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Enhancing Vehicle Ride Comfort with Optimized Tuned Mass Damper Systems

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A B S T R A C T

This study investigates the enhancement of vehicle ride comfort through the implementation of a tuned mass damper (TMD) system, specifically in a 2-degree-of-freedom (DOF) quarter-car model. The research evaluates the TMD's effectiveness in mitigating vehicle vibrations induced by two distinct road input types: random road input, simulating unpredictable surface irregularities, and sinusoidal road input, representing periodic road undulations, and focuses on optimizing TMD parameters to attenuate vibrations and improve passenger comfort according to vehicle dynamics criteria (Body acceleration, Suspension working space (SWS), and Dynamic tire load (DTL)). A comprehensive mathematical model of vehicle dynamics, incorporating the TMD, is developed and simulated by using MATLAB/Simulink program. The novelty in the research is to examine and discuss the impact of spring stiffness values on the TMD's spring stiffness and its subsequent influence on the performance of the quarter-car model. The simulations analyze the system's performance in reducing vibration amplitudes and improving ride comfort indices compared to conventional suspension systems. Results indicate significant potential for the TMD system to enhance ride comfort across different vehicle speeds and road profiles. This study provides insights into the effectiveness of TMD systems in vehicle dynamics and contributes to the development of advanced suspension technologies aimed at optimizing passenger comfort.

1. Introduction

The vehicle suspension system is fundamental in ensuring ride comfort during operation by isolating the vehicle body from road-induced vibrations. Researchers define the suspension system as the mechanism that separates the vehicle body from its wheels, a definition that highlights its critical function in vehicle dynamics. The system's effectiveness in managing dynamic maneuvers significantly influences vehicle stability and ride comfort [1, 2].

Despite advancements in suspension technologies, the complexity and cost associated with semi-active and active suspension systems remain substantial. These systems, although providing superior comfort, handling, and stability, require advanced actuators, sensors, and control algorithms, making them ideal for high-performance applications but less accessible for broader use[3-5]. Active and semi-active suspension systems redefine vehicle dynamics by offering superior comfort, handling, and stability compared to traditional passive suspensions, as these systems transform vehicle dynamics by employing advanced actuators, sensors, and control algorithms [6]. Active systems offer precise customization and dynamic response, ideal for high-performance applications. Semi-active systems provide effective damping control at a reduced cost, enhancing ride comfort and handling across different vehicle types. Both technologies mark significant advancements in improving overall driving experience, ensuring vehicles are safer, more comfortable, and better-performing on diverse road conditions [7-12].

Among the myriads of solutions proposed, and according to vehicle designers goals to achieve optimum ride comfort with lowest expense Tuned Mass Dampers (TMDs) have emerged as a sophisticated and highly effective approach to mitigating vibrations and enhancing ride comfort [13]. A TMD is a meticulously engineered device comprising a mass, spring, and damper, designed to oscillate out of phase with the primary structure be it the vehicle chassis or the seat itself. This deliberates out-of-phase oscillation facilitates the absorption and dissipation of vibrational energy, thereby substantially reducing the amplitude of oscillations [6, 14-16]. Originally conceived for architectural applications to counteract wind and seismic forces in skyscrapers and bridges, TMD and dynamic vibration absorbers (DVA) are widely implemented to suppress vibrations in various structures, including ships, wires, bridges, and buildings. While a DVA comprises a spring element and a mass, a TMD is essentially a damped DVA, augmented by the addition of a damper element parallel to the DVA [15, 17-19]. TMDs have been adapted to the automotive sector, yielding remarkable results [20, 21]. The integration of TMDs into vehicle suspension systems presents a plethora of advantages. Firstly, TMDs offer targeted vibration control, enabling precise tuning to specific frequencies that adversely affect vehicle components and passenger comfort. This precision results in a significant attenuation of vibrations transmitted to the driver and passengers, thereby enhancing overall ride quality [22, 23].

In recent years, the demand for enhanced ride comfort and stability has significantly grown in the automotive industry, driven by consumer expectations for smoother, safer, and more enjoyable driving experiences. One promising solution to achieve these objectives is the integration of Tuned Mass Damper (TMD) systems into vehicle suspension designs. Unlike traditional suspension systems, which can struggle to mitigate vibrations effectively across a wide range of road conditions, TMDs offer a versatile and efficient approach to controlling unwanted oscillations. By absorbing and counteracting vibrations, TMD systems can significantly reduce body acceleration, improve suspension of working space, and minimize dynamic tire load, contributing to both passenger comfort and vehicle durability. With advancements in materials and modeling techniques, TMDs are becoming increasingly feasible for large-scale industry adoption, presenting an opportunity to elevate vehicle design and meet the stringent demands of modern consumers. Consequently, exploring the application of TMDs in vehicle dynamics is not only timely but essential for developing next-generation suspension systems that align with the automotive industry's commitment to innovation, safety, and comfort [24].

In this study, Tuned Mass Damper (TMD) on the performance metrics of a 2-DOF (Degree of Freedom) quarter-car model equipped with a passive suspension system is investigated. MATLAB/Simulink model is established, considering two types of excitations: random road input and a sinusoidal input with a frequency of 2 Hz. Simulations were performed using TMD mass ratios of 2%, 5%, 10%, and 20% relative to the sprung mass. The investigation involved tuning the spring stiffness and adjusting various ratios to optimize ride comfort by minimizing body acceleration, suspension working space (SWS), and dynamic tire load (DTL). Moreover, the effect of a non-linear sprung mass stiffness with Tuned Mass Damper (TMD) system is investigated. Tuning of the spring stiffness to achieve better ride comfort that the effective TMD is applied.

This study focuses on the positive impact of Tuned Mass Dampers (TMD) on vehicle vibrations, specifically enhancing ride comfort performance. However, there is an opportunity for further exploration by examining how an integrated TMD could contribute to vehicle stability. Future research will aim to incorporate a TMD system into the vehicle model, allowing for a comprehensive analysis of its effects on stability and overall performance. By investigating the role of TMDs in stability enhancement, this continued work could offer valuable insights into advanced stability control measures, ultimately contributing to safer and more reliable vehicle designs.

