ISOMAse
International Society of Ocean, Mechanical and Aerospace Scientists and Engineers

Journal of Ocean, Mechanical and Aerospace
-Science and Engineering-
30th March 2025. Vol.69 No.1

March 30, 2025

Application of Numerical Study for Dynamic Vibration Absorber in Suspension System to Improve Four Wheel Vehicle Comfort

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Paper History

Received: 10-December-2024
Received in revised form: 12-February 2025
Accepted: 30-March-2025

ABSTRACT

This study aims to develop a Dynamic Vibration Absorber (DVA) for a four-wheeled vehicle, specifically the "Toyota Fortuner 4.0 V6 SR" model. The quarter vehicle structure was modeled as a two degree of freedom system with disturbances in the form of road contours and speed. The natural frequency of the DVA was designed to be equal to the lowest natural frequency of the vehicle. Hence, if the vehicle moved at a speed or road contour disturbance close to its natural frequency, the vibration energy will be transferred or received by the DVA so that the vehicle structure did not oscillate. Two DVAs were mounted on the top of the vehicle body. The results obtained the vibration that occurs when the disturbance frequency was closed to the lowest natural frequency of the ¼ vehicle structure. It was significantly reduced by the addition of two DVAs. The design model of the ¼ vehicle test structure and Dynamic Vibration Absorber (DVA) in this study can be used as a reference for the implementation of experiments. The results of experimental calculations can be compared with the results of theoretical calculations to strengthen the credibility of the results.

KEYWORDS: *Comfort, Dynamic Vibration Absorber (DVA), Natural frequency, Quarter vehicle, Oscillate.*

NOMENCLATURE

F	Force (N)
m	Mass (kg)
c	Damping (Ns/m)
k	Stiffness (N/m)
x	Displacement (m)
$\dot{x}$	Velocity (m/s)
$\ddot{x}$	Acceleration (m/s2)
f	Frequency (Hz)
ω_n	Natural frequency (rad/s)

1.0 INTRODUCTION

Passenger comfort is an important consideration in vehicle design. Vibrations that occur in the vehicle cause discomfort to passengers. The speed and contour of the road affect the amount of vibration that occurs in the vehicle. When the vehicle speed approaches the lowest natural frequency of the vehicle, the vehicle will have large vibrations, because the disturbance that occurs at the first or lowest natural frequency will cause the vehicle to oscillate with a large deviation. This causes discomfort to the driver.

In previous research, four damping design schemes were compared to reduce the vertical vibration of the car body with 9-DOF modeling, namely primary active, primary semi-active, secondary actives, and secondary semi-active suspensions [1]. The vibration isolation performance of the parallel air spring system for large equipment transportation is simulated by using finite elements. Then the vibration isolation system is fabricated and installed directly on the actual vehicle for testing. The test results show that the vibration isolation system using parallel air springs has excellent vibration isolation efficiency and acceptable lateral stability [2]. The concept of Acceleration-Driven Damping (ADD) approach with a passive Inverter-Spring-Damper (ISD) circuit is combined and applied to a four-wheeled vehicle with quarter vehicle modeling in the form of a 2-DOF system. Simulation and experimental results show that combining these two damping systems can effectively improve the suspension performance and can isolate vibrations over a wide frequency range (from lowest frequency to high frequency) significantly [3-4].

A four-wheeled vehicle suspension system is designed by applying dynamic damping. The structure is modeled as a quarter of the vehicle in the form of a 2-DOF system. Based on the optimal matching results, a hybrid control strategy is suggested for the vehicle suspension, the suspension and dynamic damper are considered as semi-active devices to adjust the hybrid control force according to the response of the vehicle structure model. The simulation results show that the optimized vehicle structure model can achieve better vibration characteristics, and the hybrid controller can effectively improve the ride comfort and stability of the vehicle [5-6]. In the experimental scale vehicle, the resonant frequency/ natural frequency of the vehicle support is analyzed. Then, the design methods of shock absorber without DVA and shock absorber with DVA are compared, the result is that shock absorber with DVA can reduce vibration at the vehicle support better than shock absorber without dynamic damper [7-8]. The next researcher tested the difference in vibration on the shock breaker with variations in the type of coil spring using a mode of shapes analyzer. From the experiment, it was found that the AHM fork spring has a graph that tends to minimize its vibration compared to the test on the ASPIRA fork spring which has a higher vibration [9-10].

The research team previously examined the effectiveness of using DVA on the two lowest modes of the building structure. Experimental and simulation results show that the use of DVA on the top floor of a two-story building structure is quite effective in reducing vibration at its two lowest modes [11-15]. In this study, the effectiveness of using DVA on the structure of a quarter of a four-wheeled vehicle is studied numerically. The vehicle structure is modeled as a two degree of freedom (2-DOF) system. Two DVAs are added to the upper part of the vehicle body. Then the effectiveness of the use of DVA is studied numerically. In addition, this study developed a design model of the ¼ vehicle test structure and Dynamic Vibration Absorber (DVA) as a reference for experiment implementation by subsequent researchers.

