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Comfort Evaluation on the Drivers using Transfer Path Analysis

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ABSTRACT

The vibration transmissibility to the seated drivers has important influences on the human body comfort, health, and safety. To establish the dynamic equation for precondition of modal analysis, Newton-Euler method is usually used as the common study. However, this method is difficult to determine generalized coordinates belong to which natural frequency for linear system. On the other hand, by using transfer path analysis (TPA) as a new technique, it is possible to determine generalized coordinates of the vibration for the car machine affected the vibration of the car. The purpose of this research is to quantify the comfort level for driver's whole body vibration in passenger car based on ISO 2631-1 and to know the level of vibration on the car using transfer path analysis. This research tries to characterize the vibration effect of different sources (variation of the car machine) at idle condition. Human as the subjects are involved in this test. The results of this study show that low frequency vibration of the seat drivers affected to the ride comfort. Through TPA, the human body comfort can be evaluated using operational force by multiplying the natural frequencies of the car with the operational acceleration.

Keywords: Comfort Evaluation on the Drivers, ISO 2631-1, Transfer Path Analysis, Whole Body Vibration.

1. INTRODUCTION

Car becomes one of the transportation that widely used by people especially in Indonesia. Based on data from the National Statistics Agency of Indonesia, the number of cars has owned by Indonesian people in 2017 reached 15.493.068 units (1). Along with the increasing use of cars as a public transportation, the factors of comfort on car drivers are being considered and become an important thing in automobile system.

Several investigations about the effect of the vibration on the driver seated position have been reported. Van Niekerk et al estimated dynamic seat comfort prediction using the Seat Effective Amplitude Transmissibility (SEAT) for 16 different automobile seats (2). Rakheja et al investigated the interactions of the seated occupants with an inclined backrest at the two driving-points using apparent mass (3). Investigation by using apparent mass was also done by Subashi et al for nonlinear subjective and dynamic responses of seated subjects exposed to horizontal whole-body vibration (4). Coyte et al presented a holistic literature survey about seated whole body vibration analysis, technology, and modelling (5). The transmissibility measures, impedance, estimation of frequency response function, and model development were surveyed as the data processing techniques for the motion of the seated body.

Transfer path analysis (TPA) is a method to solve the vibration problems by identifying their vibration sources and the contributions of each transfer path to the receiver inside the vehicle (6). TPA is commonly used in the automotive industry (7,8,9) especially for passenger cars (10,11). This research tries to implement TPA as a method to investigate the vibration behaviour on the car especially for the driver's seat comfort in automobile aspect based on ISO 2631-1. The effect of car engine as the source of vibration on seat comfort were investigated with variation of the car engine (car engine from classic car and new car) at idle condition. Experimental modal analysis by impact test is used to determine the natural frequencies as the dynamic characteristics of the car. The natural frequencies from impact test is multiplied with operational acceleration which is equal to operational force as transfer path analysis. Transfer path analysis to evaluate the vibration characteristics of the

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car related to the driver seat comfort will be discussed in section 4.1.

2. THEORETICAL ASPECTS

2.1 Whole Body Vibration According to ISO 2631-1

The ISO 2631-1 (1997) standard is usually used as the reference for vibration effects to the human body. It defines the quantified methods for whole-body vibration (WBV) with respect to vibration comfort, health, and motion sickness. The ISO 2631-1 focus on vibration comfort measured over longer time since the seat supports the driver in sitting posture related to whole body vibration. The human body is complex mechanical system and more sensitive to certain frequencies so that the frequency weighting is needed due to this fact and also according to the ISO 2631-1 standard (12). Frequency weighting of signal needs to be divided into its frequency content. Once the frequency content is calculated and the amplitude is multiplied with desirable weighting so it will change the amplitude depending on the frequency (13). The frequency weighting for human body comfort reactions to vibration environments can be seen in table 1.

