

INSPECTING A BUMP TEST IN THE MAINTENANCE OF A 1200-CC DAIHATSU SIGRA DISC BRAKE

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Abstract -- A motorized vehicle needs a system that can not only reduce the speed and stop the vehicle but can also serve as a safety tool and ensure a safe distribution which is called a braking system. It is important to conduct maintenance of a disc brake that is part of the braking system. Therefore, this study developed a technique to inspect a 1200-cc Daihatsu Sigra disc brake through the vibration measurement using a bump test. A disc brake is comprised of three fields: A (located next to the center of the disc brake), B (in the middle diameter), and C (in the outer diameter of the disc brake). Each field has ten measurement points. This study showed that disc brake damages occurring in field A were at points 1-3, and they appeared after the 250-Hz frequency. Meanwhile, in fields B and C, at all measurement points, there were no changes taking place at any frequency.

Keywords: Braking system; Disc brake; Bump test; Measurement point; Frequency

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INTRODUCTION

In a motorized vehicle, there should be a system that can not only reduce speed and stop the vehicle but can also serve as a safety tool that ensures a safe distribution. The system is called a braking system. Antara (2018) stated that driving safety required braking system that allowed the vehicle to stop at anywhere properly and safely in various conditions.

The combustion process will produce a force that leads the wheel to rotate affecting the disc brake to rotate too. When the component moves, there will be an inertial force and another force coming from the power transmission. These forces will cause the disc brake to vibrate. In a well-designed disc brake, the generated vibration is lower; however, the long period of usage will cause the vibration amplitude to be larger which will affect the disc brake's condition. The vibration generated signals have a certain frequency spectrum and certain vibration characteristics. The vibration that appears on the disc brake has a huge effect, which in turn will shorten its operating life. Ghazaly et al (2012) conducted a study on commercial disc brakes to determine their noise by using an FEA modeling and testing capital analysis. Joo et al. (2018) conducted numerical simulations and experiments on the contact

stiffness distribution and surface roughness of materials by using a Complex Eigenvalue Analysis (CAE). The noise was generated in a Disc brake resulting from the transition from a static to dynamic conditions or the other way around. Measurement of frequency response was more often used as a step to eliminate or reduce the disc brakes on experimental brake results than that on CAE according to a study conducted by Ghatwai et al (2016).

This study was aimed at conducting preventive maintenance on disc brakes using an excitation force method applied to a bump test. A bump test is a fast and economical means designed for engine vibration and structure modes. A "bump" test (or collisions) is the best way to ensure that destructive end-wall vibrations do not occur in new engines, and is usually carried out on turbine generators (Kapler et al., 2014). Delprete et al. (2010), who performed the frequency response function test in the powertrain on a cylinder engine using two excitation style methods: Impact Hammer and Exciter. The exciter is more suitable because of its ability to identify several modes with a smaller size allowing structural patterns to form as a deformation capital on a single-cylinder engine component. Pimentel-Junior et al. (2016) detected cracks in the shaft

using a bump test method. [Bhadgaonkar \(2017\)](#) research on differences in disc brake rotor frequency using Finite Element Analysis so that it can predict crack depth. The University's Mercuru Buana vibration laboratory has also conducted disc brake research using the bump test method. In that study there are personal frequencies that appear in more than one measurement point. The result shows the existence of a global vibrat mode. The global vibrat mode occurs at personal frequencies of 40 Hz, 53 Hz, 62 Hz, 116 Hz, 123 Hz, 304 Hz and 557 Hz ([Efendy et al, 2019](#)). This method was being carried out on 1200-cc Daihatsu Sibra disc brakes made in Indonesia with the hydraulic system, as [Fig. 1](#) showed us. A bump test procedure includes the measurement of the Frequency Response Function (FRF), Capital Excitation Technique, and Estimated Capital Parameter. In this study, we discussed a bump test by a measurement.

