ISSN:1991-8178

Australian Journal of Basic and Applied Sciences

Journal home page: www.ajbasweb.com

A Study and Review on Vehicle Suspension System and Introduction of a High-Bandwidth Configured Quarter Car Suspension System

[1]S.B.A. Kashem, [2]K.B. Mustapha, [3]T. Saravana Kannan, [4]Sajib Roy, [5]A.A. Safe, [6]M.A. Chowdhury, [7]T.A. Choudhury, [8]M. Ektesabi, [9]R. Nagarajah

[1,2,3,6]Faculty of Engineering, Swinburne University of Technology Sarawak, Kuching 93500, Sarawak, Malaysia.
[4]Faculty of Engineering , East West University, Dhaka, Bangladesh.
[5]Faculty of Engineering , Chittagong University of Engineering and Technology, Chittagong, Bangladesh.
[7]Faculty of Science and Technology, Federation University Australia, Churchill, VIC 3842, Australia.
[8,9]Faculty of Science, Swinburne University of Technology, Hawthorn, VIC 3122, Australia.

ARTICLE INFO

Article history:
Received 13 November 2013
Accepted 23October 2013
Available online 30November 2011

Keywords:
Vehicle; Semi-active; Suspension; Damper; Car; Adaptive; Intelligent; review; comparison.

ABSTRACT

The suspension system reduces the effect of vibration caused by the road and driving conditions. Leading automotive companies have started to use intelligent suspensions in their high-end automobiles'. But much m When travelling, vehicles experience dynamic excitations of varying magnitudes. Such excitations could lead to induced vibration or noise, which affect the vehicles' integrity and occupants. A prominent method of vibration isolation in vehicular system is the suspension system. The main objective of a car suspension system is to improve the ride comfort without compromising the ride handling characteristic. Over recent years, the massive developments in actuators, sensors and microelectronics technology have made the intelligent suspension systems more feasible to implement in automobile industry. These systems are designed and fabricated in such a way that they are able to reduce the drivers' and passengers' exposure to harmful vertical acceleration. The quarter-car suspension model is the best bench-mark to study and analyze the dynamic behavior of vehicle vertical isolation properties. This paper presents background information and a description of the quarter-car suspension model which can be used to evaluate the performance of intelligent suspension system.

To Cite This Article: S.B.A. Kashem, K.B. Mustapha, T. Saravana Kannan, Sajib Roy, A.A. Safe, M.A. Chowdhury, T.A. Choudhury, M. Ektesabi, R. Nagarajah., A Study and Review on Vehicle Suspension System and Introduction of a High-Bandwidth Configured Quarter Car Suspension System. ***Aust. J. Basic & Appl. Sci.,*** *9(30): 59-66, 2015*

INTRODUCTION

When travelling, vehicles experience dynamic excitations of varying magnitudes. Such excitations could lead to induced vibration or noise, which affect the vehicles' integrity and occupants. A prominent method of vibration isolation in vehicular system is the suspension system. The main objective of a car suspension system is to improve the ride comfort without compromising the ride handling characteristic. Over recent years, the massive developments in actuators, sensors and microelectronics technology have made the intelligent suspension systems more feasible to implement in automobile industry. These systems are designed and fabricated in such a way that they are able to reduce the drivers' and passengers' exposure to harmful vertical acceleration. The quarter-car suspension model is the best bench-mark to study and analyze the dynamic behavior of vehicle vertical isolation properties (Allen, J.A., 2008; Kashem, S.B.A., 2012). This paper presents background information and a description of the quarter-car suspension model which can be used to evaluate the performance of intelligent suspension system.

Vehicle suspension system:

A suspension system is an essential element of a vehicle to isolate the frame of the vehicle from road disturbances. Figure 1 shows a typical car suspension system. It is required to maintain continuous contact between a vehicle's tyres and the road. The most important element of a suspension system is the damper. It reduces the consequences of an unexpected bump on the road by smoothing out the shock. In most shock absorbers, vibration energy is converted to heat and dissipates into the environment. Such as, in the viscous damper, energy is converted to heat via viscous fluid. In hydraulic cylinders, the hydraulic fluid is heated up. In air

Corresponding Author: S.B.A. Kashem, Faculty of Engineering, Swinburne University of Technology Sarawak, Kuching 93500, Sarawak, Malaysia.

cylinders, the hot air is emitted into the atmosphere. But the electromagnetic damper is different; here the vibration energy is converted into electricity via an electric motor (induction machine or DC motor or synchronous machine) and stored in a condenser or battery for further use (Suda, Y., *et al*., 2004).

