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Identify damage to the mixer motor on the banbury machine using the vibration method

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Abstract: The Banburyy mixer machine is a tool used to mix materials, such as raw rubber, carbon and chemicals into a homogeneous mixture. The Banburyy engine component consists of a rotor. The rotor rotates as a result of the power transmission coming from the industrial motor, the power transmission uses a gearbox. Therefore, induction motors play an important role in Banbury engines. If damage to the induction motor is not detected early, it can result in more severe damage or even unusability. Therefore, in this paper we will analyze the characteristics of bearing damage using the Fast Fourier Transform (FFT) method and the Hilbert method on mixer motors. The bearings used in the mixer motor are SKF 6330/C3 bearings in the drive end position and SKF 6324 types in the non-drive end motor position. The measurement results were then processed using the Fast Fourier Transform (FFT) and Hilbert Transform methods. The rotation speed variations carried out in the test were 500 rpm, 1000 rpm and 1500 rpm. Bearing damage analysis uses a vibration method that is obtained based on characteristic frequency values that indicate the occurrence of damage, in the form of Ball Spin Frequency (BSF) on the ball, Ball Pass Frequency Outer (BPFO) on the outer track, Fundamental Train Frequency (FTF) that occurs on the cage, and Ball Pass Frequency Inner (BPFI) on the inner track, according to bearing specifications and motor shaft rotational speed. From the analysis results, it was found that in the SKF 6330/C3 bearing an amplitude appeared at a frequency close to the FTF value of 7.031 Hz along with its harmonic frequencies, whereas in the SKF 6324 bearing no frequency of damage appeared. This indicates that the bearing in the drive end position, namely the SKF 6330/C3 bearing, is thought to have experienced damage to the cage bearing (ball bearing cage). The action taken is to replace the bearing with a new one as soon as possible.

Keywords: Induction motor; vibration; bearing; fast fourier transform; hilbert transform

1. INTRODUCTION

The Banbury machine is a type of manufacturing machine that is used to mix raw rubber material with other chemical materials to make it homogeneous with the output product being a rubber sheet material called a compound. Compound is the basic material used to make tires. There are 4 main parts in the Banbury machine, namely up stream, mixer, roll die, and batch off. The mixer is a part that functions to mix and stir raw materials into homogeneous materials. This mixing process involves two rotors rotating at opposite speeds in a closed space. The two rotors are connected to a gearbox which is driven by an induction electric motor [1].

In the Banbury machine, the electric motor used is an ABB brand induction motor type AMI 500L4L BAFTS with a power of 2400 KW. This motorbike is built with 2 bearings, namely an SKF 6330/C3 type bearing in the drive end position and an SKF 6324 type bearing in the non-drive end position. Motor operation is carried out continuously for 24 hours. This will cause complex damage to the motorbike in the future. Monitoring by paying attention to the machine in action has so far not been successful, because the frequency conveyed by each machine component is very high and changes, making it difficult for researchers. to recognize it. Observing the state of a machine under rotating conditions often involves the use of measurable vibration signal processing methods. Because most systems operating in industry are in high-noise environments, some form of statistical averaging is usually required to analyze vibration signals [2]. If demolition is carried out, it will require a long investment and large costs. Furthermore, various strategies have been created to identify damage to engine components as soon as possible, without stopping engine operation [3].

Bearing parts are one of the cases for vibration inspection tests that must be carried out to check the condition for damage analysis. To see differences in vibration signals, there are several things that must be checked in this examination, for example, knowing how to demonstrate and break down vibration signals. Possible options for analyzing bearing damage can be seen from the results of differences in vibration signals after modeling the time domain vibration data into the frequency domain using the Fast Fourier Transform (FFT) method in the Matlab program [5]. Much research on vibrations has been carried out in the Lab. The vibe of Mercubuana University starts from automotive [6]-[9], damage to production machines and industrial equipment [10]-[13]

There has been a lot of research regarding bearing damage analysis in induction motors that has been carried out. One of them is bearing damage analysis through vibration signals [14]-[18]. The results of this research show that vibration signals in bearing components have special frequency characteristics that indicate the appearance of defects, in the form of Ball Spin Frequency (BSF), Fundamental Train Frequency (FTF), Ball Pass Frequency Inner (BPFI), and Ball Pass Frequency Outer (BPFO), according to the bearing specifications and motor shaft rotational speed [19]. However, this research was carried out only to determine the occurrence of defect frequencies by changing the vibration signal in the time domain to the frequency domain using the Fast Fourier Transform (FFT) method. Apart from that, this research cannot display non-linear vibration characteristics, so an analysis method is needed non-linear vibrations such as one of Hilbert's methods [20]-[22].