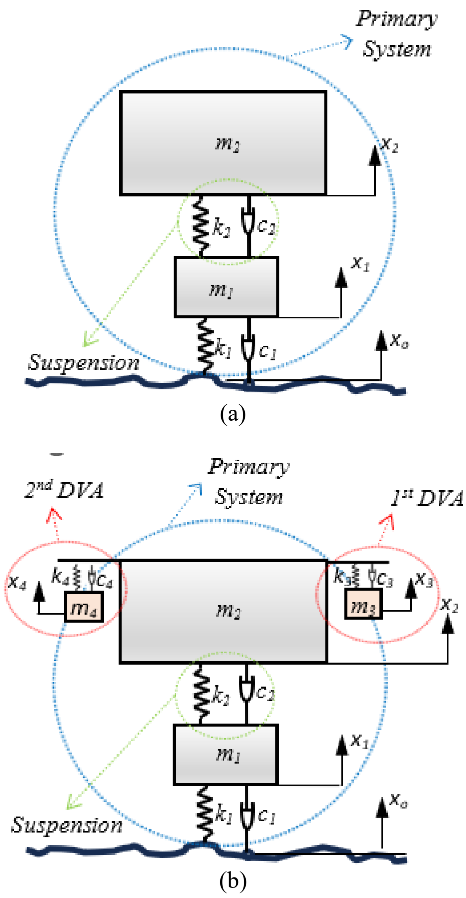

Figure 1: Modeling of ¼ vehicle structure without DVA (a) and with DVA (b)

2.0 THEORETICAL BACKGROUND

2.1 Theoretical Model of ¼ Vehicle Structure

The ¼ vehicle structure without DVA is modeled as a 2 degree of freedom vibration system (Figure 1(a)) and the ¼ vehicle structure using 2 DVAs is modeled as a 4 degree of freedom vibration system (Figure 1(b)). Where m_2 is the mass of the vehicle body, m_1 is the mass of the vehicle wheels, k_1 and c_1 are the stiffness of the vehicle tires, k_2 and c_2 are the stiffness and damping of the vehicle suspension system (shock absorber). Then on the Dynamic Vibration Absorber (DVA): m_3, k_3, and c_3 are the mass of the first DVA, the stiffness of the first DVA and the damping of the first DVA, respectively. The m_4, k_4, and c_4 are the mass of the second DVA, the stiffness of the second DVA and the damping of the second DVA, respectively.

In Figure 1(b), it can be seen that two DVAs (m_3 and m_4) in the form of a spring-mass system are installed on the vehicle body, so that it will be an additional mass on the vehicle body.

From the modeling of the ¼ vehicle structure in Figure 1, the differential equations of motion of the system can be derived as follows:

a. Structure without DVA (Figure 1(a))

1. First Mass (m_1)

$\Sigma Fx=0$

$$m_1\ddot{x}_1+c_2(\dot{x}_1-\dot{x}_2)+c_1(\dot{x}_1-\dot{x}_0)+k_2(x_1-x_2)+k_1(x_1-x_0)=0$$

$$m_1\ddot{x}_1+(c_1+c_2)\dot{x}_1-c_2\dot{x}_2+(k_1+k_2)x_1-k_2x_2=c_1\dot{x}_0+k_1x_0 \quad (1)$$

2. Second Mass (m_2)

$\Sigma Fx=0$

$$m_2\ddot{x}_2-c_2\dot{x}_1+c_2\dot{x}_2+k_2x_1+k_2x_2=0$$

$$m_2\ddot{x}_2-c_2\dot{x}_1+c_2\dot{x}_2-k_2x_1+k_2x_2=0 \quad (2)$$

The differential equations of motion written in equations (1) and (2) can be arranged in matrix form as follows:

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{Bmatrix}+\begin{bmatrix} c_1+c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix}\begin{Bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{Bmatrix}+\begin{bmatrix} k_1+k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}\begin{Bmatrix} x_1 \\ x_2 \end{Bmatrix}=$$
$$\begin{Bmatrix} c_1 \\ 0 \end{Bmatrix}\dot{x}_0+\begin{Bmatrix} k_1 \\ 0 \end{Bmatrix}x_0 \quad (3)$$

3. Structure with DVA (Figure 1(b))

1. First Mass (m_1 = Vehicle Wheel Mass)

$\Sigma Fx=0$

$$m_1\ddot{x}_1+c_2(\dot{x}_1-\dot{x}_2)+c_1(\dot{x}_1-\dot{x}_0)+k_2(x_1-x_2)+k_1(x_1-x_0)=0$$

$$m_1\ddot{x}_1+(c_1+c_2)\dot{x}_1-c_2\dot{x}_2+(k_1+k_2)x_1-k_2x_2=c_1\dot{x}_0+k_1x_0 \quad (4)$$

