PAPER REF: 4512

# VIBRATION MODAL ANALYSIS OF ROLLING ELEMENT BEARING

**Elsayed S. Elsayed[1(*)], Ahmed Elkhatib[2], Mostafa Yakoot[3]**

Department of Production Engineering, Faculty of Engineering, Alexandria University, Alexandria, Egypt
[(*)]*Email:* elsayedsaad2005@yahoo.com

## ABSTRACT

The most important factor that should be taken into consideration during the design of machinery systems is their dynamic characteristics. Rolling element bearings are fundamental components in many machinery systems that should support a high portion of the dynamic loads acting on these systems. Hence, the dynamic characteristics of bearings should be solicited during the selection of those bearings. This paper presents a criterion for a better selection of identical bearings of different makes. The paper introduces a comprehensive study of the dynamic characteristics of four bearings with the same code but from different manufacturers. Finite element modelling of these bearings was performed to determine the dynamic characteristics of each bearing. In addition, an experimental vibration modal testing was performed to determine the actual dynamic characteristics of each bearing separately. The results show that although the bearings have slight differences in their design, the variations in their dynamic characteristics are significant. This will, indeed, have a reflection on their performance and life. Hence, modal analysis and testing is an efficient tool for the selection of bearings and their manufacturers.

**Keywords:** modal testing, dynamic characteristics, rolling bearings, finite element modelling.

## INTRODUCTION

Determining the modal frequencies, modal damping and mode shapes of rolling element bearings plays a vital role in a wide range of applications. A finite difference method for modal parameters estimation has used theoretical formulae to estimate the natural frequency and damping (Yin, 2000). In a study by (Maia, 2001), modal analysis identification techniques were described. The experimental vibration modal test was used to estimate the dynamic characteristics of structural components such as railway track components (Kaewunruen, 2005). The modal damping has been described to be a good indicator for improving the fatigue life of materials (Damir et. Al., 2007). Key parameters that characterize the dynamic behaviour of bearings are the stiffness and modal damping (Ali, 2009). Other technique such as shaker testing is commonly used as a method for measuring forced input in experimental modal analysis (Peres, 2010).

In this study, vibration modal tests were performed on four different bearings with the same code of (6305zz) but from different manufacturers; A, B, C, and D. The dimensions and configuration of each bearing were studied. A comprehensive study of the dynamic characteristics of each bearing was performed. This study will help the designer to select the most appropriate bearing according to the dynamic factor that can have the longest life. Theoretical technique to get the ideal dynamic characteristics of each bearing was performed using finite element package (ANSYS™). This was used together with experimental modal analysis to validate the results obtained by FEA.

## SPECIFICATIONS OF THE TESTED BEARINGS

Four rolling element bearings from different manufacturers (A, B, C &D) were used in this study. The type of bearing is single row deep groove ball bearing with double shield of code (6305zz). Referring to their corresponding selection catalogue, the nominal dimensions of these bearings were 62 mm - outer diameter, 25 mm - shaft diameter and 17 mm - width. Table 1 shows the actual dimensions of tested bearings measured with calibrated measuring instruments and Table 2 shows the masses of elements of those bearings.

Table 1 Actual dimensions of tested bearings

| Bearing | Number of balls | Ball diam. (mm) | Outside diam. (mm) | Bore diam. (mm) | Width (mm) |
|---|---|---|---|---|---|
| **A** | 7 | 11.466 | 62.050 | 25.312 | 17.380 |
| **B** | 7 | 11.470 | 61.805 | 24.846 | 17.346 |
| **C** | 8 | 10.285 | 61.747 | 24.916 | 17.375 |
| **D** | 8 | 10.290 | 62.146 | 25.182 | 17.363 |

Table 2 Masses of elements of tested bearing in Kg

| Bearing | Ball | Outer ring | Inner ring | Shield | Bearing |
|---|---|---|---|---|---|
| **A** | 0.0062 | 0.1081 | 0.0596 | 0.0032 | 0.2309 |
| **B** | 0.0062 | 0.1041 | 0.0587 | 0.0032 | 0.2292 |
| **C** | 0.0044 | 0.1050 | 0.0687 | 0.0035 | 0.2322 |
| **D** | 0.0043 | 0.1021 | 0.0751 | 0.0037 | 0.2275 |

It is evident from the data given in Tables (1&2) that the four tested bearings are not congruent but rather different in their elements dimensions and masses. The most significant difference is the number of balls comprised in each bearing and ball size and mass. This briefly proves that the considered bearings constitute four different dynamic systems; each will have its own dynamic characteristics.

