Introducing Dual Suspension System in Road Vehicles

IMTIAZ HUSSAIN*, JAWAID DAUDPOTO**, AND ALI ASGHAR MEMON***

RECEIVED ON 24.10.2012 ACCEPTED ON 31.12.2012

ABSTRACT

The main objective of suspension system is to reduce the motions of the vehicle body with respect to road disturbances. The conventional suspension systems in road vehicles use passive elements such as springs and dampers to suppress the vibrations induced by the irregularities in the road. But these conventional suspension systems can suppress vibrations to a certain limit. This paper presents a novel idea to improve the ride quality of roads vehicles without compromising vehicle's stability. The paper proposes the use of primary and secondary suspension to suppress the vibrations more effectively.

Key Words: **Primary Suspension, Secondary Suspension, Vibrations, Road Vehicles.**

1. INTRODUCTION

The purpose of the suspension system is to improve the stability, handling and the ride quality of the vehicles. Conventional vehicle suspension systems achieve this through passive means using springs and dampers [1]. When designing vehicle suspensions, the dual objective is to minimise the vertical forces transmitted to the passenger, and to maximise the tyre-to-road contact for handling and safety [1]. For stability and handling more stiffer suspension is required whereas, for ride comfort comparatively soft suspension is suitable [2], which is a design trade-off between ride quality and the stability of the vehicle.

In order to resolve this issue semi-active and active suspension have been proposed by many researchers [3-5], which require expensive elements, such as variable dampers, sensors and actuators, and a sophisticated control mechanism that consequently increases the overall manufacturing cost of the vehicles [6, 7].

In road vehicles the suspension systems consist only of single stage, as shown in Fig. 1, which serves for all three purposes (i.e. stability, handling and ride comfort). However trains have double suspension system as shown in Fig. 2.

Fig. 2 shows the configuration of the most modern passenger-carrying railway vehicles. Each car consists of two bogies, each of which has two sets of wheels. The purpose of the bogie is to carry the weight of a vehicle along the track at the required speed and with a high degree of safety. In doing so, and as far as practicable, it isolates the vehicle from dynamic forces and vibrations resulting from motion. The car body is connected to bogies via

* Assistant Professor, Department of Electronic Engineering, Mehran University of Engineering & Technology, Jamshoro.
** Assistant Professor, Department of Mechanical Engineering, Mehran University of Engineering & Technology, Jamshoro.
*** Assistant Professor, Department of Electrical Engineering, Mehran University of Engineering & Technology, Jamshoro.

suspensions (secondary suspension), the purpose of which is to provide good ride quality by isolating the car body from vibrations induced by track irregularities. The wheelsets are connected to the bogie via primary suspensions, whose elements are much stiffer than in the secondary suspension system and are designed to satisfy the vehicle's stability and guidance requirements [2].