JOMAse | Received: 10-December-2024 | Accepted: 30-March-2025 | [(69) 1: 10-17]
Published by International Society of Ocean, Mechanical and Aerospace Scientists and Engineers, www.isomase.org., ISSN: 2354-7065 & e-ISSN: 2527-6085

2. Second Mass (m_2 = Vehicle Body Mass)
$\Sigma Fx=0$

$$(m_2+m_3+m_4)\ddot{x}_2+c_3(\dot{x}_2-\dot{x}_3)-c_2(\dot{x}_1-\dot{x}_2)+k_3(x_2-x_3)-k_2(x_1-x_2)-k_4(x_4-x_2)=0$$

$$(m_2+m_3+m_4)\ddot{x}_2-c_2\dot{x}_1+(c_2+c_3+c_4)\dot{x}_2-c_3\dot{x}_3-c_4\dot{x}_4-k_2x_1+(k_2+k_3+k_4)x_2-k_3x_3-k_4x_4=0 \quad (5)$$

3. Third Mass (m_3 = First DVA Mass)
$\Sigma Fx=0$

$$m_3\ddot{x}_3-k_3(x_2-x_3)-c_3(\dot{x}_2-\dot{x}_3)=0$$

$$m_3\ddot{x}_3-c_3\dot{x}_2+c_3\dot{x}_3-k_3x_2+k_3x_3=0 \quad (6)$$

4. Fourth Mass (m_4 = Second DVA Mass)
$\Sigma Fx=0$

$$m_4\ddot{x}_4-k_4(x_2-x_4)-c_4(\dot{x}_2-\dot{x}_4)=0$$

$$m_4\ddot{x}_4-c_4\dot{x}_2+c_4\dot{x}_4-k_4x_2+k_4x_4=0 \quad (7)$$

The differential equations of motion written in equations (4), (5), (6) and (7) can be arranged in matrix form as follows:

$$\begin{bmatrix} m_1 & 0 & 0 & 0 \\ 0 & m_2+m_3+m_4 & 0 & 0 \\ 0 & 0 & m_3 & 0 \\ 0 & 0 & 0 & m_4 \end{bmatrix}\begin{Bmatrix}\ddot{x}_1\\ \ddot{x}_2\\ \ddot{x}_3\\ \ddot{x}_4\end{Bmatrix}+\begin{bmatrix} c_1+c_2 & -c_2 & 0 & 0 \\ -c_2 & c_2+c_3+c_4 & -c_3 & -c_3 \\ 0 & -c_3 & c_3 & 0 \\ 0 & -c_4 & 0 & c_4 \end{bmatrix}\begin{Bmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\\ \dot{x}_4\end{Bmatrix}+$$

$$\begin{bmatrix} k_1+k_2 & -k_2 & 0 & 0 \\ -k_2 & k_2+k_3+k_4 & -k_3 & -k_4 \\ 0 & -k_3 & k_3 & 0 \\ 0 & -k_4 & 0 & k_4 \end{bmatrix}\begin{Bmatrix}x_1\\ x_2\\ x_3\\ x_4\end{Bmatrix}=\begin{Bmatrix}c_1\\ 0\\ 0\\ 0\end{Bmatrix}\dot{x}_0+\begin{Bmatrix}k_1\\ 0\\ 0\\ 0\end{Bmatrix}x_0 \quad (8)$$

2.2 Modal Analysis

The modal analysis of structures with Dynamic Vibration Absorber (DVA) is calculated by adding the modal damping ratio to the equations of motion of the system in modal coordinates as shown in equation (9).

$$M_{modal}\{\ddot{q}\}+C_{modal}\{\dot{q}\}+K_{modal}\{q\}=\phi^T F(t) \quad (9)$$

In this case:

$$M_{modal}=\phi^T[M_s]\phi \quad ; \quad K_{modal}=\phi^T[K_s]\phi \quad (10)$$

Where: ϕ = eigen vector, $F(t)$ = external force

3.0 METHODS

3.1 Vehicle Specifications Data

This study numerically analyzes the Toyota Fortuner 4.0 V6 SR (AT 4x4). The vehicle's specifications are provided in Table 1 [16]. These specifications detail the car's key features for the analysis.

Table 1: Specification of Toyota Fortuner Car 4.0 V6 SR

Parameter	Value
Body mass of ¼ vehicle (m_2)	450 kg
Wheel mass (m_1)	31 kg
Tire damping coefficient (c_1)	3430 Ns/m
Suspension damping coefficient (c_2)	3647.6 Ns/m
Tire stiffness (k_1)	221973.43 N/m
Suspension stiffness (k_2)	73750 N/m

3.2 Research Procedures

This research was implemented in several steps as shown in the research flowchart in Figure 2. A finite element simulation program for the ¼ vehicle structure was created using MATLAB with the structure parameters from Table 1. From the simulation results, the natural frequencies of the structure are obtained as a reference to determine the parameters (stiffness and mass) of the DVA. There are two DVA to be designed with the same type of mass-spring system. After the mass and stiffness of each DVA were calculated, a simulation of the ¼ vehicle structure using DVA was analyzed. From the results of the simulation program of the ¼ vehicle structure without DVA and using DVA, an analysis of the effectiveness of using DVA in the ¼ vehicle structure was analyzed.