## FINITE ELEMENT ANALYSIS

The tested bearings (A, B, C & D) were investigated using a modal analysis package of ANSYS™ to identify the ideal natural frequencies and modes of vibration of each bearing. It was assumed in the analysis that the system is linear and the damping is neglected. Building the model geometry of considered bearings was based on their actual measured dimensions as given in Table 1 and made using Solid Work 20™ software package.

The analysis covered a frequency range of 0-12.8 KHz. A list of suggested materials of bearing elements is shown in Table 3. This material list was not changed during the analysis of all tested bearings to eliminate the influence of bearing material on the results.

Table 3 Material list of tested bearings elements

| Bearing Element | Material |
|---|---|
| **Balls** | DIN steel tool making 1.3505 100cr6 |
| **Outer and inner rings** | DIN steel tool making 1.3505 100cr6 |
| **Cage** | ANSI 304 |
| **Shields** | Stainless steel sheet |
| **Cage bolts** | Alloy steel |

The analytical model of bearing was analysed under free-free boundary condition. Meshing was generated automatically by the program. Suggested interface setting for different elements of bearings used in the analysis are listed in Table 4.

Table 4 Suggested interface setting for different tested bearing elements

| Contact Area | Type of contact |
|---|---|
| **Balls & rings** | Frictional (coeff. of friction = 0.125) |
| **Balls & cage** | Rough |
| **Shield & outer ring** | Frictionalless |
| **Shield & inner ring** | Bonded |
| **Cage parts** | Bonded |
| **Ball & cage** | Bonded |

Figures 1-4 show the first mode of vibration of tested bearings A, B, C & D in the frequency range of 0-12.8 KHz. Illustrations of the second mode of vibration are shown in appendix A. A summary of the natural frequencies of the first two modes of vibration of tested bearings determined using finite element analysis is illustrated in Table 5.