The aim of this paper is to study the characteristics of vibration signals used to detect bearing damage on Banbury mixer motors using the Fast Fourier Transform method and display the characteristics of non-linear vibrations in bearings using the Hilbert Transform method.

2. METHOD

In this research, the subject used was an ABB induction motor type AMI 500L4L BAFTS. This motor consists of 2 bearings, namely, an SKF 6330/C3 type bearing in the drive end position and an SKF 6324 type bearing in the non-drive end position. The SKF 6330/C3 bearing has specifications of pitch diameter (Pd) 234.5 mm, outer diameter (Do) 320 mm, inner diameter (Di) 150 mm, ball diameter (Bd) 62 mm, number of balls (Nb) 12, and contact angle (α) 00. Meanwhile the SKF 6324 bearing has geometric specifications of pitch diameter (Pd) 189.6 mm, outer diameter (Do) 260 mm, inner diameter (Di) 120 mm, ball diameter (Bd) 50 mm, number of balls (Nb) 12 pieces, and contact angle (α) 0^o.

At each bearing housing, vibration signal data was collected using a JFE MK-210 HE vibrationmeter measuring instrument. The measurement point is divided into two parts, namely the front bearing of the motor (drive end) and the rear bearing of the motor (non-drive end), as shown in Figure 1.

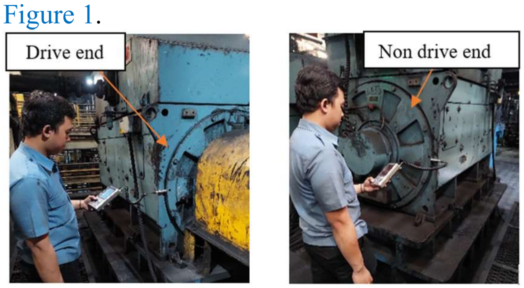

Figure 1. Measurements at the drive end and non-drive end positions

At each point, data is collected on 3 axes, namely the X axis, Y axis and Z axis, with rotation variations of 500, 1000 and 1500 Rpm. By using an inverter, the rotation variations are regulated and converted into frequencies, namely at frequencies of 8.3 Hz, 16.6 Hz and 25 Hz. The measurement data on the measuring instrument is then transferred to the laptop using a memory card. Next, the data transferred from the memory card will be analyzed using the FFT and Hilbert transform methods in the Matlab application.

3. RESULTS AND DISCUSSION

The testing process in this research is to determine vibration data and identify bearing damage on the 2400 kW Banbury mixer motor, by attaching accelerometer sensors to the front and back of the motor. Tests were carried out on the X, Y, and Z axes with variations of 500, 1000 and 1500 rotations.

The frequency value of 1x RPM in each rotation was 8.3 Hz at 500 RPM rotation, 16.6 Hz at 1000 RPM rotation, and 25 Hz at 1500 RPM rotation.

Detection of bearing damage is done by looking at the frequency of bearing damage that appears. The amplitude value does not always appear exactly at the theoretically calculated frequency, but can appear around that frequency. The following are the values for the frequency of damage to SKF 6330/C3 bearings and SKF 6324 bearings calculated theoretically, in Table 1 and Table 2.

Table 1. Frequency of damage to SKF 6330/C3 bearings

Shaft Rotor (rpm)	BPFI	BPFO	BSF	FTF
500 rpm	68,57 Hz	31,03 Hz	6,86 Hz	2,59 Hz
1000 rpm	137,14 Hz	62,06 Hz	13,72 Hz	5,17 Hz
1500 rpm	206,53 Hz	93,47 Hz	20,67 Hz	7,79 Hz

Table 2. Frequency of damage to SKF 6324 bearings

Shaft Rotor (rpm)	BPFI	BPFO	BSF	FTF
500 rpm	62,93 Hz	36,67 Hz	11,59 Hz	3,06 Hz
1000 rpm	125,87 Hz	73,33 Hz	23,17 Hz	6,11 Hz
1500 rpm	189,56 Hz	110,44 Hz	34,90 Hz	9,20 Hz

3.1 Data before repair

After obtaining theoretically calculated values for the frequency of damage to each bearing, as shown in Table 1 and Table 2, then data analysis was carried out using the frequencies that appeared on the vibration graph. First, we will analyze the frequency graph data on the front bearing or drive end, namely the SKF 6330/C3 bearing with rotation variations of 500, 1000, and 1500 RPM at the X, Y, Z axis points. As shown in Figure 2, Figure 3, and Figure 4.

