# Vibration Analysis of Suspension System for 3DOF Quarter Car Model with Tire Damping

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***Abstract*—Quarter car model is the simplest way for the analysis of vehicle suspension system. The objective of this study is to propose the natural frequency and to identify the displacement analysis of 3 Degree of Freedom (3DOF) suspension system using Matlab software. Research methodology are state space modelling method and logarithmic decrement method. From the state space modelling, seat mass, sprung mass and unsprung mass displacement have been calculated. Logarithmic decrement method is a time domain technique to obtain experimental damping ratio. The novelty of this research has demonstrated the experimental approach by using Quantum X (catman AP v5 software). Natural frequency and acceleration results are obtained from the time domain response of experimental data. The value of sprung mass natural frequency for theoretical (1.23 Hz) and experimental (1.134 Hz) result are within 1Hz to 2Hz. The maximum displacement transmissibility values for theoretical (2.7) and experimental (2.54) approach are less than the reference value (3.5) at the resonance point. The findings demonstrated that the results of natural frequency and damping ratio are within the acceptable limit. So, the obtained results are validated the suspension system of Nissan Sunny model for vehicle safety and passenger comfort.**

***Keywords*—displacement, logarithmic decrement, Matlab software, natural frequency, state space modelling**

## I. Introduction

Vehicle suspension system has the mainly two objectives; passenger comfort and vehicle control. Suspension system is the mechanism that physically separates the car body from the wheels of the car. Suspension system consists of springs, shock absorbers and linkages that connects a vehicle to its wheels. Suspension coil spring is compressed when the wheel strikes a bump. The main function of the vehicle suspension system is to minimize the vertical acceleration transmitted to the passenger which directly provides road comfort.

Zhang *et al*. [1] described the vibration performance analysis and multi-objective optimization design of a tractor scissor seat suspension system. They were focused on determining the design and optimization method of the vibration reduction of seats suspension. Wu [2] reviewed the theoretical and experimental research on active suspension system with time-delay control. He was presented about the time-delay feedback control characteristics of the system by comparing harmonic excitation with passive control and back stepping control. Zhang *et al.* [3] studied about the 3 Degree of Freedom (3DOF) active suspension systems. He was employed to handle displacement constraints problem, and the ride comfort and safety are improved at the same time. Yat Sheng Kong *et al.* [4] presented about the vibration fatigue analysis of carbon steel coil spring under various road excitations. He founded that the harshest road condition was the rural road where the spring with fatigue life of $4.47\times10^7$ blocks to failure. Devdutt [5] developed the passenger body vibration control using Hybrid ANFIS PID controller in active quarter car model. Effect on the vibration of the s`uspension system was studied by Dahi [6]. The vehicles' vibration magnitudes caused by road roughness were analysed. The vibration valves were determined with a HVM 100 device, in different field conditions.

Primarily, free vibration methods were studied by Lukasz [7]. He was recorded the accelerations of the body of an automotive vehicle equipped with a shock absorber enabling simulation of shock absorbing liquid leakage. Tan *et al.* [8] described the vibration analysis on compact car shock absorber. Natural frequencies of the shock absorber in a 2DOF system were investigated using Wolfram Mathematical, CATIA, and ANSYS. Mitra *et al.* [9] carried out the design of experiment for optimization of automotive suspension system. A quarter car test rig was developed to practically evaluate and analyze the influence of the various parameters individually as well as of their interactions, on ride comfort and road holding. Parekh *et al.* [10] discussed the transmissibility analysis of car driver's seat suspension system with an air bellow type damper. They were focused on the vibration isolation efficiency characteristics (Transmissibility analysis) for 6 different CDSSS models. It was seen that CDSSS model 5 gives approximately 63% reduction in TR compared to that of CDSSS model 1.

Patil [11] reviewed the experimental analysis of 2DOF quarter car (ASS) and (QC-Hydraulic ASS) for ride comfort. It is observed red that the ride performance

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characteristics have improved with the induction of active element in the form of hydraulic actuator in the 2DOF QC-PSS. Petal *et al*. [12] presented about the investigation and analysis of quarter car automotive suspension system using mathematical model. They were focused on the mathematical model of quarter car automotive two degrees of freedom suspension system. Rao [13] presented about base excitation model, Laplace transform and transfer function of the system. Kelly [14] described the modelling, free and force vibration of MDOF system in detail. Nouri *et al*. [15] studied about 3DOF Vehicle Suspension System. He was presented about the two applications of wavelet analysis: signal denoising based on Discrete Wavelet Transform and modal identification based on Continuous Wavelet Transform.

Yildirim [16] studied about the vibration analysis using artificial neutral. He was analyzed the effect of vibration on comfort and road holding capability of vehicles as observed in variation of suspension springs and road roughness. Dixon [17] described the standard damping ratio and damping coefficient in vibration theory. Maher and Young [18] presented about the modelling of road vehicle suspension system for performance evaluation. His experiments used a Quarter Car Rig which was built and tuned for the purpose of his research. Results experimentation were compared with simulation values obtained from a quarter car simulation run in Matlab. Shigley and Mischke [19] redesigned standard handbook of machine design to identify the formula of spring and tire stiffness. Snowdon [20] described the vibration isolation, characterization. And the performance of antivibration mountings for the control of noise and vibration.