2. Modeling and analysis of vehicle suspension system by using quarter car model

Figure 1(a) illustrates a quarter-car model equipped with a conventional passive suspension system. In this configuration, M_w denotes the un-sprung mass, while M_b represents the sprung mass. The parameters K_t, K_s, and C_b correspond to the wheel stiffness, suspension stiffness, and damping coefficients of the passive suspension system, respectively. Figure 1(b) depicts a modified version of the model where an additional mass M_{TMD} is connected to the sprung mass via spring and damper elements to create a Tuned Mass Damper (TMD). Here K_{TMD} and C_{TMD} denote the stiffness and damping coefficients of the TMD, respectively [25].

2.1. Equation of vehicle quarter car model with TMD

According to Figure (1) the Equation of motion of the 2-DOF model based on Newton's second law of motion are given below [25]

$$M_b\ddot{x}_2 + C_b(\dot{x}_1 - \dot{x}_2) + K_s(x_1 - x_2) + C_{TMD}(\dot{x}_2 - \dot{x}_3) + K_{TMD}(x_2 - x_3) = 0 \quad (1)$$

$$M_w\ddot{x}_1 + C_b(\dot{x}_2 - \dot{x}_1) + K_s(x_2 - x_1) + K_t(x_1 - x_0) = 0 \quad (2)$$

$$M_{TMD}\ddot{x}_3 + C_{TMD}(\dot{x}_3 - \dot{x}_2) + K_{TMD}(x_3 - x_2) = 0 \quad (3)$$

We could drive equations in matrix formula by following the Equation below:

$$M\ddot{X} + C\dot{X} + KX = F \quad (4)$$

Where M represents mass matrix formula by extracting $\ddot{x}_1, \ddot{x}_2, and\ \ddot{x}_3$

$$M = \begin{bmatrix} M_b & 0 & 0 \\ 0 & M_w & 0 \\ 0 & 0 & M_{TMD} \end{bmatrix} \quad (5)$$

and C represents damping matrix formula by extracting $\dot{x}_1, \dot{x}_2, and\ \dot{x}_3$

$$C = \begin{bmatrix} -C_b & C_b & 0 \\ C_b & -C_b - C_{TMD} & C_{TMD} \\ 0 & -C_{TMD} & C_{TMD} \end{bmatrix} \quad (6)$$

and K represents suspension, TMD, and tire stiffness matrix

$$K = \begin{bmatrix} K_s + K_t & -K_s & -K_t \\ -K_s & K_s + K_{TMD} & -K_{TMD} \\ -K_t & -K_{TMD} & K_{TMD} \end{bmatrix} \quad (7)$$

If x_0 represents the external force (road input) the force vector F matrix could express as the following

$$F = \begin{bmatrix} K_t x_0 \\ 0 \\ 0 \end{bmatrix} \quad (8)$$

The displacement vector matrix expresses as

$$X = \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} \quad (9)$$

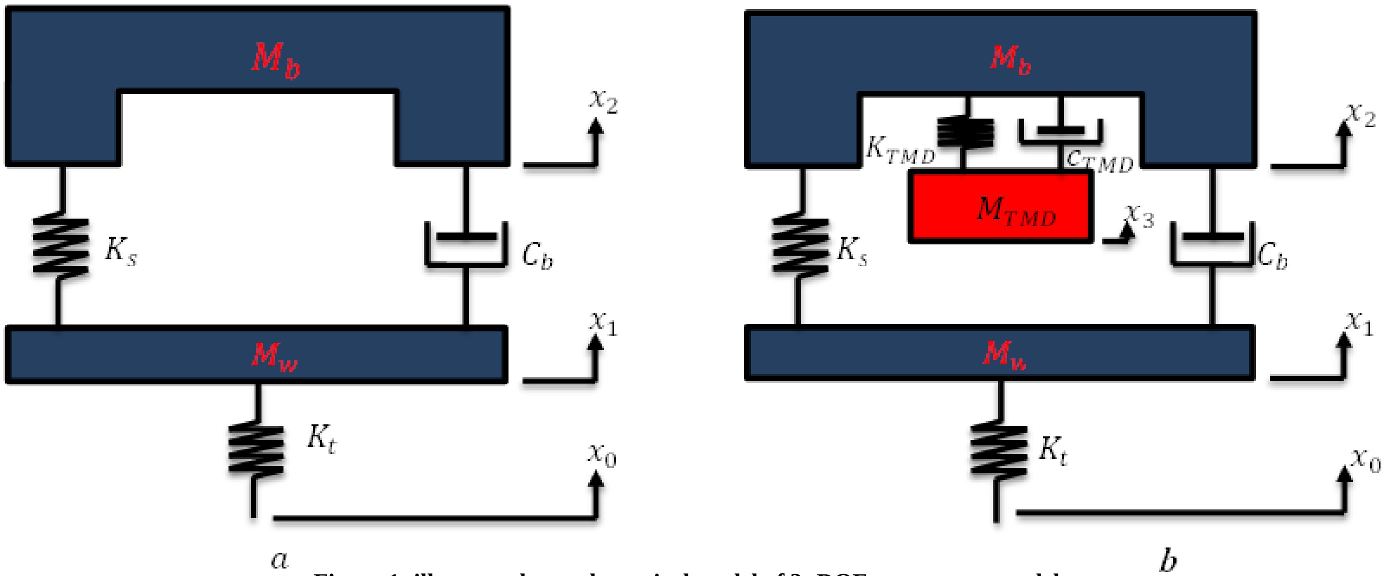

Figure 1: illustrate the mathematical model of 2- DOF quarter car model a) passive quarter car mode b) TMD quarter car mode [25]