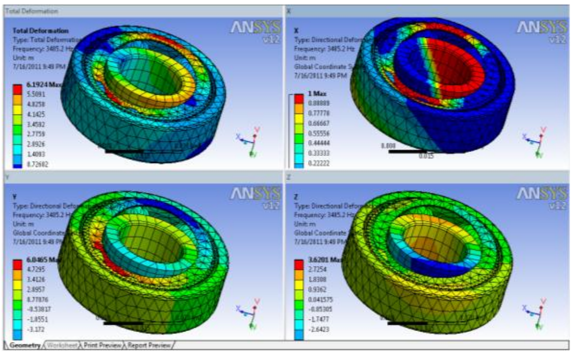

Fig. 1 First mode of vibration of bearing A

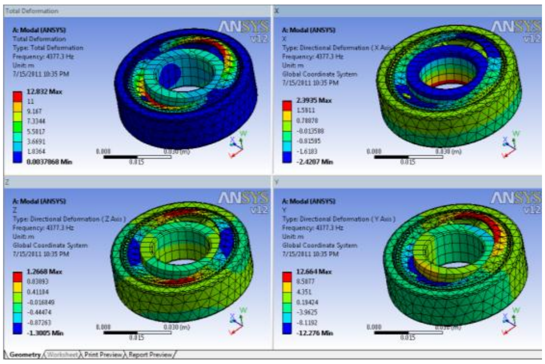

Fig. 2 First mode of vibration of bearing B

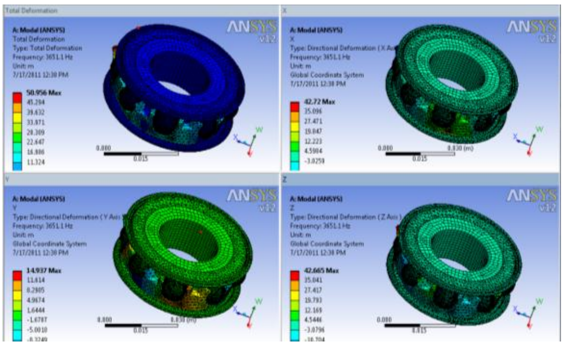

Fig. 3 First mode of vibration of bearing C

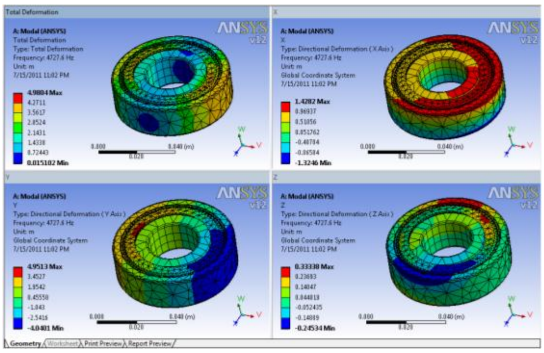

Fig. 4 First mode of vibration of bearing D

Table 5 Summary of the natural frequencies of bearings studied using FEA in Hz

| **Bearing** | **A** | **B** | **C** | **D** |
|---|---|---|---|---|
| **$1^{st}$ mode** | 3485.2 | 4377.3 | 3651.1 | 4727.6 |
| **$2^{nd}$ mode** | 8771.3 | 7224.7 | 7290.5 | 6201.8 |

## EXPERIMENTAL INVESTIGATION

Actual modal parameters of the bearings were identified using experimental modal analysis or modal testing. The bearings were excited using an impact hammer technique. The usefulness of this technique lies in the fact that the energy of an impulse is distributed continuously in the frequency domain. This impulse will excite all resonances (or modes) within its useful frequency range that depends on its shape and duration. Processing of signals of both excitation and response leads to the determination of the Frequency response function FRF of the tested bearing. The modal parameters were then experimentally determined from the measured frequency response function of the tested bearing.

To carry out the test, the considered bearing was hanged in an elastic cord to eliminate the effect of support conditions on the results. An impact hammer, equipped with force transducer was used to strike the bearing with light impacts to avoid system nonlinearity. The vibration response of the bearing due to that impulse was picked up using an accelerometer attached radially at a point on the outer ring of the bearing apposite to that of force application. Both excitation and response signals were fed to a 4-channel signal analyser for processing and determining the FRF of the bearing. The modal parameters were estimated and the FRF curve was displayed using special software installed in a host computer interfaced with the signal analyser.

Quality of the results depends basically upon the reliable procedure followed to control possible sources of noise during different stages of bearing testing and signal processing. The coherence function was also measured to monitor the quality of the FRF measurement along the specified frequency range. Figure 5 illustrates the experimental set up.

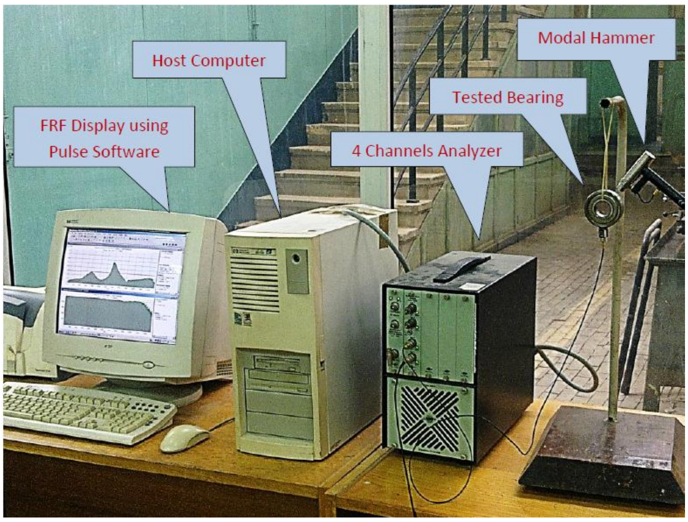


Fig. 5 Experimental set up of bearing FRF measurement

The instruments used in the present study are:

- B&K impact hammer (Type 8202) fitted with a B&K force transducer (Type 8200) having an output sensitivity of 1.01 pc/N,
- B&K accelerometer (Type 4384) having a voltage sensitivity of 0.810 $mV/m/s^2$ ,
- B&K 4-channel signal analyser (Type 2825),
- Pulse Lab shop software (Version 6.1.5),
- B&K Pulse Multi-analyser system (Type 3560) installed on a host computer interface with the signal analyser.