Table 1 – Comfort reactions to vibration environments (12)

Weighted rms acceleration (m/s^2)	Subjective response
Less than 0.315 m/s^2	Comfortable
0.315 m/s^2 - 0.63 m/s^2	a little uncomfortable
0.5 m/s^2 - 1 m/s^2	fairly uncomfortable
0.8 m/s^2 – 1.6 m/s^2	Uncomfortable
1.25 m/s^2 – 2.5 m/s^2	very uncomfortable
Greater than 2 m/s^2	extremely uncomfortable

When investigating the whole body vibration excitation, the ISO 2631-1 standard recommends several measurements which are appropriate to use. The measurable quantities use weighted root mean square acceleration (a_w) and vibration dose value. The weighted rms acceleration is the most common used to describe the acceleration time signal with a single value (14) and it is expressed in metres per second squared (m/s^2). The calculation of the weighted rms acceleration and the vibration total value are defined in equation 1 and 2.

$$a_w = \left[\frac{1}{T} \int_0^T a_w^{\,2}(t)dt \right]^{\frac{1}{2}} \tag{1}$$

where a_w is the weighted acceleration as the function of time (in m/s^2), and T is the duration of measurement (in second). ISO 2631-1 states that the vibration total value of weighted rms acceleration should be applied if the vibration comfort become the aim that shown in equation 2

$$a_{w(total)} = (k_x^{\,2} a_{wx}^{\,2} + k_y^{\,2} a_{wy}^{\,2} + k_z^{\,2} a_{wz}^{\,2})^{1/2} \tag{2}$$

where $a_{w\,(total)}$ is the vibration total value, a_{wx} is the weighted rms acceleration in x-direction, a_{wy} is the weighted rms acceleration in y-direction, a_{wz} is the weighted rms acceleration in z-direction, k_x, k_y, and k_z are multiplying factors (multiplying factors are equal to 1 for comfort evaluation).

2.2 Transfer Path Analysis

Transfer Path Analysis (TPA) is used to estimate the vibration source in the dynamic system and to identify the vibration path for complex mechanical system (15). TPA defines the vibration response at selected target as a superpostion of vectorial contributions from a set of force inputs that excite the structure (16,17,18). TPA can be obtained by calculating a set of the operational force where the operational force (F) is calculated by multipliying the natural frequency of the structure borne path (H) and operational acceleration (a) in frequency domain (15,19). The natural frequency of the structure borne path is found by using impact hammer, shaker, or impact test. There are several ways for impact test such as dropping a steel sphere in the structure according to (20). While, the operational acceleration can be measured when the vibration source is turning on (for automobile system, it is usually from car engine). Calculation of the operational force of the system based on TPA technique

is defined in equation 3.

$$F^{op} = Ha^{op} \quad (3)$$

where F^{op} is operational force, H is the natural frequency of the structure, and a^{op} is operational acceleration.

3. METHOD

This study uses ISO 2631-1 standard as the guidelines for the measurement and evaluation of human exposure to WBV. The frequency range that used in this study is 0.8 Hz to 100 Hz related to whole body vibration. The car that used in the data collection are Toyota Kijang and Toyota Avanza with specification shown in table 2. The car engine condition and suspension both of these cars are assumed still good and in accordance with the initial specifications. The car suspension, seat suspension, and ambient temperature are neglected and not being a focus for this study. One seat pad accelerometer from Svantek type SV-38V and vibrometer Svantek SV-106 are used for the measurement of whole body vibration. Four accelerometers from PCB piezotronics are used with model 352C33 and they are connected to the data acquisition from National Instrument with model NI cDAQ-9178.

Table 2 – Specifiation of the car engine

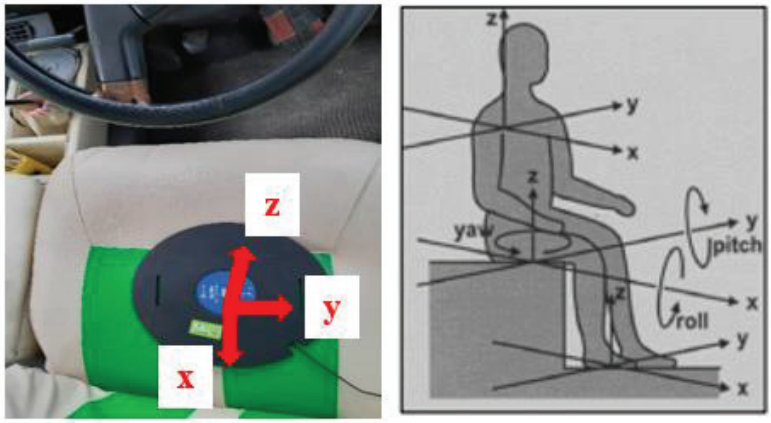
	Toyota Kijang	Toyota Avanza
Year production	1991	2018
Type of car	Super Kijang KF 40 short	Grand New Avanza 1.3L
Type of engine	Carburetor 4 cylinders in line, SOHC	4 cylinders, 16 valves, DOHC, dual VVT-i EFI
Volume of cylinder	1800 cc	1300 cc
Maximum power	73.028 PS/ 4600 rpm	95.5 PS/ 6000 rpm
Maximum torque	14.2 kgm/ 4600 rpm	12.3 kgm/ 4200 rpm