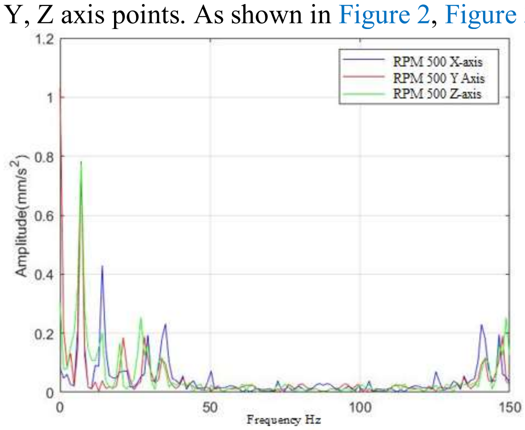

Figure 2. Graph of vibration signal of X,Y,Z axes at 500 RPM

Figure 2 is a graph of the results of measuring vibration signals on the X, Y, Z axes at a speed of 500 RPM. It can be seen that the average amplitude that appears is 0.2 mm/s^2 with the highest value being 0.78 mm/s^2 and there is no amplitude that appears close to the damage frequency value.

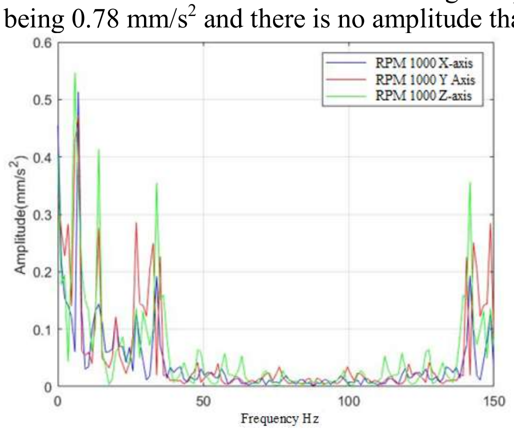

Figure 3. Graph of vibration signals for the X,Y,Z axes at 1000 RPM

Figure 3 is a graph of the results of measuring vibration signals on the X, Y, Z axes at a speed of 1000 RPM. It can be seen that the average amplitude that appears is 0.29 mm/s^2 with the highest value being 0.55 mm/s^2 and there is no amplitude that appears close to the damage frequency value.

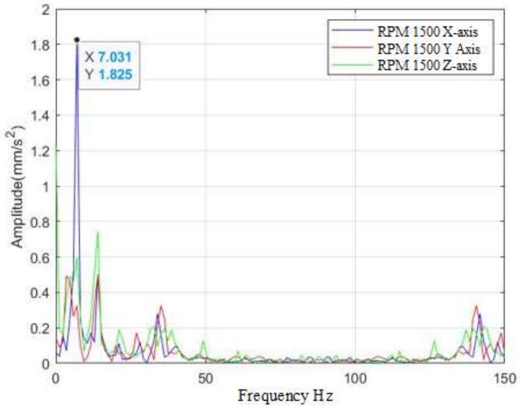

Figure 4. Graph of vibration signals for the X,Y,Z axes at 1500 RPM

Figure 4 above shows that the resulting amplitude value increases as a result of the experimental variations given. In this case it can be seen that at 1500 rpm (X Axis Point = 7,031 Hz) the amplitude graph is closest to the bearing damage limit compared to the other axis points.

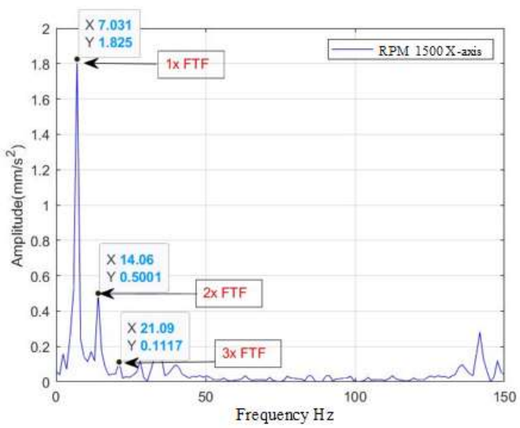

Figure 5. X-axis vibration signal graph at 1500 RPM

Figure 5, the vibration signal graphic data at point Theoretically, the calculation of 1xFTF is 7.79 as shown in Table 1, while the harmonics which include 2xFTF and 3xFTF are 14.06 Hz and 21.09 Hz respectively. This indicates that the bearing in the drive end position, namely the SKF 6330/C3 bearing, is thought to have experienced damage to the cage bearing (ball bearing cage). So the action taken is to replace the bearing with a new one as soon as possible.

