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Study on Vehicle Performance: Comparative Analysis of Passive and Semi-Active Suspension Systems for Optimal Ride and Handling

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ABSTRACT

In response to the automotive industry's demand for superior ride comfort and enhanced handling, this study explores alternatives to passive suspension systems. Specifically, Magnetorheological (MR) dampers, as semiactive components, are investigated for their potential to optimize suspension efficiency while ensuring top-notch ride quality. The research involves modelling and simulating both Passive and Semi-active suspension systems within vehicle frameworks. Evaluating the vehicle's performance during cornering and straight-line motion across various suspension setups—passive system, passive system with an anti-roll bar, Semi-Active system without and with an anti-roll bar—provides insight into their comparative efficacy. By scrutinizing key performance metrics during cornering and linear motion, the study aims to pinpoint the optimal alternative to the passive suspension system. Employing MATLAB/SIMULINK facilitates the comprehensive analysis of multiple car models, aiding in the identification of the suspension setup that best balances ride comfort and handling. Ultimately, this research endeavours to illuminate the most suitable suspension system, offering a refined equilibrium between comfort and handling, thereby aiding in enhancing overall vehicle performance across diverse driving scenarios.

Keywords: Half car, Anti-Roll Bar, MR Damper, Vehicle Handling, Ride Comfort

1. INTRODUCTION

Contemporary passive systems fail to wholly satisfy evolving customer demands for improved ride comfort [1]. To address this, alternatives like semi-active and fully active suspension systems have emerged, with fully active systems posing a risk of failure and thus remaining absent in passenger cars. Consequently, semi-active suspension systems, featuring controllability through feedback mechanisms, have gained prominence [2-5].

This study focuses on constructing a semi-active suspension system utilizing MR dampers, where voltage manipulation regulates the solenoid's magnetic flux, influencing the Magnetorheological fluid's viscosity [6-9]. Assessing MR damper performance across varied voltages (0V to 2.5V) ensures safety, as any feedback system failure defaults the suspension to passive mode without compromising basic ride quality.

The paper evaluates a half car model employing the semi-active system, comparing it to the passive system in cornering on level roads and negotiating straight paths with bumps, typical everyday driving scenarios. Simulink models for MR damper systems are formulated using the Spencer model for its practical superiority over other models [10].

This work identifies centrifugal forces during cornering and normal forces on straight paths with bumps, highlighting the significance of anti-roll bars in minimizing vehicle body roll. Combining passive and semi-active systems with anti-roll bars aims to determine the most effective configuration.

Moreover, a tool for MR damper evaluation based on operational parameters is developed [11], and a comparative optimization study employing Response Surface Methodology (RSM) and Genetic Algorithms (GA) investigates vehicle suspension parameters for ride comfort enhancement [12]. Additionally, research delves into vibrational analysis of

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automobiles on the human body [13], and model analysis of automotive suspension systems using Computer-Aided Engineering (CAE) techniques [14].

PROCEDURE

Different combinations considered to model and analyse are

1. Passive Suspension system without Anti-Roll Bar
2. Passive Suspension system with ARB
3. Semi-Active Suspension system containing MRD without ARB
4. Semi-Active Suspension system containing MRD with ARB

To examine the performance of an automobile while cornering and moving in straight path under the above stated 4 combinations of MRD and ARB, mathematical models of the half car system have been built initially using the free body diagrams of the sprung mass and unsprung masses subjected to all the possible loads. Fig. 1 shows the schematic model of half car.

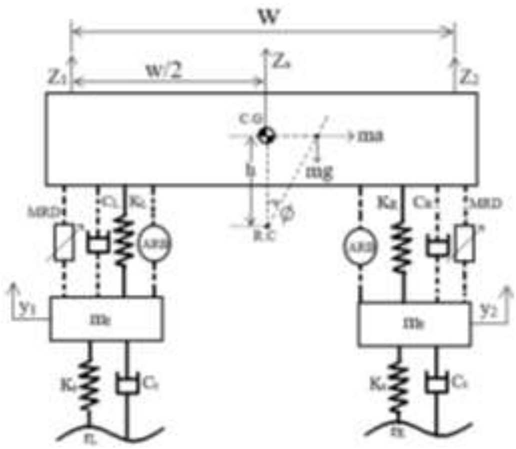

Fig. 1 Schematic Model of Half Car

The wheel alignment considered to model different conditions of cornering and straight line are the same. The front two wheels are considered as half car for the determination of the roll angle (ϕ) and the vertical displacement of the sprung mass centre of gravity (Z_s). The elements which are represented using continuous straight line present in all the 4 combinations and the elements which are represented using dashed lines used for the needed combination and setting. Table 1 gives the system parameters.