2.2. Model parameters assumption

The parameters of the Tuned Mass Damper (TMD) were selected with the assumption that the protected structure corresponds to the sprung mass of the vehicle. Consequently, the mass ratio of the TMD system is defined as follows in Equation (4).

$$\mu = \frac{M_{TMD}}{M_b} \tag{10}$$

The stiffness coefficient was determined through optimization for the 2-DOF TMD system based on H_2 optimization, as specified in the Equation below [21, 25]

$$K_{TMD} = \frac{1}{2} \times \frac{(\mu+2)\times K_s \times \mu}{(\mu+1)^2} \tag{11}$$

Table 1 investigate quarter car model assumptions[21] and TMD parameters assumption investigated by Equation below:

Table 1: Quarter car model parameters

Parameter	Symbol	Value	unit
Sprung mass	M_b	332	Kg
Un-sprung mass	M_w	40	Kg
Spring stiffness	K_s	25000	N/m
Suspension damper	C_b	1500	N.s/m
Tire stiffness	K_t	200000	N/m

3. Analysis and Results of TMD quarter car model

Using the equations of the 2-DOF quarter-car model described above, MATLAB/Simulink was employed to analyze the performance of the passive suspension system. This analysis aimed to evaluate the impact of the Tuned Mass Damper (TMD) on the suspension characteristics of the model. The performance criteria, including body acceleration, dynamic tire load (DTL), and suspension working space (SWS), were assessed under two types of external forces: random road input and sinusoidal input with a frequency of 2 Hz. Table 2 investigates the parameter of TMD according to μ in Equation (10).

Table 2: TMD parameters assumption [25]

Parameter	Symbol	Value	Unit
Tuned mass damper mass	M_{TMD}	$\mu * M_b$	kg
Tuned mass damper stiffness	K_{TMD}	$\frac{1}{2} \times \frac{(\mu+2)\times K_s \times \mu}{(\mu+1)^2}$	Kg
Tuned mass damper damper value	C_{TMD}	$\delta * M_{TMD} * K_{TMD}$	N/m

where δ is damping co-efficient and it assumed to be 0.2. in Table 3 investigation of mass, stiffness and damper value according to the assumption that illustrates previous.

Table 3: TMD parameters value

Ratio (μ)	M_{TMD}	K_{TMD}	C_{TMD}
2%	6.64	485	12
5%	16.6	1160	27
10%	33.2	2170	54
20%	66.4	3820	100

4. Road response of quarter car model with passive and TMD suspension system

4.1. Random road input excitation

To define the vehicle's dynamic response during maneuvering, it is essential to introduce disturbance inputs. The road surface serves as the primary input for investigating vehicle dynamics. In this context, a random road profile was selected as a theoretical representation of road roughness with a length of 400 meters. The random road surface, characterized by its Power Spectral Density (PSD), is further described figure (2-a) and (2-b) presented road profile elevation and power spectral density (PSD) algorithm for road input type[26]. The Equation below describes the rando road input according to velocity 20 m/sec.

$$PSD(f) = \frac{G * V^{p-1}}{f^p} \tag{12}$$

where G is the road roughness coefficient, V is the vehicle speed in m/s, f is the road frequency, and p is the constant.

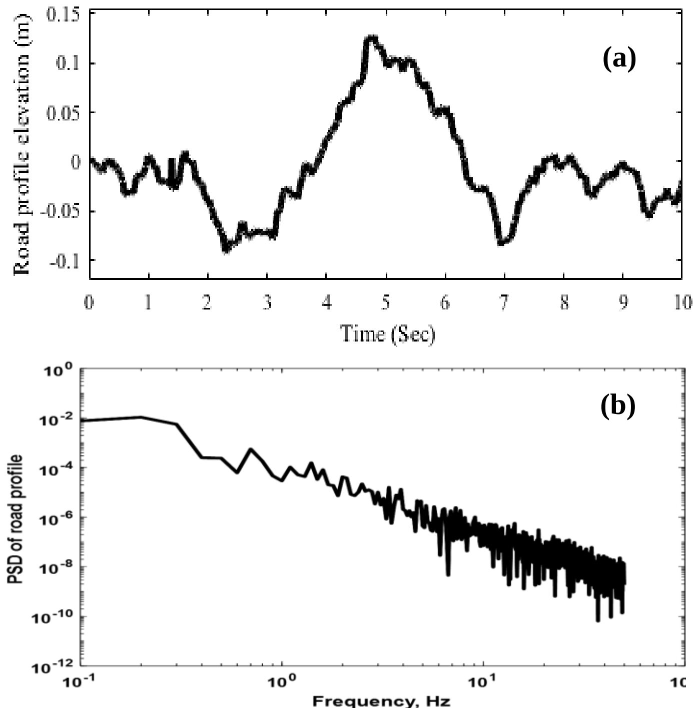

Figure 2: Random road input

The 2-DOF quarter car model was subjected to random road input as explained in figure (3). The simulation is based on a comparison between passive suspension system characteristics and when we applied TMD system to it. The figures are presented according to PSD algorithm to show the characteristics of suspension system in frequency domain and the improvement of TMD system the criteria of the system. The figures show a perfect improvement in vehicle performance (Body acceleration, SWS, DTL) according to the changing in TMD parameters that exposed with ratios μ that explained in Table-3. Figures from 3 to 6 showed the model performance according to ratio $2\%, 5\%, 10\%, and\ 20\%$ respectively. The optimum ride comfort characteristics showed at ratio 20%. Table 4 discusses the improvements in ride comfort criteria due to TMD system according to root mean square (RMS).

