Enhancing Motorcycle Suspension Dynamics: A Comparative Study Of Dual And Triple Spring Suspensions Using A Numerical Approach

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Abstract: This study investigates the potential improvements offered by a passive triple spring damper suspension system as a replacement for the traditional dual spring damper mechanism in the rear wheel of motorcycles. The research aims to enhance rider safety and comfort by analyzing the performance of this innovative suspension configuration using a four-degree-of-freedom (4DOF) numerical model. By simulating the behavior of dual and triple spring systems under standard road disturbances, such as bumps and dips, the study evaluates key suspension characteristics, including comfort, road-holding, and stability. Results indicate that the triple spring system significantly reduces peak accelerations and displacements at the motorcycle's centroid while improving rear tire road-holding capabilities. Furthermore, the addition of a medium stiffness spring facilitates a smoother transition between soft and hard spring states, leading to enhanced suspension performance. The findings underscore the potential of triple spring configurations to improve the safety and user experience of motorcycles, contributing valuable insights to the field of motorcycle dynamics and suspension design.

Key words – **Motorcycle, Numerical model, Suspension, Dual Spring, Triple Spring, Road holding.**

I. INTRODUCTION

The advancement of motorcycle technology has continually sought to enhance performance, safety, and comfort for riders. One critical component that plays a significant role in achieving these goals is the suspension system, which directly affects the bike's handling, stability, and rider experience. Depends on the motorcycle, the importance of these parameters varied. For example, racing motorcycles aims to achieve better road holding, whereas, off-road vehicle focus on comfort. In standard motorcycle, safer ride is a key requirement. Safer ride consists of smooth ride with better handling for the driver and passenger, irrespective of road conditions.

Currently, most motorcycles utilize passive dual spring suspension systems, due to their simplicity, robustness, and cost-effectiveness. However, with evolving demands for safety, there is a need for innovative suspension designs that offer superior performance without adding significant complexity or cost. This thesis investigates the potential improvements offered by a passive triple spring damper suspension system, by replacing dual spring damper mechanism in the rear wheel of motorbikes.

The primary focus of this research is to model the triple spring linear suspension system and compare its performance to that of the traditional dual spring setup. By employing numerical modelling techniques, this study seeks to simulate the behavior of both suspensions to analyze and quantify the comfort and road-holding capabilities under various conditions. The scope is confined to the rear wheel suspension, as it is crucial for stability and handling in motorbikes, which are inherently different from four-wheeled vehicles in terms of dynamics and weight distribution.

The design and development of this advanced passive suspension system contributes to the existing body of knowledge in motorcycle dynamics and suspension design. While there have been studies on dual spring systems and various suspension technologies, there is a notable gap when it comes to the application of triple spring configurations in motorcycle design. By exploring this uncharted territory, this thesis aims to provide

insights that could lead to enhanced motorcycle safety and rider comfort, aligning with the industry's ongoing pursuit of innovation and improved user experience.

II. LITERATURE SURVEY

Dual Spring Suspension System

The dual spring suspension system has been widely adopted in motorcycles due to its ability to balance comfort and road-holding. Research by Guglielmino et al. (2010) emphasized the importance of dual spring configurations in reducing rider fatigue and improving vehicle handling under varied road conditions. The study highlighted that using springs with distinct stiffness levels allows for better shock absorption during low-intensity disturbances and enhanced stability during high-intensity events. Similarly, a study by Cossalter et al. (2013) demonstrated that incorporating a stopper mechanism in dual spring systems further improved stability by preventing excessive compression during extreme loading conditions.

However, while dual spring systems offer an improvement over single spring setups, limitations remain in their ability to smoothly transition between stiffness levels, leading to abrupt changes in suspension characteristics. This limitation is particularly pronounced during scenarios involving sudden shocks or bumps.

Innovations in Passive Suspension Systems

Recent research has explored the potential of multi-rate spring systems to address the limitations of traditional dual spring suspensions. A comprehensive analysis by Ahmadian and Jalili (2016) proposed the use of variable stiffness mechanisms to achieve smoother transitions between different damping states. The study underlined the benefits of such systems in reducing peak accelerations and enhancing rider comfort without compromising handling performance.

Moreover, the concept of introducing additional springs, as discussed by Poozesh et al. (2018), demonstrated significant promise in improving suspension dynamics. By adding a medium stiffness spring, the transition between soft and hard spring states was made more gradual, resulting in a more progressive suspension response. This setup effectively mitigated the harshness associated with dual spring systems while maintaining the benefits of variable stiffness.

