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MODELING AND ANALYSIS OF VIBRATION AMPLITUDE REDUCTION IN AN IN-WHEEL ELECTRIC VEHICLE USING A REGENERATIVE TUNE MASS DAMPER (TMD)

Margiasih Putri Liana[1]*, I Kadek Warjaya[1]

[1] *Mechanical Engineering, Faculty of Engineering, Universitas Cahaya Surya, Kediri (64129), Indonesia*

*Corresponding Author Email: putriliana@ucs.ac.id

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ABSTRACT

This is a digest of the paper. One of the features of in-wheel electric vehicles is an increase in unsprung mass due to the integration of the motor into the wheel. It's resulting in both increased vibration amplitude and reduced Vehicle comfort and stability. The purpose of this study is to model and analyze the reduction in vibration amplitude in the suspension system of in-wheel electric vehicles using an electromagnetic-based Regenerative Tuned Mass Damper (TMD). A dynamic model was developed using the quarter-car approach and transformed to state-space form for simulation in MATLAB. The parameters used were TMD masses of 5 to 15 kg with an increase of 1 kg, with road excitation testing conducted using a sinusoidal wave with an amplitude of 0.02 m and a frequency of 5 Hz. The results were then evaluated based on the Root Mean Square (RMS) value of vehicle unsprung mass acceleration as an indicator of vibration-damping performance. The results show that implementing TMD improves vibration attenuation compared to the baseline system, increasing vibration reduction from 16.97% to 18.81%. The system performs best at a TMD mass of around 8 kg, while achieves maximal damping effectiveness. However, increasing TMD mass beyond the ideal point decreases vibration attenuation efficacy, indicating a detuning impact between the TMD and the primary system. In contrast, the regenerative TMD generates electrical energy that increases with mass, with output power increasing from 5.04 W to 11.17 W. This study contributes to the development of adaptive suspension system designs to minimize the risk of failure at the in-wheel motor of electric vehicles while generating energy recovery.

Keywords: *Regenerative Tuned Mass Damper; Dynamic Model; A Quarter car in-wheel; Vibrating Amplitude Reduction.*

I. INTRODUCTION

The automotive industry is quickly moving toward electric vehicles (EVs) because fossil fuel reserves are running out and pollution is getting worse (Deepak et al., 2023). Electric vehicles with an in-wheel motor (IWM) system offer several advantages over central drive systems, such as higher transmission efficiency, simplified mechanical structure, and control flexibility through independent wheels (Nichiţelea & Unguritu, 2022). Despite its great potential, one of the main drawbacks of IWM technology is a significant increase in un sprung mass resulting from the addition of motor weight directly to the wheels (Kim et al., 2024). This increase in un sprung mass negatively impacts the vehicle's vertical dynamics, leading to greater vibration amplitude, and risks reducing ride comfort and driving stability(Arman Haditiansyah et al., 2025).

This problem can be solved by using a Tuned Mass Damper (TMD) in the suspension system, which is an effective method for reducing forced vibrations caused by the addition of un sprung mass (Ibrahim et al., 2025). In this study, the TMD was modified into an electromagnetic Regenerative Tuned Mass Damper (TMD) with a back-iron architecture. The design not only used as a shock absorber but is also capable of converting mechanical oscillation energy into electrical energy through the principle of the electromotive force (EMF). Therefore, this study aims to model and analyze the reduction of vibration amplitude in the IWM electric vehicle suspension system using an Electromagnetic Regenerative Tuned Mass Damper (TMD) by varying the TMD mass to evaluate the trade-off between damping performance and the potential regenerative power generated.

Many studies on vehicle suspension systems have been conducted to reduce vibrations caused by road surface profiles. The quarter-car model approach is frequently used due to its simplicity and its ability to accurately represent the vertical dynamics of the sprung mass (vehicle body) and un-sprung mass (wheels and axles) (Deepak et al., 2023). To minimize vehicle body acceleration, various active and semi-active suspension control methods have been extensively evaluated, ranging from General Algorithm (Desai et al., 2021), Proportional-Integral-Derivative (PID) control (Mohammed Matrood & Ahmed Nassar, 2021) and Linear Quadratic Regulator (LQR) to hybrid methods based Magneto-rheological damper lookup table (Turnip & Panggabean, n.d.). In the case of in-wheel motor (IWM) electric vehicles, the addition of motor mass to the wheel increases the unsprung mass (Beauduin et al., 2017; Zhao & Fan, 2023), which significantly affects the dynamic characteristics of the system, particularly shifting the resonance frequency and deteriorating ride comfort (Chen et al., 2016; Nie et al., 2018; Wu et al., 2025). The experimental testing and simulation analysis in the thesis by Mahmood (Mahmood, 2023) showed that an increase in the un sprung mass from the IWM motor caused a surge in the Root Mean Square (RMS) value of the body vibration acceleration, which directly degrades passenger comfort and safety.