3.1 Subjects

Seven people (men) as the test subjects took part in this study with average age 25 years ± 6.5 (21 to 39 years old), the average height 173 cm ± 6.7 (163 cm to 180 cm), and the average weight 79 kg ± 20.6 (65 kg to 125 kg). The subjects were in good physical health and did not have any history of neurological, musculoskeletal (back pain and spinal pain), cardiovascular, and respiratory diseases. The subjects also did not used any controlled medicines and have not been submitted to recent surgery according to recommendations of (21).

3.2 Procedure

This study used one seat pad accelerometer for the driver's seat car for whole body vibration analysis. This seat pad accelerometer was connected to vibro-analyzer Svantek SV-106 and placed at the seat cushion as shown in figure 1(a). Three axes indicated on the seat pad should be coincided with the specified axes of vibration shown in figure 1(b). The measurement used translational movement as the vibration direction where the fore and aft direction is defined as the x-axis, lateral as the y-axis, and vertical direction as the z-axis (22). Four shear accelerometers were also used and connected to portable DAQ. We conducted the measurement on zero condition (car engine off) and idle speed condition (car engine on). These shear accelerometers were attached on the some selected node positions : near car engine, steering wheel, driver seat cushion, and foot floor (near driver position). The comfort reactions on the drivers are evaluated using the weighted rms acceleration and 1/3 octave band frequency analysis using equation 1 and 2. Transfer Path Analysis is used to analyze the vibration characteristics on the car. The natural frequencies were determined using modal analysis by dropping a steel sphere based on the experiment by Qi and Yin (20) and the value of operational accelerations were determined by measuring the acceleration when the car engine is turning on at idle condition and car placed on concrete floor. TPA method were used to identify and trace the flow of vibration on the car using operational force as in equation 3.



(a) (b)

Figure 1 – (a) Installation seat pad accelerometer on the driver's seat cushion, (b) specified axes of vibration (23)

4. RESULTS

4.1 Vibration Characteristics of the Car using Transfer Path Analysis

The operational acceleration of the car engine was measured at low frequency engine idle speed vibrations between 0 Hz to 100 Hz. The value of operational accelerations when the car engine running at idle speed as the vibration source has frequency of 49.61 Hz for Toyota Kijang and 59.57 Hz for Toyota Avanza as shown in figure 2. Table 3 shows the weighted rms displacement from car engine as the vibration source at idle speed to some receivers : steering wheel, driver seat cushion, and foot floor. We can see in table 3 that the highest rms displacement value both Toyota Kijang and Toyota Avanza were on the position of near car engine, followed by foot floor, steering wheels, and driver seat cushion. The energy of vibrations that transmitted from the car engine to some selected positions decreased their magnitudes. On the other words, the vibration that received by the driver that seat on the driver seat cushion has smaller amplitude compare to vibration amplitude near the car engine. The car engine rms displacement of Toyota Kijang was higher than Toyota Avanza. It was caused by the amplitude of operational vibration for Toyota Kijang has higher value 0.015 m/s^2 compared to Toyota Avanza which has amplitude 0.002 m/s^2 (see figure 2). It means the vibration amplitude of Kijang was 7.5 times higher than the vibration amplitude of Avanza.