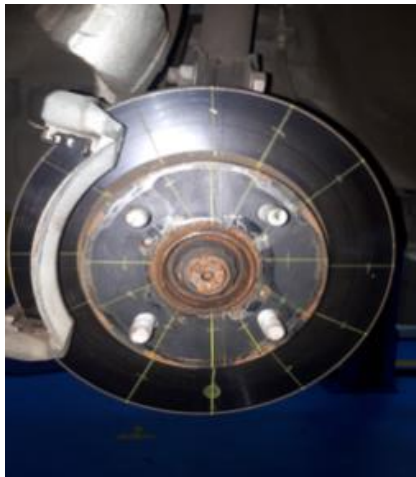


Figure 1. Sibra's Disc Brake (Repair manual of Daihatsu Sibra, 2016)

The FRF measurements have been carried out by several researchers such as [Masahiro \(2015\)](#) comparing the FRF method to multibody dynamic that showed a similar trend in bolt stiffness, both at the amplitude and the phase occurring in the steering system. The FRF method was also employed to determine the comfort level of passengers related to the intensity of vibrations in the bus frame ([Dahil et al., 2016](#)). The FRF is the basis of measurement to find out the dynamic characteristics that exist in a mechanical structure. Experimental capital parameters (frequency, attenuation, and shape mode) can be obtained by measuring the FRF ([Bilošová, 2011](#)). The FRF method was also employed to detect the level of damage or cracks occurring in gears ([Mohamad et al., 2016](#)). [Chen, et al. \(2017\)](#) studied the effect of a friction coefficient between tires and road to the

steering system using the FRF method, and then he validated the experiment and simulation. The performance of a vehicle attenuation measured by the FRF method obtained from the excitation force was then compared to the vibration obtained from the speed level with measurements at the same point ([Saha, 2017](#)). Through Fourier transforms, the data obtained from the measurement and transformation is in the frequency domain ([Broch, 1984](#)). [Subekti \(2018\)](#) tested the dynamic characteristics of a cylindrical piston motor, and they resulted in data about global vibration mode frequency. [Chen et al. \(2017\)](#) studied the effects of a friction coefficient resulted from tires and road to the steering system with the FRF method, which was then validated between the experiment and simulation. The performance of a vehicle attenuation measured by the FRF method obtained from the excitation force was then compared to that of the vibration obtained from the speed level with measurements at the same point ([Saha, 2017](#)). A nonlinear Identification using FRF method and analysis using wavelet packet decomposition was made by [Subekti et al. \(2018\)](#). In addition to the application on the vehicle, FRF measurement was also employed to detect damage to a structure ([Homaei et al., 2015](#)) and to detect cracks occurring in a beam ([Lin, 2015](#)). This study was aimed at developing a technique to inspection a 1200-cc Daihatsu Sibra Disc brake through vibration measurement using a bump test.

MATERIAL AND METHOD

The Frequency Response Function (FRF) is a transfer function, expressed in the frequency domain. It is a complex function with real and imaginary components. It may also be represented in terms of magnitude and phase. FRFs can be formed from data measurement or analytical functions. The FRF expresses a structural response to the force applied as a function of frequency. Responses can be given to the displacement, velocity or acceleration. The relationship between time and response is depicted in [Fig. 2](#).

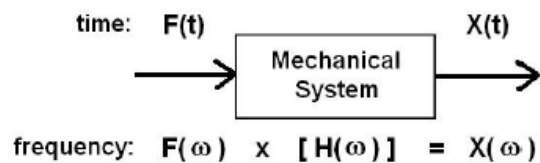


Figure 2. Block Diagram of an FRF

$F(\omega)$ is the input force as a function of the frequency of the angle w . $H(\omega)$ is the transfer function. $X(\omega)$ is the displacement response function. Each function is a complex function and

can be represented in the forms of magnitude and phase. Each function is thus spectral. There are many types of spectral functions, yet, for a simple reason, we should consider each of the Fourier transforms.

Normally, squeal brakes occur in a 1-to-20-kHz frequency. The squeal is a complex phenomenon because of its strong dependence on many parameters on the one hand, and because of mechanical interactions in the brake system on the other hand. Thus, the Frequency response, in this case, is deemed to be between 1 to 16,500 Hz.

The damage that occurred on the disc brake was identified by measuring the FRF. The excitation force given to the disc brake was a bump test which was then measured using a vibration analyzer. The bump test style was applied to the surface of the disc brake in a vertical or perpendicular direction. The response of vibration measured was carried out at three points, namely point A (located in the disc brake diameter and in line with the sensor and bump test, as shown in the picture), field B (located in the middle of the diameter line), and field C (located in the outer diameter line which is also the FRF point). Each field consisted of ten points, as shown in Fig. 3.