During bearing testing, the analyser was set to a frequency range of 0-12.8 KHz. Transient window was selected for sampling of impulse signal while exponential window was selected for sampling of response signal. Since both impulse and response signals are transient in nature, both input and output channels of the analyser were set to trigger mode. Both the frequency response and coherence functions were computed from the averaged power and cross spectra.

## RESULTS

The results of the FRF measurement of tested bearings A, B, C and D using Modal testing are shown in Figs. 6-9. A summary of the identified modal parameters (natural frequency $f_n$, damping ratio $\zeta$ and FRF amplitude) of tested bearings at the $1^{st}$ mode of vibration as well as the value of coherence function $\gamma^2$ are shown in Table 6, while those at the $2^{nd}$ mode are shown in Table 7.

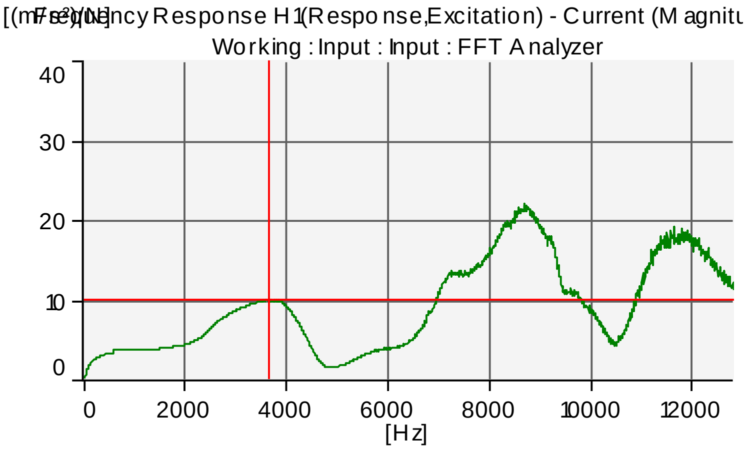


Fig. 6 Frequency response function FRF of bearing A

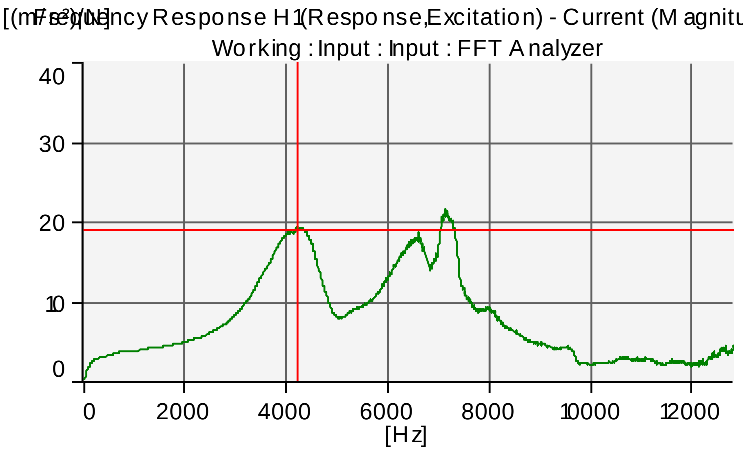


Fig. 7 Frequency response function FRF of bearing B

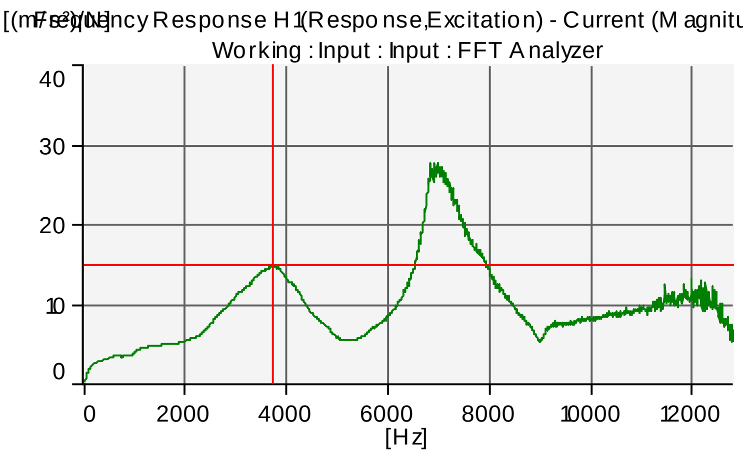


Fig. 8 Frequency response function FRF of bearing C

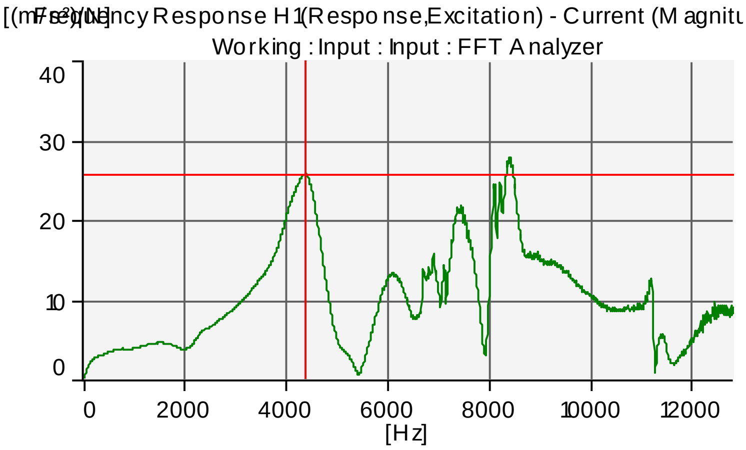


Fig. 9 Frequency response function FRF of bearing D

Table 6 Summery of modal parameters of bearings at the 1st mode

| Bearing | A | B | C | D |
|---|---|---|---|---|
| Natural Frequency $f_n$ (Hz) | 3188 | 4260 | 3424 | 4240 |
| Damping Ratio $\zeta$ % | 14.9 | 11.2 | 18.2 | 13.2 |
| FRF Amplitude ($m/s^2/N$) | 10.2 | 18.9 | 14.8 | 25.5 |
| Coherence Function $\gamma^2$ | 0.993 | 0.982 | 0.995 | 0.974 |

Table 7 Summery of modal parameters of bearings at the 2nd mode

| Bearing | A | B | C | D |