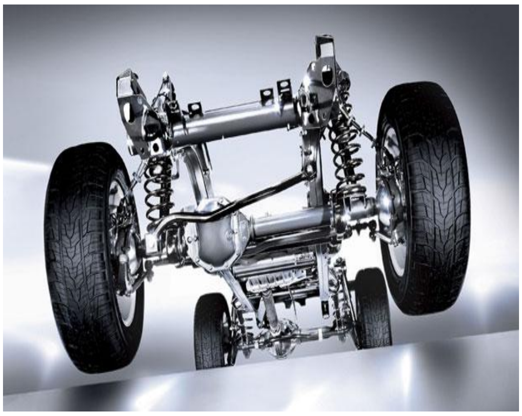


Fig. 1: Vehicle suspension.

Suspension systems are categorized as passive, active and semi-active considering their level of controllability. Although all the types of the suspension systems have different advantages and disadvantages, all of them utilize the spring and damper units.

2.1 Passive suspension system:

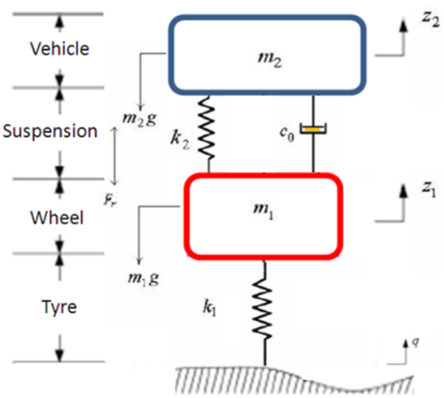


Fig. 2: Passive suspension system.

Passive suspension systems are composed of conventional springs and oil dampers with constant damping properties (Figure 2). .In this model m_1 and m_2 represent the un-sprung mass and sprung mass respectively, k_1 is the tyre stiffness coefficient or tyre spring constant, k_2 is the suspension stiffness or suspension spring constant. c_0 and c_t are the suspension damping constant and the tyre damping constant respectively, Fr is friction of suspension, q, z_1 and z_2 represents road profile input, displacement of un-sprung mass and displacement of sprung mass respectively.

In most instances, passive suspension systems are less complex, more reliable and less costly compared to active or semi-active suspension systems. The constant damping characteristic is the main disadvantage of passive suspension systems. For a passive suspension, the use of soft springs and moderate to low damping rates is needed but the use of stiff springs and high damping rates is needed to reduce the effects of dynamic forces. Designers utilize soft springs and a damper with low damping rates for applications that need a smooth and comfortable ride such as in a luxury automobile.

On the other hand, sports cars incorporate stiff springs and a damper with high damping rates to gain greater stability and control at the expense of comfort. Therefore, the performance in each area is limited for the two opposing goals (Gillespie, T., 2006). There is always a compensation need to be made between ride comfort and ride handling in the passive suspension system as spring and damper characteristics cannot be changed according to the road profile.

2.2 Semi-active suspension System:

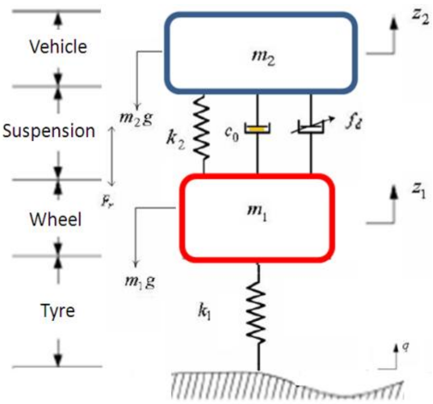


Fig. 3: Semi-active suspension system.

The semi-active suspension system was first proposed by Karnopp *et al*. in 1973. In this model, Figure 3 is a semi-active suspension model. Here f_d can generate an active actuating force by an intelligent controller. Since then, semi-active suspension systems have continued to acquire popularity in vehicular suspension system applications, due to their better performance and advantageous characteristics over passive suspension systems. In semi-active suspension systems, the damping properties of the damper can be changed to some extent. The adjustable damping characteristics in semi-active dampers are achieved through a variety of technologies, such as: Electro-Rheological (ER) and Magneto-Rheological (MR) fluids, solenoid-valves and piezoelectric actuators. It has been widely recognized that a semi-active suspension system provides better performance than a passive system. As it is safe, economical and does not need a large power supply, semi-active suspension has recently been commercialized for use in high-performance automobiles (Irmscher, S. and E. Hees, 1966; Konik, D., *et al*., 1996; Nakayama, T., et al., 1966; Yi, K. and B.S. Song, 1999; Sankaranarayanan, V., et al., 2008). However, there still exist many challenges that have to be overcome for these technologies to achieve their full potential. MR degradation with time, sealing problems and temperature sensitivity are some crucial issues of the MR dampers that need development.

