, the half-power bandwidth method will be implemented (Guo 2007). It is worth noting that this method needs a great

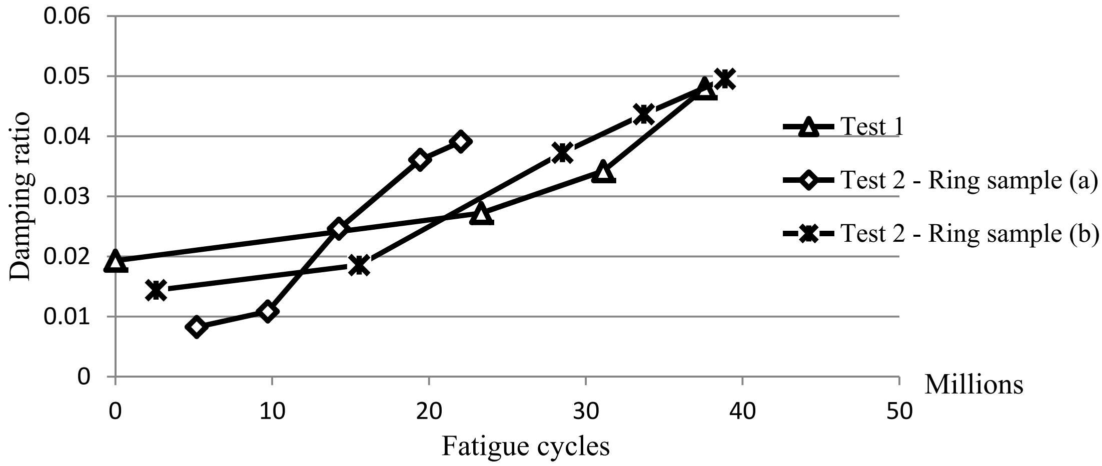


Fig. 8 Damping ratio variations for four ring samples (1st test campaign and ring samples (a) and (b) (2nd test campaign)

precision in damping measurement. Fig. 8 displays the average damping ratio of four different rings as a function of cycles, and clearly highlights growing damping values as the components fatigue level increases. They are consistent to steel damping ratios (commonly below 0.05). Up to 25 M cycles, linear curve remain almost constant. Damping factor varies between 0.02 and 0.025. From 25 M cycles on, the curve greatly increases to reach damping factor of 0.05.

The same figure presents damping ratio variations for two different thrust ball bearings as described above. It is noticed that values are very similar to those obtained in the first case. Some differences may occur due to experimental imprecision. It is also noticed that the shape of curves is almost the same. For case (a), the curve remains constant up to 10 M cycles. After that, it increases quickly. For case (b), curve increases from 14 M cycles. Damping ratio reaches 0.04 for 23 M cycles in the first case and 0.05 for 40 M cycles in the second case. The samples behavior may may partially be related to manufacturing process and experimentation uncertainties, which could explain these differences.

4. Dynamic experimentations

In dynamic mode, many techniques were developed over the last years to determine modal parameters such as damping ratio. They are based mainly on time domain as it is considered more suitable for operational modal analysis. Thus, such methods use complex mathematical models based on the fitting of response correlation functions or the parametric models. Vu *et al*. (2011) proposed a multivariable autoregressive (AR) model using the least squares method to estimate modal parameters which seems to be suitable for this kind of work. However, such precision is not required as the focus is on evolution trends of damping ratio. Therefore, the same method as in static mode will be implemented.

In this section, an algorithm to extract thrust ball bearings normal modes and damping ratios will be presented. The key function in this algorithm is the Random-Decrement (RD) technique. The RD will be presented in the next paragraph. The experimental test and the algorithm schematic will be presented in paragraph 4.2 before analyzing the results in paragraph 4.3.

4.1 Random-Decrement

Random-Decrement is a signal processing technique which transforms the response of a resonant system to random excitation, into its impulse response. It was developed by H.A Coles at NASA during the late 60s and early 70s (Cole 1968) in order to detect space structure damage from the measured response. Since then, it has been applied to a wide variety of structures subjected to immeasurable ambient excitations, in order to extract the modal parameters and eventually to detect failures.