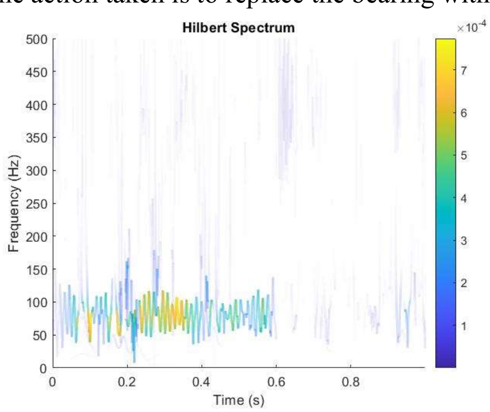

Figure 6. X-axis Hilbert spectrum at 1500 RPM

Figure 6, is a graphic spectrum of the X axis at a speed of 1500 RPM which was analyzed using the Hilbert method with the Matlab application. It can be seen that the average frequency of the energy produced is 98 Hz, and it can also be seen that the resulting graph is a non-linear graph which indicates that at that point damage has occurred.

Frequency graph data analysis was carried out on the rear or non-drive end bearings, namely SKF 6324 bearings with rotation variations of 500, 1000 and 1500 RPM at the X, Y and Z axis points. As shown in Figure 7.

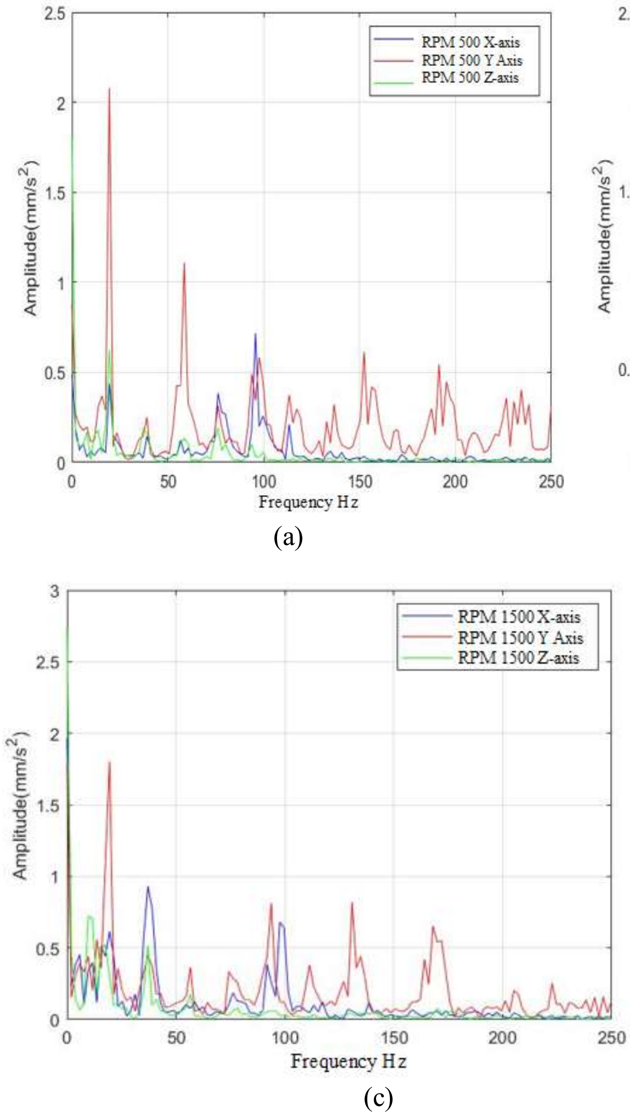

Figure 7. (a). Vibration signal graph at 500 RPM, (b). Vibration signal graph at 1000 RPM, (c). Vibration signal graph at 1500 RPM

From the results of the graphic analysis in Figure 7, overall it can be seen that several amplitudes with high values appear. However, from this amplitude there are no frequencies that appear at the theoretically calculated bearing damage frequency as in Table 2, or around that frequency. This is because not all high amplitudes indicate that damage has occurred to the bearing. The bearing damage frequency value is estimated to only appear around the damage frequency value that has been calculated theoretically. This high amplitude may be caused by dust entering the bearing components, causing friction between the steel balls or other noise. Therefore, it can be concluded that the overall condition of the rear or non-drive end bearings, namely SKF 6324 bearings, is still in good condition and there has been no damage.