2.3 Active suspension system:

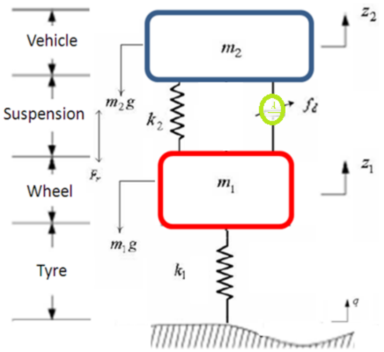


Fig. 4: Active suspension system.

The active suspension system (Figure 4) actuates the suspension system links by extending or contracting them through an active power source as required. Conventionally, automotive suspension designs have been a compromise between the three contradictory criteria of road handling, suspension travel and passengers comfort. In recent years the use of active suspension systems has allowed car manufacturers to achieve all three desired criteria independently. A similar approach has also been used in train bogies to improve the curving behaviour of the trains and decrease the acceleration perceived by passengers. But this makes the system expensive and increases the design complexity and energy demands.

From the above discussion, it is apparent that a semi-active suspension system is more appropriate for implementing and evaluating the performance of various control strategies.

Quarter-car suspension model:

Quarter car suspension system is wide used to investigate the performance of intelligent suspension system. In this paper, a two degree of freedom quarter-car model has been described. A quarter-car model imitates the heave or the vertical motion of the vehicle alone. As the design goal of most semi-active suspension system is to reduce the vertical acceleration, the quarter-car model is sufficient for evaluating the performance of control strategies (Hrovat, D., 1997). The sprung mass, suspension components, un-sprung mass and a wheel are the basic components of a quarter-car model. For a quarter-car model, sprung mass means the body or chassis of the car and it represents almost one fourth of the weight of the whole body of the car. The suspension system bridges the connection between the wheel and body of the car and consists of many parts, and varies according to the type of the suspension system such as passive, semi-active or active suspension (described in the previous section). Un-sprung mass includes the weight of everything geometrically below the suspension system, such as axle, wheel and rim. The wheel denotes the tyre, which incorporates the spring and damping characteristics.

A two degree of freedom quarter-car model as shown in Figure 5 (a) is known as an ideal model and used by some researchers (Zhang, H., 2009; Abdalla, M.O., 2009; Fateh, M.M. and S.S. Alavi, 2009). Faheem et al., (2006) presented an insight on the suspension dynamics of the quarter car model with a complete state space realisation. In the ideal case the sprung mass and un-sprung mass is free only to bounce vertically. In this model m_1 and m_2 represent the un-sprung mass and sprung mass respectively, k_1 is the tyre stiffness coefficient or tyre spring constant, k_2 is the suspension stiffness or suspension spring constant. f_d can generate an active actuating force by an intelligent controller. c_0 and c_t are the suspension damping constant and the tyre damping constant respectively, Fr is friction of suspension, q, z_1, z_2 represents road profile input, displacement of un-sprung mass and displacement of sprung mass respectively.

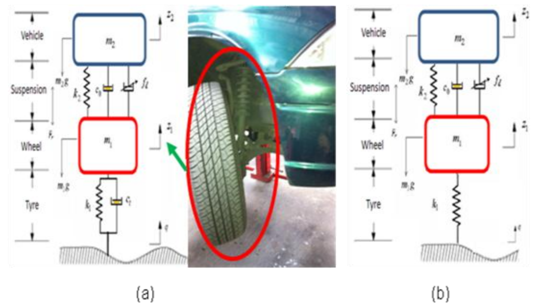


Fig. 5: (a) Ideal quarter-car model, (b) simplified quarter-car model.