Vibration characteristics of the Toyota Kijang and Toyota Avanza decreased their amplitudes from car engine position to some selected positions. This was caused by the car structure which has the damping coefficient of the material to reduce and absorb the vibration. Beside that, the system both of those cars have no resonance effect as proven in transfer path analysis method where resonance is a condition which occurs when a natural frequency is excited by an external forcing frequency. The natural frequencies both Toyota Kijang and Toyota Avanza which obtained through impact test by dropping a steel sphere are shown in figure 3. Equation 3 was used to determine transfer path analysis by multiplying the natural frequencies of the car with operational acceleration at idle condition. Figure 4 and 5 show the transfer path analysis where the forcing frequency of operational condition at idle condition did not coincide the natural frequencies both Kijang and Avanza in the frequency range from 0 Hz to 100 Hz. Therefore, the operational frequency at idle condition would not be amplified by the natural frequencies which can cause the frequency at which a machine will naturally amplify vibration.

Table 3 – RMS displacement values with car engine as the source of vibration at idle condition

Selected node positions	RMS displacement for Kijang	RMS displacement for Avanza
Near car engine	6.0 mm	5.6 mm
Steering wheel	2.9 mm	2.3 mm
Driver seat cushion	1.8 mm	1.1 mm
Foot floor	4.1 mm	5.5 mm

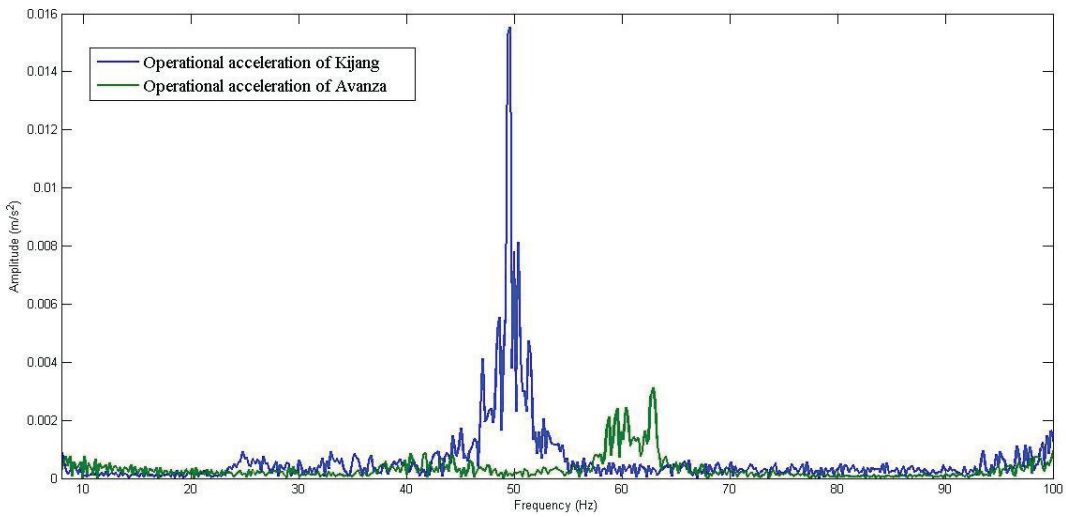


Figure 2 – Operational acceleration of the car engine as the vibration source at idle condition for Toyota Kijang and Toyota Avanza

