

DYNAMIC CHARACTERIZATION OF BALL BEARING IN TURBINE PROPELLER USING BUMP TEST METHOD

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Abstract

Bearing is an essential component in a mechanical rotating equipment system. It is no less important than lubrication to prevent wear is very important to consider in the mechanical maintenance system of rotating equipment. Bearing wear is one of the problems in wind turbines that will increase maintenance costs, shorten the wind turbines' lifespan, and cause the component or overall damage to the wind turbine. The latest technology has provided instruments for analyzing the damage of elements in a bearing according to the caused vibrations. Therefore, this study was performed on Ball Bearing Turbine Propeller to identify the dynamic characteristics of Ball Bearing with and without lubrication. The test was carried out using the Bump Test method applied in three measured parts: X, Y, and Z axes. The measuring instrument which was used was Fast Fourier Transform (FFT) Analyzer (Ono Sokki) and the data were analyzed using MATLAB. It was identified that the application of oil could reduce the amplitude and decrease the frequency. Personal frequency appearing more than once indicates the existence of global vibration modes. The frequency which only appears once in the measurement spot indicates local vibration modes. The highest frequency both after and before the application of oil was found in the Y-axis.

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INTRODUCTION

Bearing is a very important component in a mechanical rotating equipment system. One of the causes for decreasing bearing performance is the friction in the bearing that causes wear due to the bearing's lack of lubrication. Bearing wear is one of the problems in wind turbines that will increase maintenance costs, shorten the wind turbines' lifespan, and cause the component or overall damage to the wind turbine.

Damage in the bearing can be indicated and analyzed visually using naked eyes after detaching it from shaft or housing. Some faults can be analyzed visually, including overload, overheating, incorrect installation, contamination, lubricating errors, etc. The latest technology has provided instruments for

analyzing the damage of elements in a bearing according to the caused vibrations.

Measuring vibration on the bearing can be done using a vibration test. A bump test is a quick and economic vibration test designed for vibration mode and machine structure.

Several researchers have carried out some studies, including Matos who did the bump test that used an instrumented force hammer to analyze failure diagnosis in generator endwindings [1]. The bump test (or collision test) is the best way to ensure the damaging end wall vibration that does not happen in the new engine and it is usually used for turbine generators [2]. Therefore, the generators' vibrational analysis as rotating machines will be beneficial for the generator design in the initial stage and online monitoring and faults

diagnostics during generator operation. The bump test (hammer impact testing) of stators endwinding vibrations with an accent on the physical parameters' influence temperature [3]. The bump test on the turbine side and the exciter side of the endwinding generator identify natural frequency (resonance) around 95 Hz to 115 Hz. The bump test result showed a lot of end winding ties structure looseness [4].

Through vibration analysis that bumps test method, Torie analyzes resonance mitigation of cooling tower. Based on the research, it was found that the gearbox's natural frequency coincided with the cooling fans blade pass frequency (BPF), which an indication of damage. [5]. In ASME, it does not provide a clear guideline about the vibration problem of thermowell that may be operating transverse vibration zone that made the thermowell vibration amplitude increase until eventual failure. Prasad et al. gave a method that is suggested to quantify the damping in the system by utilizing actual site vibration measurement of thermowell. Damping plays a very crucial role in amplitude when thermowell is operated in a critical zone. Hence, it is very important to estimate the damping factor [6]. Moreover, the bump test method can be used the vehicle as excitation for medium-and small-span bridges. The vehicle bump test's dynamic response can provide theoretical support for the dynamic load test [7].

The brief test (the bump test) procedure includes the Frequency Response Function (FRF) test. It compares the vibration response signal received by a mechanic structure due to vibration excitation from a frequency domain system. Signals of vibration responses and excitation are measured simultaneously using vibration sensors and bump test as excitation force. By utilizing Fourier transform, data from the measurement and results of the transformation are obtained from the frequency domain. The FRF is the basis of measurement to determine the dynamic characteristics in a mechanical structure. Experimental capital parameters (frequency, attenuation, and shape mode) can be obtained by measuring the FRF [8].

The following are some studies about the FRF test. Conducted an FRF test on a powertrain's cylinder machine using two excitation force methods: Impact Hammer and Exciter. FRF is a fundamental measurement to know the dynamic characteristics of the mechanic structure [9].