The ideal dynamic equations of motion of un-sprung and sprung masses which satisfy Newton's second law of motion are given by the equation 1.

$$m_1\ddot{z}_1 = -c_0\left(\dot{z}_1-\dot{z}_2\right)-k_2\left(z_1-z_2\right)-c_t\left(\dot{z}_1-\dot{q}\right)-k_1\left(z_1-q\right)+f_d-F_r+m_1g$$
$$m_2\ddot{z}_2 = -c_0\left(\dot{z}_2-\dot{z}_1\right)-k_2\left(z_2-z_1\right)-f_d+F_r+m_2g \quad (1)$$

The simplified model as shown in Figure 5 (b) has been used in most recent studies (Gupta, A., et al., 2006; Guo, D., 2004; Nguyen, L.H., et al., 2009; Priyandoko, G., 2009; Scheibe, F. and M.C. Smith, 2009; Bin Abul Kashem, S., 2014; Kashem, S.B.A., 2008; Bakar, S.A.A., et al., 2008; Yan, B., 2012;

Australian Journal of Basic and Applied Sciences, 9(30) Special 2015, Pages: 59-66

Zhu, X., 2012; Hu, H., et al., 2012; Jiang, X., 2012; Chen, S.Z., et al., 2012; Wang, W.R., et al., 2012; Soliman, A.M.A., et al., 2012; Shisheie, R., et al., 2012; Shiri, A., 2012; Zhang, J.J., 2012; Choudhury, S.F. and D.M.A.R. Sarkar, 2012; Kruczek, A., et al., 2011; Kruczek, A., et al., 2011; Mourad, L., 2011; Roqueiro, N., 2011; Edelmann, J., 2011; Pellegrini, E., et al., 2011; Collette, C. and A. Preumont, 2010; Du, F., et al., 2010) as the effect of the tyre damping coefficient c_t is negligible compared to the tyre stiffness coefficient. So omitting the tyre damping force $c_t(\dot{z}_2 - \dot{q})$, the equation (1) becomes equation (2).

$$m_1\ddot{z}_1 = -c_0\left(\dot{z}_1-\dot{z}_2\right)-k_2\left(z_1-z_2\right)-k_1\left(z_1-q\right)+f_d-F_r+m_1g$$
$$m_2\ddot{z}_2 = -c_0\left(\dot{z}_2-\dot{z}_1\right)-k_2\left(z_2-z_1\right)-f_d+F_r+m_2g \qquad (2)$$

Explanation of motion equations of quarter-car:

To understand the motion equations for the quarter-car suspension, it is better to start from ideal mass-spring-damper motion equations, which are well known. First one considers horizontal motion as shown in the Figure 6.

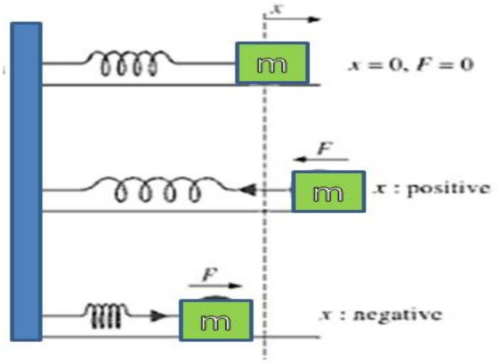


Fig. 6: Mass spring characteristics.

In this figure, x is the position of the square block in meters, m is the mass of the block in kilograms, k is the spring stiffness in Newton's per meter and F_{spring} is the spring Force in Newton's. When a spring is stretched from its equilibrium position due to an external force, the spring itself acts as a force proportional to the length it is stretched and this force acts in the opposite direction to the stretch.

$F_{spring} \propto -$ stretch

Or

$F_{spring} = -k \times$ stretch

If x = 0 at the position where the spring is in equilibrium, then x is equal to the stretch of the spring. So the force of the spring becomes

$F_{spring} = -k\,x$

In addition, there is a force that opposes the motion of the mass as shown in the Figure 7.

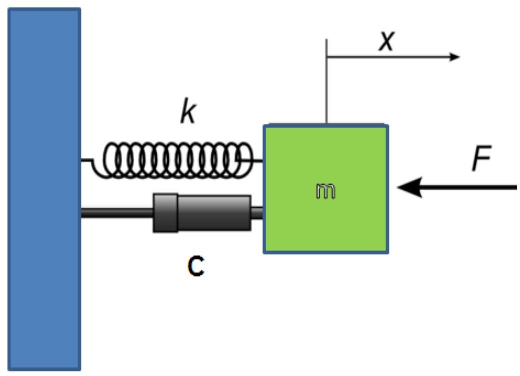


Fig. 7: Mass-spring-damper configuration.