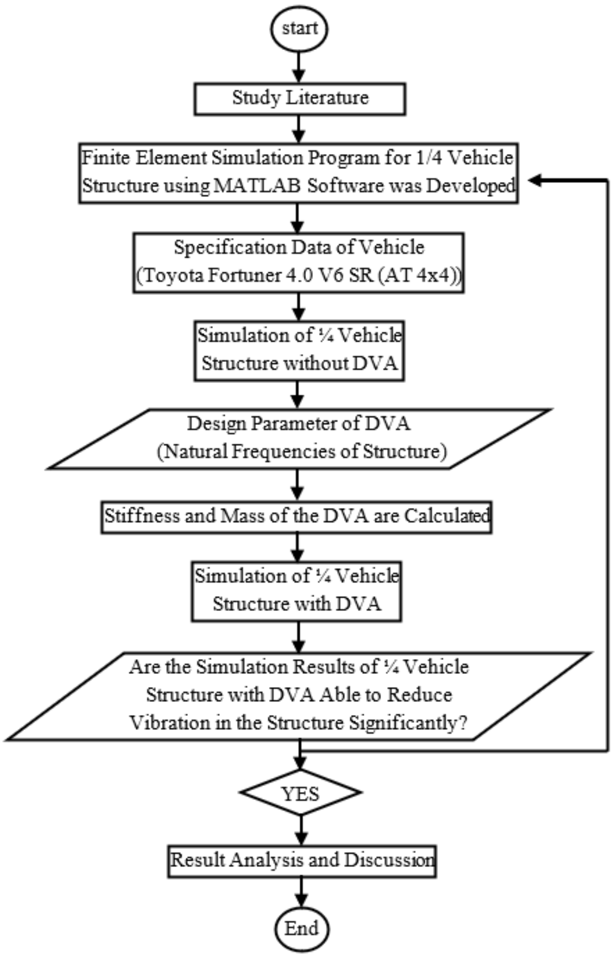

Figure 2: Research Flowchart

JOMAse | Received: 10-December-2024 | Accepted: 30-March-2025 | [(69) 1: 10-17]
Published by International Society of Ocean, Mechanical and Aerospace Scientists and Engineers, www.isomase.org., ISSN: 2354-7065 & e-ISSN: 2527-6085

4.0 RESULT AND DISCUSSION

4.1 Finite Element Program Simulation for ¼ Vehicle Structure (2-DOF and 4-DOF system)

A simulation program for the ¼ vehicle structure was developed to numerically determine the natural frequency values of the structure. This program was developed using MATLAB software with data input parameters from table 1 (Toyota Fortuner car specifications). The *Dynamic Vibration Absorbe*r (DVA) is designed based on the natural frequency values of the simulated ¼ vehicle structure. From the simulation results of the ¼ vehicle structure program without DVA, the lowest natural frequency value of the structure is obtained as shown in Table 2.

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Table 2: Natural frequencies value from simulation result

Natural Frequency	Without DVA (Hz)
1	1.76
2	15.58

4.2 Determining the Mass and Stiffness Value of DVA

Two DVAs were used. Both DVAs are of the same type, which is a spring-mass system. The first Dynamic Vibration Absorber (1st DVA) is designed to reduce the vibration at the lowest natural frequency of the vehicle structure (vehicle body). As shown in Table 2, the lowest natural frequency of the structure is 1.76 Hz, so the natural frequency of the 1st DVA is designed to be equal to the lowest natural frequency of the structure which is 1.76 Hz.

The second Dynamic Vibration Absorber (2nd DVA) is designed to reduce the vibration at the second lowest natural frequency of the structure (vehicle wheel). As shown in Table 2, the natural frequency of the second structure is 15.58 Hz, so the natural frequency of the 2nd DVA is designed to be equal to the lowest natural frequency of the second structure which is 15.58 Hz. To determine the mass and stiffness of each DVA, it can be calculated by the following equation:

1. For the 1st DVA:

$$f_1 = 1.76\,Hz \rightarrow \omega_{n,1^{st}\mathrm{DVA}} = \sqrt{\frac{k_3}{m_3}}$$

$$2 \times \pi \times f_1 = \sqrt{\frac{k_3}{m_3}} \tag{11}$$

2. For the 2nd DVA:

$$f_2 = 15.58\,Hz \rightarrow \omega_{n,2^{nd}\mathrm{DVA}} = \sqrt{\frac{k_4}{m_4}}$$

$$2 \times \pi \times f_2 = \sqrt{\frac{k_4}{m_4}} \tag{12}$$

Where k_3 and k_4 are the spring stiffness of 1st DVA and the spring stiffness of 2nd DVA, respectively, m_3 and m_4 are the mass of 1st DVA and the mass of 2nd DVA. In order to effectively absorb vibrations, the mass of the 1st DVA was designed to be 10% of the vehicle wheel mass. Then, the mass of the 2nd DVA was designed to be 10% of the vehicle body mass [11]. Thus, the mass value for the 1st DVA was 10% of the 1/4 vehicle body mass (450 kg), which was 45 kg and the mass for the 2nd DVA of 10% of the wheel mass (31 kg), which was 3.1 kg. Then the stiffness value of each DVA was obtained as follows:

1. Stiffness for 1st DVA, from equation (11) obtained:

$$2 \times \pi \times f_1 = \sqrt{\frac{k_3}{m_3}}$$

$$2 \times \pi \times 1.76_1 = \sqrt{\frac{k_3}{45}}$$

$$k_3 = 5511.73 \ {}^{N}\!/\!_{m}$$

2. Stiffness for 2nd DVA, from equation (12) obtained:

$$2 \times \pi \times f_2 = \sqrt{\frac{k_4}{m_4}}$$

$$2 \times \pi \times 15.58 = \sqrt{\frac{k_4}{3.1}}$$

$$k_4 = 29700.73 \ {}^{N}\!/\!_{m}$$

Thus, the DVA specifications are obtained as shown in Table 3.

Table 3: Specification of Dynamic Vibration Absorber (DVA)

DVA	Specification	Value
1st DVA	Mass (m_3)	45 kg
	Spring stiffness (k_3)	5511.73 N/m
2nd DVA	Mass (m_4)	3.1 kg
	Spring stiffness (k_4)	29700.73 N/m

After obtaining the DVA specifications, simulations were carried out using the MATLAB program so that a comparison of the natural frequency values of the structure without DVA and using DVA was obtained as shown in Table 4.

Table 4: Values of natural frequency from simulation results of ¼ vehicle structure without DVA and with DVA

Natural frequency	without DVA (Hz)	with DVA (Hz)
1	1.76	1.47
2	15.58	2.00
3	-	15.55
4	-	15.65

Furthermore, the simulation was performed by applying an external force to the ¼ vehicle structure, with a frequency equal to its lowest natural frequency of 1.76 Hz. This approach was used to evaluate the effectiveness of using the 1st Dynamic Vibration Absorber (DVA) on the vehicle's structural response.

The response of the structure to the applied time function was then obtained and analyzed. The simulation results are presented in Figures 3 and 4. These figures highlight the dynamic behavior of the structure under the applied force. Additionally, they demonstrate the effectiveness of the Dynamic Vibration Absorber (DVA) in reducing vibrations.

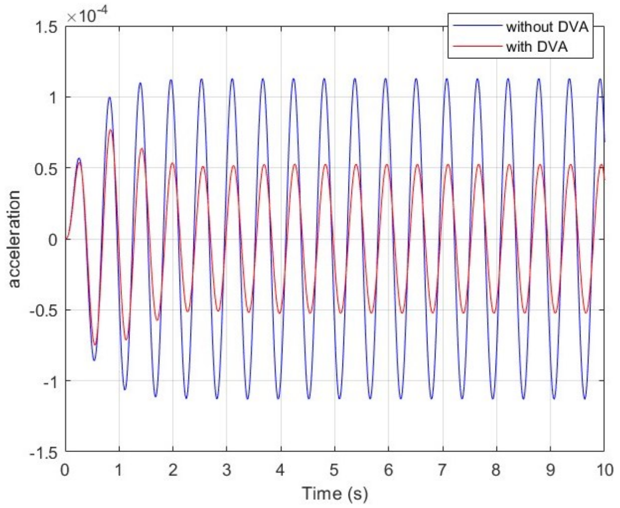

Figure 3: Response of ¼ vehicle structure as a function of time for frequency 1.76 Hz

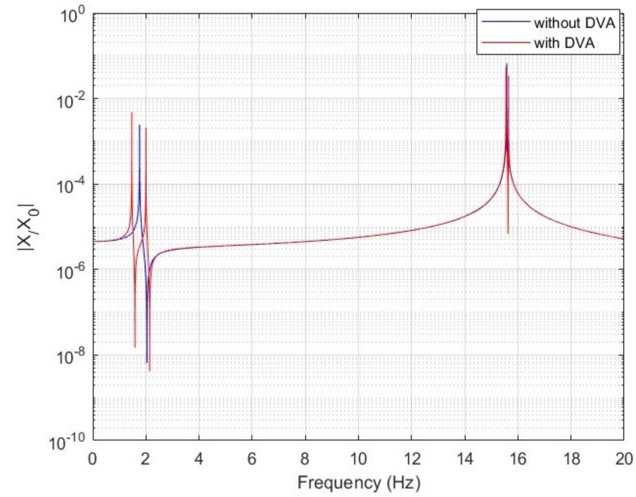

Figure 5: Simulation Results of Vibration Modes of Structures without and with Dynamic Damping (DVA damping (c_{dva}) = 0)