Additionally, Stefan and Sayantan (2020) conducted a study in which they varied the position and angle of the rear suspension to evaluate the performance of single spring-damper system. Their findings revealed the maximum reduction in vertical acceleration could be achieved by positioning the rear suspension closer to the pivot point of the swingarm and orienting it at a steeper angle. This configuration optimizes the suspension's ability to absorb vertical forces. Enhancing overall performance.

Numerical Modeling and Simulation in Suspension Design

Numerical modelling has become a cornerstone in the design and evaluation of advanced suspension systems. Studies such as those by Stefan and Sayantan (2020) utilized 4DOF model to study the dynamic behavior of motorcycles under various road disturbances. These models provided critical insights into the interaction between sprung and unsprung masses, enabling the optimization of suspension parameters for enhanced comfort and stability.

In particular, modelling efforts have focused on simulating real-world conditions, such as road bumps and dips, to evaluate the performance of different suspension configurations. For instance, Kumar and Singh (2021) used a combination of Matlab and Simulink to analyze the time response of dual and triple spring systems under standard road profiles. Their findings revealed that multi-spring configurations, when properly tuned, could significantly reduce peak-to-peak displacements and improve road holding.

Triple Spring Suspension System

While the dual spring system has been extensively studied, research on triple spring configurations remains relatively scarce. However, preliminary investigations, such as those by Zuo et al. (2022), suggest that introducing a third spring can enhance suspension performance by bridging the gap between comfort and road-holding. By incorporating a medium stiffness spring, the suspension system achieves a more linear response, reducing the abruptness in stiffness transitions observed in dual spring setups.

Triple spring systems also offer greater flexibility in tuning, allowing manufacturers to tailor suspension characteristics to specific performance requirements, such as racing, off-road riding, or standard commuting. The addition of stoppers in such systems further enhances their ability to adapt to varying load conditions, ensuring stability and comfort across a wide range of operating scenarios.

The evolution of motorcycle suspension systems reflects an ongoing effort to balance comfort, stability, and cost-effectiveness. While dual spring systems have laid the foundation for improved suspension dynamics, innovations like triple spring configurations hold the potential to revolutionize the field. By leveraging advanced modelling techniques and exploring new design paradigms, researchers can pave the way for safer and more comfortable motorcycles, addressing the diverse needs of riders worldwide.

III. NUMERICAL MODELING AND SIMULATION OF SYSTEM DYNAMICS

The study utilizes a 4-DOF (degrees of freedom) motorcycle model, which includes two unsprung masses (2 DOF corresponding to vertical displacements, z_1 and z_2) and a sprung mass (2 DOF corresponding to vertical displacement, z and angular displacement, φ). Figure 1 illustrates the mathematical representation of the motorcycle, incorporating rigid body masses such as the engine, chassis, and passenger. The combined mass, m_3 and moment of inertia, I are centered at the centroid, which is located at a distance l_1 from the front wheel and l_2 from the rear wheel.

The front and rear tires, with masses m_1 and m_2, respectively, and stiffness constants k_1 and k_2, are connected to the chassis through distinct suspension systems. The front suspension is mounted to the chassis at an angle ε_1 relative to the wheel center, while the rear suspension is attached to the swingarm at an angle ε_2, positioned at a distance $L - L_1$ from the center of the rear wheel, where L is the length of the swingarm.

From the Newton's law of motion, linear differential equation for above model is written as,

$$m_1\ddot{z}_1 + k_1(z_1 - z_{Road,f}) - c_{3,eq}(\dot{z} + l_1\dot{\varphi} - \dot{z}_1) - k_{3,eq}(z + l_1\varphi - z_1) = 0 \quad (1)$$

$$m_2\ddot{z}_2 + k_2(z_2 - z_{Road,r}) - c_{4,eq}(\dot{z} - l_2\dot{\varphi} - \dot{z}_2) - k_{4,eq}(z - l_2\varphi - z_2) = 0 \quad (2)$$

$$m_3\ddot{z} + c_{3,eq}(\dot{z} + l_1\dot{\varphi} - \dot{z}_1) + c_{4,eq}(\dot{z} - l_2\dot{\varphi} - \dot{z}_2) + k_{3,eq}(z + l_1\varphi - z_1) + k_{4,eq}(z - l_2\varphi - z_2) = 0 \quad (3)$$

$$I_3\ddot{\varphi} + c_{3,eq}l_1(\dot{z} + l_1\dot{\varphi} - \dot{z}_1) + c_{4,eq}l_2(\dot{z} - l_2\dot{\varphi} - \dot{z}_2) + k_{3,eq}l_1(z + l_1\varphi - z_1) + k_{4,eq}l_2(z - l_2\varphi - z_2) = 0 (4)$$