Several attempts have been made to isolate these vibrations (Liu et al., 2017) demonstrated that the addition of an in-wheel vibration absorber proved highly effective in reducing wheel and motor vibrations. To fill this research gap, this study adopts and further analyzes the Electromagnetic Regenerative Tuned Mass Damper (TMD) design previously proposed by (Kopylov et al., 2020). The back-iron architecture in the TMD was chosen because it provides the best power output as an energy harvester while maintaining a compact form for implementation in a vehicle's suspension. Even though studies on suspension control and TMDs exist, there remains a research gap regarding how variations in the TMD's counterweight mass specifically affect the effectiveness of reducing the wheel motor RMS acceleration in IWM electric vehicles, which are equipped with additional un-sprung mass compared to internal combustion engine vehicles and conventional electric vehicles.

Even though studies on suspension control and TMDs exist, there remains a research gap regarding how variations in the TMD's counterweight mass specifically affect the effectiveness of reducing the wheel motor RMS acceleration in IWM electric vehicles, which are equipped with additional unsprung mass compared to internal combustion engine vehicles and conventional electric vehicles. To better identify the research gap and position the contribution of this study within the existing body of knowledge, a systematic comparison of previous studies is presented in Table 1.

Table 1. Comparison of Previous Study

Author	**Method**	**Main Findings**	**Limitation**
Liu et al. (2017)	In-wheel vibration absorber	Reduced wheel vibration	No energy harvesting
Kopylov et al. (2020)	Electromagnetic Regenerative TMD	Vibration reduction and power generation	No optimization of TMD mass
Zhao & Fan (2023)	Dynamic vibration absorber	Improved ride comfort	No regenerative mechanism
Wu et al. (2025)	Unsprung mass evaluation	Identified comfort degradation due to IWM	No vibration mitigation strategy
Proposed Study	Regenerative TMD with mass variation	Simultaneous vibration attenuation and energy harvesting	-

As shown in Table 1, previous studies have extensively investigated vibration reduction techniques, vibration absorbers, and regenerative damping systems for vehicle suspension applications. However, limited attention has been given to evaluating the influence of regenerative TMD mass variation on both vibration attenuation and energy harvesting performance in in-wheel motor electric vehicles. Regenerative TMD systems have been studied in the literature, but the effect of TMD mass variation in in-wheel motor electric vehicles has not been widely studied. This study aims to determine the optimal TMD mass which provides a compromise between vibration attenuation and energy harvesting performance under harmonic road excitation.