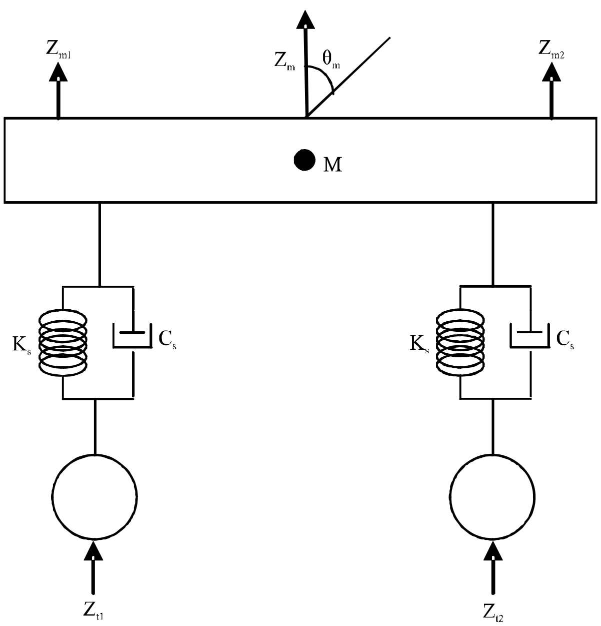

FIG. 1. CONVENTIONAL SUSPENSION SYSTEM

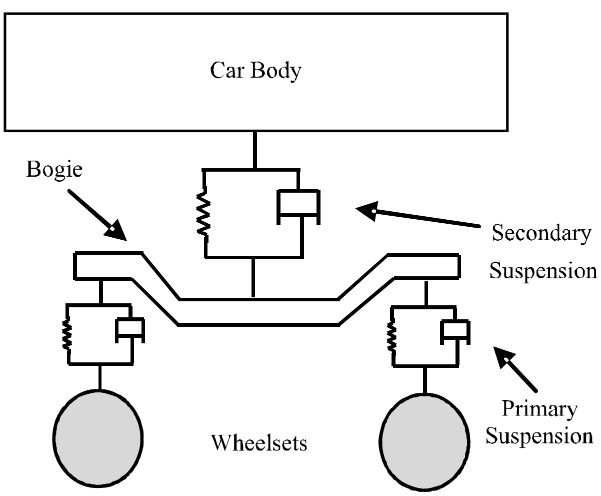

FIG. 2. PRIMARY AND SECONDARY SUSPENSION IN RAILWAY VEHICLES

In this paper a suspension system similar to the railway suspension, as shown in Fig. 3, is proposed for road vehicles. The one end of primary and secondary suspension elements are mounted on an intermediate mass called body frame. The purpose of the primary suspension is to improve the stability of the vehicle by ensuring maximum tyre-road contact on sharp curves and the purpose of secondary suspension is to isolate the vehicle body from road irregularities.

The potential of this research work is presented using a simple model with 4 DoF (Degrees of Freedom). The work can further be extended for a full vehicle model by introducing more DoF.

2. MODELLING OF SYSTEM DYNAMICS

In this study only those DoF are considered which are relevant to ride comfort (i.e. Bounce motion and Pitch Angle) other DoF have very small effect on ride comfort therefore, are ignored in this study.

A detailed diagram of the road vehicle with primary and secondary suspension is shown in Fig. 4, where Z_{t1} and Z_{t2} are the bounce motions applied by the road to the front and rear wheels respectively. Z is the resultant

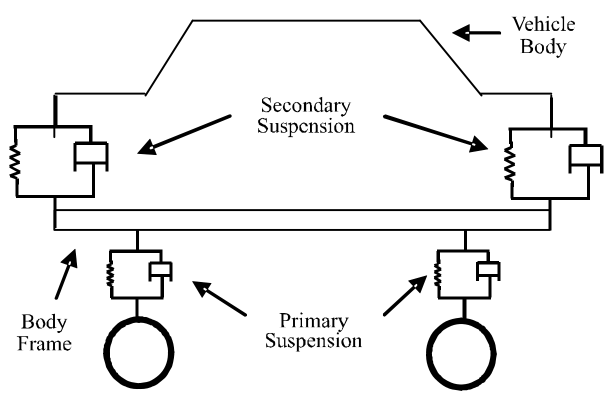

FIG. 3. PROPOSED PRIMARY AND SECONDARY SUSPENSION IN ROAD VEHICLES

bounce motion of the vehicle body and θ is the pitch angle in radians. L is the distance between front and rear wheels, Z_3 and Z_4 are the bounce motions of the body frame. M is the mass of the vehicle body, m_1 and m_2 are the masses of the body frame projected to front and rear wheels respectively. Z_1 and Z_2 are the vertical motions of the vehicle body due to pitch angle. The model is represented by Equations (1-10) and symbols are explained in Appendix-I.