Equation 1 and 2 derives the displacement, velocity and acceleration of masses m_1 and m_2 with respects to the road disturbance. Equation 3 derives the motion of mass, m_3 in vertical direction and equation 4 used to simulate the angular movement of mass, m_3.
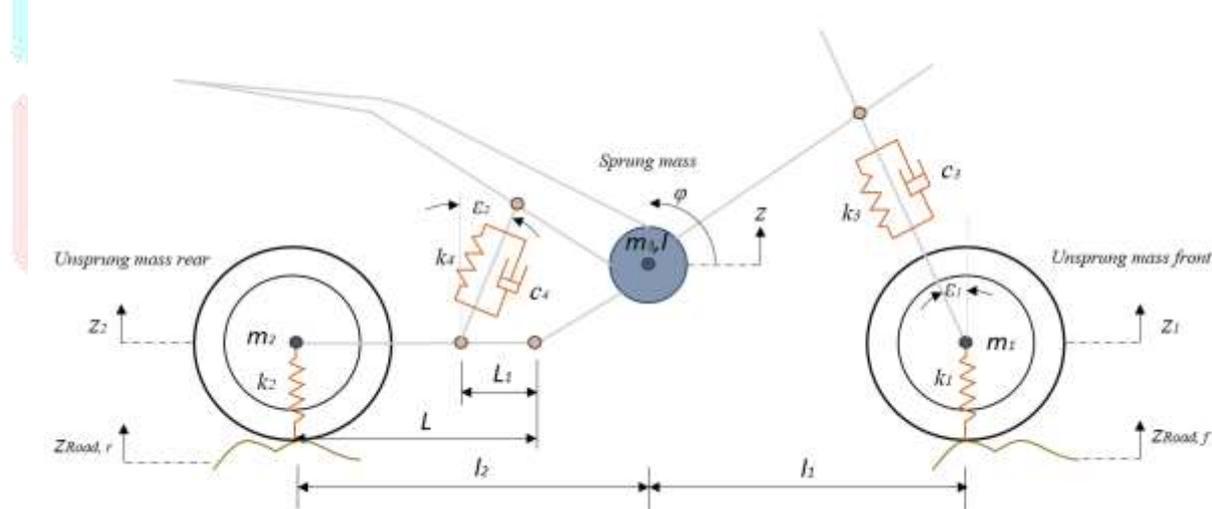

Figure 1: Numerical model of motor bike

Equivalent Stiffness and Damping Coefficients of Front and Rear Suspension

The rake angle in the front suspension plays a critical role in enhancing stability by generating a trail effect, particularly at higher speeds. Additionally, it contributes to rider comfort by absorbing and redirecting forces away from the rider. For simulation purposes, however, the stiffness and damping coefficients of the front suspension must be decomposed into their vertical and horizontal components. Since this study focuses on vertical dynamics, horizontal forces are excluded. Consequently, the equivalent stiffness coefficient for the front suspension is expressed as (Stefan and Sayantan, 2020):

$$k_{3,eq} = k_3 cos^2\varepsilon_1 \quad (5)$$

Similarly, equivalent damping coefficient for the front suspension system is given by:

$$c_{3,eq} = c_3 cos^2\varepsilon_1 \quad (6)$$

In rear suspension, swing arm displacement influence the displacement of the spring-damper mechanism. The equivalent stiffness and damping coefficients for rear suspension are calculated using the following expressions:

$$k_{4,eq} = k_4 \left(\frac{L_1}{L}\right)^2 cos^2 \varepsilon_2 \tag{7}$$

And,

$$c_{4,eq} = c_4 \left(\frac{L_1}{L}\right)^2 cos^2 \varepsilon_2 \tag{8}$$

Dual Spring Suspension

In a dual spring suspension system, two springs are arranged in series to provide distinct stiffness characteristics, thereby improving ride comfort, enhancing handling, and increasing suspension travel. One spring is softer, offering lower stiffness to enhance comfort, while the other is stiffer to improve road-holding and handling performance. In certain dual-rate suspension designs, a stopper nut plays a critical role in preventing the spring from bottoming out and ensuring proper alignment of suspension components. The mathematical model of such a dual-rate suspension system, incorporating a stopper mechanism, is illustrated in figure 2, for three different spring configurations. Notably, the stopper is engaged only during compression and remains disengaged during expansion.
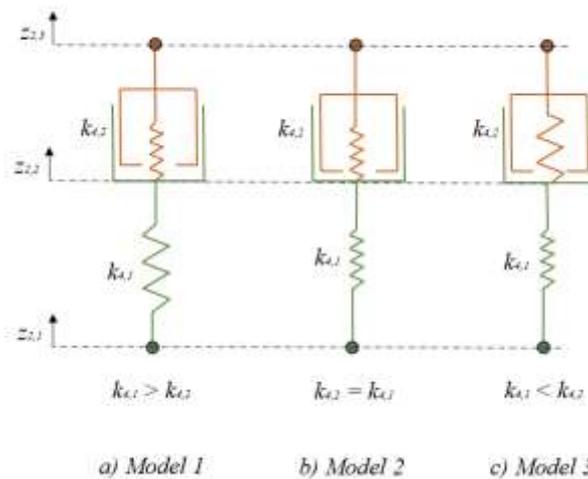

Figure 2: Dual spring setup

Model 1 features a conventional stopper configuration, where the stopper limits the displacement of the soft spring. Conversely, in Model 3, the arrangement is reversed, with the stopper limiting the displacement of the stiffer spring. In Model 2, a mono suspension system is transformed into a dual-rate suspension by incorporating a stopper mechanism between the two springs.

For dual rate suspension simulation, the stiffness change of k_4 is modelled in the form,

$$k_4 = \begin{cases} \dfrac{k_{4,1}k_{4,2}}{k_{4,1} + k_{4,2}}, & if \ (z_{2,3} - z_{2,2}) > a \\ k_{4,1}, & if \ (z_{2,3} - z_{2,2}) \le a \end{cases} \tag{9}$$

Where, $k_{4,1}$ and $k_{4,2}$ represents the stiffness coefficients of dual spring suspension system, $z_{2,1}$ denotes the relative displacement of swing arm at a distance L_1 from the pivot, $z_{2,2}$ corresponds to the displacement of first spring, and $z_{2,3}$ indicates the total deformation of dual spring system. The dual spring operates only when the relative displacement of the top spring is greater than 'a'. once this limit is reached, only the bottom spring remains active, resulting in an increased overall stiffness. When $a=0$, there is no allowance for compression and it operates only during expansion. For positive value of 'a', a preload effect is introduced. To eliminate this, 'a' is typically selected as less than or equal to zero.

Triple Spring Suspension

In addition to the soft and hard spring, a medium spring is introduced in the triple spring system as shown in figure 3. The primary purpose of the middle spring is to reduce the transition effects between the soft and hard springs, thereby achieving a more gradual progress in spring stiffness, approximating a progressive spring rate in linear manner. As the number of springs increases, the system incorporates the additional displacement, $z_{2,4}$, enabling the model to up to four distinct spring rates, depending on the shock conditions induced by the road profile.
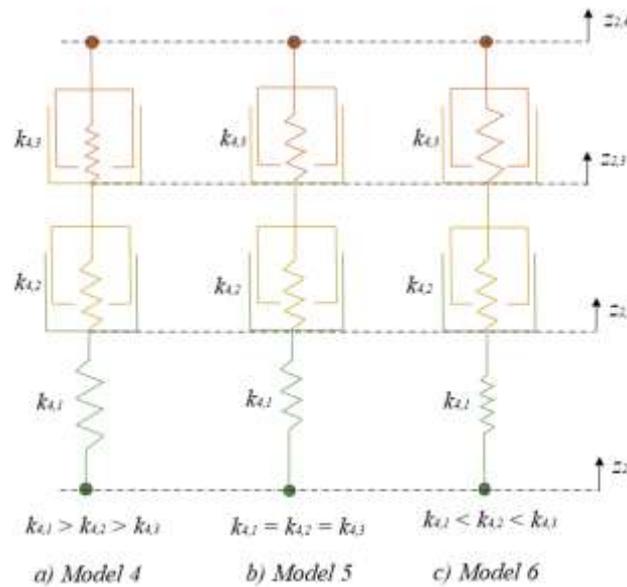
Figure 3: Triple spring setup