II. METHOD

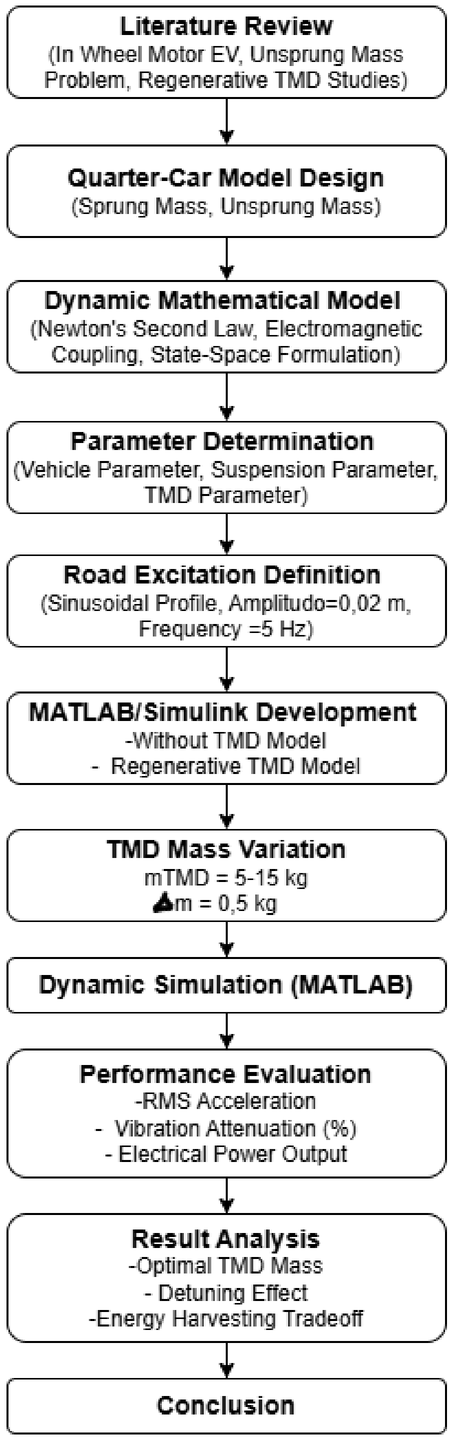


Figure 1. Research Methodology Flowchart

A systematic research methodology was developed to investigate the effectiveness of the proposed regenerative Tuned Mass Damper (TMD) in reducing vibration amplitude and harvesting energy in an in-wheel motor electric vehicle. The methodology involves several steps like quarter car model development, mathematical formulation of suspension dynamics, parameter determination, road excitation modelling, implementation in MATLAB/Simulink and performance evaluation under different TMD mass variations. Figure 1 shows the overall framework of the research procedure.

A. Model Design of Car Suspension

This research uses a mathematical modeling and numerical simulation approach to analyze the dynamic response of a vehicle suspension system. Vehicle dynamics are modeled using a quarter-car approach integrated with Regenerative TMD.

Customized design of the TMD application is integrated into the vehicle's suspension system and therefore requires a compact electromagnetic device. Its compact design helps save space and enables an electromagnetic approach. Consequently, a back-iron design was selected to provide these features.

This back-iron design was chosen because the back-iron construction allows the total mass of the oscillating part to be adjusted by modifying the dimensions of the back-iron section independently of the magnet section. Consequently, the back-iron architecture was adopted for the current implementation, as shown in Figure 2(a).

Figure 2(a) illustrates a ring magnet configuration placed between two ferromagnetic components: the upper and lower back-iron sections. The "T"-shaped structure of the upper component is designed to form an air gap with the inner surface of the lower component. The solenoid is then positioned within the annular air gap, which is filled with a homogeneous magnetic flux. In this design, the magnetic induction vector is directed radially outward from the core of the structure. Based on Ampere's Law, the relative motion between the solenoid and the back-iron will induce an electromotive force (EMF) in the wire, which simultaneously generates a viscous damping force. The electric current generated by the EMF is supplied directly to the electric vehicle system or stored in a battery. The implementation of the ETMD design in vehicle suspension is carried out in parallel, as shown in Figure 2(b).

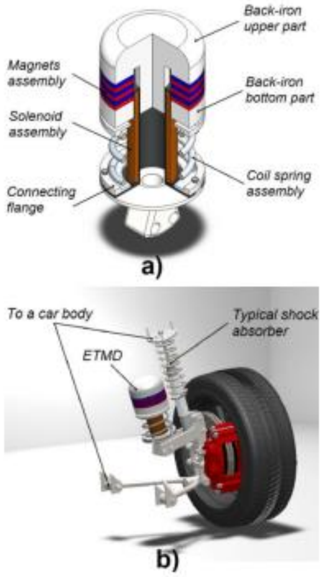


Figure 2. (a). Electromagnetic TMD based back iron design, (b) ETMD implementation in a car suspension. (Kopylov et al., 2020)

B. Dynamic Modelling

The analysis of the effectiveness of the Regenerative Tuned Mass Damper (TMD) system focuses on reducing vibration amplitude in in-wheel electric vehicles. The primary focus of this methodology is on the dynamic modeling of the suspension system and the integration of the TMD to address the issue of increased un-sprung mass resulting from the installation of the motor on the wheel. The system under study is modeled using Free Body Diagram with a modified quarter-car model approach, incorporating the TMD unit, see figure 3.

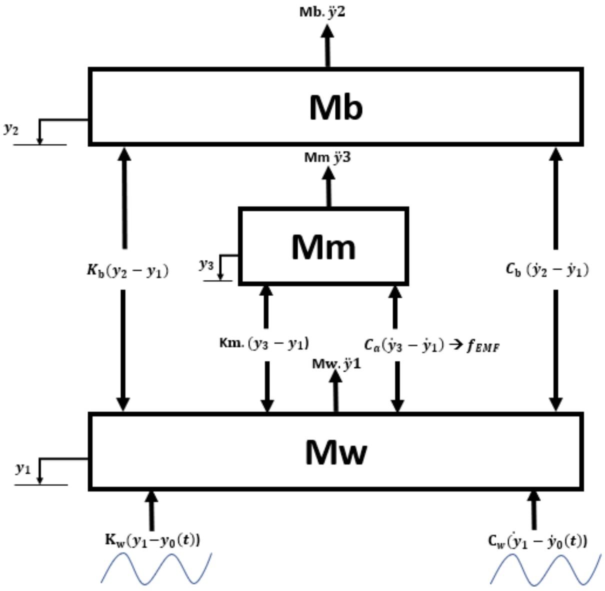


Figure 3. Quarter Free Body Diagram Car of the Regenerative TMD Model.