$$f_1 = C_1(\dot{z}_1 - \dot{z}_3) + K_1(z_1 - z_3) \tag{1}$$

$$f_2 = C_2(\dot{z}_2 - \dot{z}_4) + K_2(z_2 - z_4) \tag{2}$$

$$m_3\ddot{z}_3 = C_1(\dot{z}_1 - \dot{z}_3) + K_1(z_1 - z_3) \tag{3}$$

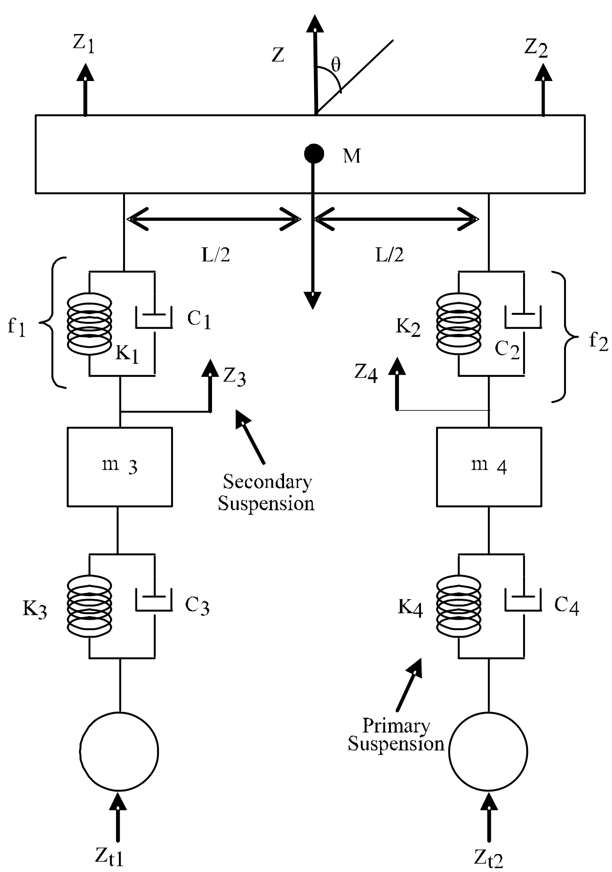

FIG. 4. PROPOSED PRIMARY AND SECONDARY SUSPENSION (4 DOF)

$$\begin{aligned} m_4\ddot{z}_4 &= C_2(\dot{z}_2 - \dot{z}_4) + K_2(z_2 - z_4) \\ &\quad + C_4(\dot{z}_{t2} - \dot{z}_4) + K_4(z_{t2} - z_4) \end{aligned} \tag{4}$$

It is assumed that the vehicle is travelling on straight road therefore the pitch angle would be a small value. Using the small angle approximation rest of the equations can be written as follows.

$$z_1 = z + \frac{L}{2}\theta \tag{5}$$

$$z_2 = z - \frac{L}{2}\theta \tag{6}$$

$$T_1 = \frac{L}{2} f_1 \tag{7}$$

APPENDIX-I. LIST OF SYMBOLS

K_1	Secondary stiffness of front wheel
K_2	Secondary stiffness of rear wheel
C_1	Secondary damping coefficient of front wheel
C_2	Secondary damping coefficient of rear wheel
K_3	Primary stiffness of front wheel
K_4	Primary stiffness of rear wheel
C_3	Primary damping coefficient of front wheel
C_4	Primary damping coefficient of rear wheel
m_1	Bogie mass projected to front wheel
m_2	Bogie mass projected to rear wheel
M	Mass of the vehicle body
Z	Bounce motion of vehicle
Z_1	Bounce motion of front part of the vehicle
Z_2	Bounce motion of rear part of the vehicle
Z_3	Bounce motion of front part of the bogie
Z_4	Bounce motion of rear part of the bogie
J	Moment of inertia of the vehicle
L	Distance between front and rear wheels
Z_{t1}	Road disturbance applied to front wheel
Z_{t2}	Road disturbance applied to rear wheel
θ	Pitch angle of the vehicle
T_1	Torque acting on front part of the vehicle
T_2	Torque acting on rear part of the vehicle

$$T_2 = \frac{L}{2} f_2 \tag{8}$$

$$J\ddot{\theta} = T_2 - T_1 \tag{9}$$

$$M\ddot{z} = -(f_1 + f_2) \tag{10}$$

It is expected that the bounce motion Z of the vehicle body would be much smaller than the vertical disturbances (Z_{t1} and Z_{t2}) applied by the road. It is also expected that in response to any sudden disturbance, such as a pothole in the road, the body motion would be smooth as compare to conventional suspension system.

3. EXPERIMENTS AND RESULTS

A Simulink model is developed of both conventional (Fig. 1) and proposed (Fig. 3) suspension systems. The parameter values used in simulations are summarized in Table 1 and the simulation results are presented in Figs. 5-8. Figs. 5-6 show responses against step disturbances and Figs. 7-8 show responses against road roughness.

Fig.5 shows the response of the conventional and proposed vehicle suspension system when the vehicle is subjected to a step disturbance of 5 milli-meters after 1 second of the simulation. The bounce motion of the vehicle body overshoots and undershoots few times before it is settled down, whereas the response of the body of the proposed model is comparatively smoother.