According to the literature review, suspension system is one of the most important systems of the automobile for vehicle safety and passenger comfort. During the study, many researchers were considered the 2DOF suspension system of quarter car model without tire damping. Hence, 3DOF suspension system with tire damping was considered in this current research. In this study, vibration analysis of the suspension system is conducted by theoretical and experimental approach. Theoretical analysis is performed by changing the damping ratio using Matlab program with state space modelling method. Experimental test of the suspension system is conducted by using accelerometers and data acquisition system Quantum X.

## II. Materials and Methods

The research methodology is mainly divided into two portions: theoretical and experimental approach. State space modelling method is used in theoretical analysis as it is one of the best ways of presenting differential equation. Logarithmic decrement method is a time domain technique that exploits the exponential decay property of the free responses of a damped system to obtain the experimental damping ratio. In this research, Matlab program is applied to evaluate the natural frequency and displacement of the suspension system. Furthermore, the experimental test is designed by using the single axis accelerometer and DAQ (Quantum X).

### *A. Theoretical Calculation of Natural Frequency of Suspension System*

Every elastic object has a certain speed of oscillation that will occur naturally when there are zero outside forces or damping applied. Natural vibration occurs only a certain frequency, known as the natural frequency. Natural frequency is also known as eigen frequency. The suspension frequency varies in inverse proportion to the spring load and the amount of static deflection produced by that load. Quarter car model is used for suspension analysis because it is simple. Specification data of Nissan Sunny model are as follows:

Gross weight = 1435 kg
Pay load = 408 kg
Net weight = 1027 kg
Unsprung weight = 38.5 kg
Sprung weight = 358.6 kg
Spring stiffness, $k_s$ = 23.26 kN/m
Tire dimension (185/65R15), $S_N$ =185, AR=65, $D_R$ =15

$$k_t = 0.00028P\sqrt{(-0.004AR + 1.03)S_N \times \left(\frac{S_N \times AR}{50} + D_R\right)} + 3.45$$

Equivalent stiffness, $k_e = \frac{k_s + k_t}{k_s \times k_t}$

Natural frequency of sprung mass is calculated by using these equations.

Natural frequency, $\omega_n = \frac{1}{2\pi}\sqrt{\frac{k_e}{m_s}}$

The theoretical results of sprung mass natural frequency are 1.23 Hz. The sprung mass natural frequency of suspension system must be within 1 Hz to 2 Hz [11]. So, the result is within acceptable limit.

### *B. Theoretical Calculation of 3DOF Suspension System*

3DOF suspension system is included the seat and drive mass ($m_{se}$) sprung mass ($m_s$) and unsprung mass ($m_u$). The proposed suspension system is modelled by interconnecting three mass spring damper system.

Critical damping, $c_c$ can be calculated as $c_c = 2\sqrt{\mathrm{km}}$

Damping coefficient, c is also calculated by using these equation $c = \xi c_c$

In this study, three degree of freedom quarter model with tire damping case is calculated by using Matlab software. Specification data of the 3DOF suspension system are as follows:

Person and Seat mass, $m_{se}$ = 70 kg
Sprung mass, $m_s$ = 358. 6kg
Unsprung mass, $m_u$ = 38.5 kg
Seat mass stiffness, $k_{se}$ = 6.099 kN/m
Spring stiffness, $k_s$ = 23.26k N/m
Tire stiffness, $k_t$ = 224.387k N/m
Damping coefficient of seat mass, $c_{se}$ = 468.6 N.s/m
Damping coefficient of spring, $c_s$ = 1770N.s/m
Damping coefficient of tire, $c_t$ = 1866.5N.s/m

Three degree of freedom quarter car model with tire damping from the actual car structure diagram is shown in Fig. 1.

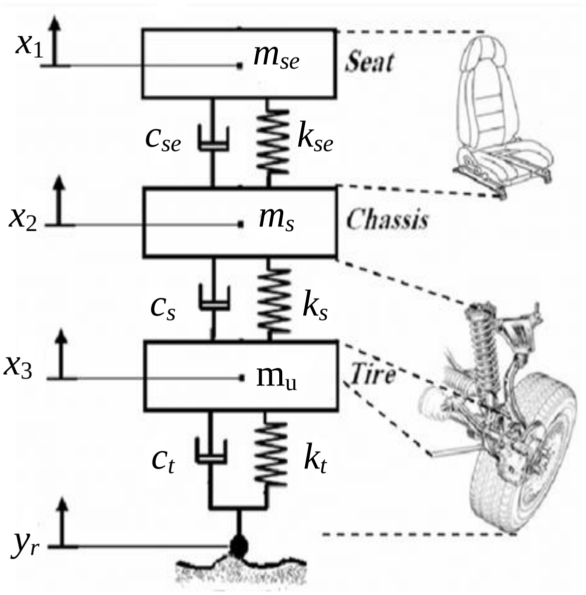



Fig. 1. 3DOF quarter car model with tire damping [15].