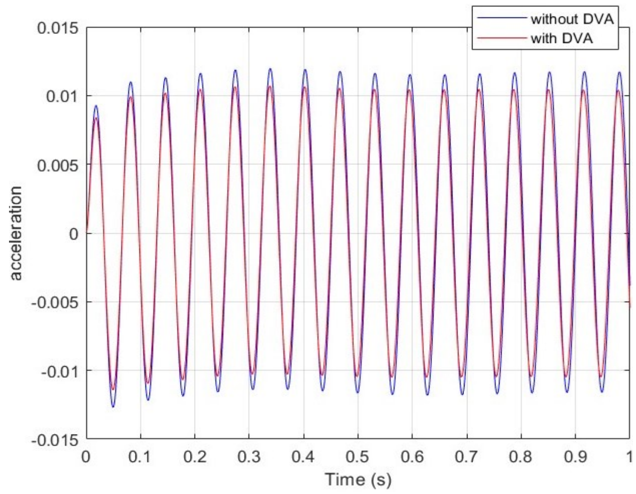

Figure 4: Response of ¼ vehicle structure as a function of time for frequency 15.58 Hz

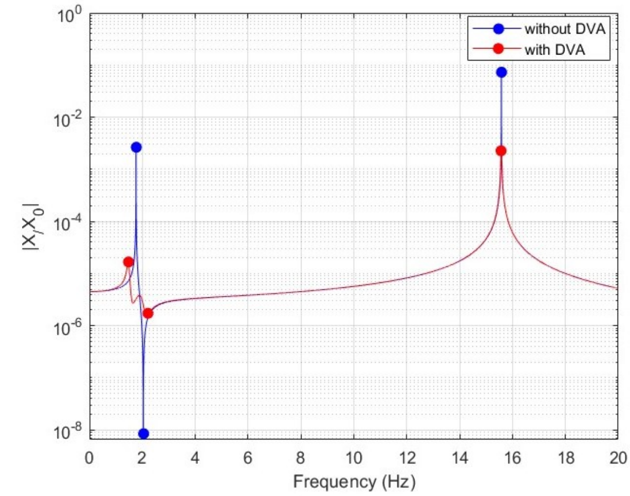

Figure 6: Simulation results of vibration modes of structures without and with dynamic damping (DVA damping (c_{dva}) ≠ 0)

From Figure 3, it can be seen a comparison of the response of the ¼ vehicle structure without DVA and using DVA when the structure is disturbed at its lowest natural frequency of 1.76 Hz. The use of the 1st DVA is able to significantly reduce vibration in the main structure, which is the vehicle body. In Figure 4 is a comparison of the structural response of ¼ vehicles without DVA and by using DVA when the structure is disturbed at the second lowest natural frequency of 15.58 Hz. It can be seen in Figure 4, the use of the second DVA is able to reduce the vibration of the main structure, but the reduced vibration is not very significant. This may be because the second personal frequency of the structure (car tires) is quite large and car tires also have a large damping value, so the use of DVA for the second lowest personal frequency of the structure is not too effective.

Then by modal analysis, a comparison of the response of the ¼ vehicle structure without DVA and with the use of DVA is obtained as shown in Figure 5 and Figure 6.

From Figure 5, it can be seen that if the DVA has no damping (DVA damping = 0), the vibrations that occur in the ¼ vehicle structure are still quite large. Whereas in Figure 6 it can be seen that with the damping value in the DVA (DVA damping ≠ 0), the vibrations that occur in the ¼ vehicle structure are significantly reduced.

4.3 Potential Experimental Validation

4.3.1 Design Plan of ¼ Vehicle Test Structure

The laboratory-scale ¼ vehicle structure was designed with a ratio of 1:100 of the actual vehicle weight (Table 1) as presented in Table 5. The modeling of the ¼ vehicle test structure is designed based on the theoretical model of ¼ vehicle structure without DVA (Figure 1(a)). The test structure components consist of body mass of ¼ vehicles (m_2) and wheel mass (m_1) which can be produced from brass. Then for the suspense/ shock absorber system (k_2, c_2) and tire stiffness, tire damping (k_1, c_1) can be produced from coil springs.

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-Science and Engineering-
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Table 5: Specifications of the laboratory-scale ¼ vehicle structure

Specification	Value
Body mass of ¼ vehicle (m_2)	4.5 kg
Wheel mass (m_1)	0.31 kg
Tire stiffness (k_1)	2219.73 N/m
Suspension stiffness (k_2)	737.5 N/m

Next, the components of the test structure are assembled in such a way on a stand made of aluminum as can be seen in Figure 7. On m_1, 6 wheels are mounted and on m_2, 4 wheels are mounted in the form of bearings so that m_1 and m_2 can move freely on their respective trajectories. The suspense/ shock absorber system (k_2, c_2) is mounted between the body mass of ¼ vehicle (m_2) and the wheel mass (m_1), then the tire stiffness and damping (k_1, c_1) are mounted on the wheel mass (m_1) and pinned on a vertical stand as can be seen in Figure 7.