In this figure, c is the damping constant in Newton-second per meter and v is the velocity of the block in meters per second. This force is the damping force and it is proportional to the mass velocity which also opposes the mass velocity, such as

$F_{damping} \propto -v$

Or

$F_{damping} = -c\,v$

So the total force acting on spring-mass-damping system is

$F = F_{spring} + F_{damping} = -k\,x - c\,v$ (3)

According to Newton's law of motion F = m a. From the definition of acceleration, the first derivative of position x is equal to the velocity v and the acceleration a is equal to the second derivative of position x.

$a = \ddot{x}$

And

$v = \dot{x}$

Now the differential equation becomes,

$$m\ddot{x} = -c\dot{x} - kx \qquad (4)$$

The simple mass-spring-damper model described above is the foundation of vibration analysis. This is defined as the single degree of freedom (SDOF) model, since it has been assumed that the mass only moves up and down in the same axis. The Figure 8 is a more complex system involving more mass which is free to move in more than one direction adding degrees of freedom.

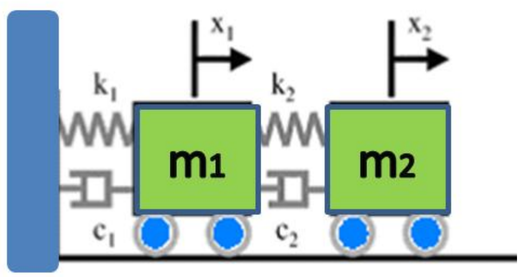


Fig. 8: Two degree of freedom horizontal multiple mass spring damper.

Australian Journal of Basic and Applied Sciences, 9(30) Special 2015, Pages: 59-66

In this model, the two springs act independently, so it is easy to figure out the forces acting on the two blocks. It is assumed that the connection of the spring and damper to the wall is the origin of this suspended system. Here x_1, x_2 are the position (left edge) of the blocks, m_1, m_2 are the mass of blocks and k_1, k_2 are the spring constants. So the motion equations would be

$$
\begin{aligned}
m_1\ddot{x}_1 &= -c_2\left(\dot{x}_1-\dot{x}_2\right)-c_1\dot{x}_1-k_2\left(x_1-x_2\right)-k_1x_1 \\
m_2\ddot{x}_2 &= -c_2\left(\dot{x}_2-\dot{x}_1\right)-k_2\left(x_2-x_1\right)
\end{aligned}
\qquad (5)
$$

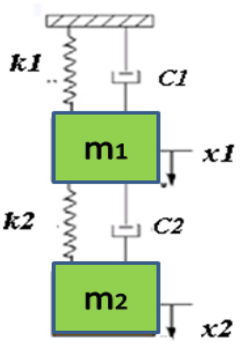


Fig. 9: Vertical multiple mass spring damper configuration.

Now the vertical linear motion has been considered as shown in the above figure. Here a new force strikes due to gravitation g (m/s2) which acts in the same direction (downward) as the mass velocity and equals the product of mass and gravity, so the differential equation becomes

Now, considering a two degree of freedom quarter-car suspension model having an actuator which delivers a force f_d as shown in the Figure 9 and the corresponding motion equation is the equation (6).

$$
\begin{aligned}
m_1\ddot{x}_1 &= -c_2\left(\dot{x}_1-\dot{x}_2\right)-c_1\dot{x}_1-k_2\left(x_1-x_2\right)-k_1x_1-m_1g \\
m_2\ddot{x}_2 &= -c_2\left(\dot{x}_2-\dot{x}_1\right)-k_2\left(x_2-x_1\right)-m_2g
\end{aligned}
\qquad (6)
$$

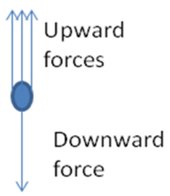


Fig. 10: Forces acting at a point.

If one considers the forces acting on the un-sprung mass m_1 then the forces acting downward is the m_1g force due to gravitation and actuating force f_d. According to Figure 10, force due to the acceleration of the un-sprung mass $m_1\ddot{z}_1$ is acting in the upward direction. If the displacement $z_1>q$ is positive then the spring force $k_1(z_1-q)$ and the damping force $c_1(z_1-q)$ is negative in the downward direction according to Figure (10). This is same for a damping force of co and a spring force of k_2 if $z_1>z_2$ is positive. The friction force F_r is acting negatively in the downward direction.

Again for sprung mass m_2, the forces acting downward is the m_1g force due to gravity and friction force F_r. The force due to the acceleration of the sprung mass $m_2\ddot{z}_2$ is acting in the upward direction. The actuating force f_d is acting negatively downward. Damping force of co and spring force of k_2 is negative in the downward direction under the condition that displacement $z_2>z_1$.