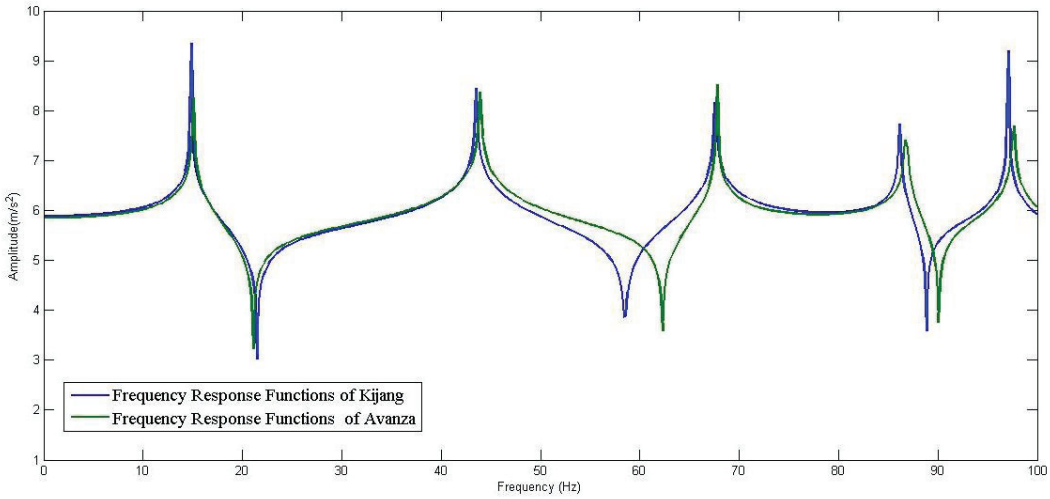


Figure 3 – The natural frequencies for Kijang and Avanza

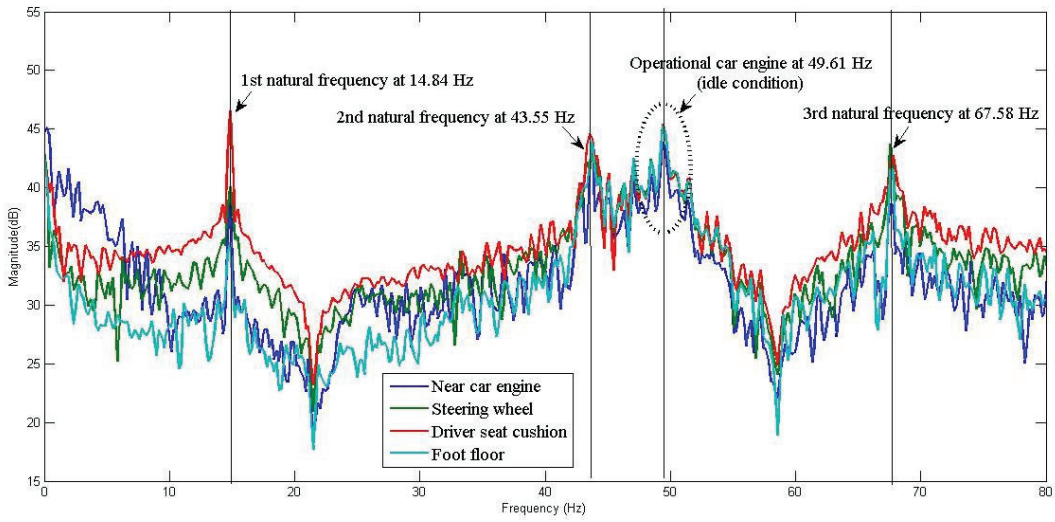


Figure 4 – Transfer path analysis at idle condition for Kijang

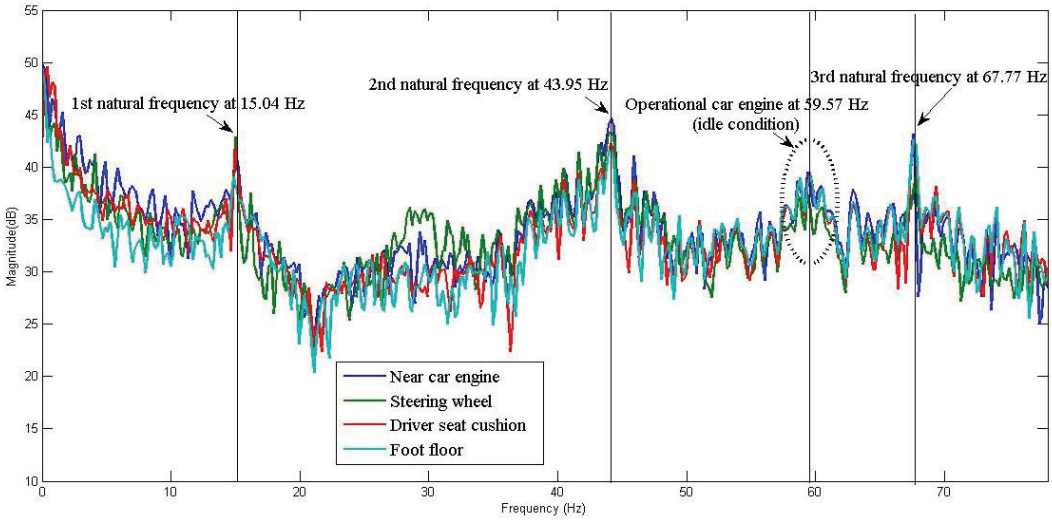


Figure 5 – Transfer path analysis at idle condition for Avanza

4.2 Evaluating Method of WBV using the Total Weighted RMS Acceleration

As shown in figure 6(b), we can see that the mean of the total weighted rms acceleration of Toyota Kijang a_w *(kijang)*= 0.1 m/s^2 has greater value compared to Toyota Avanza a_w *(avanza)*= 0.003 m/s^2 at idle speed condition althought both of them are still in the comfortable zone (< 0.315 m/s^2) based on comfort zone criteria by ISO 2631-1 stated in table 1. This was caused by engine car vibration of Toyota Kijang as the source of vibration has an amplitude 7.5 times higher compared to engine vibration from Toyota Avanza as discussed in chapter 4.1 (see figure 2). The Toyota Kijang has amplitude of 0.015 m/s^2 and the Toyota Avanza has amplitude of 0.002 m/s^2. Beside the car engine, the higher value of the total weighted rms acceleration from Toyota Kijang was caused by the vibration transmitted to the seat's driver from the car engine which has greater rms displacement value compared to the Toyota Avanza as stated in table 3 where Toyota Kijang has displacement of 1.8 mm and Toyota Avanza has displacement of 1.1 mm on the driver seat cushion position. The driver seat cushion was used to place the seat pad accelerometer sensor for whole body