The selection of measurement points was conducted as a follow-up study of the characteristics of vibrations occurring in those three regions, especially on the disc brake component, when the disc brake worked in a silent condition and received a vibration input. When measuring the FRF in this study, we applied frequencies ranging from 1 to 1000 Hz to identify the range of frequency which could be used for testing purposes. Fig. 3 showed photos of the tests set to obtain the experimental data. A bump test was carried out on the disc brake, specifically in part C, which was then read by the vibration analyzer. The data obtained from the measurement were then analyzed using MATLAB.

Fig. 4 showed arrangements of FRF measurement devices and the types of equipment used in the research. The significant component used is an accelerometer. The accelerometer used in this case is the piezoelectric accelerometer made by Rion Japan Corporation type CCLD type, PV-571. Accelerometer served to measure response vibration. The component has a 100-Hz-span frequency with a 1600-line analysis using a linear function window and actual sensitivity Num 510 and actual sensitivity magnify x 0.01.

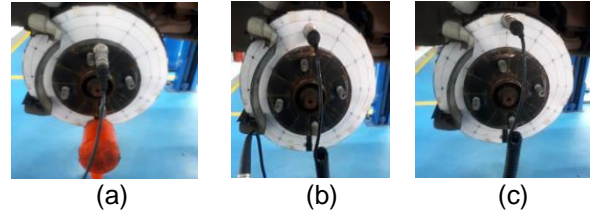


Figure 3. The measurement of the FRF

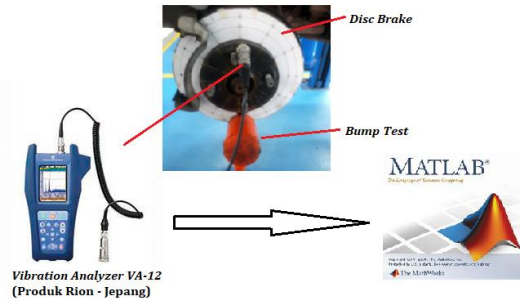


Figure 4. Photo of Set-up testing of FRF

The test object in this study was a Daihatsu Sibra brake disc made in Indonesia as shown in Fig. 5, with specifications as follows: outside diameter with 234 mm, inner diameter with 148 mm, disc thickness with 44 mm, tightening point with four positions and air conditioner.

RESULTS AND DISCUSSION

By using Fast Fourier Transform (FFT), we found out that that at points A10-A10, the global frequencies in a good condition were 40 Hz, 53 Hz, 116 Hz, 125 Hz, 304 Hz, and 504 Hz respectively, while in a damaged condition, the global frequencies were 35 Hz, 48 Hz, 53 Hz, 125 Hz, 503 Hz, and 504 Hz respectively. The analysis of damages on the Disc Brake was conducted by comparing the frequency, which was shown in the forms of FRF as shown in Fig. 5 of the Log-log plot of the same FRF.

Fig. 5(a) showed that in a damaged condition, the initial frequency appeared to be 70 Hz with an amplitude amounting to 19.67 m/s². In good condition, the frequency appeared to be 80 Hz with an amplitude amounting to 25 m/s². In a damaged condition, the initial frequency appeared to be 70 Hz with an amplitude amounting to 18 m/s², while in good condition, the frequency appeared to be around 80 Hz with an amplitude amounting to 13 m/s² as shown in Fig. 5(b).

In Fig. 5(c), there appeared to be no new frequency, but there was a frequency shift in the disc brake in a damaged condition. In addition, it was found out that the frequency in the damaged condition was around 70 Hz with an amplitude amounting to 33 m/s². While the frequency in good condition appeared to be 80 Hz with an amplitude amounting to 15 m/s², this was the same as what

happened at point A6 as shown in Fig. 5(d); moreover, then frequencies in good and damaged conditions occurred at an 80-Hz and 70-Hz frequencies where the amplitudes were 15 m/s² and 37 m/s².