The stopper is added in medium spring as well to achieve four spring rates which is modelled in the form,

$$
k_4 = \begin{cases}
\dfrac{k_{4,1}k_{4,2}k_{4,3}}{k_{4,1}k_{4,2} + k_{4,2}k_{4,3} + k_{4,1}k_{4,3}}, & if\ (z_{2,4} - z_{2,3}) > a\ and\ (z_{2,3} - z_{2,2}) > b \\[2ex]
\dfrac{k_{4,1}k_{4,2}}{k_{4,1} + k_{4,2}}, & if\ (z_{2,4} - z_{2,3}) \le a\ and\ (z_{2,3} - z_{2,2}) > b \\[2ex]
\dfrac{k_{4,1}k_{4,3}}{k_{4,1} + k_{4,3}}, & if\ (z_{2,4} - z_{2,3}) > a\ and\ (z_{2,3} - z_{2,2}) \le b \\[2ex]
k_{4,1}, & if\ (z_{2,3} - z_{2,2}) \le a\ and\ (z_{2,3} - z_{2,2}) \le b
\end{cases}
\tag{10}
$$

Where, 'a' and 'b' used to limit the top and middle spring compression, respectively. This facilitates the tuning of ride comfort and road holding performance for the given road profile.

Road Disturbance Profile

As a vehicle traverses a road, it encounters various irregularities that significantly influence its motion and dynamic behavior. To ensure the suspension system performs effectively under such conditions, an appropriate disturbance profile is essential. For this study, a standard highway speed breaker was designed in accordance with IRC:SP:84-2014 and utilized for simulation purposes. Additionally, a road dip was introduced to evaluate the dynamic response of the front and rear suspension systems.

The time-dependent displacement profile for the instantaneous bump and dip is described as follows:

$$
z_{road}(t) = \begin{cases}
\dfrac{A}{2}\left(1 - \cos\dfrac{2\pi V}{d}t\right), & if\ t \ge 1\ and\ t \ge 1 + \dfrac{d}{V} \\[2ex]
-\dfrac{B}{2}\left(1 - \cos\dfrac{2\pi V}{d}t\right), & if\ t \ge 5\ and\ t \ge 5 + \dfrac{d}{V} \\[2ex]
0, & Otherwise
\end{cases}
\tag{11}
$$

Here, the bump begins at $t = 1s$ and spans a distance $d=5m$, reaching a maximum amplitude of $A=0.01m$. Similarly, the dip starts at $t = 5s$ and extends over the same distance, d, with a maximum amplitude of $B=0.11m$.

The above equation represents the disturbance experienced by the front wheel. The disturbance on the rear wheel is delayed by a time, $T = \dfrac{l_1 + l_2}{V}$, where $l_1 + l_2$ is the distance between the front and rear wheels, and the vehicle velocity is $V=10m/s$.
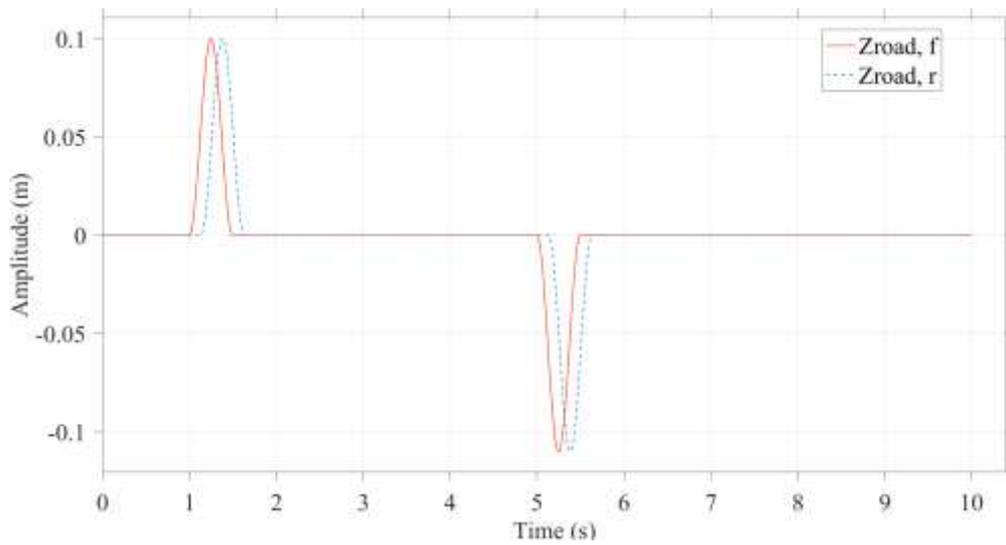
Figure 4: Road bump and dip profile for front and rear wheel at vehicle velocity, 10m/s, with respect to time