Table 1 Half-Car System Parameters

Parameter	Description	Value
W	Track Width	1.2192 m
m	Total Sprung mass	450 Kgs
a	Centrifugal Acceleration	10.92 m/s^2
g	Acceleration due to Gravity	9.81 m/s^2
h	Distance between Roll Centre & Centre of Gravity	0.33528 m
K_R	Stiffness of right end Spring	30000 N/m
K_L	Stiffness of left end Spring	30000 N/m
K_T	Stiffness of tire	200000 N/m
C_R	Damping Co-efficient of right end Passive Damper	750 N-s/m
C_L	Damping Co-efficient of left end Passive Damper	750 N-s/m
C_T	Damping Co-efficient of tire	125 N-s/m
m_t	Un-Sprung mass (tire)	45 Kgs

2. SPENCER MODEL

The schematic diagram of Spencer model is presented in Fig.2.

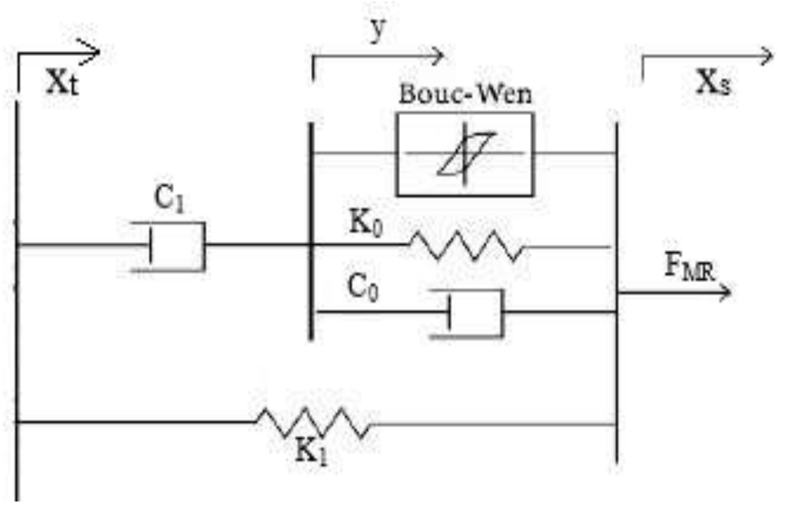

Fig. 2 Spencer Model Schematic Diagram

Spencer Model is a modified and updated version of Bouc-Wen model. Spencer model has been used to model mathematically and simulate the MR damper system which works as the

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semi active system element. The control is only limited to the dampers but not to the anti-roll bar. The Spencer model is the best model that gives results that are very practical and close to the physical test results when compared to other models like Bingham and Bouc-Wen.

Spencer Model Equations

$$F_{MR} = \alpha z + C_0(\dot{x}_s - \dot{y}) + k_0(x_s - y) + k_1(x_s - x_t - x_0)$$

$$F_{MR} = C_1(\dot{y} - \dot{x}_t) + k_1(x_s - x_t - x_0)$$

$$\dot{z} = -\gamma|\dot{x}_s - \dot{y}|z|z|^{n-1} - \beta(\dot{x}_s - \dot{y})\,|z|^n + \delta(\dot{x}_s - \dot{y})$$

$$\alpha = \alpha(u) = \alpha_a + \alpha_b u$$

$$c_1 = c_1(u) = c_{1a} + c_{1b} u$$

$$c_0 = c_0(u) = c_{0a} + c_{0b} u$$

$$\dot{u} = -\eta(u - v)$$

$$\dot{y} = \frac{1}{(c_0 + c_1)}[\alpha z + C_0\dot{x}_s + C_1\dot{x}_t + k_0(x_s - y)]$$