Table 4: The improvement in suspension performance due to TMD changing ratioμ for random road input

Ratio (μ)	***Body acceleration***	***SWS***	***DTL***
2%	4%	5%	2.2%
5%	8%	11%	4%
10%	11%	16.2%	6%
20%	13.3%	21.4%	*8%*

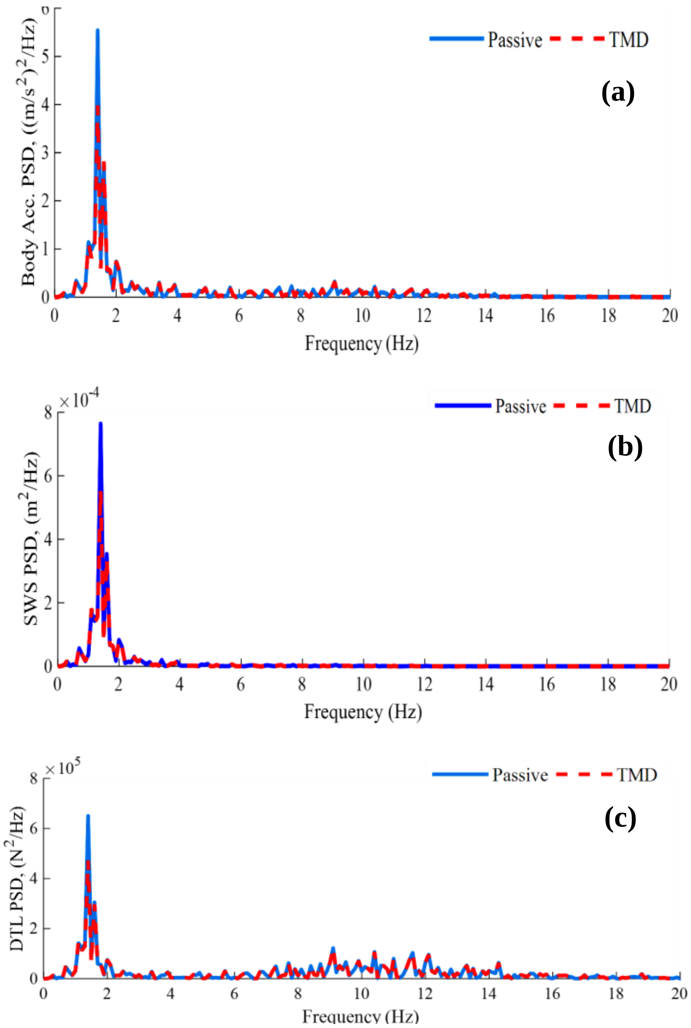

Figure 2: Spectrum for Ratio (μ) =2% (a) Sprung mass acceleration, c) Suspension working space, (c) Dynamic tire load

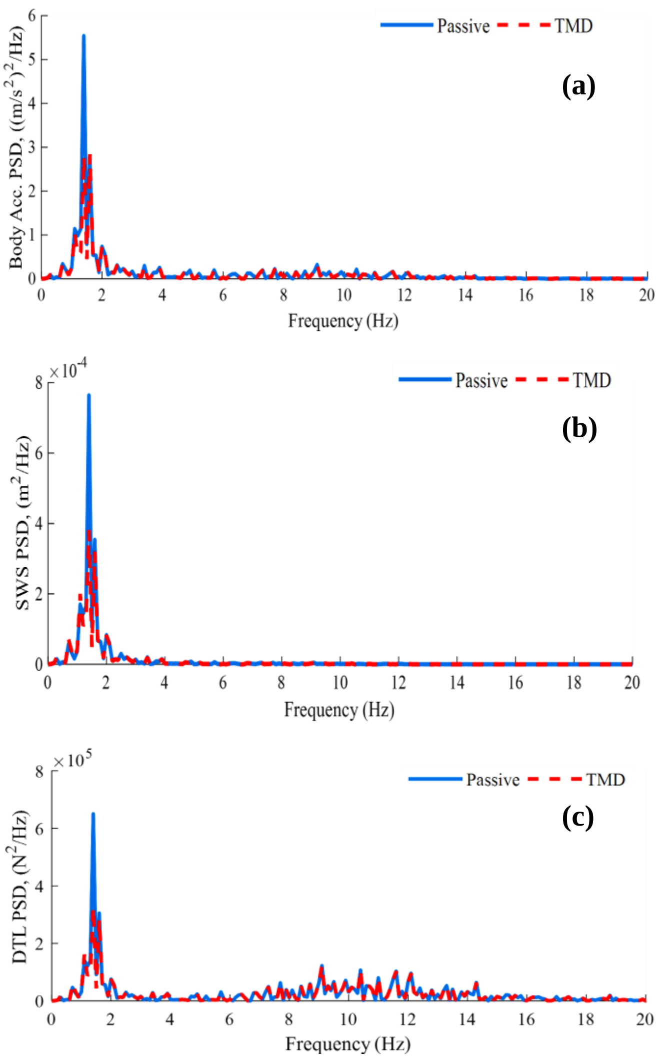

Figure 4: Spectrum for Ratio (μ) =5% (a) Sprung mass acceleration, c) Suspension working space, (c) Dynamic tire load