The damping performance in a vehicle measured using the FRF method is obtained from the excitation force applied. It is compared

with vibration according to the speed level measured at the same point [10]. Identification of nonlinearity using the FRF method and analysis using wavelet packet decomposition [11]. Research a dynamic characterization test on the piston-cylinder motor to find global vibration modes frequency. The FRF method to know the dynamic character of a single cylinder and found global and local vibration modes [12]. FRF measurement was also employed to detect cracks occurring in a beam [13], to detect damage a structure on vehicle [14] and to detect the degree of damage or to crack on the gears [15]. FRF method can also be applied to study the damage at the endmill based on vibration response [16]. The FRF method to study the investigation of endmill feeds on the CNC router machine based on vibration response. Frequency's broken endmill was far higher than the normal endmill because the broken endmill results in the unperfect feed [17]. The investigation of vibration damping in the passenger seat constructions. The research aims to determine the passengers' comfort levels to reduce the vibrations caused by the bus frame used the FRF method [18]. The FRF test investigates a mechanic structure's dynamic character in identifying damage on x, y, and z axes of tapered bearing with harmonic vibration from handphones [19]. Dynamic characteristics of disc brake using the bump test [20], then the method of damage to the disc brake [21] and detected crack in shaft [22] using the bump test method. The Eigenvalue of a structure can be obtained from the FRF test and to verify the frequency modes, it is compared using CEA [23].

Bearing analyses have been extensively carried out. Based on vibration analysis, FRF was used to monitor the condition of roller bearing [24]. The research about damages occurring detection in the raceway cone on tapered roller bearing was analyzed based on vibration analysis that measured the FRF with the excitation force method given through the harmonic method. The response of bearing vibration was analyzed in three directions [25].

The involving different types of bearing on the rotating shaft in which tapered roller bearing produced the highest vibration amplitude among three bearings. In contrast, the ball bearing shows the lowest amplitude under the same operation [26]. Dynamic characters result in dynamic performance as scroll bearing is significantly affected by internal radial distances [27]. The nonlinear correlation between friction and natural vibration and the effect of lubricant specification on nonlinear natural vibrations. Different lubricant specification results in different

vibration modes according to the damping ratio because of differences among damping mechanisms [28].

The research objects were ball bearing of the propeller turbine with five blades with winglets [29]. In this case, the winglet is attached to the propeller turbine's tip and the simulation result was compared using CFD while the trial results using Wind Tunnel. In the research, vibration analysis was not carried out on the propeller turbine's design, so this paper aims to identify the dynamic characteristics of ball bearing after adding oil or lubricant using the bump test method for knowing the effect of lubrication caused by vibrations.

MATERIAL AND METHOD

The Frequency Response Function (FRF) is the ratio of the vibration response signals received by a mechanical structure resulting from a system's excitation force in the frequency domain. The vibration response and the vibration excitation signals are measured simultaneously using a vibration sensor and load cell, which acted as the exciter force to measure the instruments. We obtained the data and the frequency domain transformation by applying a Fourier Transform, as shown in Figure 1.

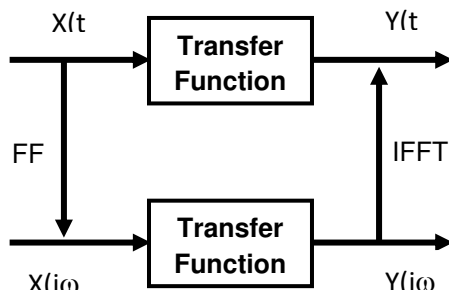


Figure 1. The relationship between the time and frequency domain

A Fast Fourier Transform (FFT) is used to change the signal time domain into the frequency domain, while Invert Fast Fourier transform (IFFT) is used to change the signal frequency domain into the time domain. Matlab 2013 software is a data analysis and simulation software that already provides FFT functions [5].

The influence of the lubrication on vibration was identified by measuring the FRF. The excitation force with impact hammer given to ball bearing was a bump test measured using a vibration analyzer. The vibration response measured is carried out on three axes, as shown in Figure 2, the points measured before and after the tool applied with oil.

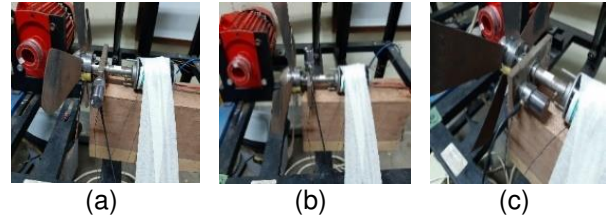


Figure 2. Points measured with FRF:
(a) X-axis, (b) Y-axis, (c) Z-axis

The range of applied frequencies was 0-20 KHz. The application of frequency in that range allows us to get the graphical results of measurement using a vibration analyzer in the silent mode, which was then processed using Matlab Software. The adjustment of the test to get the experimental data is shown in Figure 3. The test results were analyzed using Matlab, which was then processed again using Microsoft Excel to identify the personal frequency of each position measured. While a gallery set up of testing to obtain experimental data can be seen in Figure 3.