4.3.2 Dynamic Vibration Absorber (DVA) Design Plan

This DVA is designed based on the theoretical model of ¼ vehicle structure with DVA (Figure 1(b)). The DVA used is 2 mass-spring systems installed in such a way on the vehicle body (m_2). In accordance with the design of the ¼ vehicle test structure, the DVA is also designed with a ratio of 1:100 from the calculation results of the DVA specifications in Table 3. So, the laboratory-scale DVA design specifications can be seen in Table 6.

The DVA component in the type of mass-spring system consists of masses (m_3 and m_4) which can be made of wood and a coil spring (k_3 and k_4) on each mass (m_3 and m_4). These two mass-spring systems (DVA) are installed in such a way above the body mass of ¼ vehicle (m_2) using a stand made of aluminum as can be seen in Figure 8 (a) and 8 (b). On m_3 and m_4, four wheels are each mounted in the form of bearings so that m_3 and m_4 can move freely on their respective trajectories and can absorb vibrations from disturbance forces.

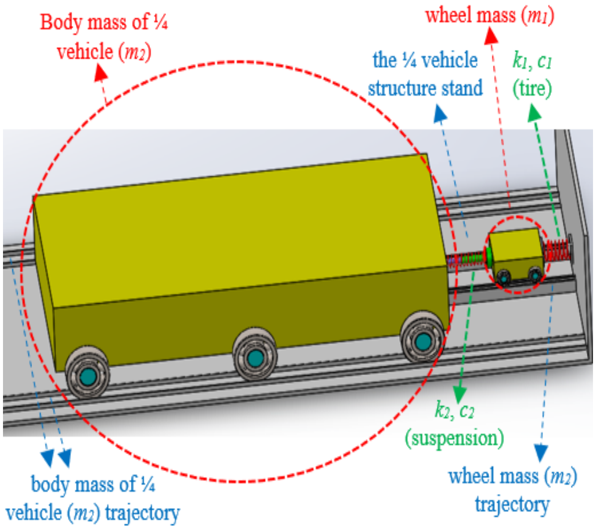

Figure 7: Design sketch of ¼ vehicle test structure without DVA

Table 6: Laboratory-scale DVA specifications

DVA	Specification	Value
DVA 1	Mass (m_3)	0.45 kg
	Spring stiffness (k_3)	55.11 N/m
DVA 2	Mass (m_4)	0.031 kg
	Spring stiffness (k_4)	297 N/m

4.3.3 DVA Effectiveness Test Plan on ¼ Vehicle Structure

The test can be carried out in two conditions. The first condition was the ¼ vehicle structure that was tested without DVA. The second condition the ¼ vehicle structure was tested with DVA. The first DVA was designed to reduce vibration at the lowest natural frequency of the structure. The second DVA was designed to reduce the vibration at the second natural frequency of the structure. In Figure 7 is shown the model of the ¼ vehicle test structure without DVA. In Figure 8(b) the model of the ¼ vehicle test structure with DVA is shown. The parameters of the ¼ vehicle structure and DVA are shown in Table 5 and Table 6.

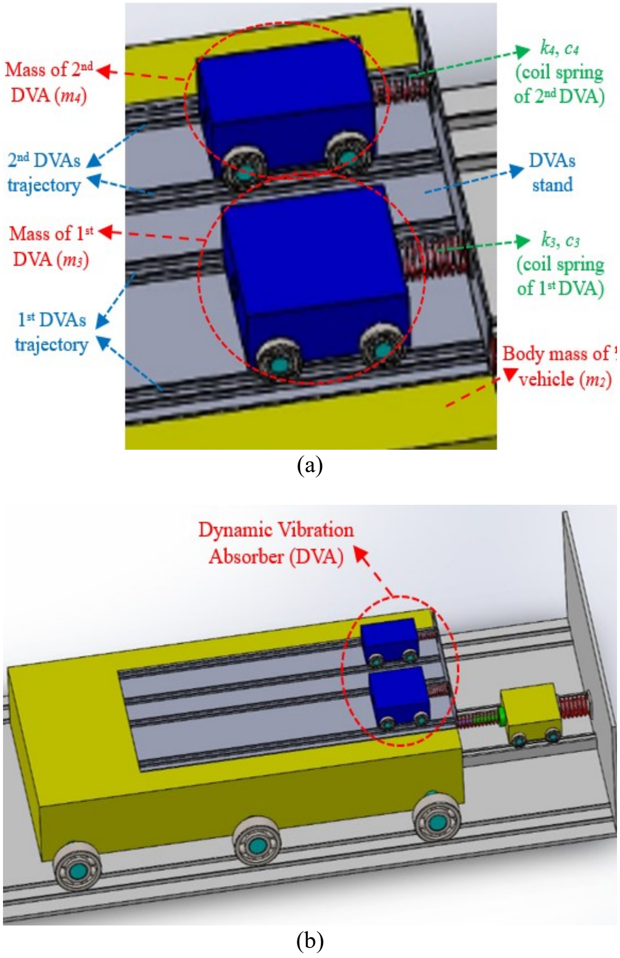

Figure 8: (a) Design sketch of dynamic vibration absorber (DVA) and (b) design sketch of ¼ vehicle test structure with DVA

The tests were conducted using the Accelerometer sensor. Accelerometer sensor served to measure the acceleration of vibration that occurs in the structure. The tool used to extract data from the Accelerometer sensor was an Oscilloscope. To amplify the signal, a voltage amplifier was used. The exciter was used as a disturbance force actuator on the ground structure, whose frequency can be adjusted on the power amplifier. The processing of the signal from the accelerometer is shown in Figure 9.