As shown in Fig. 1 model, each mass has a stiffness k constant, damping constant, c and a vertical displacement x. The vertical displacement $y_r$ denotes the change of road surface elevation. The equations of motion can be written as the free body diagram. EOM for 3DOF suspension system is written as follows:

For person and seat mass, $m_{se}$

$$m_{\mathrm{se}}\ddot{x}_1 = -k_{\mathrm{se}}(x_1 - x_2) - c_{\mathrm{se}}(\dot{x}_1 - \dot{x}_2)$$

For sprung mass, $m_s$

$$m_s\ddot{x}_2 = k_{\mathrm{se}}(x_1 - x_2) + c_{\mathrm{se}}(\dot{x}_1 - \dot{x}_2)$$

$$-k_s(x_2 - x_3) - c_s(\dot{x}_2 - \dot{x}_3)$$

For unsprung mass, $m_u$

$$m_u\ddot{x}_3 = k_s(x_2 - x_3) + c_s(\dot{x}_2 - \dot{x}_3)$$

$$-\mathrm{k}_t(x_3 - y_r) - c_t(\dot{x}_3 - \dot{y}_r)$$

Let $\dot{x}_1 = V_1,\ \dot{x}_2 = V_2,\ \dot{x}_3 = V_3$

$$\Delta_1 = (x_1 - x_2), \qquad \dot{\Delta}_1 = (V_1 - V_2)$$

$$\Delta_2 = (x_2 - x_3), \qquad \dot{\Delta}_2 = (V_2 - V_3)$$

From person and seat mass equation

$$\dot{V}_1 = -\frac{k_{\mathrm{se}}}{m_{\mathrm{se}}}\Delta_1 - \frac{c_{\mathrm{se}}}{m_{\mathrm{se}}}(V_1 - V_2)$$

Let, $\dot{T} = \dot{V}_3 - \frac{c_t}{m_u}\dot{y}_r$

From sprung mass equation

$$\dot{V}_2 = \frac{k_{\mathrm{se}}}{m_s}\Delta_1 + \frac{c_{\mathrm{se}}}{m_s}V_1 - \left[\frac{c_{\mathrm{se}}}{m_s} + \frac{c_s}{m_s}\right]V_2$$

$$-\frac{k_s}{m_s}\Delta_2 + \frac{c_s}{m_s}T + \frac{c_s c_t}{m_s m_u}y_r$$

From unsprung mass equation

$$\dot{T} = \frac{k_s}{m_u}\Delta_2 + \frac{c_s}{m_u}V_2 - \left[\frac{(c_s + c_t)}{m_u}\right]T$$

$$+\left[-\frac{c_s c_t}{m_u^2} - \frac{c_t^2}{m_u^2} + \frac{k_t}{m_u}\right]y_r - \frac{k_t}{m_u}x_3$$

State space model is the mathematical model of a physical system. A set of input, output and state variables are related to differential equation.

State space model

$$X = AX + Bu$$

$$Y = CX + Du$$

State variable are $x_3$, $\Delta_1$, $\Delta_2$, $V_1$, $V_2$, $T$

where

$A =$

$$\begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 1 & -1 & 0 \\ 0 & 0 & 0 & 0 & 1 & -1 \\ 0 & -\frac{k_{\mathrm{se}}}{m_{\mathrm{se}}} & 0 & -\frac{c_{\mathrm{se}}}{m_{\mathrm{se}}} & \frac{c_{\mathrm{se}}}{m_{\mathrm{se}}} & 0 \\ 0 & \frac{k_s}{m_s} & -\frac{k_s}{m_s} & \frac{c_{\mathrm{se}}}{m_s} & -\left[\frac{(c_{\mathrm{se}} + c_s)}{m_s}\right] & \frac{c_s}{m_s} \\ -\frac{k_t}{m_u} & 0 & \frac{k_s}{m_u} & 0 & \frac{c_s}{m_u} & -\left[\frac{(c_s + c_t)}{m_u}\right] \end{bmatrix}$$

$$X = \begin{bmatrix} \dot{x}_3 \\ \dot{\Delta}_1 \\ \dot{\Delta}_2 \\ \dot{V}_1 \\ \dot{V}_2 \\ \dot{T} \end{bmatrix}, \qquad B = \begin{bmatrix} \frac{c_t}{m_u} \\ 0 \\ -\frac{c_t}{m_u} \\ 0 \\ \frac{c_s c_t}{m_s m_u} \\ -\frac{c_s c_t}{m_u^2} - \frac{c_t^2}{m_u^2} + \frac{k_t}{m_u} \end{bmatrix}$$

Output matrix is

$$\begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 & 0 & 0 & 0 \\ 1 & 0 & 1 & 0 & 0 & 0 \\ 1 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}\begin{bmatrix} x_3 \\ \Delta_1 \\ \Delta_2 \\ V_1 \\ V_2 \\ T \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix}[y_r]$$

The above equation is calculated by applying Matlab program to analyze the displacement of 3DOF quarter car model. The seat mass, sprung mass and unsprung mass displacement of suspension system with tire damping based on the damping ratio 0.2 is shown in Fig. 2.