In Figure 3, tools and materials occupied at test included:

- Accelerometer sensor to measure the response from vibration.
- Type: Piezoelectric accelerometer made by ono Japanese sokki.
- Accelerometer cable: 1.5m
- FFT portable type analyzer CF-3600A (4-ch) with a touch screen completed with analyzer simulation and data recorder. The maximum frequency that can be analyzed was 40kHz. The product is from Ono Sokki Japan.
- Matlab R2019.
- The specification of the ball bearing on the propeller turbine can be seen in Figure 4.

The specification of the ball bearing on the propeller turbine is:

- Outer Diameter: 35 mm
- Inner Diameter: 17 mm
- Width: 10 mm



Figure 3. Set-Up of FRF Measurement Test

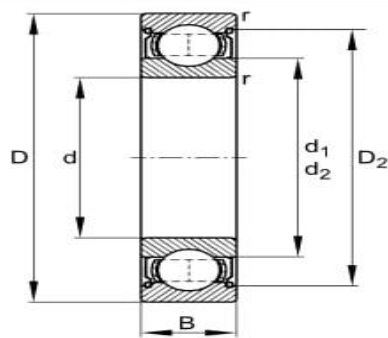


Figure 4. Ball Bearing 6003 – 2Z FAG

RESULTS AND DISCUSSION

In Figure 5, Figure 6, and Figure 7 show the data obtained from measurement using FRF. They offer the results of measurements performed at X, Y, and Z axes with and without oil. The applied frequency was 20 kHz and the number of samples was 4096. Data presented on the graphs applied the frequencies between 0 – 50 Hz.

The FRF measurement results in each axis can be seen in Figure 5, Figure 6, and Figure 7. In Figure 5, it can be that at X-axis, the application of lubricant can lower the amplitude leading to the reduction of frequency. By using lubricant, the maximum frequencies were only 4 and 12 Hz. While without lubricant, they can reach 4, 12, 20, 28 and 36 Hz.

Due to the absence of lubricant, on Y-axis, amplitude rose and increased frequency. Without lubricant, the maximum frequencies emerging in ball bearing were 4, 6, 8, 12, 14, 16, 23, 25, 28, 32 and 36 Hz. While the maximum frequencies in ball bearing added with lubricants were 4, 6, 8, 12, 14, 16 and 20 Hz. For more detail, the results of FRF measurement on the Y-axis can be seen in Figure 6.

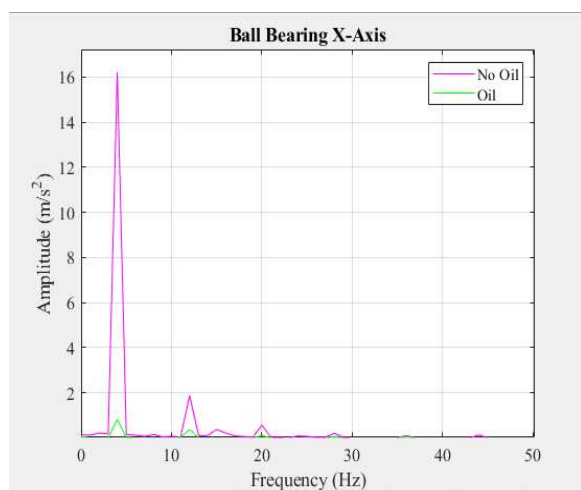


Figure 5. Results of FRF Measurement on X-axis

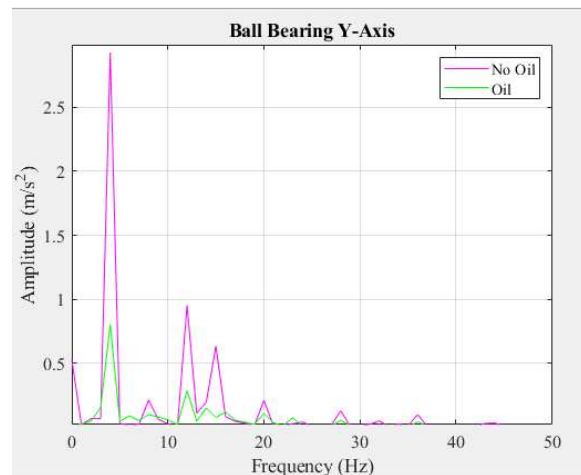


Figure 6. Graph of FRF measurement at Y-axis

Z-axis added with lubricant could reduce the amplitude it results, thus minimizing the frequency. When using a lubricant, the maximum frequencies were only 4 and 12 Hz, while without lubricant, it can reach 4, 6, 8, 12, 20, 28 and 36 Hz. In more detail, it can be seen in Figure 7.