vibration. Therefore, the vibration displacement and vibration amplitude of the Toyota Kijang were greater than the Toyota Avanza at idle condition. We can observe in figure 6(d) that the VDV values at idle condition both Kijang and Avanza were less than 9.1 m/s$^{1.75}$ for Exposure Action Value (EAV) and less than 21 m/s$^{1.75}$ for Exposure Limit Value (ELV) which meant it was safe and healthy from vibration exposure that caused potential health risks to the drivers according to the European Directive standard (24).

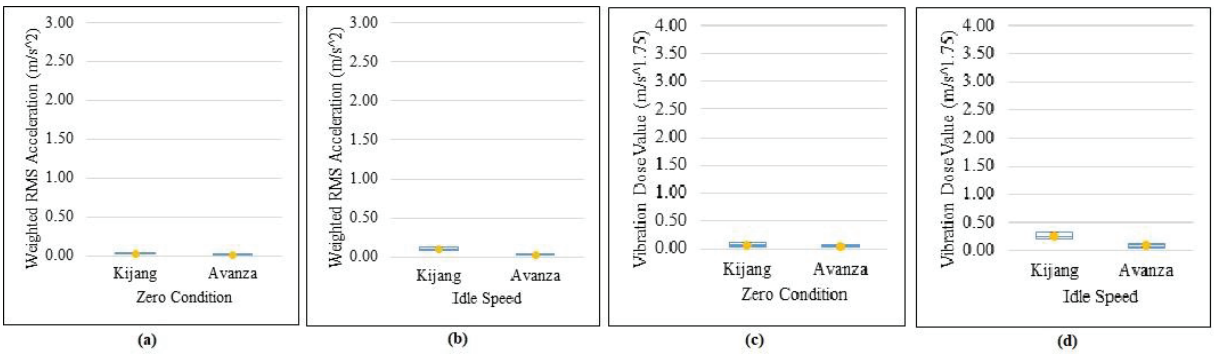


Figure 6 – Comparison between Kijang and Avanza for the total weighted rms acceleration on : (a) zero condition (b) idle, and vibration dose value on : (c) zero condition (d) idle

4.3 Evaluating Method of WBV using 1/3 Octave Band Frequency Analysis

According to ISO 2631-1, the frequency weighting that used for the measurement of whole body vibration are principal weightings in one-third octaves (*Wd* for the x and y directions, and *Wk* for the z direction). These frequency weightings were applied for seated driver on the effect of vibration on comfort with multiplying factor *k*=1 for all of the directions on supporting seat surface vibration. The

dominant frequency was identified as the 1/3 octave band frequency in the range from 0.8 Hz to 100 Hz associated with the highest frequency weighted acceleration values. The relation between frequency weighted rms acceleration and 1/3 octave band frequency in the vertical direction (z- axis) were shown in figure 7.