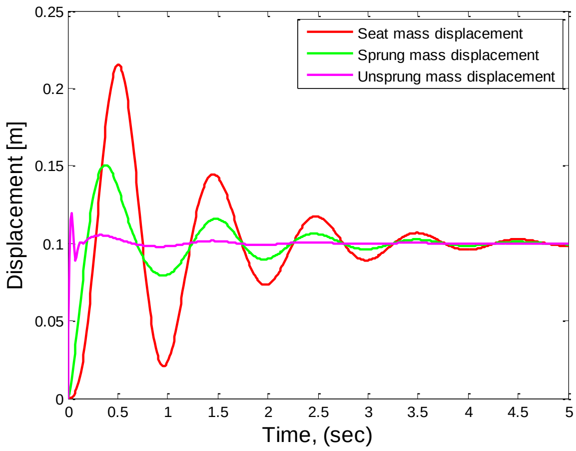



Fig. 2. Comparison of seat mass, sprung mass and unsprung mass displacement.

According to Fig. 2, the maximum displacement of 3DOF suspension system is occurred in seat mass displacement in the system. The displacement analysis of seat mass, sprung mass and unsprung mass for suspension system are analysed by changing damping ratio (0.2–0.4) and the increasement is 0.5. All of the following results are calculated with matlab software.

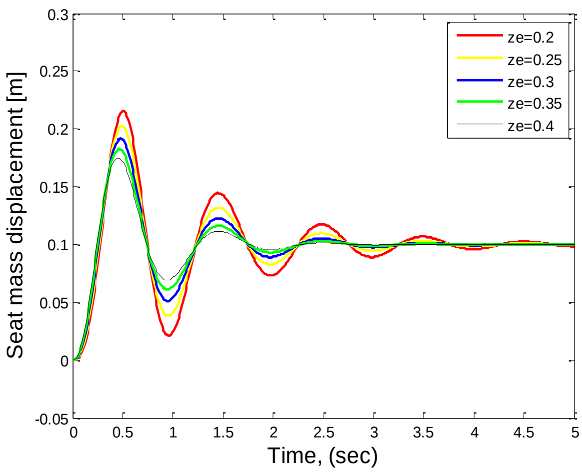



Fig. 3. Seat mass displacement by changing damping ratio 0.2–0.4.

The seat mass displacement of suspension system with tire damping changes the damping ratio from 0.2 to 0.4 shown in Fig. 3. From these curves, it can be determined that the maximum overshoot of sprung mass is occurred at damping ratio 0.2 and amplitude 0.214 m and settling time occurs at 4.84 sec.

As shown in Fig. 4, the maximum overshoot of sprung mass displacement of suspension system with tire damping is 0.15 m and settling time occurs at 4.56 s.

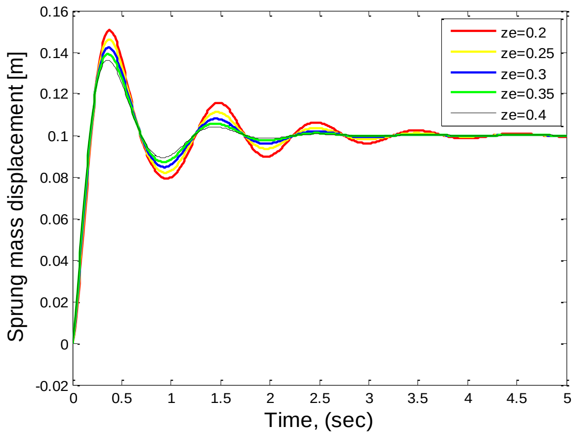



Fig. 4. Sprung mass displacement for damping ratio 0.2–0.4.

Fig. 5 shows the unsprung mass displacement of suspension system with tire damping by changing the damping ratio 0.2 to 0.4. According to Fig. 5, the maximum overshoot of unsprung mass is 0.12 m and settling time occurs at 2.56 s.