IV. MODEL PARAMETERS

Table 1: Numerical values for the model parameters

Parameter	Value	Unit
m_1	15	Kg
m_2	18	Kg
m_3	194	Kg
k_1	180000	N/m
k_2	180000	N/m
k_3	15600	N/m
c_3	785	Ns/m
c_4	1550	Ns/m
I	38	Kgm^2
l_1	0.64	m
l_2	0.7	m
ε_1	27	Deg
ε_2	20	Deg
L	0.6	m
L_1	0.25	m

These values are adopted in the model and called during simulation stage by software itself. But the stiffness values of rear suspension are modified depends on the type of model as listed in Table 2.

Table 2: Spring stiffness values for dual and triple spring models

Stiffness	Model 1 (N/m)	Model 2 (N/m)	Model 3 (N/m)	Model 4 (N/m)	Model 5 (N/m)	Model 6 (N/m)
$k_{4,1}$	80000	60000	40000	80000	60000	40000
$k_{4,2}$	24000	30000	34287	60000	60000	60000
$k_{4,3}$	-	-	-	40000	60000	80000

V. RESULTS AND DISCUSSION

The six spring models were simulated to investigate the effects of linear variable stiffness on ride comfort and road-holding performance. Table 3 presents a comparative analysis of the results obtained from the simulations of dual and triple spring configurations. It is evident that different spring setups have varying impacts on the system's motion. Among the dual spring configurations, Model 3 demonstrates superior ride comfort, as indicated by lower RMS acceleration and reduced total displacement of mass m_3. Similarly, in the triple spring configurations, Model 6 provides a smoother ride experience.

In this study, the preload on the tires was determined by modelling the motorcycle as a simply supported beam, with the resultant force at the supports used to estimate the precompression of the front and rear tires. Under static conditions, the front and rear tires compress by *0.0063m* and *0.006m*, respectively. While these values provide insights into road-holding capabilities for a stationary vehicle, they are subject to variation under dynamic conditions.

Effective road-holding requires continuous contact between the tire and the road surface. In the dual spring configurations, Model 2 exhibits higher equivalent stiffness, which enhances front tire road-holding. Conversely, in Model 3, the delayed stiffness transition results in improved rear tire road-holding. This behavior can be attributed to the stopper being attached to the higher stiffness spring, delaying engagement with the stopper mechanism.

Similarly, for the triple spring configurations, better front and rear tire road-holding were observed in Models 5 and 6, respectively.

Table 3: Motorcycle body centroid RMS acceleration and displacement, relative displacements of front and rear tire with respect to road for different spring configuration

Models	$\ddot{z}_{rms}$ (m/s^2)	$z_{Peak-peak}$ (m)	$z_1 - z_{road,f}$ (mm)		$z_2 - z_{road,r}$ (mm)	
			Max	Min	Max	Min
Model 1	1.437	0.2116	3.933	-3.484	6.623	-4.990
Model 2	1.338	0.2031	3.906	-3.624	5.909	-4.822
Model 3	1.101	0.1864	3.933	-3.908	4.801	-4.247
Model 4	1.229	0.1908	3.651	-4.158	4.205	-4.700
Model 5	1.184	0.1901	3.723	-4.099	4.365	-4.463
Model 6	1.055	0.1809	3.830	-4.159	3.185	-3.938