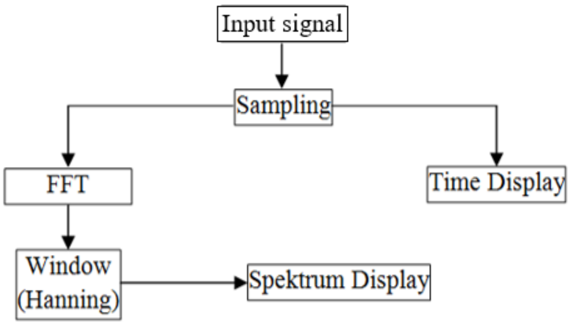

Figure 9: Digital signal processing

The schematic of the test is shown in Figure 10. The exciter applies an excitation force to the base of the structure. The force applied by the exciter was derived from a signal on a power amplifier whose frequency can be adjusted. The frequency of the disturbance was varied from 1-10 Hz. The acceleration of the structure was measured by an accelerometer sensor attached to the body mass of ¼ vehicles (m_2). The signal from the accelerometer sensor was amplified by the signal amplifier and passed to the oscilloscope in the form of an analog signal. The data was stored on a flash disk in the form of digital signals, which were further processed using MATLAB software on a computer to display communicative graphs.

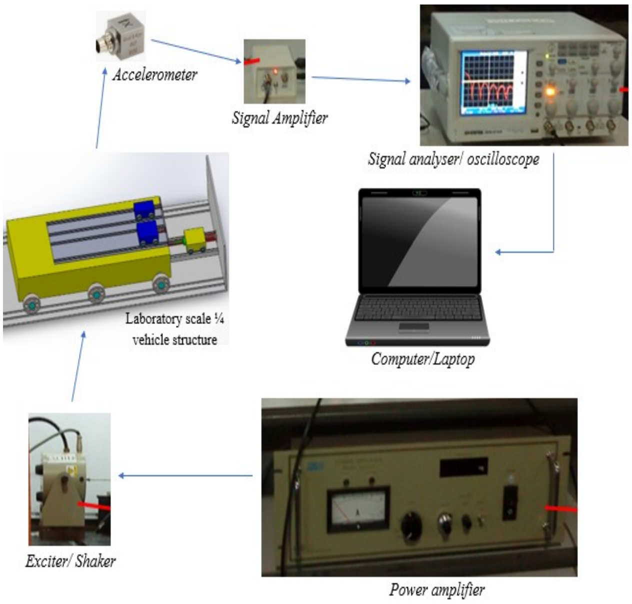

Figure: 10 Test of implementation scheme

Due to funding limitations, this experimental plan could not be implemented. The results of theoretical calculations of the effectiveness of using DVA on the ¼ vehicle structure in this article can be used as a reference for the implementation of experiments by subsequent researchers. Then the results of experimental calculations can be compared with the results of theoretical calculations to strengthen the credibility of the results.

4.0 CONCLUSION

By designing the natural frequency of the first mass-spring system (1st DVA) equal to the lowest natural frequency of the ¼ vehicle structure, the vibration occurring in the vehicle body during the disturbance frequency near the lowest natural frequency of the ¼ vehicle structure can be effectively reduced. By designing the natural frequency of the second spring-mass system (2nd DVA) equal to the second natural frequency of the ¼ vehicle structure, the vibration occurring at the wheels of the vehicle during the disturbance frequency near the second natural frequency of the structure can be reduced, but not significantly. The use of DVA in the form of a mass-spring system (1st DVA) in the ¼ vehicle structure is able to absorb the vibration energy that occurs in the structure during the disturbance frequency at the lowest natural frequency of the structure and the use of DVA in the form of a mass-spring system (2nd DVA) in the ¼ vehicle structure is able to absorb the vibration energy that occurs in the vehicle structure during the disturbance frequency near the second lowest natural frequency of the structure. Designing the mass value for DVA above 10% of the mass of the structure can displace the natural frequency of vehicle structure. So that the mass of the DVA can be ignored (does not affect the value of the natural frequency of the vehicle structure) then the mass for the DVA should be designed with a value below 10% of the mass of the vehicle structure. The design plan of the ¼ vehicle test structure and DVA in subsection 4.3 can be considered as a reference for further research in the form of experiments to validate the results of this numerical study.

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