We can observe in figure 7 that idle condition has higher value compared to zero condition both Kijang and Avanza. Generally, Toyota Kijang has higher amplitude of the weighted rms acceleration as the comfort parameter compared to Toyota Avanza for the driver's whole body vibration. Kijang has the highest amplitude at frequency 20 Hz with the weighted rms acceleration value 0.155 m/s^2 and Avanza has the highest amplitude at frequency 4 Hz with the weighted rms acceleration value 0.041 m/s^2. This was confirmed with the results of the vibration characteristics on the car as discussed in section 4.1 where Toyota Kijang has higher amplitude and also rms displacement compared to Toyota Avanza on the driver seat cushion as the position of the driver (see table 3). Therefore, the vibration amplitudes that received by drivers for Toyota Kijang were greater than Toyota Avanza. However, Toyota Kijang and Toyota Avanza were still in the comfortable zone at idle condition because the value of the total weighted rms accelerations were less than 0.315 m/s^2 based on comfort zone criteria classified by ISO 2631-1.

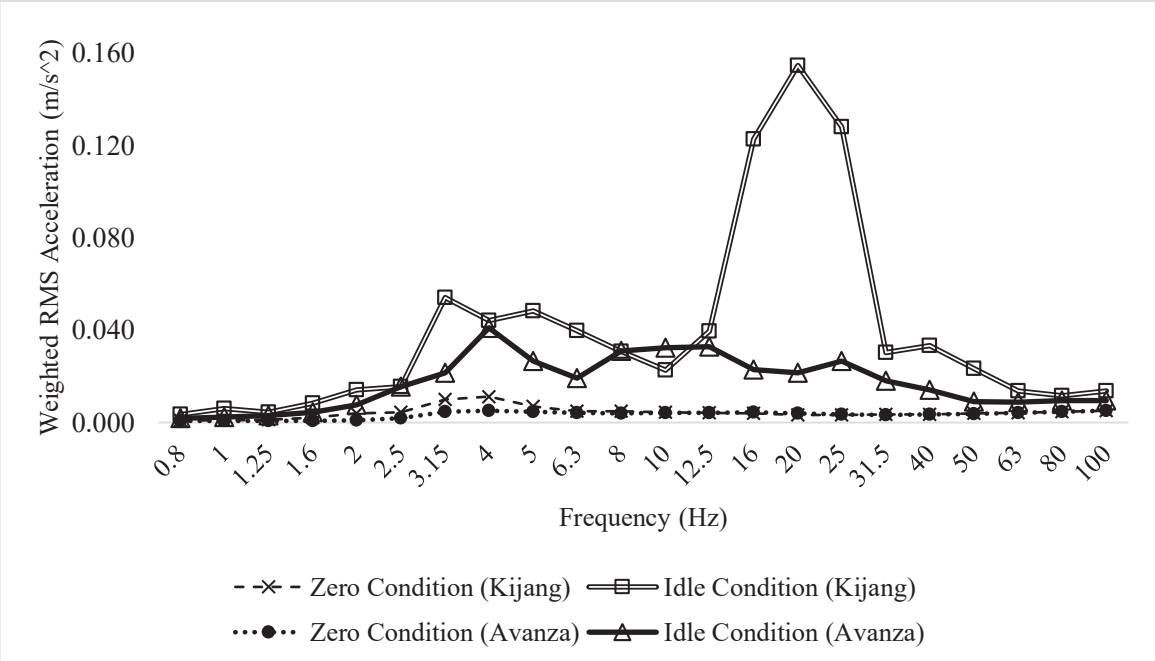


Figure 7 – One third octave band frequency analysis for Kijang and Avanza

5. CONCLUSIONS

The vibration level and vibration characteristics that affected by different vibration source (variation of the type of car) were analyzed. The results showed that low frequency vibration of the seated drivers affected to the comfort of ride both Toyota Kijang and Toyota Avanza using weighted rms acceleration and 1/3 octave band frequency analysis according to ISO 2631-1 standard. At idle condition, the driver seat comfort were in the comfortable zone both of those cars. Transfer path analysis has shown the level of vibration on the car. Through TPA, the human body comfort can be evaluated by multiplying the natural frequencies of the car with the operational acceleration.