High vs. low-bandwidth suspension system:

A semi-active suspension system has two sections: semi-active and passive. The semi-active part usually gets damping force from an external energy source to control the suspension system (in regenerative type system, it may differ). The passive part has a spring and a damper or similar devices. In some systems this part is rigid but it can be omitted as well. This can be distinguished as low-bandwidth and high-bandwidth suspension systems (Kruczek, A. and A. Stribrsky, 2004).

Low-bandwidth configuration (LBC) represents the series connection between the active and passive components of the suspension system (Figure 11 (a)). In the mathematical modeling, the differential motion equations are as follows,

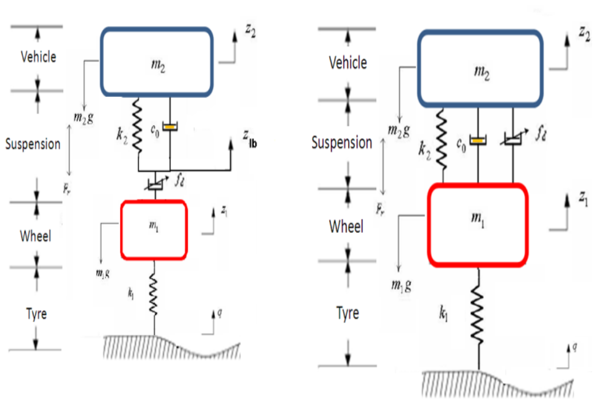
$$
\begin{aligned}
m_1\ddot{z}_1 &= -c_0\left(\dot{z}_{lb}-\dot{z}_2\right)-k_2\left(z_{lb}-z_2\right)-k_1\left(z_1-q\right)-F_r+m_1g \\
m_2\ddot{z}_2 &= -c_0\left(\dot{z}_2-\dot{z}_{lb}\right)-k_2\left(z_2-z_{lb}\right)+F_r+m_2g
\end{aligned}
\qquad (7)
$$

where $c(dz_{lb} - dz_1) = f_d$ is the actuating force. Through LBC configuration, the active suspension system can control the car body (sprung mass) height. But the actuator cannot be omitted or turned off as it carries the static load. Another disadvantage of this system is that it is good only in the low frequency range.

On the other hand, in a high-bandwidth configuration (HBC), it is possible to control at higher frequencies than for LBC and also the passive part can work alone in case of failure of the active part. The only drawback of HBC is that practically it can't control the vehicle height. In a HBC configuration, active and passive components are linked in parallel (Figure 11 (b)). The motion equations of HBC are almost similar to that of LBC but an extra term is added which is an actuator force f_d.

For the research of intelligent suspension system, a two degree of freedom HBC semi-active suspension system should be used, mainly because there is no requirement of a static load force.

Australian Journal of Basic and Applied Sciences, 9(30) Special 2015, Pages: 59-66



(a) (b)

Fig. 11: (a) Low-bandwidth suspension model, (b) high-bandwidth suspension model.

Conclusions:

In this paper, the vehicle suspension system has been categorised and discussed briefly. It has been explained that the semi-active suspension system is the most suitable for road vehicles. A brief description of the quarter-car model has been given as well as an explanation of the motion equations used in the model. High and low bandwidth suspension systems have also been discussed. As there is no requirement of a static load force in this research of a quarter car suspension system, a two degree of freedom HBC semi-active suspension system would be the best bench mark to investigate different semi-active or active suspension systems' control algorithms.

REFERENCES

A broadband controller for shunt piezoelectric damping of structural vibration. Smart Mater. Struct., 2003. 12(1): 18.

Abdalla, M.O., N. Al Shabatat and M. Al Qaisi, 2009. Linear matrix inequality based control of vehicle active suspension system. Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, 47(1): 121-134.

Active LQR and H 2 shunt control of electromagnetic transducers. Proc. 42nd IEEE Conf. on Decision and Control, 2003: p. 2294.

Active vibration control of a suspension system using an electromagnetic damper. Proc. Inst. Mech. Eng., 2001. 215(8): 865.

Allen, J.A., 2008. Design of active suspension control based upon use of tubular linear motor and quarter-car model. Texas A&M University.

Bakar, S.A.A., *et al*., 2008. Vehicle ride performance with semi-active suspension system using modified skyhook algorithm and current generator model. International Journal of Vehicle Autonomous Systems. 6(3-4): 197-221.