In field B, at point B1, there was no new frequency, but there was a frequency shift where the disc brake was in a broken condition. By using

Fast Fourier Transform (FFT), we found out that at points B1-B10, the global frequencies in a good condition that we found were 40 Hz, 53 Hz, 62 Hz, 116 Hz, 125 Hz, 504 Hz, and 566 Hz respectively, whereas in a damaged condition, they were 35 Hz, 48 Hz, 53 Hz, 125 Hz, 502 Hz respectively.

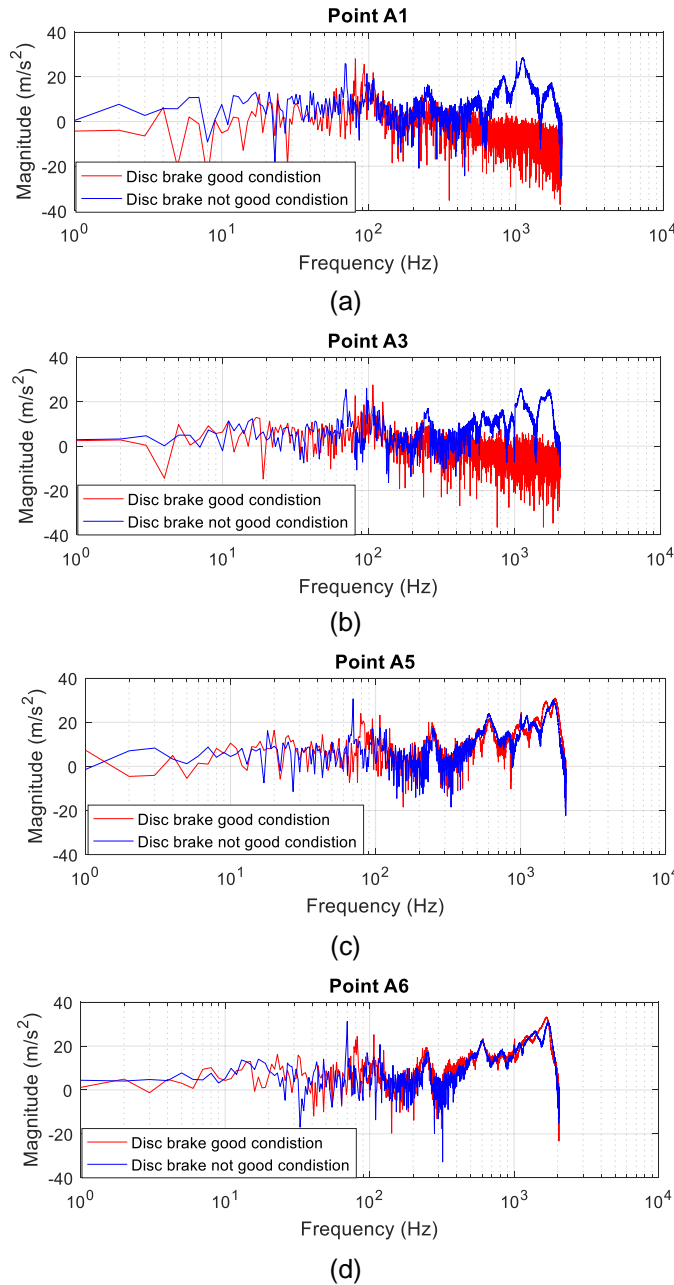


Figure 5. FRF measurement in field A

Fig. 6(a) showed that in a damaged and good conditions, the initial frequencies appeared to be 70 Hz and 80 Hz with the same amplitude amounting to 28 m/s². Fig. 6(b) showed that at point B3, there was a frequency shift like what happened at point B1. We found out that in a

damaged and good condition, there were no changes in the frequency appearance; however, the amplitude in the damaged condition changed the amplitude height amounting to about 32 m/s², while the good condition experienced a decrease in amplitude amounting to 19 m/s².

Fig. 6(c) showed a relation between the frequency and the amplitude. The amplitude in good condition amounted to around 10 m/s², and the amplitude in bad condition amounted to around 24 m/s²; meanwhile, the frequencies in both conditions remained the same. They did not change. Then, there appeared to be no new

frequency, but there was a frequency shift for the disc brake in a damaged condition like what happened at that of point A6 as shown in Fig. 6(d). In Fig. 6(d), the frequencies in good and damaged conditions were 80 Hz and 70 Hz respectively, while the amplitudes were 16 m/s² and 38 m/s² respectively.