Where,

Z_1 = Sprung mass Displacement at Left Wheel

Z_2 = Sprung mass Displacement at Right Wheel

Z_s = Sprung mass Displacement

C.G = Centre of Gravity

Φ = Roll angle

R.C= Roll Centre

ARB = Anti-Roll Bar

MRD = Magnetorheological Damper

y_1 = Left wheel Displacement

y_2 =Right wheel Displacement

r_L = Road Profile of left wheel

r_R = Road Profile of right Wheel

To find the optimum solution between MR damper system and passive system, a working simulated model of MR damper, with parameters as given in Table 2, is incorporated into the half car model. The graphs are plotted to verify the functioning of MR damper at different voltages. From the graphs, it can be interpreted that the increase in voltage of the damper increases the viscosity of the MR Fluid and so the vibrations are damped at a fast pace. Thus, taking it into consideration, the model is verified to be functioning at different

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voltages and can be incorporated into the half car model so that it can be compared with the passive system and optimum ARB and MRD combination can be found.

Table 2 System Parameters of Spencer Model

System parameters	Values
$\mathbf{C_{0a}}$	784 NS/m
$\mathbf{C_{0b}}$	1803NS/m
$\mathbf{K_0}$	3610N/m
$\mathbf{C_{1a}}$	14649Ns/m
$\mathbf{C_{1b}}$	34622Ns/m
$\mathbf{K_1}$	840N/m
$\mathbf{x_0}$	0.0245m
$\boldsymbol{\alpha_a}$	12441N/m
$\boldsymbol{\alpha_b}$	38430N/Vm
$\boldsymbol{\gamma}$	$136320m^{-2}$
$\boldsymbol{\beta}$	$2059020m^{-2}$
$\boldsymbol{\delta}$	58
η	$190s^{-1}$
n	2

Modelling and simulation of MR damper using Spencer model.

The plots in Fig. 3 represent the MR damper response when cornering forces act on the sprung mass and body roll angle is recorded. The roll angle is recorded for the same force but the voltages of the MR damper is varied from 0V to 2.5V with a step of 0.5V. Anti-roll bar is not incorporated in this simulation.

From these plots, it is evident that the damping effect is increasing with an increase in voltage. This shows the variable damping forces can be generated using the same damper system via controlling the voltages. The 0V simulation is very much identical to the passive system simulation without an anti-roll bar. Table 3 depicts the maximum roll angles with respect to the voltages and is showing the controllability of the MR damper system during cornering.

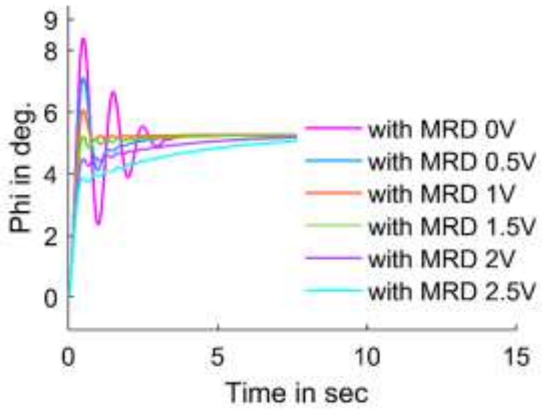

Fig. 3 Sprung Mass Roll angle (ϕ) at Different Voltages while Cornering

Table 3 Maximum roll angle corresponding to different voltages

Voltage	**Maximum roll angle**
0	8.397
0.5	7.122
1	6.083
1.5	5.270
2	4.491
2.5	3.871

The plots in Fig. 4 represent the MR damper response when bump and pothole forces act on the sprung mass and body displacement is recorded. The displacement of the sprung mass, as depicted in Table 4, is recorded for the same force but the voltages of the MR damper is varied from 0V to 2.5V with a step size of 0.5V. Anti-roll bar is not incorporated in this simulation. Table 4 states the displacement of the sprung mass cg with respect to the datum. This shows that the MRD is efficient even in straight path.