TABLE 1. PARAMETER VALUES

Parameter	Value
K_1	479482 N/m
K_2	479482 N/m
K_3	239741 N/m
K_4	239741 N/m
C_1	33923 Ns/m
C_2	33923 Ns/m
C_3	18961 Ns/m
C_4	18961 Ns/m
M	3000 Kg
m_1	1500 Kg
m_2	1500 Kg

Fig. 6 shows the pitch motion of the vehicle body for same step input of Fig. 5. Again, the vehicle with conventional suspension system moves like a see saw whereas the proposed model again cops with the applied disturbance smoothly.

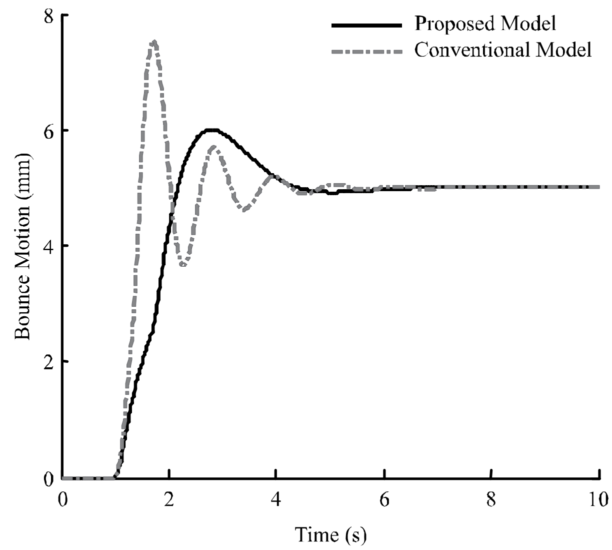

FIG. 5. STEP RESPONSE OF CONVENTIONAL AND PROPOSED MODEL TO 5 mm STEP DISTURBANCE

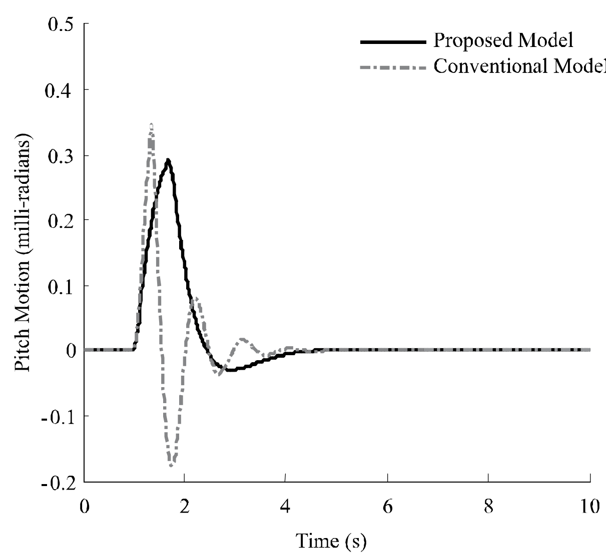

FIG. 6. STEP RESPONSE OF CONVENTIONAL AND PROPOSED MODEL TO 5 mm STEP DISTURBANCE

A road is usually an irregular surface with induces disturbances in all directions. The disturbances only in vertical direction are relevant to this study. Therefore, other inputs from the road are not included in the simulation model. The vertical irregularities are modeled by a random signal and are applied to the simulation model as shown in Figs. 7-8. The response of proposed suspension model to the road's roughness is again much