The technique was later given a mathematical basis (Asmussen 1997). Let $X(t)$ be a stochastic process, the RD function is defined as the mean value of a stochastic process on condition, T, of the process itself

$$D_{XX}(\tau) = E\left\lfloor X(t+\tau) \middle| T_{X(t)} \right\rfloor \tag{1}$$

The condition T is the triggering condition.

In order to accurately estimate the conditional mean value from a single observation, it is necessary to assume that the stochastic process is not only stationary but also ergodic. In this case the RD function can be estimated as the empirical conditional mean value from a single realization

$$\hat{D}_{XX}(\tau) = \frac{1}{N}\sum_{i=1}^{N} x(t_i+\tau)\Big|T_{x(t_i)} \tag{2}$$

Where N is the number of points in the process which fulfills the triggering condition and $x(t)$ is a realization of $X(t)$. The triggering condition used in this work is

$$T_{x(t_{i+1})} = \{x(t_{i+1}) > 0 > x(t_i)\} \tag{3}$$

i.e., a positive going zero crossing.

In damage failure detection the basic idea is that an incipient failure will change the stiffness and the damping characteristics of the structure (Ait Sghir 2007). The RD function can be used to estimate the FRF (Asmussen 1996) of the whole system and through a filtering process, the ring FRF can be extracted. The last step of the process is the damping calculation.

4.2 Experiments

In this section, the damping ratios of several thrust ball bearings will be estimated in dynamic mode in situ using real-time analysis. A DJB accelerometer (sensitivity 1.025 mV/m/s²) was set in the axial direction. All boundary conditions used in the previous experiments were maintained on the dynamic test campaign so that test conditions are identical. In this part of our work, only pre-spalled (indented) components will be tested. This technique was successfully implemented in many previous studies to investigate components spalling evolution *in situ* (da Mota *et al.* 2008, Christophe 2001, Rosado *et al.* 2010, Branch *et al.* 2010).

In fact, ball bearings raceways undergo contact surface hardening over the first cycles. Natural spalling is difficult to obtain and the experimentation can take a long time. Mainly, for time cost reasons, indents were created on bearing raceways to accelerate the crack propagation phenomenon. They must be initiated in sub-layer according to Hertz theory (Rosado *et al.* 2010).