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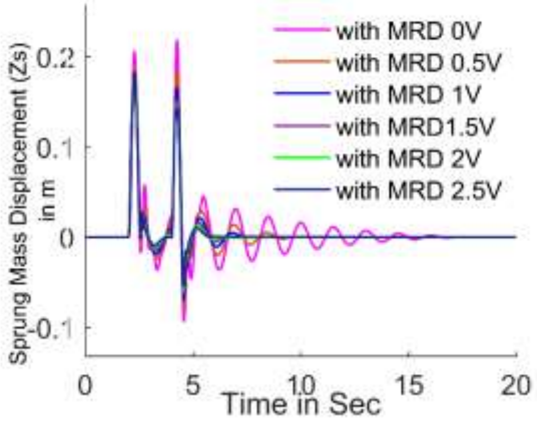

Fig. 4 Sprung Mass Displacement (Z_s (m)) at Different Voltages while moving in a straight path

Table 4 Displacement values as resultant of different voltages

Voltage	Displacement
0V	2.182 e-1 m
0.5V	1.890 e-1 m
1V	1.819 e-1 m
1.5V	1.800 e-1 m
2V	1.812 e-1 m
2.5V	1.830 e-1 m

3. SIMULATION

The mathematical models are computed using MATLAB/SIMULINK to check the performance of the automobile under the conditions of cornering and straight path.

Equations of a passive half car are

$$Z_1 = Z_s + \frac{w\emptyset}{2}$$

$$Z_2 = Z_s - \frac{w\emptyset}{2}$$

$$F_1 = K_L\,(y_2 - Z_2) + C_L(\dot{y}_1 - \dot{Z}_1)$$

$$F_2 = K_R(y_2 - Z_2) + C_R(\dot{y}_2 - \dot{Z}_2)$$

$$\dot{Z}_1 = \dot{Z}_S + \frac{w\dot{\emptyset}}{2}$$

$\dot{Z}_2 = \dot{Z}_S - \frac{w\dot{\emptyset}}{2}$

$I_{xx}\ddot{\emptyset} = mah + mgh\emptyset + F_1\frac{w}{2} - F_2\frac{w}{2}$

$m\ddot{Z}_s = F_1 + F_2$

$m_t\ddot{Y}_2 = K_T(r_r - y_2) + C_T(\dot{r}_r - \dot{y}_2) - F_2$

$m_t\ddot{y}_1 = K_T(r_r - y_1) + C_T(\dot{r}_r - \dot{y}_1) - F_1$

The force equations are modified from passive system mathematical equations to incorporate forces from Anti-Roll bar.

$F_1 = K_L(y_1 - Z_1) + C_L(\dot{y}_1 - \dot{Z}_1) - \frac{K_\gamma wi^2\emptyset}{2r^2}$

$F_2 = K_R(y_2 - Z_2) + C_R(\dot{y}_2 - \dot{Z}_2) + \frac{K_\gamma wi^2\emptyset}{2r^2}$

These force equations are modified from passive system mathematical equations to incorporate forces from MR Damper.

$F_1 = K_L(y_1 - Z_1) + F_{M1}$

$F_2 = K_R(y_2 - Z_2) + F_{M2}$

These force equations are modified from passive system mathematical equations to incorporate forces from MR Damper and Anti Roll Bar.

$F_1 = K_L(y_1 - Z_1) + F_{M1} - \frac{K_\gamma wi^2\emptyset}{2r^2}$

$F_2 = K_R(y_2 - Z_2) + F_{M2} + \frac{K_\gamma wi^2\emptyset}{2r^2}$

The Simulink model in Fig. 5, which consists of both ARB and MRD mathematical models along with conventional passive half-car mathematical model, gives better results among the four built models. This Simulink model provides the output in the form of both Roll Angle (Ø(Degs)) and Sprung mass Displacement (Zs (m)), which are considered as the desired output for this work.

From the output obtained for the models which are built with the combinations like, without ARB and MRD, only with ARB, only with MRD, and lastly with both MRD and ARB, it is observed that fourth combination provides better desired output which will be discussed in

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the inference of the graph shown in Fig.6.