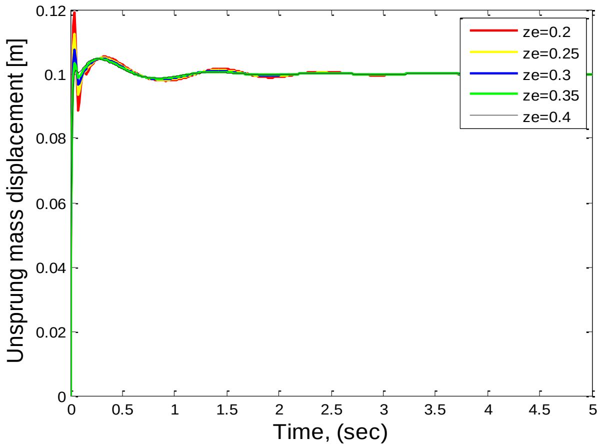



Fig. 5. Unsprung mass displacement for damping ratio 0.2 to 0.4

*C. Theoretical Calculation of Displacement Transmissibility*

Response of a Damped system under the harmonic motion of the base is analyzed. Viscous damped support on a based excitation system is considered under the harmonic motion of the base. The foundation is subject to harmonic vibration Y= sin ωt and required to determine the response, *x*, of the body. Let $y(t)$ denote the displacement of the base and $x(t)$ denotes the displacement of mass from its static equilibrium position at time $t$ [13]. The vehicle travels over a road with harmonic profile, having amplitude and wavelength $\lambda$ = 6 m. Fig. 6 shows the road bump excitation model.

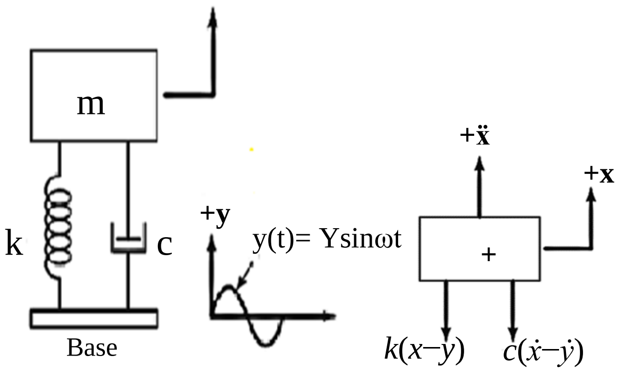



Fig. 6. Base excitation model and free body diagram [13].

Equation of motion is written by using Newton's method.

$$m\ddot{x} = -k\,(x-y) - c\,(\dot{x}-\dot{y})$$

$$\text{If } y(t) = Y\sin\,\omega\, t,\ \ x\ = X\sin\,(\omega\, t)$$

The ratio of the amplitude response $X(t)$ and base of motion $Y(t)$ is called the displacement transmissibility. The variation of $X/Y$ is given by following equation

$$\text{Transmissibility}, \frac{X}{Y} = \sqrt{\frac{1+(2\,\xi\,r)^2}{(1-r^2)^2+(2\,\xi\,r)^2}}$$

Vibration transmissibility is calculated by using Matlab program. The value of vibration transmissibility with frequency ratio by changing the damping ratio (0.2 to 0.4) is shown in Fig. 7.

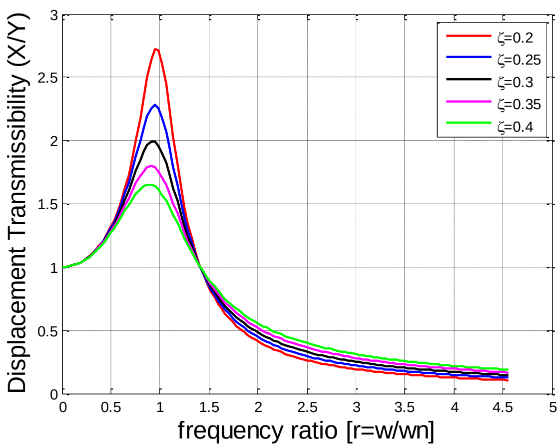



Fig. 7. Displacement transmissibility for damping ratio 0.2 to 0.4.

According to Fig 7, if the damping ratio increases, the displacement transmissibility value will decrease. The maximum displacement transmissibility is occurred near frequency ratio 1. Rao [13] determined the displacement transmissibility value with frequency ratio for damping ratio ($\zeta$=0 to 1). Maximum acceptable value of transmissibility for damping ratio (0.2 to 0.4) is 1.5 to 3.5 and it occurs in resonance case [13]. Theoretical calculation of maximum displacement transmissibility value is 2.7. So, it is less than the acceptable value.

## D. Experimental Vibration Test of Suspension System

Experimental analysis is performed to determine the natural frequency and damping ratio. Many equipment is required to perform the experimental test of vibration system for quarter car model. Accelerometer and Data acquisition system are the main components of this experimental test. Experimental procedure of vibration system is as follow:

- AC to DC converter is connected with the 12V battery
- Data acquisition system (Quantum X) is connected with AC to DC converter
- Accelerometer is mounted in automobile
- Accelerometer and data acquisition system (Quantum X) are connected via USB interface.
- Digital dynamic analyzer and PC are connected with LAN network cable.
- Natural frequency and acceleration response are obtained from catman AP V5.02 software.

The sketch of experimental setup figure is as shown in Fig. 8.