This model consists of three main masses: Sprung mass ***Mb*** represents the vehicle body. Un-sprung Mass ***Mw*** consists of the wheels, brakes, and in-wheel electric motors, which are the sources of the added mass. TMD mass ***Mm*** is additional mass designed to oscillate, functioning to absorb vibration energy from the unsprung mass.

C. *Mathematical Model*

In order to quantitatively analyze the physical behavior of the designed quarter-car model, the dynamic scheme is then translated into mathematical equations. Using Newton's Second Law, the interactions between the three main masses, namely the vehicle body ***Mb***, the wheel assembly with the in-wheel motor ***Mw*** and the TMD unit ***Mm*** are modeled to map the distribution of forces working on the system. Therefore, the equation can be written as follows:

$$L\frac{di}{dt} + (R_c + R_r).\mathrm{i} = e_{EMF}$$

$$L\frac{di}{dt} + (R_c + R_r).\mathrm{i} = (NBla)(\dot y_3 - \dot y_1)$$

$$(R_c + R_r).\mathrm{i} = \{(NBla)(\dot y_3 - \dot y_1)\} - L\frac{di}{dt}$$

$$\mathrm{i} = \frac{\{(NBla)(\dot y_3 - \dot y_1)\} - L\frac{di}{dt}}{(R_c + R_r)}$$

$$f_{EMF} = (NBla)i \ldots\ldots\ldots\ldots\ldots(1)$$

$$R_r.\mathrm{i} = e_0$$

$$\begin{aligned} Mw.\ddot y1 + K_w(y_1 - y_0(t)) + C_w(\dot y_1 - \dot y_0(t)) &= C_b\,(\dot y_2 - \dot y_1) + f_{EMF} \\ &+ K_b(y_2 - y_1) + \mathrm{Km}.(y_3 - y_1) \end{aligned}$$

$$Mb.\ddot y2 + C_b(\dot y_2 - \dot y_1) + K_b(y_2 - y_1) = 0$$

$$Mm.\ddot y3 + f_{EMF} + K_m(y_3 - y_1) = 0$$

$$Mm.\ddot y3 + (NBla)i + K_m(y_3 - y_1) = 0 \ldots(2)$$

Assuming that the electromechanical coupling coefficient ($\boldsymbol{NBla}$) is denoted as (kt) and ***Rc***+***Rr***=***R***, the equations for the dynamic system and the circuit diagram of the Regenerative TMD can be written as follows:

$$\mathrm{i} = \frac{k_t(\dot y_3 - \dot y_1) - L\frac{di}{dt}}{R}$$

$$f_{EMF} = k_t.i$$

$$R_r.\mathrm{i} = e_0$$

$$\begin{aligned} Mw.\ddot y1 + K_w(y_1 - y_0(t)) + C_w(\dot y_1 - \dot y_0(t)) &= C_b\,(\dot y_2 - \dot y_1) + f_{EMF} \\ &+ K_b(y_2 - y_1) + \mathrm{Km}.(y_3 - y_1) \end{aligned}$$

$$Mb.\ddot y2 + C_b(\dot y_2 - \dot y_1) + K_b(y_2 - y_1) = 0$$

$$Mm.\ddot y3 + f_{EMF} + K_m(y_3 - y_1) = 0$$

$$Mm.\ddot y3 + k_t.i + K_m(y_3 - y_1) = 0 \ldots(3)$$

The dynamic equations that have been formulated are then converted into state variables so they can be further processed using a simulation block diagram. This conversion aims to map the system's behavior in the time domain more systematically. The following is the set of state variables for the vertical plane vibration analysis.

$$\dot y_1 = v_1$$

$$\begin{aligned} \dot v_1 = \frac{1}{Mw}\{&K_w y_0(t) + C_w y_0(t) - (C_b + C_w).\dot y_1 + C_b\dot y_2 - (K_m + K_b + \\ &K_w).y_1 + K_b y_2 + K_m y_3 + k_t.i\} \end{aligned} \quad .(4)$$

$$\dot y_2 = v_2$$

$$\dot v_2 = \frac{1}{\mathrm{Mb}}(-C_b.\dot y_2 + C_b\dot y_1 - K_b.y_2 + K_b y_1)(5)$$

$$\dot y_3 = v_3$$

$$\dot v_3 = \frac{1}{\mathrm{Mm}}(-C_m.\dot y_3 + C_m\dot y_1 - K_m.y_3 + K_m y_1)$$

$$\dot v_3 = \frac{1}{\mathrm{Mm}}(k_t.i - K_m.y_3 + K_m y_1)\ldots\ldots..(6)$$

with $\dot v_1$ is velocity of wheel, $\dot v_2$ is velocity of body and $\dot v_3$ velocity of TMD with used to model simulation in MATLAB or Simulink.