|---|---|---|---|---|
| Natural Frequency $f_n$ (Hz) | 8680 | 7120 | 6904 | 6072 |
| Damping Ratio $\zeta$ % | 7.56 | 3.18 | 5.86 | 4.69 |
| FRF Amplitude ($m/s^2/N$) | 22.1 | 20.9 | 27.7 | 13.8 |
| Coherence Function $\gamma^2$ | 0.966 | 0.981 | 0.972 | 0.924 |

## DISCUSSION

The results show good correlation between the experimental and theoretical natural frequencies. There is a significant variation in both natural frequency and damping ratio for different bearings at first and second modes of vibration. Maximum variation of natural frequency reaches 34% between bearings B and A at the first mode and 43% between bearings A and D at the second mode. Maximum variation of damping ratios reaches 64% between bearings C and B at the first mode and 138% between bearings A and B at the second mode. Maximum variation of FRF amplitude reaches 151% between bearings D and A at the first mode and 108.2% between bearings C and D at the second mode. Bearing A has the lowest value of Frequency response function amplitude at the 1st mode, while bearing D has the highest value of Frequency response function amplitude at the same mode. This proves that bearing A can have the longest expected life, provided that the bearings will subjected to same mounting and operating conditions in practice.

## CONCLUSIONS

This study shows that there are substantial differences between the dynamic characteristics of bearings of different makes. The dynamic characteristics of bearing should be considered during the selection of those bearings. Also, modal testing is an efficient tool for the selection of the most appropriate bearing that can have the longest expected life. This technique is valid only for non-separable bearings types.