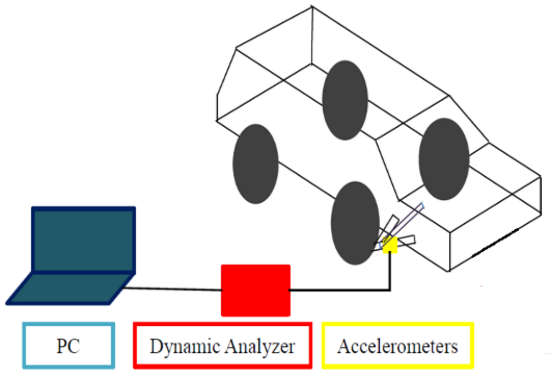



Fig. 8. Schematic diagram of experimental test.

Fig. 9 shows the connection of 12 V battery and AC to DC converter. Battery is joined with AC to DC converter.

Fig. 9. 12V battery and AC to DC converter.

Battery is connected with AC to DC converter. Converter is also connected with the data acquisition system (Quantum X). Data acquisition system (Quantum X) is attached to the accelerometer sensor with channel cable. The receiving data of the accelerometer sensors are recorded in the data recorder of the data acquisition system.

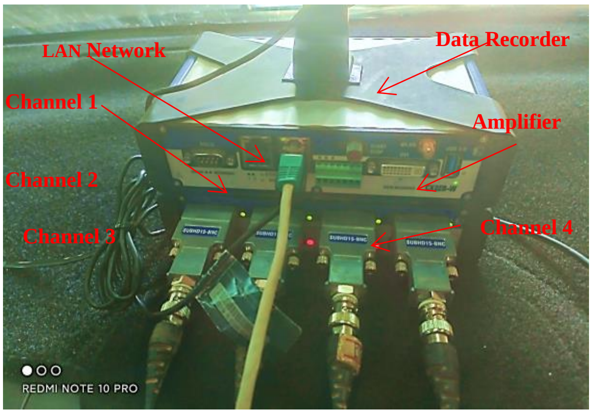



Fig. 10. Data acquisition system (Quantum X).

Fig. 10 shows the data acquisition system applied with accelerometer sensors. Data acquisition system (Quantum X) is an interface between the accelerometer sensor and PC. An accelerometer is a device that measures the acceleration of a body. Single axis accelerometer is used in this experiment which is fixed with safety form and tape to protect the accelerometer. Acceleration and frequency response have been taken by fixing accelerometer in the quarter model of automobile. Accelerometer measures the vertical vibration which is produced on bumps. Fig. 11 shows the accelerometer mounting on automobiles.

Fig. 11. Location of accelerometer.

The experiment is performed while running a car on urban road and passing the car on bumps. Accelerometer is connected to Digital dynamic analyzer via channel cables. PC and data acquisition system Quantum X are connected with LAN network cable. The measure data is directly saved in catmanAP V5.02 software of PC. Frequency and acceleration responses are obtained from catmanAP V5.02 software. The experimental damping ratio can be obtained by using the logarithmic decrement method. Logarithmic decrement method is used to extract damping ratio from the time domain responses of the experimental test. These responses are plots of time and acceleration amplitude.

## III. Result and Discussion

In this section, we discuss the vibration analysis of 3DOF quarter car model and we present the practical implications of our research. Theoretical analysis of natural frequency of the suspension system for quarter car model is 1.24 Hz. Seat mass, sprung mass and unsprung mass displacement are analyzed by changing the damping ratio (0.2 to 0.4) using Matlab software. For the experimental analysis, acceleration and frequency responses are obtained from Catman AP V5 software. Fig. 12 shows the natural frequency with time domain response for channel 1 to 4.