D. Road Excitation Profile

The dynamic response of a vehicle suspension system is strongly influenced by the characteristics of the road surface profile. In real conditions, road irregularities are inherently random and are commonly represented using stochastic approaches such as power spectral density (PSD). However, in vibration studies a deterministic sinusoidal profile is often used for simplification and controlled analysis.

According to Priyambada and Sutantra (Priyambada & Sutantra, 2016), the conditions of the urban road may be approximated to a sinusoidal profile of 2 cm amplitude and comparatively longer wavelength (see figure 4). This is indicative of the more even nature of the urban road as opposed to the rural position where the surface irregularities are generally not so severe and more widely spaced.

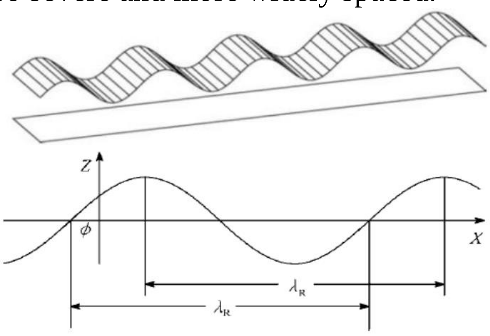


Figure 4. Sinusoidal Road Profile (Priyambada & Sutantra, 2016)

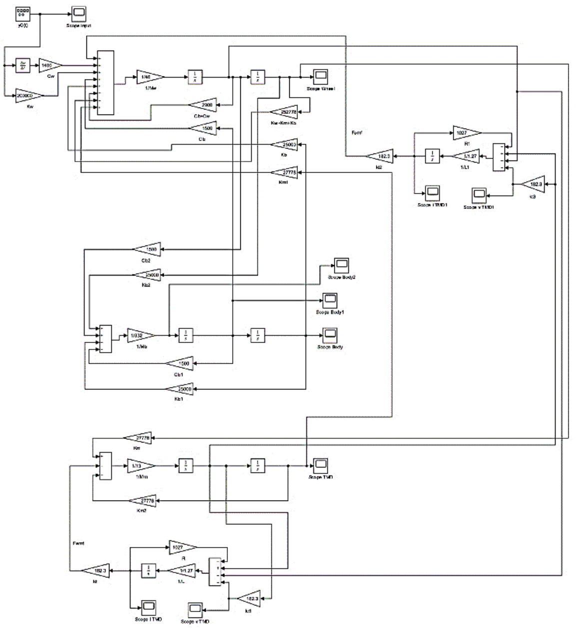
In this paper, the road excitation is modelled as sinusoidal input with an amplitude of 0.02 m (20 mm) based on the urban road profile as described by Priyambada. The excitation frequency is kept constant at 5 Hz, which is representative of a typical operating condition of the vehicle when it is driven at moderate speed. The amplitude was 0.01414 RMS of road excitation. The harmonic excitation allows a more controlled evaluation of the suspension system response, in particular to identify resonance behaviour and the effectiveness of the TMD.

Sinusoidal excitation has several advantages, such as ease of implementation and the ability to isolate the effects of system parameters, such as TMD mass and stiffness. The constant amplitude and frequency allow the system response to be directly related to the change in TMD configuration, providing a more straightforward interpretation of the vibration attenuation performance.

However it should be noted that this method does not fully reflect the complexity of real road conditions, which have a wide band of frequencies and amplitudes. Hence, the sinusoidal model is appropriate for parametric and comparative analysis, but further studies are suggested, such as those using stochastic road profiles based on standards such as ISO 8608, to increase the realism of the simulation.

E. Simulation

Following the theoretical formulation of the state variable model, the equations must be implemented using a dynamic system block diagram in MATLAB Simulink. The goal of this translation is to convert intricate mathematical relationships into a visual simulation that can process real-time data related to traffic disturbances. The following block diagram architecture was created to illustrate the suspension's and the regenerative TMD's working mechanisms:



. **Figure 5.** Simulink Diagram Block of Regenerative TMD

To compare the performance of the regenerative TMD, two Simulink models were developed: one with the regenerative TMD and one without it. Figure 5 shows the model with the regenerative TMD, while Figure 6 presents the model without the regenerative TMD.