The personal frequencies that were obtained in X, Y and Z axes of bearing with and without lubricant (oil) are illustrated in Table 1 and Table 2.

Table 1 shows that the personal frequencies were found in more than one position which were measured. It shows that the personal frequency appeared due to the existence of global vibration modes. It happened in the personal frequencies of between 4 Hz and 12 Hz.

In contrast, the bearing without lubricant showed that the global vibration modes appeared at personal frequencies of 4 Hz, 6 Hz, 12 Hz, 20 Hz, 28 Hz, and 36 Hz. The more detailed data are presented in Table 2.

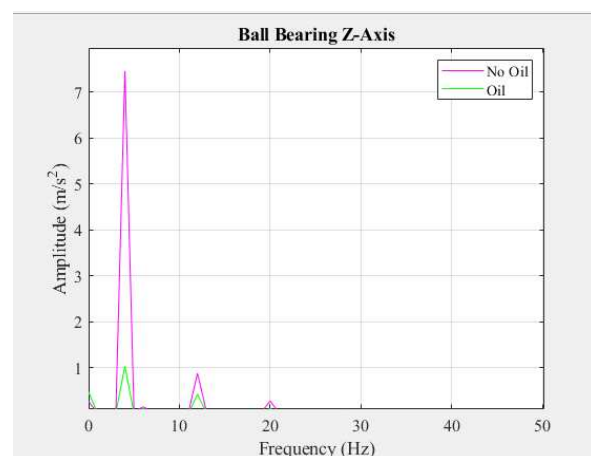


Figure 7. Graph of FRF measurement at the Z-axis

Table 1. Personal Frequency with the application of oil

Personal Frequencies	Axes		
	X	Y	Z
1	4	4	4
2	-	6	-
3	-	8	-
4	12	12	12
5	-	14	-
6	-	16	-
7	-	20	-

Table 2. Personal Frequencies on the measurement without the application of oil

Personal Frequencies	Axes		
	X	Y	Z
1	4	4	4
2	-	6	6
3	-	-	8
4	12	12	12
5	-	14	-
6	-	-	-
7	-	16	-
8	20	20	20
9	-	23	-
10	-	-	-
11	-	25	-
12	28	28	28
13	-	32	-
14	-	-	-
15	36	36	36

By utilizing the FRF method, it can be identified that the ball bearing's personal frequencies only appeared once while the global frequencies were found more than once.

The personal frequencies that always appear in X, Y and Z axes of bearing both with and without lubricant are 4 Hz and 12 Hz. In Figure 5, The lubrication of bearing will reduce the amplitude of vibrations that appear at the same of the personal frequencies. At the personal frequencies is 4 Hz, the amplitude runs into a reduction from 16.2 m/s² without lubricant to 0.81 m/s² with lubricant. Likewise, at the personal frequencies is 12 Hz, the amplitude runs into a reduction from 1.88 m/s² without lubricant to 0.377 m/s² with lubricant.

In Figure 6, the personal frequency is 4 Hz, the amplitude runs into a reduction from 2.925 m/s² without lubricant to 0.8 m/s² with lubricant. Likewise, at the personal frequency is 12 Hz, the amplitude runs into a reduction from 0.212 m/s² without lubricant to 0.1 m/s² with lubricant.

In Figure 7, the personal frequency is 4 Hz, the amplitude runs into a reduction from 7.45 m/s² without lubricant to 1.039 m/s² with lubricant. Likewise, at the personal frequency is 12 Hz, the amplitude runs into a reduction from 0.88 m/s² without lubricant to 0.425 m/s² with lubricant.

CONCLUSION

The ball bearing is obtained that personal frequencies always appear both before and after the addition of oil in 4 Hz and 12 Hz. It is indicating the existence of global vibration modes. The addition of lubricant can reduce the amplitude that appears before applying the lubricant at the same frequency in each part measured. The personal frequency that most frequently appears both before and after the addition of oil was at Y-axis.

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