Bin Abul Kashem, S., S. Roy and R. Mukharjee, 2014. A modified skyhook control system (SKDT) to improve suspension control strategy of vehicles. in Informatics, Electronics & Vision (ICIEV), 2014 International Conference on.

Chen, S.Z., *et al*., 2012. Magnetorheological Semi-Active Suspension Demonstration for Off-Road Vehicles. Advanced Science Letters, 12(1): 1-6.

Choudhury, S.F. and D.M.A.R. Sarkar, 2012. An approach on performance comparison between automotive passive suspension and active suspension system (pid controller) using matlab/simulink. Journal of Theoretical and Applied Information Technology, 43(ISSN: 1992-8645): 295-300.

Collette, C. and A. Preumont, 2010. High frequency energy transfer in semi-active suspension. Journal of Sound and Vibration, 329(22): 4604-4616.

Damping of structural vibrations with piezoelectric materials and passive electrical networks. J. Sound Vib., 146(2): 243.

Du, F., *et al*., 2010. Robust Control Study for Four-Wheel Active Steering Vehicle. in International Conference on Electrical and Control Engineering, ICECE. IEEE.

Edelmann, J., M. PlÃ¶chl and P. Lugner, 2011. Modelling and analysis of the dynamics of a tilting three-wheeled vehicle. Multibody System Dynamics, pp: 1-19.

Electromagnetic shunt damping. Aim 2003: Proc. 2003 IEEE/ASME Int. Conf. on Advanced Intelligent Mechatronics, 2003: 1145.

Faheem, A., F. Alam and V. Thomas, 2006. The suspension dynamic analysis for a quarter car model and half car model, in 3rd BSME-ASME International Conference on Thermal Engineering. 20-22 December: Dhaka.

Fateh, M.M. and S.S. Alavi, 2009. Impedance control of an active suspension system. Mechatronics, 19(1): 134-140.

Faulhaber DC MOTOR Technical information, Faulhaber DC motor specs, Germany. 2011.

Gillespie, T., 2006. Development of semi-active damper for heavy off-road military vehicles.

Guo, D., H. Hu and J. Yi, 2004. Neural network control for a semi-active vehicle suspension with a magnetorheological damper. Journal of Vibration and Control, 10(3): 461-471.

Gupta, A., *et al*., 2006. Design of electromagnetic shock absorbers. International Journal of Mechanics and Materials in Design, 3(3): 285-291.

Hrovat, D., 1997. Survey of advanced suspension developments and related optimal control applications. Automatica, 33(10): 1781-1817.

Hu, H., *et al*., 2012. Design, modeling, and controlling of a large-scale magnetorheological shock absorber under high impact load. Journal of Intelligent Material Systems and Structures.

Irmscher, S. and E. Hees, 1966. Experience in semi-active damping with state estimators. in proceeding of AVEC.

Jiang, X., J. Wang and H. Hu, 2012. Semi-active control of a vehicle suspension using magneto-rheological damper. Journal of Central South University, 19(7): 1839-1845.

Karnopp, D., M.J. Crosby and R.A. Harwood, 1973. Vibration Control Using Semi-Active Force Generators. ASME Pap, (73 -DET-122).

Kashem, S.B.A., M. Ektesabi and R. Nagarajah, 2012. Comparison between different sets of suspension parameters and introduction of new modified skyhook control strategy incorporating varying road condition. Vehicle system dynamics, 50(7): 1173-1190.

Kashem, S.B.A., S. Raju and I. Raza, 2008. Modified physical configuration to compensate parasitic effects in high speed systems, in Proceedings of ICECE 2008 - 5th International Conference on Electrical and Computer Engineering. Dhaka, Bangladesh, pp: 741-744.

Konik, D., *et al.*, 1996. Electronic damping control with continuously working damping valves (EDCC)â€"system description and functional requirements. in International Symposium on Advanced Vehicle Control. Aachen, Germany.

Kruczek, A. and A. Stribrsky, 2004. A full-car model for active suspension - Some practical aspects. in Proceedings of the IEEE International Conference on Mechatronics 2004, ICM'04.

Kruczek, A., *et al.*, 2011. Active suspension - Case study on robust control. World Academy of Science, Engineering and Technology, 78: 411-416.

Liu, S.L. and S.Y. Zheng, 2011. Improved passive electromagnetic damper and its application. Journal of Vibration and Shock, 30(9): 94-97.