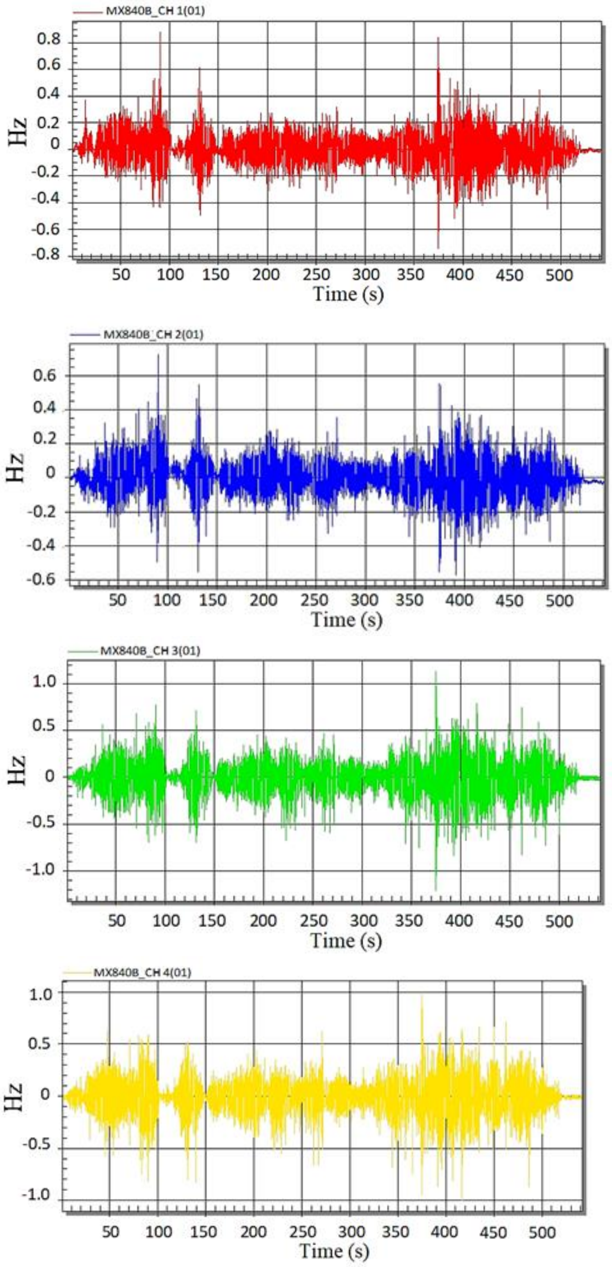
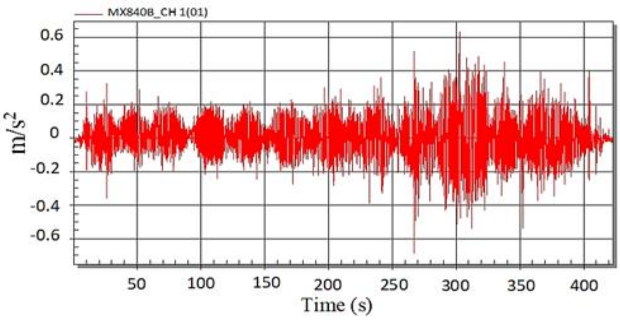


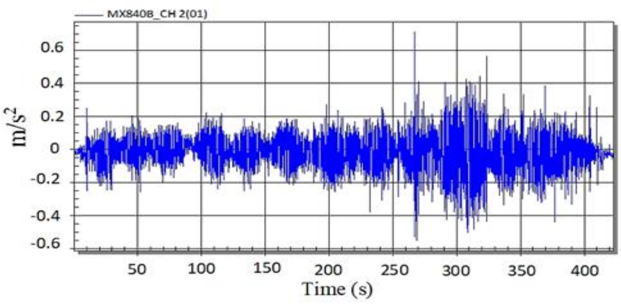

Fig. 12. Frequency response with time domain for all channels

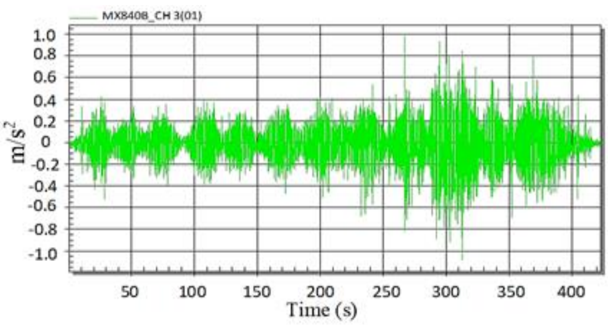
Experimental test results of the natural frequency for channel 1 to 4 is shown in Table I.

TABLE I. EXPERIMENTAL RESULTS OF NATURAL FREQUENCY FOR ALL CHANNELS

| Name | MX840B_CH1 | MX840B_CH2 | MX840B_CH3 | MX840B_CH4 |
|---|---|---|---|---|
| In preview | √ | √ | √ | √ |
| Unit | Hz | Hz | Hz | Hz |
| Samples | 108313 | 108313 | 108313 | 108313 |
| Min | **–0.7403** | **–0.5683** | **–1.208** | **–1.002** |
| Max | **0.8827** | **0.7251** | **1.134** | **1.009** |
| Mean | 0.00901 | 0.00599 | –0.00464 | –0.00581 |
| STD | 0.08360 | 0.07450 | 0.1200 | 0.1117 |
| Channel comment | – | – | – | – |
| Sample rate | 200Hz | 200Hz | 200Hz | 200Hz |
| Sensor | Hz1004R1 | Hz1004R2 | Hz970R3 | Hz966L1 |



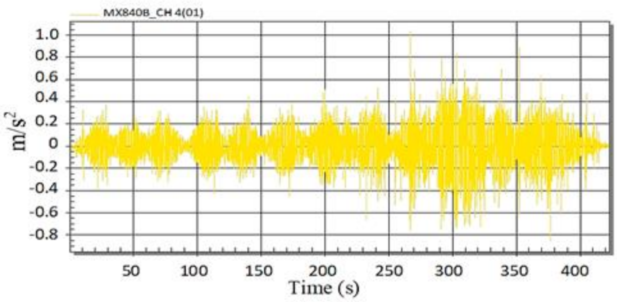




Fig. 13. Acceleration response with time domain for all channels

From these experimental test results, the maximum natural frequency of the sprung mass is 1.134 Hz. So, the result is within acceptable limit. Experimental test results of the acceleration response based on the time domain that is channel 1 to 4 shown in Fig. 13.