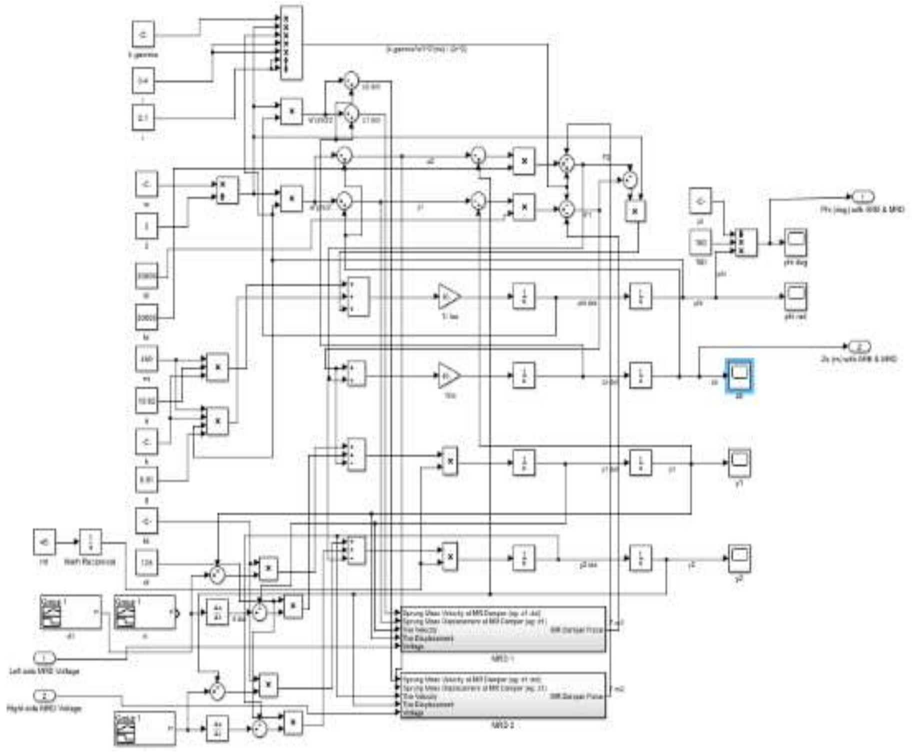

Fig. 5 Half Car Simulink model with ARB & MRD for reference

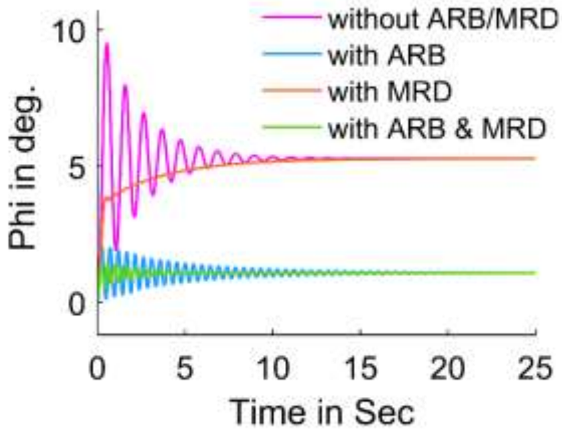

Fig. 6 All the phi(deg) curves of both semi active and passive systems with and without anti-roll bar

According to Figure 6, when the car undergoes a turn with specific acceleration, the inertia of motion results in centripetal forces acting radially outward. These forces, generated at the unsprung mass, induce rolling in the Sprung mass—a fundamental aspect of the underlying physics. The crucial observation pertains to the varying amplitudes of roll across different combinations. Firstly, in the combination lacking both an Anti-Roll Bar (ARB) and Magnetorheological Damper (MRD), handling is notably deficient compared to the other three configurations. The roll amplitude reaches approximately 9 degrees at specific accelerations, progressively increasing. Despite stabilizing around 5 degrees, this outcome is far from desirable.Secondly, the combination featuring only an ARB demonstrates the advantage of its installation in enhancing vehicle handling. The roll amplitude in this scenario is approximately 2 degrees, a notable improvement compared to the configuration without an ARB.