## ACKNOWLEDGMENTS

The authors gratefully acknowledge the efforts of production engineering BSc. students in carrying out the tests and Machine Dynamic Laboratory, Facuty of Engineering, Alexandria University for providing the necessary testing facilities.

## REFERENCES

Ali NJ, Garcia JM. Experimental studies on the dynamic characteristics of rolling element bearings. Journal of Engineering Tribology, 2010, 224, p. 659-666.

Damir AN, Elkhatib A, Nassef G. Prediction of fatigue life using modal analysis for grey and ductile cast iron, International Journal of Fatigue, 2007, 29, p. 499-507.

Kaewunruen S, Remennikov A. Applications of experimental modal testing for estimating dynamic properties of structural components, Australian Structural Engineering Conference (ASEC 2005), Newcastle, Australia, 2005, p. 50-55.

Maia NMM, Silva JMM. Modal analysis identification techniques. Philosophical Transactions A of the Royal Society, 2001, 359, p. 29-40.

Peres MA, Bono RW, Brown DL. Practical Aspects of Shaker Measurements for Modal Testing. International Conference on Noise and Vibration Engineering (ISMA2010) including Uncertainty in Structural Dynamics (USD2010), 2010, p. 2539-2550.

Yin HP, Duhamel D. Finite Difference Formulation for Modal Parameter Estimation. Journal of Sound and Vibration, 2000, 231(2), p. 259-275.

## Appendix A

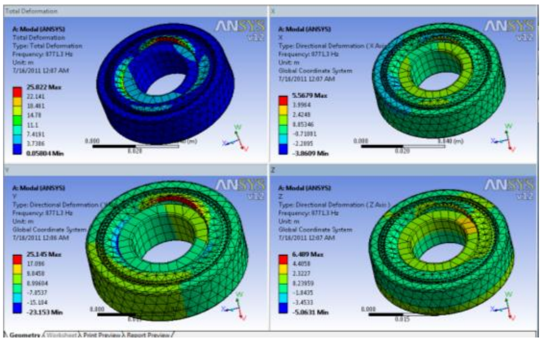

Fig. A1 Second mode of vibration of bearing A

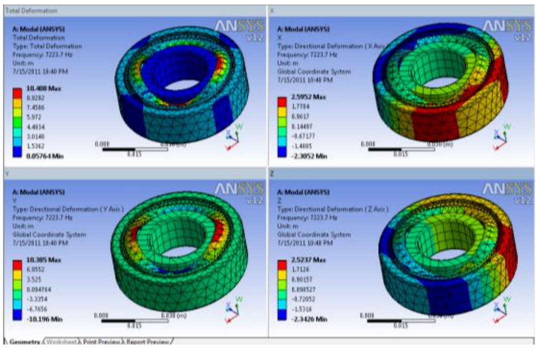

Fig. A2 Second mode of vibration of bearing B

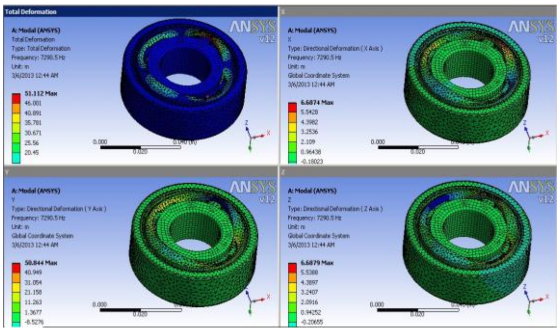

Fig. A3 Second mode of vibration of bearing C

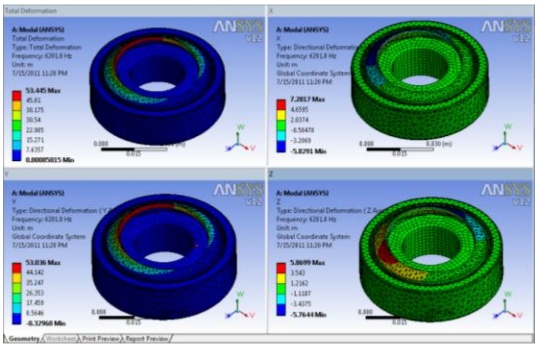

Fig. A4 Second mode of vibration of bearing D