Mourad, L., F. Claveau and P. Chevrel, 2011. A Lateral Control Strategy for Narrow Tilting Commuter Vehicle Based on the Perceived Lateral Acceleration. in 18th IFAC World Congress Milano (Italy), September.

Multimodal passive vibration suppression with piezoelectric materials and resonant shunts. J. Intell. Mater. Syst. Struct., 1994. 5(1): 49.

Nakayama, T., *et al.*, 1966. Development of semi-active control system with PUDLIS. in In Proceedings of International Symposium on Advanced Vehicle Control, AVEC'96. Aachen University of Technology.

New method for multiple-mode shunt damping of a structural vibration using a single piezoelectric transducer. Proc. SPIE, 2001. 4331: 239.

Nguyen, L.H., *et al.*, 2009. Modified skyhook control of a suspension system with hydraulic strut mount. in ICCAS-SICE 2009 - ICROS-SICE International Joint Conference 2009, Proceedings.

Passive vibration control via electromagnetic shunt damping. IEEE/ASME Trans. Mechatronics, 2005. 10(1): p. 118.

Pellegrini, E., *et al.*, 2011. A dynamic feedforward control approach for a semi-active damper based on a new hysteresis model.

Priyandoko, G., M. Mailah and H. Jamaluddin, 2009. Vehicle active suspension system using skyhook adaptive neuro active force control. Mechanical Systems and Signal Processing, 23(3): 855-868.

Roqueiro, N., M.G. de Faria and E.F. Colet, 2011. Sliding Mode Controller and Flatness Based Set-Point Generator for a Three Wheeled Narrow Vehicle. in 18th IFAC World Congress Milano (Italy), September.

Sankaranarayanan, V., *et al.*, 2008. Semiactive suspension control of a light commercial vehicle. Mechatronics, IEEE/ASME Transactions on, 13(5): 598-604.

Scheibe, F. and M.C. Smith, 2009. Analytical solutions for optimal ride comfort and tyre grip for passive vehicle suspensions. Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, 47(10): 1229-1252.

Shiri, A., 2012. Robust Sliding Mode Control of Electromagnetic Suspension System with Parameter Uncertainty. Journal of Applied Sciences, Engineering and Technology, 10(1): 1677-1683.

Shisheie, R., *et al.*, 2012. Linear Quadratic Regulator Time-Delay Controller for Hydraulic Actuator. Journal of Basic and Applied Scientific Research.

Soliman, A.M.A., *et al.*, 2012. Fuzzy-Skyhook Control for Active Suspension Systems Applied to a Full Vehicle Model. International Journal of Engineering and Technology, 2(2): 01-12.

Suda, Y., *et al.*, 2004. Study on electromagnetic damper for automobiles with nonlinear damping force characteristics:(Road test and theoretical analysis). Vehicle System Dynamics, 41: 637-646.

Synthetic impedance for implementation of piezoelectric shunt-damping circuits. Electron. Lett., 2000. 36(18): 1525.

Vibration isolation using a shunted electromagnetic transducer. SPIE: Smart Structures and Materials 2004: Damping and Isolation, 2004: p. 506.

Wang, W.R., *et al.*, 2012. Analysis of Vane Hydraulic Damp Static Seal Mechanism and Performance Influencing Factors. Applied Mechanics and Materials, 152: 1774-1778.

Web22. ttp://www.mbusa.com/vcm/MB/DigitalAssets/Vehicles/Models/2014/G63/Features/2014-G-CLASS-G63-AMG-SUV-010-MCFO.jpg. June, 2015.

Yan, B., X. Zhang and H. Niu, 2012. Design and test of a novel isolator with negative resistance electromagnetic shunt damping. Smart Materials and Structures, 21: 035003.

Yi, K. and B.S. Song, 1999. A new adaptive sky-hook control of vehicle semi-active suspensions. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 213(3): 293-303.

Australian Journal of Basic and Applied Sciences, 9(30) Special 2015, Pages: 59-66

Zhang, H., H. Winner and W. Li, 2009. Comparison between Skyhook and Minimax Control Strategies for Semi-active Suspension System. World Academy of Science, Engineering and Technology, 624-627.

Zhang, J.J., B.A. Han and R.Z. Gao, 2012. Multi-Body Model Identification of Vehicle Semi-Active Suspension Based on Genetic Neural Network. Applied Mechanics and Materials, 121: 4069-4073.

Zhu, X., X. Jing and L. Cheng, 2012. Systematic design of a magneto-rheological fluid embedded pneumatic vibration isolator subject to practical constraints. Smart Materials and Structures, 21: 035006.