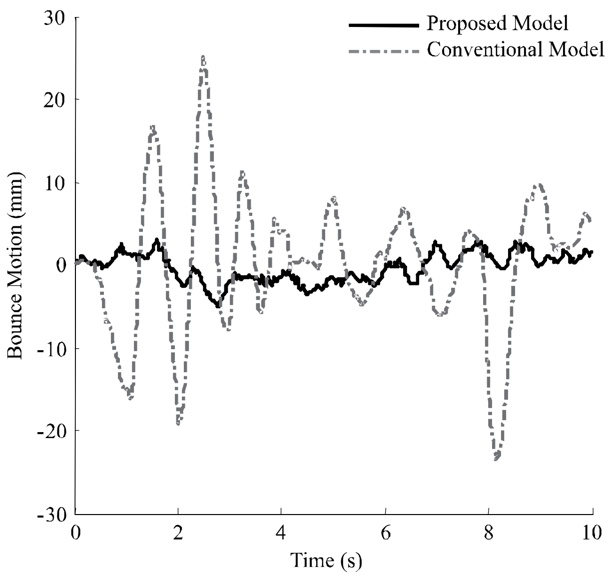

FIG. 7. RESPONSE OF CONVENTIONAL AND PROPOSED MODEL TO ROAD'S ROUGHNESS

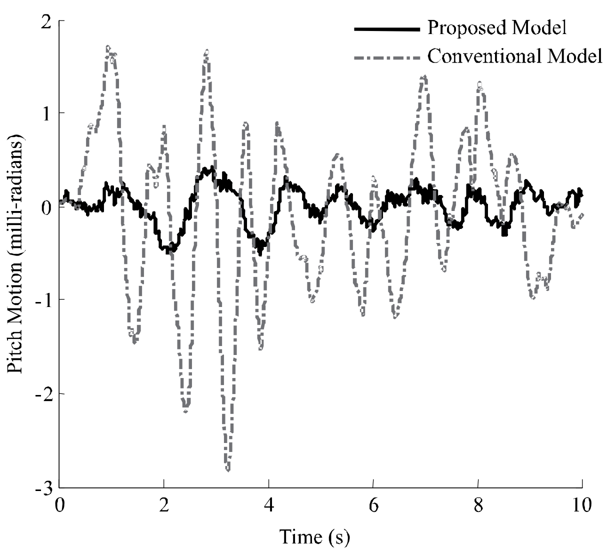

FIG. 8. RESPONSE OF CONVENTIONAL AND PROPOSED MODEL TO ROAD'S ROUGHNESS

better than the conventional one. The bounce motion of the does not exceed ±5mm, whereas, the bounce motion of the conventional suspension system sometimes exceed 20mm.

4. CONCLUSION

This paper presents a new technique for improving the ride comfort of road vehicle without compromising the vehicles' stability. The proposed idea eliminates the use of expensive sensors and actuators and avoids the use of active control to keep the overall design of the vehicle economically feasible. Although further work is required before the idea can be put into implementation but the potential of the research is evident from the simulation results presented. In this paper only 4 DoF are taken into account. The model can be extended to include more DoF.

ACKNOWLEDGEMENTS

Authors are very thankful to Mehran University of Engineering & Technology, Jamshoro, Pakistan, for providing the facilities to carry out this research. First author is also grateful to Prof. T.X. Mei, University of Salford, UK, for his endless support during this work.

REFERENCES

[1] Dowds, P., and O'Dwyer, A., "Modelling and Control of a Suspension System for Vehicle Applications", Proceedings of the 4th Wismarer Automatisierungs Symposium, Wismar, Germany, September 22-23, 2005.

[2] Hussain, I., "Multiple Model Based Estimation of Wheel-Rail Contact Conditions", Ph.D. Thesis, University of Salford, UK, August, 2012.

[3] Rao, M.V.C., and Prahlad, M.V.C.., "A Tunable Fuzzy Logic Controller for Vehicle-Active Suspension Systems", Fuzzy Sets and Systems, Volume 85, No. 1, pp. 11-21, January, 1997.

[4] BenLahcene, Z., Faris, W.F., Khan, Md. R., and Ihsan, S.I., "Semi-Active Suspension System for Off-Road Vehicles", 3rd International Conference on Mechatronics, Kuala Lumpur, Malaysia, 18-20 December, 2008.

[5] Fialho, I., and Balas, G.I., "Road Adaptive Active Suspension Design Using Linear Parameter Varying Gain-Scheduling", IEEE Transactions on Control System Technology, Volume 10, No. 1, pp. 43-54, January, 2002.

[6] Martins, I., Esteves, J., Marques, G.D., and Pina Da Silva, F., "Permanent Magnet and Linear Actuators Applicability in Automobile Active Suspensions", IEEE Transactions on Vehicular Technology, Volume 55, No. 1, pp. 86-94, January, 2006.

[7] Donald, L., and Margolls, "The Response of Active and Semi-Active Suspensions to Realistic Feedback Signals", International Journal of Vehicle Mechanics and Mobility, Volume 11, Nos. 5-6, pp. 267-282, July, 2007.