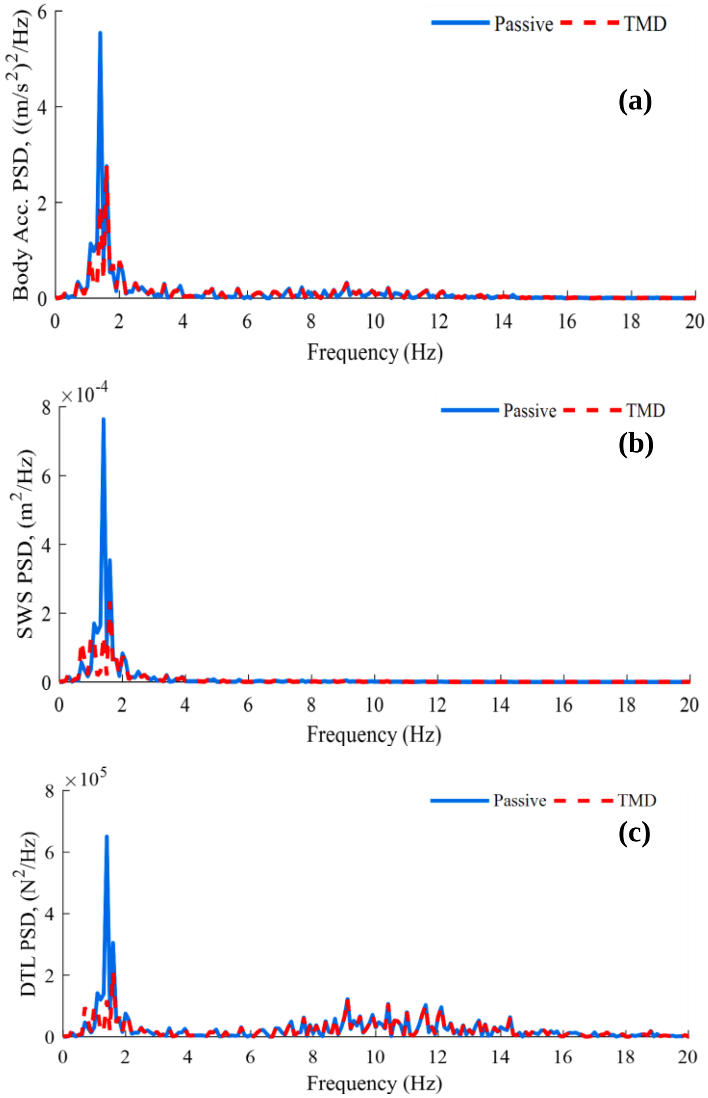

Figure 5: Spectrum for Ratio (μ) =10% (a) Sprung mass acceleration, c) Suspension working space, (c) Dynamic tire load

4.2. Sinusoidal input excitation

In this section investigates the influence of TMD in performance of suspension system characteristics according to the ratio of TMD parameters assumptions in table 3. Figure 7 illustrates input according to time with an amplitude of 0.06 m and frequency 2Hz.

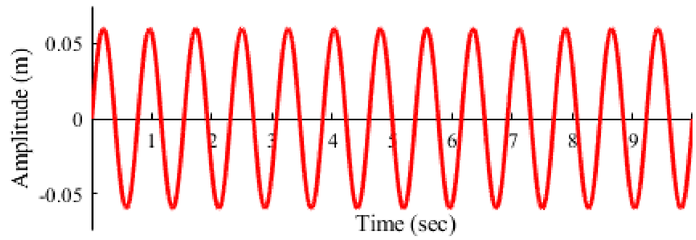

Figure 3: Sinusoidal input excitation

Based on research goal to illustrate the effect of TMD`s parameters change according to ratio in equation (10) the figures below investigate the improvement in performance when we applied TMD`s system to suspension system. Figures 8 to 11 show the difference between passive system without TMD and with TMD at different ratio μ as discussed in table-3, also table-5 presents the improvement percentage due to changes in parameters according to the ratios case of study. And according to data discussed in table-3 the percentage for $\mu = 20\%$ shows best improvement for all performances (body acceleration, SWS, and DTL).

Table 5: The improvement in suspension performance due to TMD changing ratio μ for sinusoidal input

Ratio (μ)	Body acceleration	SWS	DTL
2%	14%	9.5%	10%
5%	31.5%	24%	25%
10%	49%	42%	43%
20%	60%	59%	61%

4.3. The influence of tuning spring stiffness on TMD performance

The idea of TMD system is to eliminate vibration due to external excitation, the aim of that part is to achieve the optimum suspension performance by changing the value of spring stiffness which affects TMD spring stiffness. The tuning or changing will be applied by ratio between sprung mass and spring stiffness according to the equation below

$$ratio = \frac{spring\ stiffness}{sprung\ mass} \quad (13)$$

As shown in histogram below in Figures (12), (13), and (14) exposed by sinusoidal input. The ratio between mass and spring stiffness influences body acceleration, SWS, and DTL. However, the tuning in spring stiffness didn`t affect TMD improvement significantly in random road input applied on the that case of study as the sinusoidal input and the improvement is closely to the results of optimized value of spring stiffness in the typical model assumptions for ratio $\mu = 2\%$ but slowly the improvement increased for body acceleration to reach to the