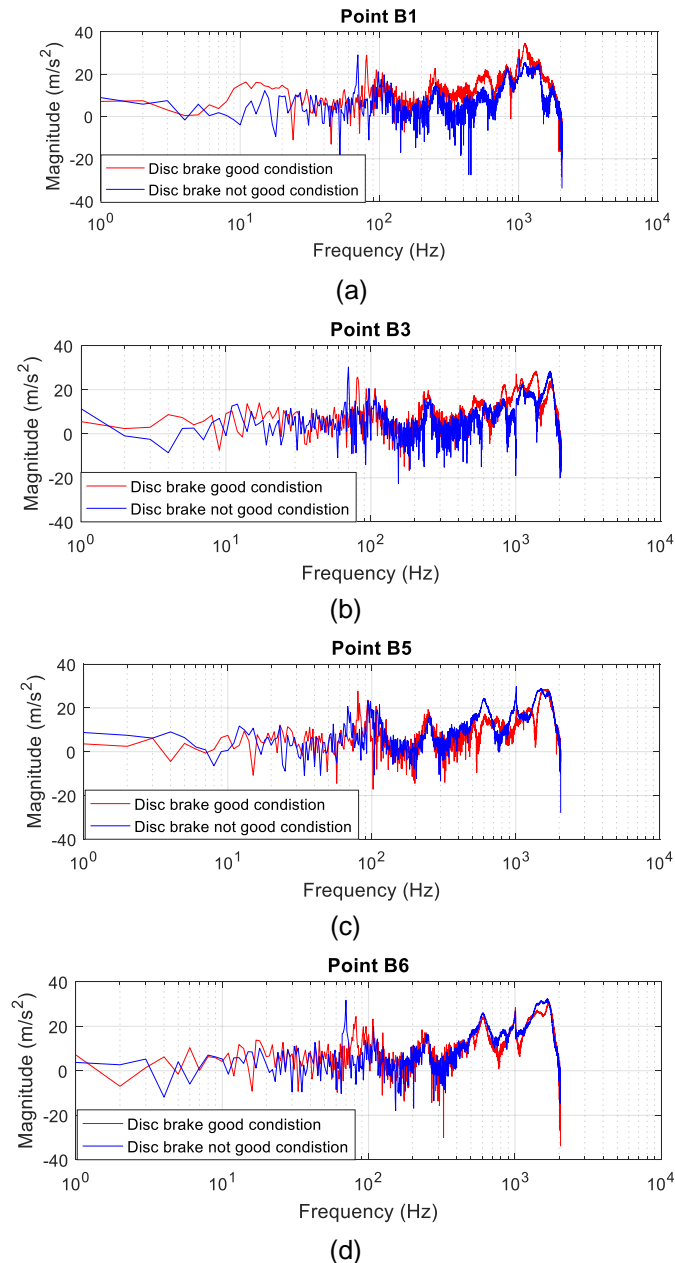


Figure 6. FRF measurement in field B

In field C, by employing the Fast Fourier Transform (FFT), we found out that at points B1-B10, the global frequencies in good condition were 40 Hz, 53 Hz, 62 Hz, 107 Hz, 116 Hz, 123 Hz, 253 Hz, 257 Hz, 302 Hz, 306 Hz, 406 Hz, 411 Hz, 504 Hz, 557 Hz respectively. While in a damaged

condition, the global frequencies were 35 Hz, 48 Hz, 53 Hz, 124 Hz, 128 Hz, 211 Hz, 289 Hz, 413 Hz, 504 Hz, 557 Hz respectively. Fig. 7(b) showed that at point C3, there was a frequency shift like what happened at point C1.

Fig. 7(a) showed that in a damaged condition, the initial frequency appeared to be 70 Hz with an amplitude amounting to 19.67 m/s², while in good condition, the frequency appeared to be 80 Hz with an amplitude amounting to 25 m/s².

In a damaged condition, the initial frequency appeared to be 70 Hz with an amplitude amounting to 18 m/s², while in good condition, the frequency appeared to be around 80 Hz with an amplitude amounting to 13 m/s² as shown in Fig. 7(b).