Table II shows the acceleration response of the experimental test for channel 1 to 4. Experimental damping ratio is obtained from the acceleration response. The response spectrum is possible to plot the maximum response system in term of acceleration, relative velocity and relative displacement using logarithmic scale. First and second amplitude are 1.031 and 0.25 from the channel 4 of acceleration and time response. The experimental damping ratio ($\zeta$) is 0.22 from logarithmic decrement method. The damping ratio of suspension system must be between 0.2 and 0.4 [13]. So, the experimental damping ratio is within the acceptable limit. Comparison of theoretical and experimental results of natural frequency, seat mass, sprung mass and unsprung mass displacement and transmissibility values are shown in Table III.

TABLE II. EXPERIMENTAL RESULTS OF ACCELERATION FOR ALL CHANNELS

| Name | MX840B_CH1 | MX840B_CH2 | MX840B_CH3 | MX840B_CH4 |
|---|---|---|---|---|
| In preview | √ | √ | √ | √ |
| Unit | $m/s^2$ | $m/s^2$ | $m/s^2$ | $m/s^2$ |
| Samples | 84572 | 84572 | 84572 | 84572 |
| Min | –0.6866 | –0.5484 | –1.083 | –0.8555 |
| Max | 0.6317 | 0.7121 | 0.9821 | 1.031 |
| Mean | 0.00503 | -0.01010 | -0.00931 | 0.00210 |
| STD | 0.08085 | 0.07617 | 0.1200 | 0.1122 |
| Channel comment | – | – | – | – |
| Sample rate | 200Hz | 200Hz | 200Hz | 200Hz |
| Sensor | Hz1004R1 | Hz1004R2 | Hz970R3 | Hz966L1 |

TABLE III. COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS

| Name | Theoretical | Experimental | Deviation (%) |
|---|---|---|---|
| Natural frequency (Hz) | 1.23 | 1.134 | 7% |
| Seat maas displacement (m) | 0.214 | 0.209 | 2% |
| Sprung mass displacement (m) | 0.15 | 0.148 | 1.3% |
| Unsprung mass displacement (m) | 0.12 | 0.117 | 2.5% |
| Displacement transmissibility | 2.7 | 2.54 | 6% |

According to the results obtained, the most relevant findings of the investigation are presented below:

- Sprung mass natural frequency for theoretical and experimental test are within the acceptable limit (1 to 2 Hz).
- It was performed the displacement analysis based on the state space modelling method using Matlab software.
- Theoretical and experimental results of displacement transmissibility have been found that the minimum percent deviation and less than the acceptable limit.
- The percent deviation is caused because theoretical and time domain response of damping ratio is slightly difference.

But all of these results are less than the acceptable value of the under damped case. So, this suspension system design is satisfied for Nissan Sunny model.

## IV. CONCLUSION

In this research, vibration analysis of suspension system for quarter car model is analyzed by theoretical and experimental approach. The main contributions of this research are showed as follows:

- The review of the past literature, 2DOF suspension system were studied. Compared with 2DOF, 3DOF suspension system with tire damping generally exist in vehicle and made it easier for vehicle to study the ride comfort.
- This work studies the transmissibility and displacement constraints of 3DOF suspension system using Matlab software by changing damping ratio.
- The novelty of this research is demonstrated the experimental test by using Catman AP V5 software.

Sprung mass natural frequency of Theoretical (1.23 Hz) and experimental (1.134 Hz) results are within 1 Hz to 2 Hz. Experimental damping ratio is obtained from the acceleration response based on the logarithmic decrement method. Experimental damping ratio (0.22) is within the acceptable limit (0.2 to 0.4). Seat mass, sprung mass and unsprung mass displacement of the suspension are calculated based on the theoretical and experimental damping ratio by using Matlab software. And then displacement transmissibility of the suspension system is analyzed by theoretical and experimental approach. The displacement transmissibility values for theoretical (2.7) and experimental (2.54) results are less than the reference value (3.5) of resonance point. In the comparison of theoretical and experimental results of 3DOF suspension system, it has been seen that the minimum percent deviation and less than the acceptable limit of under damped case. So, this suspension system is safe and comfortable for Nissan Sunny model. Recommendations will be established for the realization of future related projects, considering the aspects of design methodology, analysis and interpretation of the obtained data. In the future, further research will be conducted the vibration analysis as well as the Matlab Simulink method to analyze the displacement, velocity and acceleration of the suspension system.

## CONFLICT OF INTEREST

The authors declare no conflict of interest.

## AUTHOR CONTRIBUTIONS

YYA designed and analyzed the data. YYA wrote the manuscript. HHW, WWMS and AKL reviewed and edited the manuscript. All authors have been read and approved the final version of the manuscript.

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