Figure 5 illustrates the time response of the acceleration of mass m_3, highlighting a significant reduction in peak acceleration when the vehicle traverses a pit. Additionally, Figure 6 demonstrates better displacement control at the centroid, with triple spring models exhibiting lower overall peak-to-peak displacement values compared to dual spring models. Although positive displacement is smaller in dual spring configurations, the peak-to-peak displacement of the triple spring system is notably better.
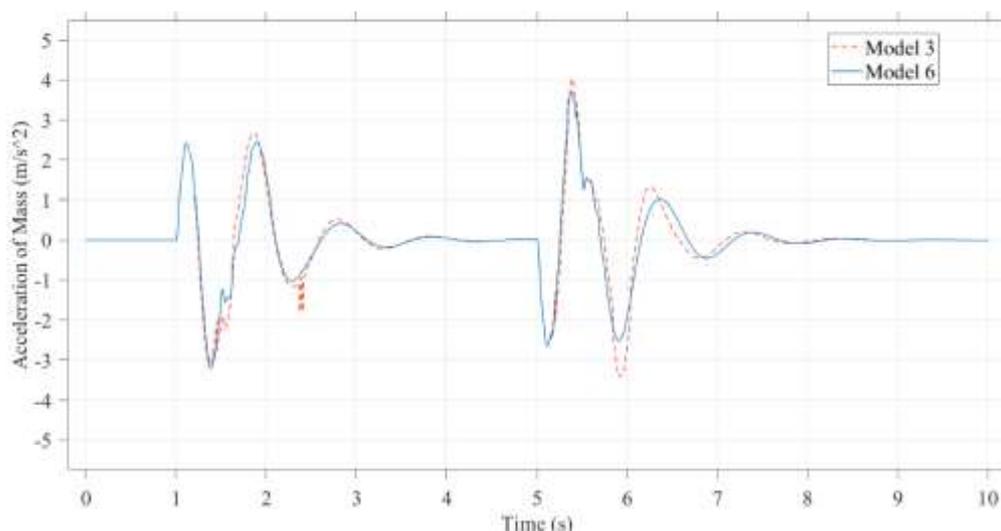

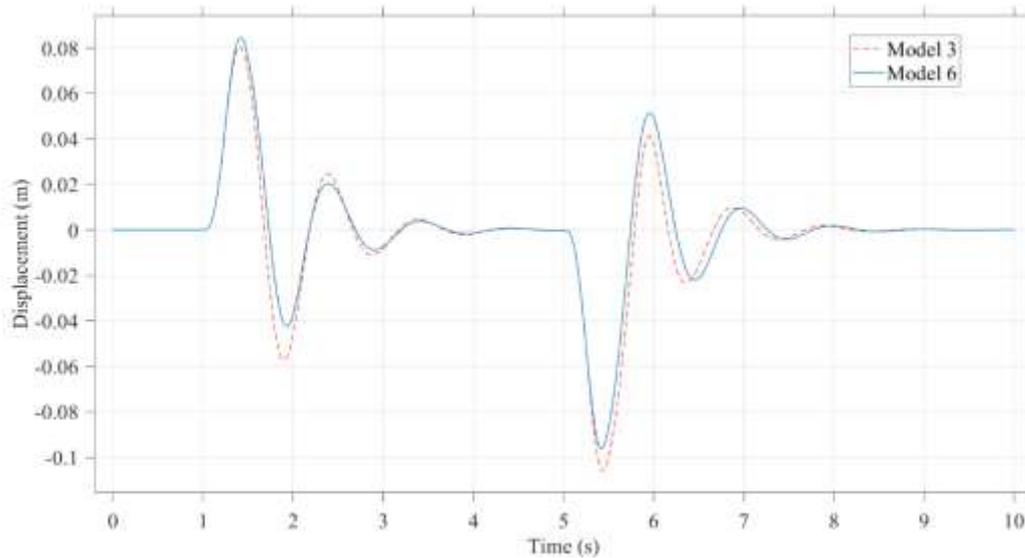Figure 5: Time response plot for motorcycle centroid acceleration