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REFERENCES

1. National Statistics Agency of Indonesia. Development of the number of vehicle in Indonesia 1949-2017. 2017. Accessed in https://www.bps.go.id/linkTableDinamis/view/id/1133
2. Niekerk JV, Pielemeier W, Greenberg J. The use of seat effective amplitude transmissibility (SEAT) values to predict dynamic seat comfort. Journal of Sound and Vibration (Vol. 260). 2003.
3. Rakheja S, Stiharu I, Zhang H, Boileau PÉ. Seated occupant interactions with seat backrest and pan, and biodynamic responses under vertical vibration. Journal of Sound and Vibration. 2006; 298(3), 651–671.
4. Subashi GHMJ, Nawayseh N, Matsumoto Y, Griffin MJ. Nonlinear subjective and dynamic responses of seated subjects exposed to horizontal whole-body vibration. Journal of Sound and Vibration. 2009; 321(1), 416–434.
5. Coyte JL, Stirling D, Du H, Ros M. Seated whole-body vibration analysis, technologies and modelling: a survey. IEEE Transactions on Systems, Man and Cybernetics, Part A: Systems and Humans. 2016; 46 (6) p. 725-739.
6. Stan AC, Yenerer H, Sendur P, Basdogan I. Experimental transfer path analysis for a heavy duty truck. Proc 12th Biennial Conference on Engineering Systems Design and Analysis (ESDA 14 by ASME); 25-27 July 2014; Copenhagen, Denmark. 2014.
7. Genuit K, Poggenburg J. The design of vehicle interior noise using binaural transfer path analysis. society of automotive engineers Inc. NCV19. 1999.
8. Sturesson PO, Svensson C, Weckner J, Karlsson R, Sohr P. N&V integration and optimization of driveline using transfer path analysis. SAE International. 2012.
9. Koizumi T, Tsujiuchi N, Nakamura Y, Kido I, Hashioka M. A measures planning method by analysis of contribution of the vibration transfer path. Proc SAE Noise and Vibration Conference and Exhibition. 2009.
10. Knapen PL. Transfer path analysis related to booming, performed on a car. DCT rapporten; Vol. 1999.018. Eindhoven: Technische Universiteit Eindhoven.1999.
11. Plunt J. Strategy for transfer path analysis (TPA) applied to vibro-acoustic systems at medium and high frequencies. International Conference on Noise and Vibration Engineering. Leuven, Belgium. 1998.
12. International Standard of Organization. ISO 2631-1 - Mechanical vibration and shock- evaluation of human exposure to whole-body vibration. 1997.
13. Darth J. Engine start / stop vibrations in truck. Master Thesis of KTH Unversity (Sweden). 2014.
14. Brandt A. Noise and vibration analysis. Chichester: Johan Wiley & sons. 2011.Edition 1, p.438.
15. Kim BL, Jung JY, Oh IK. Modified transfer path analysis considering transmissibility functions for accurate estimation of vibration source. Journal of Sound and Vibration. 2017; 398, 70–83.
16. Plunt J. Finding and fixing vehicle NVH problems with transfer path analysis. Journal of Sound and Vibration, 2005: p. 12–16.
17. Wyckaert K, Auweraer H. Operational analysis, transfer paths analysis, modal analysis: tools to understand road noise problems in cars. Proceedings of SAE Noise and Vibration Conference. 1995; p. 139–143.
18. Oktav A, Yilmaz C, Anlas G. Transfer path analysis : Current practice , trade-offs and consideration of damping. Journal of Mechanical System and Signal Processing. 2017; 85 (760-772).
19. Taufan I, Susation J, Arifianto D. Transfer Path Analysis as a Feature for Damage Detection on the Pump-Beam System. Proceeding of ICSV25. 2018.
20. Qi X, Yin X. Experimental Studying Multi-Impact Phenomena Exhibited During The Collision of a Sphere Onto a Steel Beam. Journal of Advanced in mechanical Engineering. 2016.
21. Griffin MJ. Handbook of human vibration. London: Academic Press. 1990.
22. Mansfield NJ. Human response to vibration. London : CRC Press. 2005.
23. British Standards Institution, BS 6841: Measurement and evaluation of human exposure to whole body mechanical vibration and repeated shock, London, 1987.
24. 2002/44/EC, D. European Parliament and the Council of the European Union Directive 2002/44/EC (EU Directive) on the minimum health and safety requirements regarding the exposure of workers to the risks arsing from physical agents (vibration). Off. J. Eur. Communities OJ L177. 2002.