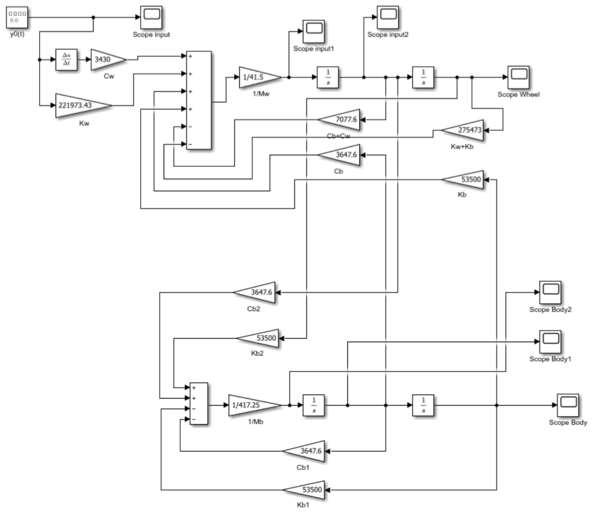


Figure 6. Simulink Diagram Block without Regenerative TMD

To ensure that the simulation results are highly accurate and reflect actual physical conditions, this model was developed based on precise technical specifications. The operational design of the Regenerative TMD system includes variables such as mass, spring constants, and electromagnetic characteristics, the details of which are summarized in the table 2 below:

Table 2. Parameter Regenerative TMD

Parameter	*Symbol*	*Value*	*Unit*	*Source*
Quarter car parameters				
Sprung mass	Mb	417.25	*Kg*	(Kopylov et al., 2020)
Unsprung mass	Mw	41.5	*Kg*	(Warjaya et al., 2025)
Suspension damping	Cb	3647.6	*N.s/m*	(Priyambada & Sutantra, 2016)
Suspension stiffness	Kb	53500	*N/m*	(Priyambada & Sutantra, 2016)
Tire damping	Cw	3430	*N.s/m*	(Yudhyadi et al., 2025)
Tire stiffness	Kw	221973.43	*N/m*	(Kruczek & Stribrsky, 2004)
TMD parameters				
Electromechanical coupling coefficient	Kt	182.3	*T.m*	(Kopylov et al., 2020)
TMD Spring stiffness	Km	39000	*N/m*	(Kopylov et al., 2020)
Resistance of the coil	$\boldsymbol{Rc}$	513.5	*Ohm*	(Kopylov et al., 2020)
Resistance of the load	$\boldsymbol{Rr}$	513.5	*Ohm*	(Kopylov et al., 2020)
Inductance of the coil	L	1.27	*H*	(Kopylov et al., 2020)

Unsprung mass is derived from earlier studies conducted by the author (Warjaya et al., 2025). The wheel mass value is taken from Kopylov's study (Kopylov et al., 2020). Priyambada and Sutantra's work (Priyambada & Sutantra, 2016) provides the suspension spring stiffness and damping coefficients, while Yudhyadi's study (Yudhyadi et al., 2025) and Philips' experimental tire damping data (Kruczek & Stribrsky, 2004) provide the tire stiffness and damping values.

MATLAB Simulink simulation using predetermined settings was used for the testing step. A road wave profile $\mathbf{y_0(t)}$ with an amplitude of 0.02 m (20 mm) and a constant frequency of 5 Hz was used to excite the system. The mass of the Tuned Mass Damper (TMD) changed from 5 kg to 11 kg throughout load situations in order to assess the system's performance. The purpose of mass variation was to monitor the Root Mean Square (RMS) value on the vehicle body in order to assess the efficacy of vibration dampening.

III. RESULT AND DISCUSSION

A. *Model Validation*

The simulation results were compared with the patterns documented in earlier research on regenerative tuned mass dampers for vehicle suspension systems in order to confirm the accuracy of the generated MATLAB/Simulink model. The electromechanical configuration and important parameters such as the electromagnetic coupling coefficient, electrical circuit representation, and back-iron regenerative TMD architecture reported by (Kopylov et al., 2020). are used by the suggested model.

According to the simulation results, adding a regenerative TMD consistently lowers the RMS vibration response as compared to the baseline suspension system while also producing electricity. These results are in line with the findings of Tang and Zuo (2012), Zhu et al. (2019), and Kopylov et al. (2020), who showed that regenerative TMD systems may simultaneously collect energy from structural oscillations and attenuate vibration.

Additionally, the current analysis demonstrates that there is an ideal absorber mass that maximises vibration attenuation, beyond which performance declines when the absorber mass is raised. This behaviour is consistent with Zuo and Nayfeh's (2006) traditional tuned mass damper theory, which states that excessive mass fluctuation results in detuning between the absorber and the primary system. Therefore, trust in the validity of the constructed simulation model is provided by the similarity of the obtained dynamic behaviour with existing theoretical and experimental findings in the literature.