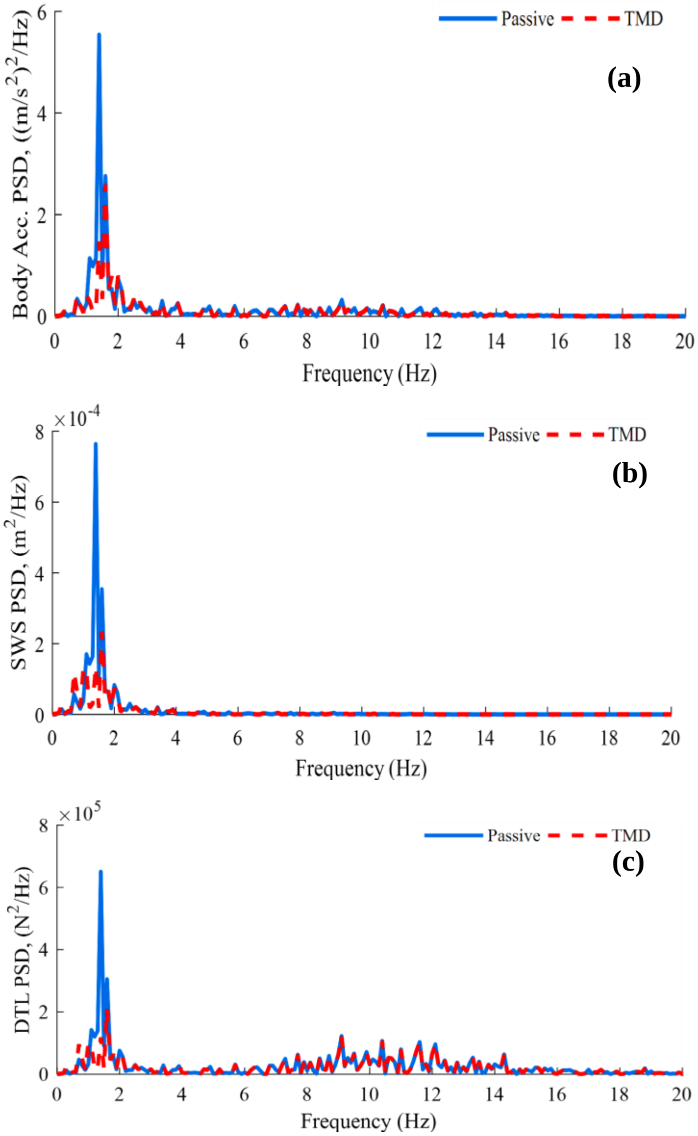

Figure 6: Spectrum for Ratio (μ) =20% (a) Sprung mass acceleration, c) Suspension working space, (c) Dynamic tire load

optimum at ratio $\mu = 20\%$ however in SWS the optimum was at ratio $\mu = 20\%$ at ratio 80 of sprung mass and started to decrease according to the ratio of sprung mass, on the other hand, DTL value tuning it starts at lowest value then increase to reach the perfect improvement at ratio $\mu = 20\%$ at sprung mass ratio 100 then started to decrease according to sprung mass ratio as it shown in figures from (12) to (17).

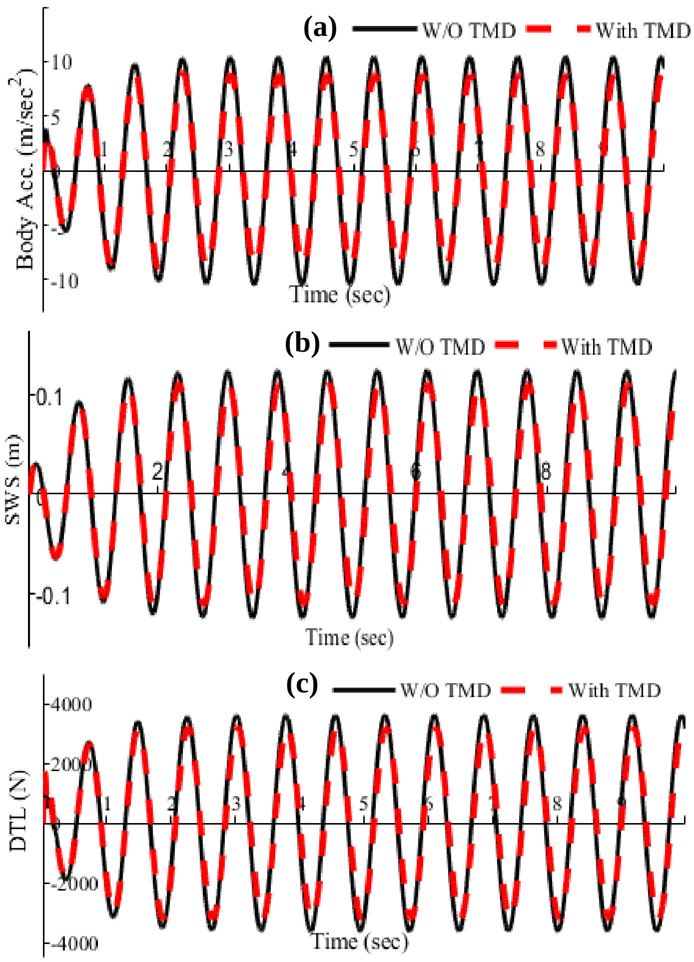

Figure 8: Responses for Ratio (μ) =2% (a) Sprung mass acceleration, (b) Suspension working space, (c) Dynamic tire load

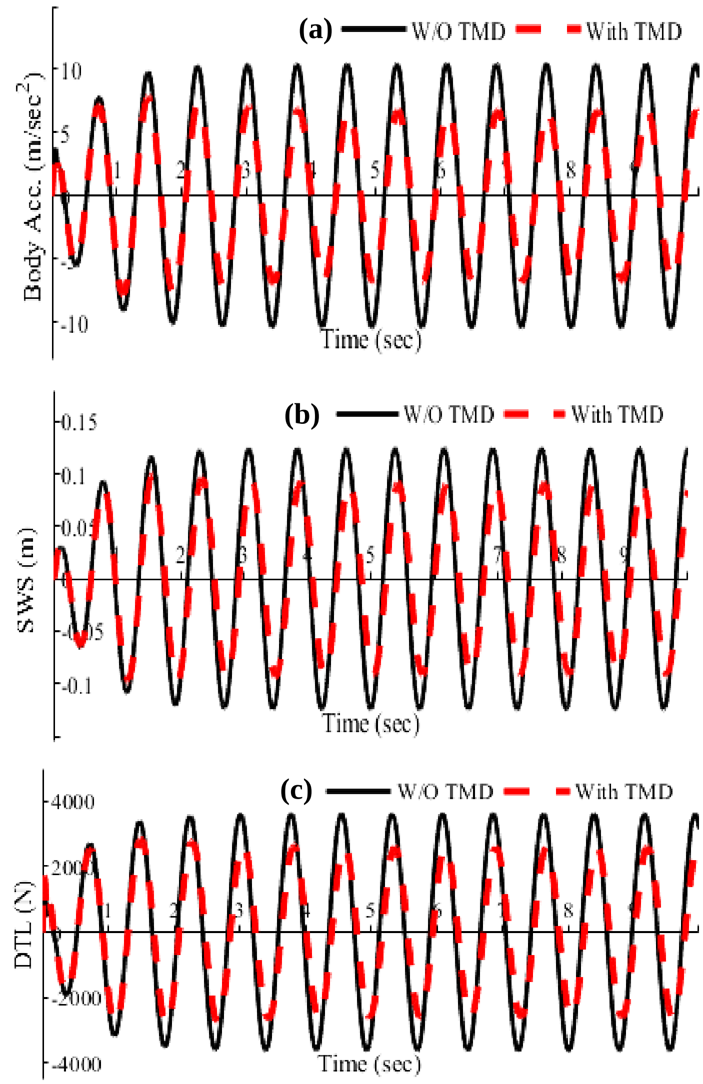

Figure 4: Responses for Ratio (μ) =5% (a) Sprung mass acceleration, (b) Suspension working space, (c) Dynamic tire load