Figure 6: Time response plot for motorcycle centroid displacement

When comparing front tire displacements (Figure 7), a marginal improvement in road-holding is observed with triple springs. However, the rear tire displacement shows a more pronounced difference, with Model 6 providing better control compared to Model 3. Furthermore, the triple spring configuration effectively reduces sharp peaks in tire displacement, which benefits the vehicle's control during acceleration and deceleration. Overall, the triple spring system demonstrates significant potential as a viable alternative to dual suspension systems, offering enhanced comfort, road-holding, and dynamic performance.
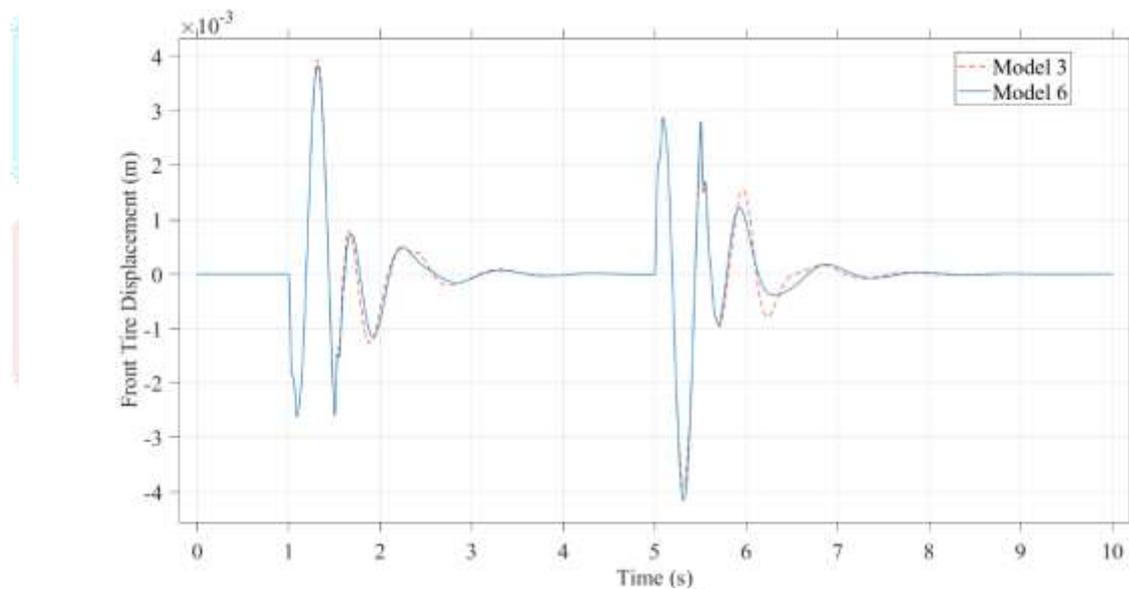

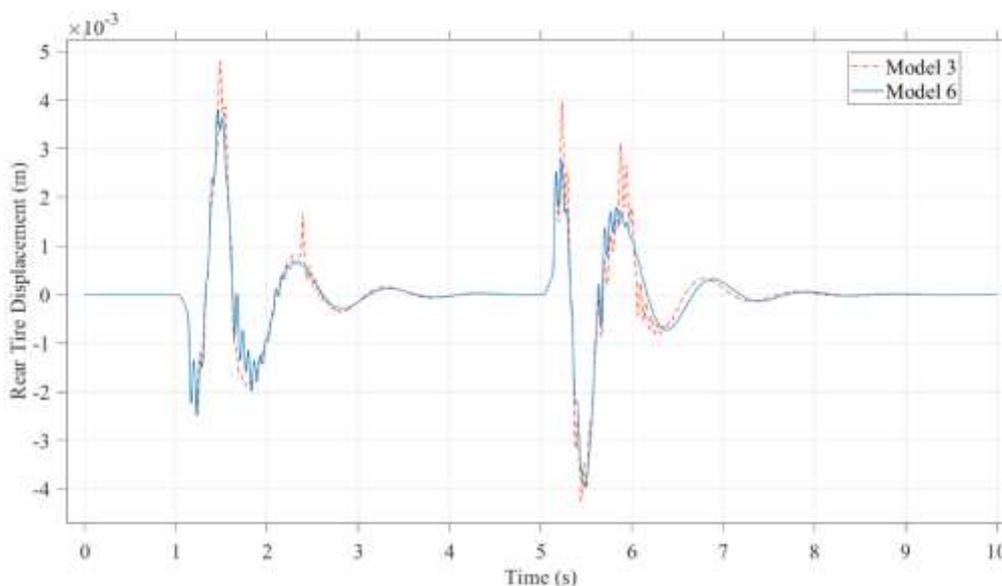Figure 7: Time response plot for front tire relative displacement

Figure 8: Time response plot for rear tire relative displacement

VI. CONCLUSION

This study presents a numerical simulation of a passive triple spring motorcycle suspension system, with its performance compared to dual spring configurations. A four-degree-of-freedom motorcycle model was developed in MATLAB Simulink, and suspension characteristics were evaluated under standard road bump and dip profiles. Time-response plots of acceleration and displacement at the centroid, as well as relative displacement of the front and rear tires, were analyzed to assess key performance metrics, including ride comfort, road-holding, and stability.

The simulation results indicate a significant improvement in centroid acceleration and displacement with the triple spring configuration. Notably, this improvement is achieved by delaying the stiffness transition through the incorporation of a stopper mechanism for the hard spring. Modest enhancements in road-holding were observed for the front tire displacement when compared to the dual spring Model 3. More importantly, a substantial reduction in peak-to-peak displacement of the rear tyre was observed with the Model 6 triple spring configuration.

Overall, the proposed linear variable stiffness triple spring suspension system demonstrates enhanced performance in terms of comfort and stability, offering a promising approach to improving rider safety.

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