Interestingly, the configuration with solely an MRD, expected to offer superior output, surprisingly falls short of providing an optimal solution. This shortfall leads to the fourth combination, utilizing both ARB and MRD. Remarkably, the graph indicates a significantly reduced roll amplitude of just 0.9 degrees compared to the ARB-only configuration, marking this combination as the most desirable among the four.Detailed curve statistics for comparison are provided in Table 5 for comprehensive analysis and comparison purposes.

Table 5 Roll angle Comparison of Curve at all 4 combinations

Curve statistics	**Passive system without Anti-roll bar (Deg)**	**Passive system with – Anti-roll bar (Deg)**	**Semi active system without anti-roll bar (Deg)**	**Semi-active system With anti-roll bar (Deg)**
Maximum roll angle	9.941	2.107	5.269	1.444
Minimum roll angle	0.00	0.00	0.00	0.00
Mean value	5.143	1.049	4.713	1.037
RMS value	5.314	1.105	4.813	1.056

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Hence, based on the curve statistics, it is evident that the most favorable outcomes are achieved when utilizing MR damper systems in conjunction with anti-roll bars, particularly excelling in cornering conditions. Considering that cornering stability isn't the sole parameter, the MR damper system's performance in straight-line motion is also crucial for proposing an optimal solution. Therefore, a mathematical model for simulating straight-line motion is employed to assess the MR damper system's functionality compared to the passive system.

Figures 7 and 8 illustrate the sprung mass displacement for two combinations: one with an ARB and another with both MRD and ARB. These plots indicate superior vehicle handling in the combination incorporating MRD and ARB, surpassing the performance observed in the configuration featuring only an ARB. Comprehensive curve characteristics extracted from these plots are detailed in Table 6 for further analysis and comparison.

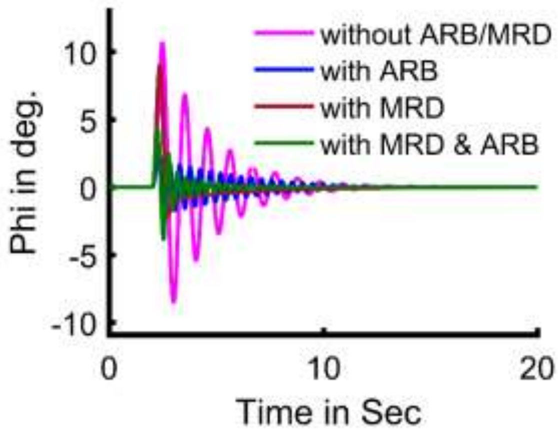

Fig. 7 Half car St. Line Phi (deg) for all the 4 combinations.

These plots indicate that this combination with MRD and ARB has better vehicle handling when it is compared to combination with only ARB. Table 6 represents the curve characteristics from the above plots.

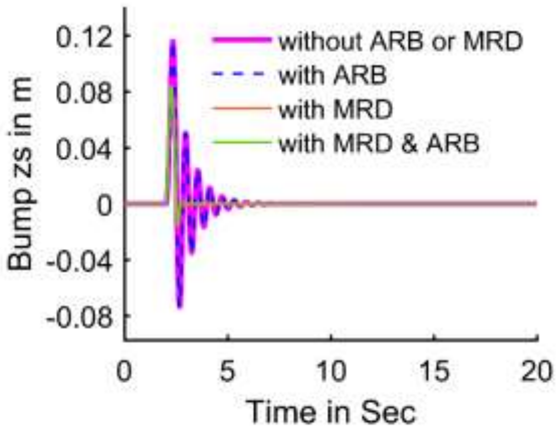

Fig. 8 Half Car Straight path with a bump z_s (m) vs time graph with all the 4 combinations

Table 6 Displacement comparison between passive and semi-active suspension system

Curve statistics	**Passive system with anti-roll bar**	**Semi active system with anti-roll bar**
Z displacement maximum (in m)	1.160e-1	8.428e-2
Z displacement minimum (in m)	-7.348e-2	-1.630e-2
Mean of the curve (in m)	5.576e-3	6.014e-3
RMS of the curve	2.693e-2	2.004e-2

From the statistical data, it can be observed that MR damper system with ARB is showing much better results than that of the passive system.