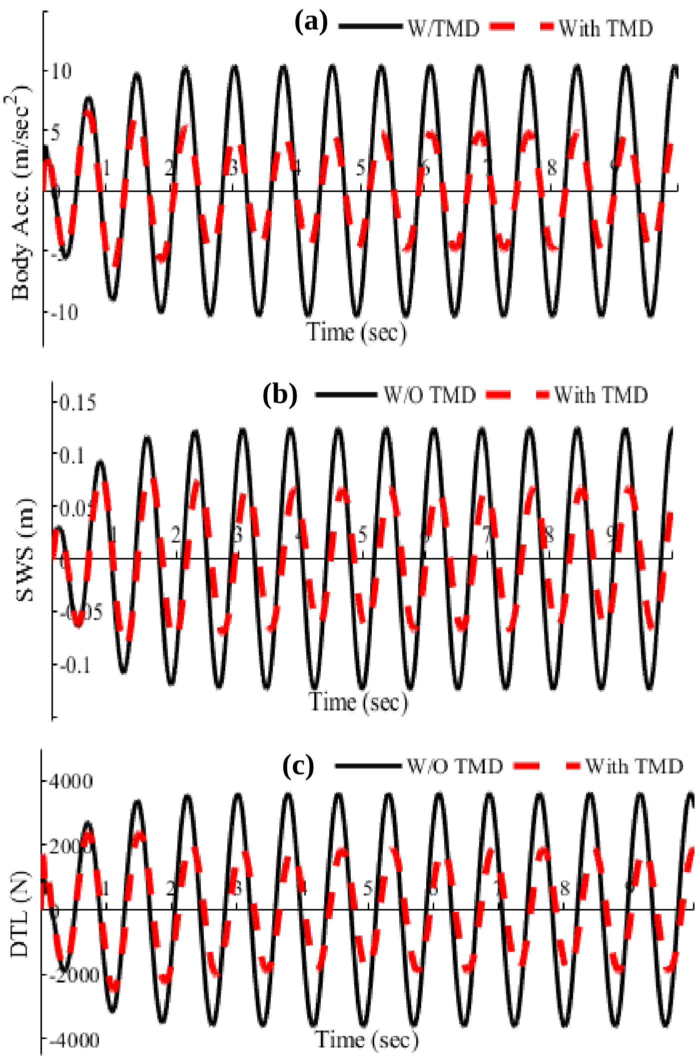

Figure 10: Responses for Ratio (μ) =10% (a) Sprung mass acceleration, (b) Suspension working space, (c) Dynamic tire load

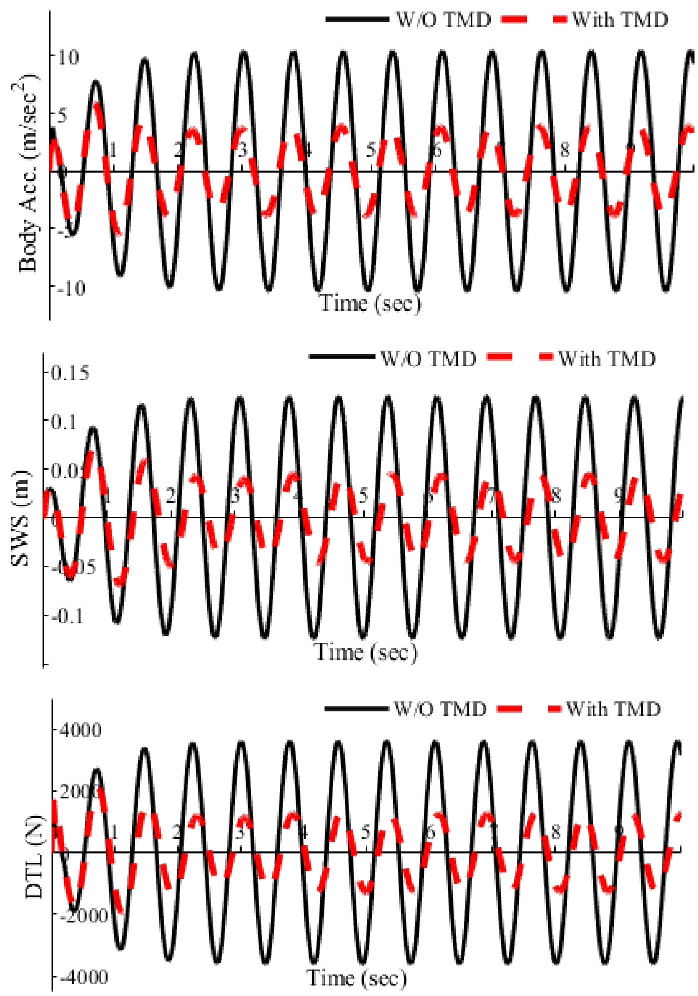

Figure 11: Responses for Ratio (μ) =20% (a) Sprung mass acceleration, (b) Suspension working space, (c) Dynamic tire load