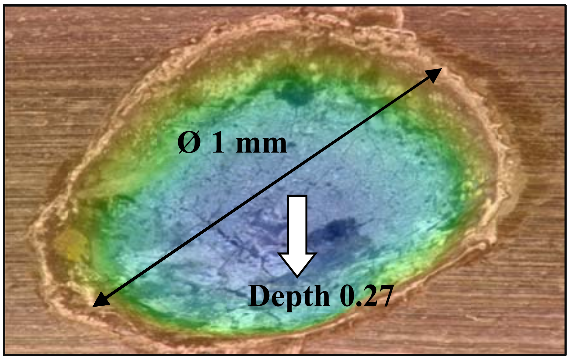


Fig. 9 Microscopic picture of the indented area

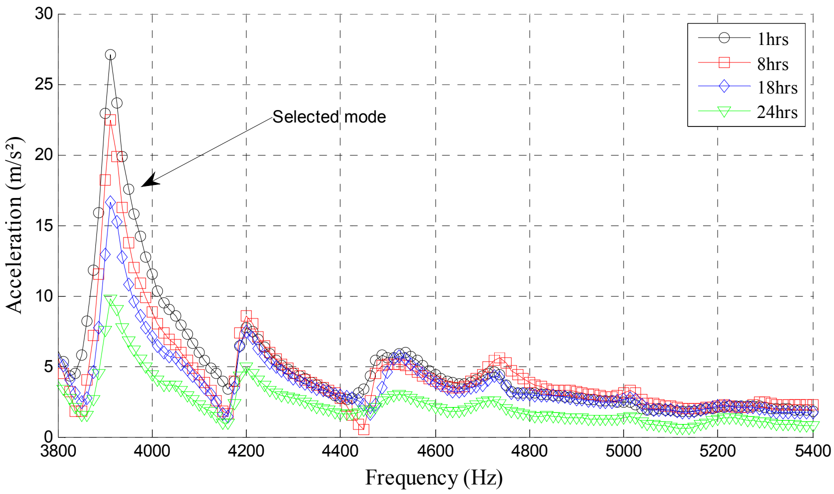


Fig. 10 RD spectra of indented rings

In our case, depth of indents is 0.27 mm and diameter is 1 mm. An electro-erosion machine was employed to minimize stress concentration effects. Fig. 9 presents a microscopic picture of indented raceway.

For this test campaign, four thrust ball bearings were subjected to fatigue tests during 1, 8, 18 and 24 hours respectively. They were subsequently indented in the same conditions. All data acquisitions (40 seconds each) were performed using dynamic system OROS. To determine the thrust ball bearings damping ratios, the algorithm presented in Fig. 11 was applied. First, Random Decrement method (window size 2048, averaging 50000) was applied to extracted frequency spectra for the whole system normal modes. As a result of the static study, it was identified that the thrust ball bearings normal mode is around 4800 Hz. Nevertheless, dynamic study showed that this frequency decreased to 3900 Hz. After studying all modes, this was the best reacting mode with fatigue tests (Fig. 10).

A filtering (from 3850 Hz to 4200 Hz) was applied to isolate this mode. Then FFT was applied to the filtered impulse response. Finally, the Half Power Bandwidth method was used to determine damping factors.