B. Result

The performance of the regenerative Tuned Mass Damper (TMD) is evaluated under a sinusoidal road excitation with an amplitude of 0.02 m and a frequency of 5 Hz, representing an urban road condition. The simulation results are summarized in Table 3, which presents the RMS response, vibration attenuation percentage, and regenerative output for various TMD masses.

Table 3. Result of Regenerative TMD

TMD Mass (Kg)	*RMS (m/s^2)*	*Vibration Attenuation Percentage (%)*	*Power Output (Watt)*
Without TMD	0.01174	16.97%	-
5	0.01155	18.32%	5.04
5.5	0.01156	18.25%	5.20
6	0.01158	18.10%	5.36
6.5	0.01155	18.32%	5.58
7	0.01153	18.46%	5.81
8	0.01148	18.81%	6.31
9	0.01149	18.74%	6.79
10	0.01150	18.67%	7.33
11	0.01150	18.67%	7.95
12	0.01151	18.60%	8.63
13	0.01154	18.39%	9.37
14	0.01159	18.03%	10.18
15	0.01160	17.96%	11.17

According to Table 3, the unsprung mass's RMS value in the absence of TMD is 0.01174, which translates to a vibration attenuation of 16.97% in relation to road excitation. All evaluated mass variations show a similar decrease in RMS response upon the introduction of the TMD. For example, the RMS value drops to 0.01155 for a TMD mass of 5 kg, yielding an attenuation of 18.32%. In comparison to the baseline condition without TMD, this shows a 1.34% improvement. Even though the RMS acceleration is reduced by about 1.34%, neither active control nor external energy input are required to obtain this outcome. As a result, the improvement is achieved with very little extra system complexity.

C. Discussion

The tuned mass damper's vibration absorption process explains why there is an ideal TMD mass of about 8 kg. A portion of the vibrational energy from the original system is transferred to a secondary oscillating mass by a TMD. Resonance energy transfer takes place and the primary structure's vibration amplitude is reduced when the TMD's natural frequency gets closer to the suspension system's major excitation frequency. The 8 kg absorber mass in this investigation offers the tuning condition that is closest to the 5 Hz Road excitation, which produces the lowest RMS response.

The absorber's ability to resist the vibration of the unsprung mass is limited for masses under 8 kg because of its relatively tiny inertial force. On the other hand, the natural frequency of the TMD moves away from the excitation frequency when the absorber mass surpasses the ideal value. The progressive increase in RMS acceleration seen for masses greater than 8 kg is explained by this detuning effect, which lowers the efficiency of energy transfer. Zuo and Nayfeh (2006) reported similar behaviour and showed that the mass ratio and tuning frequency between the absorber and the primary system had a significant impact on TMD performance. Additionally, the decrease in vibration levels noted in this investigation is consistent with earlier studies by Marian and Giaralis (Giaralis & Taflanidis, 2018) and Yasin (Yasin et al., 2023), where it has been demonstrated that the use of a tuned mass damper efficiently attenuates system response by transferring vibrational energy to a secondary mass, lowering both amplitude and RMS vibration levels.

The vibration attenuation first gets better as the TMD mass increases, peaking at about 8 kg. At this moment, the RMS value decreases to

0.01148, indicating the greatest attenuation of 18.81% and a 1.84% improvement over the system without TMD. This behaviour demonstrates that the TMD reaches optimal tuning at this mass, when effective energy transfer from the primary system is made possible by its natural frequency nearly matching the excitation frequency.

Nevertheless, after this ideal threshold, additional TMD mass increases cause the damping performance to gradually decline. The attenuation drops to 17.96% and the RMS value rises to 0.01160 for a mass of 15 kg. This pattern suggests a detuning effect, in which the TMD's dynamic properties shift away from the excitation frequency, making it less effective.

The electrical output produced by the regenerative TMD exhibits a monotonic increase with respect to the TMD mass, in contrast to the damping performance. The power output increases from 5.04 W at 5 kg to 11.17 W at 15 kg, a boost of approximately 55%. This increase is explained by the heavier TMD absorbing more kinetic energy, which is then transformed into electrical energy via the electromagnetic coupling mechanism.

This pattern is consistent with research by Tang and Zuo (Tang & Zuo, 2012), Zhu (Zhu et al., 2019), and Kopylov (Kopylov et al., 2020), who found that the mass of the absorber has a positive impact on the energy harvesting performance of regenerative TMD systems because a larger mass allows for greater vibration energy absorption and subsequent conversion into electrical energy.