4. CONCLUSION

This study has meticulously compared the ride characteristics of the MR damper, functioning as a semi-active system, with both the passive system and the passive system equipped with an anti-roll bar in cornering, and with an anti-roll bar in straight-line paths. The analysis of the obtained results leads to a decisive conclusion: the integration of the MR damper system with a passive anti-roll bar constitutes an outstanding configuration within the realm of semi-active suspension systems. This combined setup effectively fulfills the requisite standards for

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ride control and passenger comfort, highlighting its potential as a superior solution in meeting these demands.

REFERENCES

[1] Dukkipati RV, Pang J, Qatu MS, Sheng G, Shuguang Z. Road vehicle dynamics. Warrendale (PA): Society of Automotive Engineers; 2008.

[2] Cronjé, P. H., & Els, P. S. (2010). Improving off-road vehicle handling using an active anti-roll bar. Journal of Terramechanics, 47(3), 179–189.

[3] Zulkarnain N, Imaduddin F, Zamzuri H, Mazlan SA. Application of an active anti-roll bar system for enhancing vehicle ride and handling. IEEE Colloquium on Humanities, Science and Engineering. 2012. p. 260-265.

[4] Yuen TJ, Rahizar R, Azman ZAM, Anuar A, Afandi D. Design optimization of full vehicle suspension based on ride and handling performance. Proceedings of the FISITA 2012 World Automotive Congress: Springer; 2013. p. 75-86.

[5] Chumjun C, Chantalakhana C, Koetniyom S. A Compromise of comfort and handling in automotive vertical dynamics. The 20th Conference of Mechanical Engineering Network of Thailand Nakhon Ratchasima, Thailand 2006.

[6] DVA RamaSastry, K V Ramana, N Mohan Rao, SVR Siva Kumar & T G L Priyanka (2016). Analysis of Train Suspension System Using MR dampers. IOP Conference Series: Materials Science and Engineering, Volume 149(1), 012137.

[7] RamaSastry, DVA, Ramana, KV, Mohan Rao, N, Pruthvi, P & Santhosh, DUV (2017). Analysis of MR Damper for Quarter and Half Car Suspension Systems of a Roadway Vehicle. International Journal of Vehicle Structures and Systems, 9(1).

[8] Rama Sastry, D., Ramana, K., Mohan Rao, N., Phani Kumar, M., & Rama Chandra Reddy, V. (2018). Evaluation of Human Exposure to Vibrations using Quarter Car Model with Semi-Active Suspension. International Journal of Vehicle Structures and Systems,10(4).

[9] RamaSastry, DVA, Ramana, K V, Mohan Rao (2019). Analysis and Prediction of Performance of MR damper at different currents and control strategies for Quarter

ISSN PRINT 2319 1775 Online 2320 7876

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Suspension System of a Roadway Vehicle. International Journal of Vehicle Structures & Systems, 11(1).

ISSN PRINT 2319 1775 Online 2320 7876

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[10] Spencer, B. F., Dyke, S. J., Sain, M. K., & Carlson, J. D. (1997). Phenomenological Model for Magnetorheological Dampers. Journal of Engineering Mechanics, 123(3), 230-238.

[11] Duduku M.R., Reddy N.S., Vishnuvardhan, Sai K.M., Chowdary B.P. (2017),'Development of the soft tool for evaluation of MR Damper based on the operational parameters', International Journal of Civil Engineering and Technology,8(5),PP.267-271.

[12] Reddy M.B.S.S., Vigneshwar P., Ram M.S., Sekhar D.R., Harish Y.S. (2017),'Comparative Optimization Study on Vehicle Suspension Parameters for Rider Comfort Based on RSM and GA',Materials Today: Proceedings,4(2),PP.1794-1803

[13] Dama K.K., Srinivasulu, Sai T., Swaroop, Dileep (2018), 'Design for crashworthiness of an automotive sub-system using CAE techniques', International Journal of Mechanical Engineering and Technology ,9(4), PP. 21-2

[14] Deepak Kumar G., Kommuri N., Kota H., Prasanth V., Hameed S.(2018), 'Vibrational analysis of a human body on an automobile', Journal of Advanced Research in Dynamical and Control Systems ,10(6), PP. 39-4