3.2 Data after repair

After repairing the replacement of the front bearing, namely the SKF 6330/C3 bearing, the motorbike was set up again as before. Then the vibration data is taken again using the same method. The following are the results of vibration data after bearing replacement repairs:

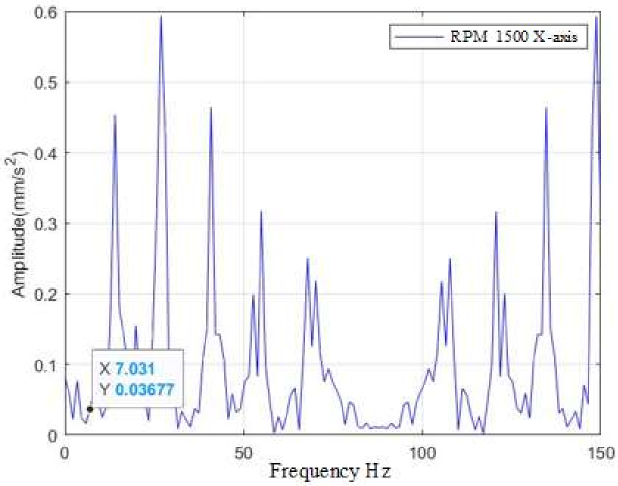

Figure 8. X-axis vibration signal graph at 1500 RPM after repair

The picture above you can see vibration data on the X axis at 1500 rpm after repair. As a result of carrying out repairs to the bearing replacement, the peak amplitude value experienced a very significant decrease from the initial amplitude before the repair of 1.825 Hz (seen in Figure 5) to a value of 0.0367 Hz (Figure 8). And at the values of 1x FTF, 1x BSF, 1x BPFO, and 1x BPFI, by observing Table 1, nothing is close to the bearing damage limit. So that the repair process in the form of bearing replacement is successful.

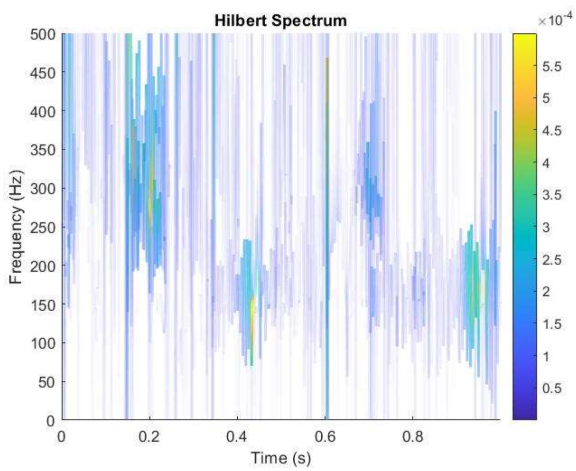

Figure 9. X-axis Hilbert spectrum at 1500 RPM after improvement

Figure 9, a graph of the x-axis Hilbert spectrum is shown at a speed of 1500 with conditions after bearing replacement. A spectrum pattern can be seen that is different from the condition before repair (Figure 1), where the resulting graph is a linear graph which indicates that at that point there is no indication of damage.

3 CONCLUSION

The PCP on the surface of RCC specimens is greatly influenced by the amount of CEG layers. The PCP tends to rise with the number of CEG layers applied. The number of CEG layers and STS have a favorable correlation. The STS of RCC specimens rises with the application of additional CEG layers. On the other hand, PCP rises in tandem with an increase in CEG layers. This suggests that improving tensile strength and increasing damage patterns are mutually exclusive. Increasing the amount of CEG layers will result in an increase in specimen crack patterns but will also improve tensile strength. This finding implies that a comprehensive analysis is needed to choose the appropriate number of CEG layers in a design or application, taking into account both increasing tensile strength and allowing for a certain amount of damage in the form of fracture patterns.

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