These findings unequivocally show a trade-off between energy harvesting efficiency and vibration attenuation. The dual function of regenerating TMD is the source of the observed trade-off. To get the best dynamic tuning, the absorber mass must be chosen from the standpoint of vibration control. However, because it stores more kinetic energy during vibration, a bigger oscillating mass is advantageous from the standpoint of energy harvesting. Therefore, even when vibration attenuation starts to worsen, increasing the TMD mass constantly enhances the electrical power generation capabilities.

This behaviour suggests that a regenerative TMD's design goal is different from a traditional passive. The regenerative TMD necessitates a multi-objective design approach that concurrently considers ride comfort, vibration reduction, and electrical energy recovery, whereas a traditional TMD is optimized only for vibration suppression. As a result, in real-world vehicle applications, the ideal absorber mass is determined by how much weight is given to these performance parameters. The electrical output is improved by increasing the TMD mass, however the damping performance is not consistently improved. Rather, the efficiency of the TMD decreases beyond an ideal mass that maximizes vibration reduction.

The potential advantages of the suggested regenerative TMD are highlighted by a comparison with existing vibration control strategies documented in the literature. Because the control force may be continuously modified based on road conditions, active suspension systems based on PID, LQR, or intelligent control algorithms typically offer higher vibration attenuation. Nevertheless, these techniques raise system complexity, implementation costs, and power consumption because they call for sensors, actuators, control gear, and an external energy source.

The suggested regenerative TMD, on the other hand, functions as a passive electromechanical device and doesn't need a specialized control system. Technology simultaneously suppresses vibration and harvests energy, despite the study's relatively small gain in vibration attenuation. Compared to traditional passive TMDs, which release vibration energy as heat without recovering useful electrical power, this dual capability offers a substantial advantage. For in-wheel motor electric cars, where simplicity, dependability, and energy efficiency are crucial design factors, the regenerative TMD can be viewed as an appealing substitute.

Consequently, these goals should be considered when choosing the TMD mass. Instead of focusing on just one performance metric, an ideal design should strike a compromise between the necessity for efficient energy generation and good vibration suppression. The suggested method has several benefits. First, the regenerative TMD increases overall system efficiency by reducing vibration and recovering some of the vibrational energy at the same time. Second, compared to active suspension systems, the system is simpler and possibly more dependable because it does not require sophisticated control algorithms. Third, integration with car suspension systems with constrained installation space is made easier by the small back-iron electromagnetic architecture.

However, it is important to recognize a few restrictions. The quarter-car model and harmonic road excitation used in the analysis are insufficient to accurately capture the intricate multidirectional dynamics of a real vehicle operating on erratic road surfaces. Furthermore, only the TMD mass parameter was changed; the electromagnetic damping coefficient, load resistance, and spring stiffness were all kept constant. Additionally, the achieved vibration reduction improvement is still quite limited, indicating that additional electromechanical parameter tuning may be necessary to attain better performance. In order to enhance both vibration attenuation and energy harvesting efficiency, full-vehicle models, stochastic road profiles based on ISO 8608, and multi-objective optimization techniques are being investigated. Additionally, experimental validation has not yet been carried out and is still a crucial path for future work.

IV. CONCLUSION

The performance of a regenerative tuned mass damper applied to a quarter-car suspension system under harmonic road excitation with an amplitude of 0.02 m and a frequency of 5 Hz is examined in this study. The findings show that, in comparison to the baseline state without TMD, the addition of the TMD enhances the system's ability to attenuate vibrations. The unsprung mass's RMS response decreases from 0.01174 to a minimum value of 0.01148, which is equivalent to an increase in attenuation from 16.97% to 18.81%.

According to the investigation, the system achieves its best vibration suppression performance at an ideal TMD mass of about 8 kg. The efficient transfer and dissipation of vibrational energy is made possible by this condition, which represents an efficient tune between the TMD and the excitation frequency. Nevertheless, attenuation performance gradually declines as the TMD mass is increased beyond this threshold, suggesting the presence of detuning effects that lower the absorber's effectiveness.

On the other hand, the system's electrical energy output steadily rises as the TMD mass does. A higher TMD mass improves the system's capacity to absorb and transform vibration energy into electrical energy, as seen by the output power's notable increase over the studied range. This tendency draws attention to a basic trade-off: ideal vibration reduction may not always follow advancements in energy collecting.

Overall, the results highlight the fact that a single performance goal cannot be the basis for designing a regenerative TMD system. Rather, a balanced strategy is needed to accomplish both efficient energy harvesting and good vibration control. To further evaluate the usability of the suggested system in real-world situations, future research should take into account more accurate road excitation models and a range of operational conditions.

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