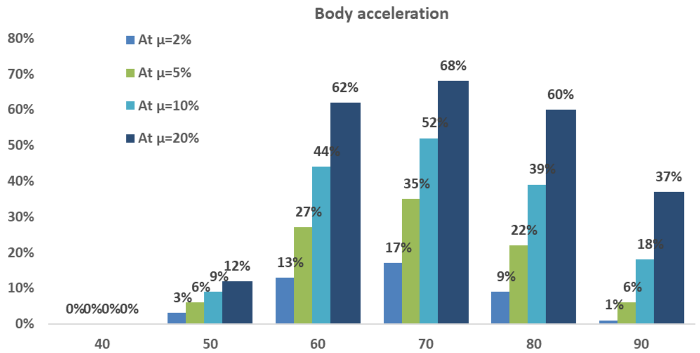

Figure 5: Percentage of sprung mass acceleration improvement

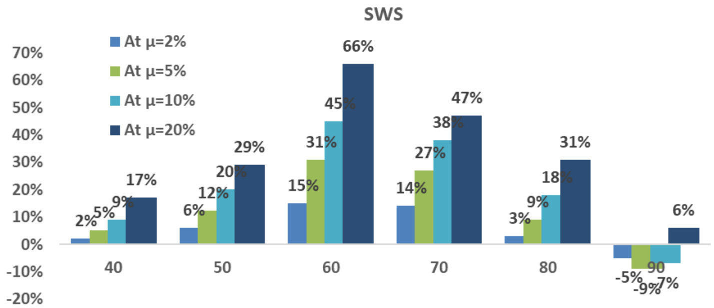

Figure 13: Percentage of suspension working space improvement

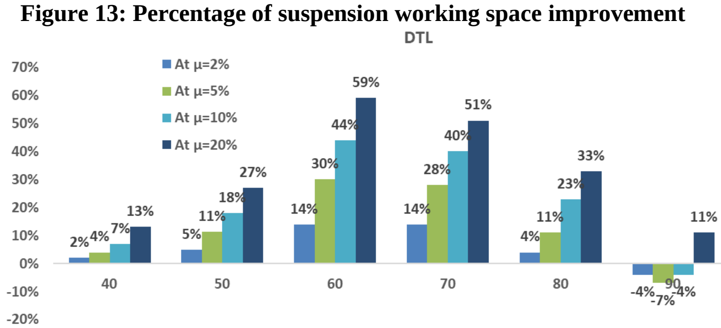

Figure 14: Percentage of dynamic tire load improvement

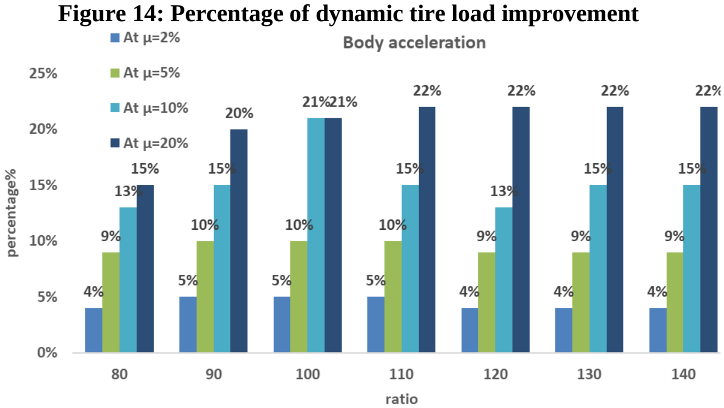

Figure 6: Percentage of sprung mass acceleration improvement

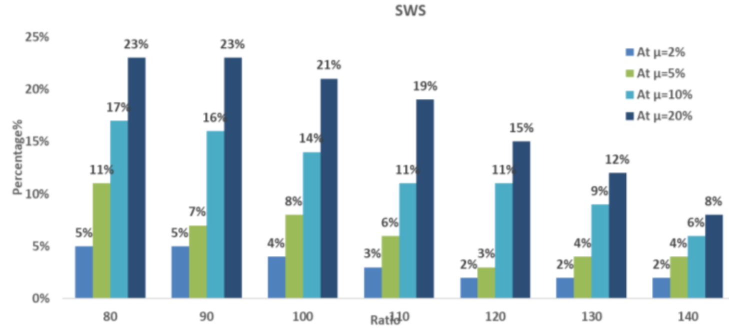

Figure 16: Percentage of suspension working space improvement

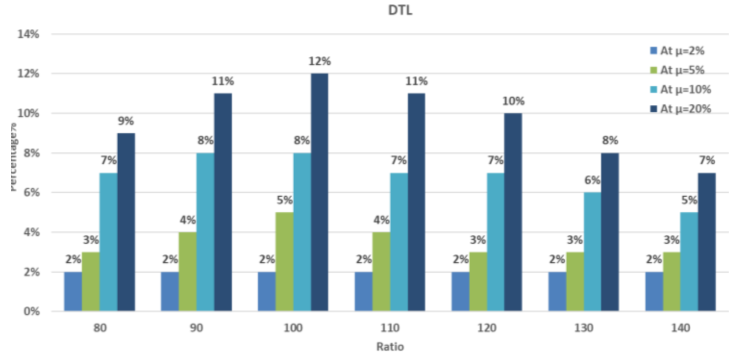

Figure 17: Percentage of dynamic tire load improvement

5. Conclusions

This work presents the quarter-car model with the released TMD, which validates that TMD stabilizes in-vehicle vibrations and improved overall dynamics equivalent to ride comfort improvement while passing through irregular terrains.The test was done based on two types of road inputs random and sinusoidal with the different percentages (say 2 to 20%) of TMD parameters.

Under random road input, 20% proved to be the best TMD parameter with body acceleration (13.3%), suspension travel range (21.4%) and dynamic tire load (8%) improvements through the whole frequency spectrum. For sinusoidal input, the 20% TMD parameter did offer significant improvements (on average: 60% in body acceleration; up to 50 kN/m/rad in suspension working space and dynamic tire loads).

Studies on variable spring stiffness showed an increase in sprung mass acceleration percentage for random road input, of 22% when μ=20%; also working space for suspension dropped from 23% to 8%. Tire load dynamic rise went up to 12% at μ=20% but then down to 7%. The results show that the suspension system with TMD worked best at 60:1 as a ratio between spring stiffness and sprung mass to be and μ=20% especially under sinusoidal input

Conflict of Interest

The authors declare no conflict of interest.

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