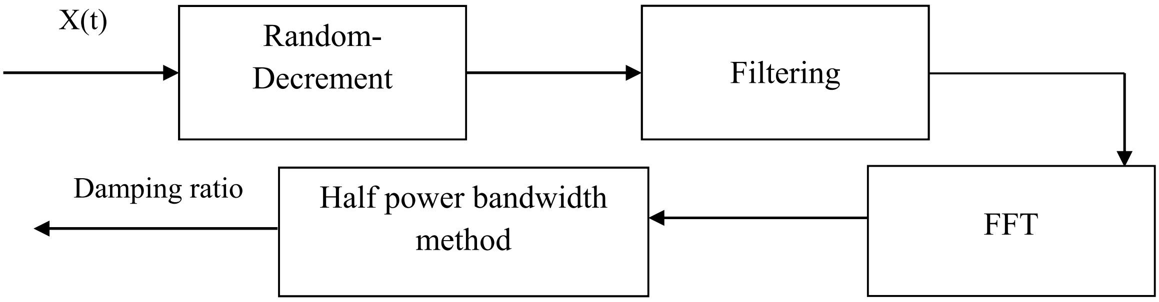


Fig. 11 Damping ratio extraction algorithm

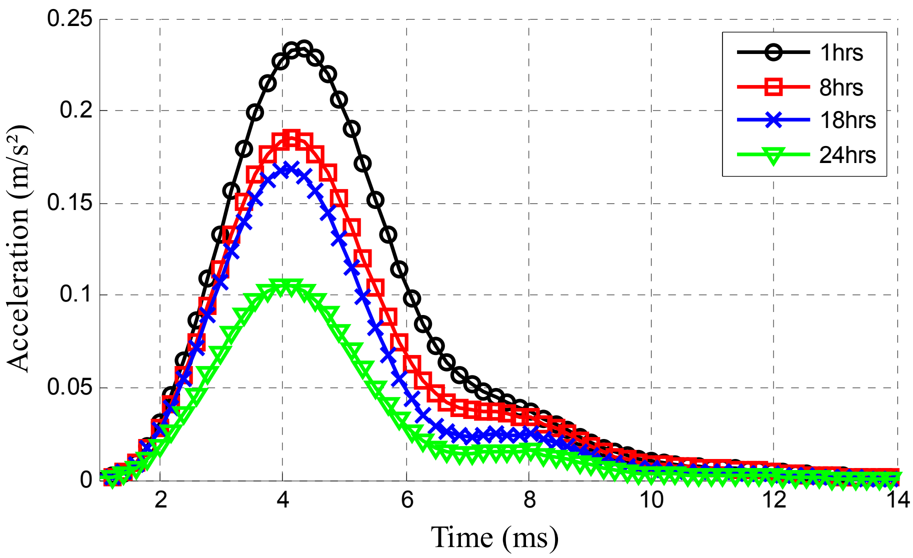


Fig. 12 Impulse response envelope in dynamic mode

4.3 Results

Fig. 12 shows the envelope curves of the studied thrust ball bearings *in situ*. Hilbert Transform was applied to filtered FFT spectra. It is noticed that signals are damped proportionally to fatigue level.

Damping ratios determined using the half power bandwidth method are shown on Fig. 13.

These results are in good agreement with those determined in static mode. It is worth noting that the vibrations propagation path is not the same as in static test, since it will excite the whole structure that supports the bearing. This explains why values are different from those previously estimated. However, the evolution trend is very similar. As a consequence, this procedure is considered suitable to estimate damping ratios in both static and dynamic modes. Damping factor can also be considered as an indicator for thrust ball bearings fatigue damage.

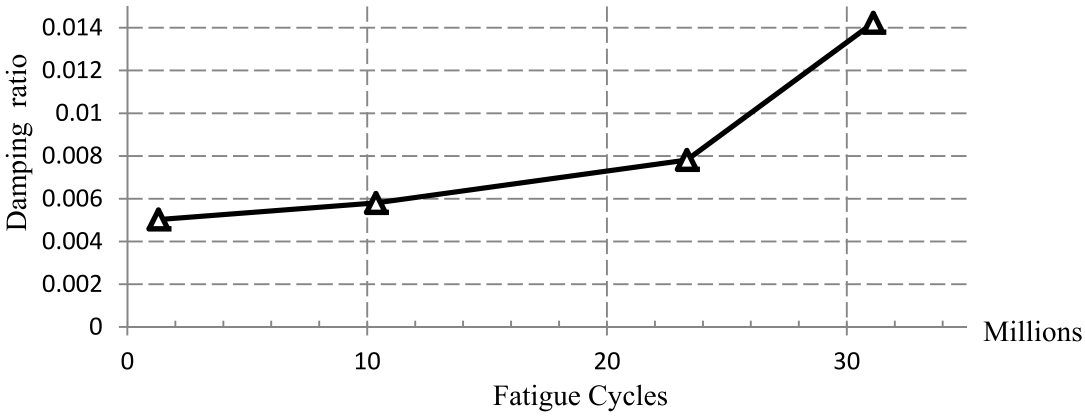


Fig. 13 Damping ratio of four rings versus operating cycles in dynamic mode

5. Conclusions

In this paper, a technique using the modal damping ratios was developed to estimate thrust ball bearings damage variation. Test campaigns were carried out in both static and dynamic mode. In static mode, impact hammer tests were used to characterize normal modes, then to extract damping values from normalised frequency spectra. Two test campaigns corresponding to two different kinds of experimentations were made. Both are clearly similar and show that damping ratios vary greatly due to components damage level.

In dynamic mode, indented components were studied *in situ*. Real-time vibration spectra were extracted and treated using the Random Decrement method. Estimated damping ratios evolution was quite similar to that highlighted in static mode. In this kind of work, it seems clear that damping is a suitable indicator to estimate structural damage level.

For both static and dynamic mode, it is shown that damping ratio increases with the fatigue damage level of tested thrust ball bearings. Thus, the damping ratio determined from the measured vibrations seems to be a good indicator of the damage level of mechanical components subjected to rolling contact fatigue such as thrust ball bearings.

Further test campaigns are planned to study the effects of load, velocity and indent size on damping ratios evolution in order to determine a fatigue damage time domain indicator. This one is going to be used later to readjust a numerical model of thrust ball bearing fatigue damage.

Acknowledgements

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