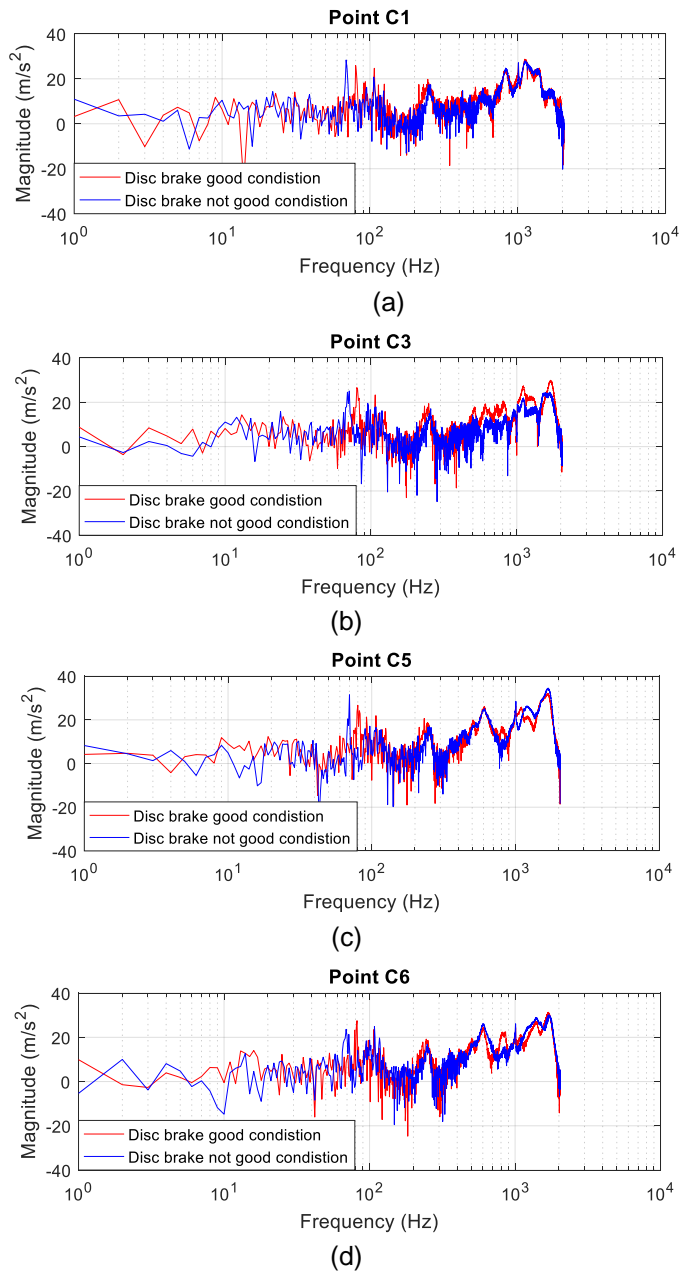


Figure 7. FRF Measurement in field C

In Fig. 7(c), there appeared to be no new frequency, but there was a frequency shift for the disc brake in a damaged condition. Moreover, we found out that the frequency in a damaged condition was around 70 Hz with an amplitude amounting to 33 m/s². While the frequency in good condition appeared to be 80 Hz with an amplitude

amounting to 15 m/s² like what happened at point C6 Fig. 7(d) showed us. Furthermore, then frequencies in a good and damaged conditions were 80 Hz and 70 Hz respectively, while the amplitudes were 15 m/s² and 37 m/s² respectively.

In field C, at points C5 and C6, there were no significant changes happened in the disc brake in good condition and a normal condition.

By employing an FRF method, we compared both the good and bad conditions of the disc brake. Moreover, it is required that we conduct a further study aimed at measuring vibration characteristics when the disc brake was subject to loading and identifying the effects of speed.

CONCLUSION

The results of the bump test on the disc brake showed that Disc Brake damage occurred on almost throughout the Disc Brake surface, especially at points A1, A2, and A3, with the appearance of new frequencies that occurred at both points. There was also a personal frequency that appeared at more than one measurement point. The result showed the existence of a global vibrational mode. The global vibrational mode occurred at several personal frequencies, namely 70 Hz, 90 Hz, 100 Hz, 120 Hz, 130 Hz, 250 Hz, 600 Hz, and 1000 Hz.

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