American Journal of Science and Technology
2014; 1(2): 77-81
Published online May 20, 2014 (http://www.aascit.org/journal/ajst)

Keywords
Fully Conventional Suspension Systems,
Effect of Damping,
Comfort Level

Received: April 23, 2014
Revised: May 04, 2014
Accepted: May 05, 2014

Effect of damping on comfort level of a fully conventional suspension systems of an automobile

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Citation
A. Bala Raju, R. Venkatachalam. Effect of Damping on Comfort Level of a Fully Conventional Suspension Systems of an Automobile. *American Journal of Science and Technology.* Vol. 1, No. 2, 2014, pp. 77-81.

Abstract
Suspension system of an automobile not only supports the body of the vehicle, engine and passengers but also absorbs shocks arising from the roughness of the road. It has been a practice from the beginning to use conventional suspension system in the automobiles, which involves a chassis supported by the axles through springs and dampers. When the automobile is moving, the roughness of the road keeps giving excitations to the suspension system through tyres. The stiffness and damping of the suspension system play an important role in absorbing the shocks and provide comfort to the passengers. In this paper, an attempt is made to study the effect of damping on the comfort level.

1. Introduction

The suspension system is one of the most important systems of an automobile. Its main purpose is not only to support the engine, its components, passengers, but also to isolate them from shocks arising due to roughness of the road. It has been a practice from the beginning to have a frame called *chassis* which is being supported through springs and dampers by the front and rear axles. This type of suspension system is called *Conventional Suspension System*. There is yet another type of suspension called *Independent Suspension*, in which the axle of a wheel is hinged to the body and is held in position by springs and dampers which are placed in between axle and the body. There is no separate chassis and the body of the vehicle itself acts as chassis. Many of the present day cars use independent suspension for the front wheels and conventional suspension for the rear wheels. Such a system may be referred as *Semi-independent Suspension System*.

The study of suspension systems has been a subject of interest for many researchers. A suspension system may be modelled as a quarter car model or a half car model or a full car model. The quarter car model or half car model yield the results very quickly but they are not accurate because they do not represent the system in a realistic way because the roll and/or pitch motions cannot be taken into account by these models. Full car model considers the entire vehicle as it is. The results can be considered to be accurate and realistic. However, the analysis becomes more complex.

Hedrick [1] considered a quarter car model with hydraulic actuator acting under the effect of coulomb friction. An absorber based nonlinear controller and adaptive nonlinear controller are proposed. Employing two sensors, one for displacement and other for velocity measurements, Majjad [2] considered a quarter car model and estimated the nonlinear damping parameters. Gobbi and Mastin [3], Wei Gao et al. [8] studied dynamic behaviour of passively suspended vehicles running on rough roads. The road profile is considered to give random inputs to the suspension system. Rajalingam and Rakheja [4] studied the dynamic behaviour of quarter car model under nonlinear suspension damper. Ahmed Faheem [5] studied the dynamic behaviour using quarter car model and half car model for different excitations given by the road. Jacquelien et al. [6] used electrical analogy in conjunction with quarter car model and studied the control scheme of the suspension system. Wei Gao et al. [7] also studied the dynamic characteristics considering the mass, damping and tyre stiffness as random variables. Kamalakannan et al. [9] tried adaptive control by varying damping properties according to the road conditions. Sawant et al. [10] developed an experimental procedure for determining the suspension parameters using a quarter car model. Thite [11] refined the quarter car model to include the effect of series stiffness. State space equations are employed to calculate the natural frequency and model damping ratios. Gadhia et al. [12] analysed quarter car model for rear suspension using ADAMS software. Wei Gao et al. [13] investigated dynamic response of cars due to road roughness treating it as random excitation. Lin [14] performed a time domain direct identification for vehicle mass, damping and stiffness.

Husiyno Akcay [15] studied multi objective control of half car suspension system. It is observed that when the tyre damping coefficients are precisely estimated, the road holding quality of the suspension system can be improved to some extent. Li-Xing Gao [16] considered a half car model in conjunction with pseudo-excitation for the road conditions and studied the dynamic response of the vehicle. Thite et al. [17] used a frequency domain method for estimating suspension system parameters. Roberto Barbosa [18] studied the frequency response of half car model due to pavement roughness. Roberto Barbosa [19] also investigated vibrations of vehicles subjected to a long wave measured pavement irregularity.

Attempts are being made to analyze the four wheeler with fully independent suspension system. Libin Li [20] performed computer simulation studies through multi body model, identifying twenty degrees of freedom. Pater Gaspar [21] considered full car model and proposed a method for identifying suspension parameters taking into account nonlinear nature of the components. Anil Shirahatt et al. [22] attempted to maximize the comfort level considering a full car model. Genetic algorithms have been employed to perform optimization to arrive at optimum values of suspension parameters. Hajkurami et al. [23] studied the frequency response of a full car model as a system of seven degrees of freedom. Ikbal Eski [24] obtained neural network base control system for full car model. Guidaa et al. [25] proposed a method of identifying parameter of a full car model. The analysis has been developed for designing an active suspension system. Balaraju and Venkatachalam analysed the dynamic behaviour of an automobile using full car model for both, fully conventional suspension systems [26] and fully independent suspension systems [27].

In this paper, an attempt is made to study the effect of the damping on the comfort level offered by a fully conventional suspension system of an automobile.

Nomenclature

B, L_1, L_2	Dimensions of the main body see Figure 1
c_2 and c_3	Damping coefficients of dampers in suspension system, $N.s/m$
G_1 and G_2	Centres of mass of front and rear axles
I_1 and I_2	Mass moment of inertia of the front and rear axles, $kg.m^2$
I_r and I_p	Mass moments of inertia of the main body about roll and pitch axes, respectively, $kg.m^2$
k_1, c_1	Stiffness and damping of the tyre, N/m and $N.s/m$
k_2 and k_3	Stiffness's of springs in the suspension system, N/m
m	Mass of chassis and various things attached to the chassis, such as engine, body of the vehicle, kg
m_1 and m_2	Masses of the front and rear axles, kg
x	Vertical linear displacement of the main body
x_1 and x_2	Vertical linear displacements of masses m_1 and m_2
y_i	The vertical displacements caused by roughness of the road at different wheels, $i = 1$ to 4, m
γ_1 and γ_2	Roll motions of the front and rear axles
λ_1 and λ_2	Pitch motions of the front and rear axles
γ and λ	Roll and Pitch motions of the main body

2. Formulation

Figure 1 shows the arrangement of a conventional suspension system. (k_1, c_1) represent stiffness and damping properties of the tyres. (k_2, c_2) and (k_3, c_3) represent the properties of the shock absorbers, at the front suspension and rear suspension, respectively. m_1 and m_2 represent the masses of the front and rear axles, respectively. m represents the mass of the main body. The centre of mass G of the main body is located at distances, L_1 and L_2 from the front and rear ends, respectively. To describe the system, seven independent coordinates are chosen as, (x_1 and γ_1) to represent vertical linear motion and roll motion of the front axle, (x_2 and γ_2) to represent vertical linear motion and roll motion of the rear axle, and (x, γ and λ) to represent vertical linear motion of G and roll and pitch motions of the main body, respectively. The vertical displacements caused by the road roughness may be represented by the variables y_i, i = 1 to 4, as shown in the Figure 1. Figure also shows the absolute linear displacements of the four corners of the main body through the variables z_i, i = 1 to 4. The equations of motions may be derived as

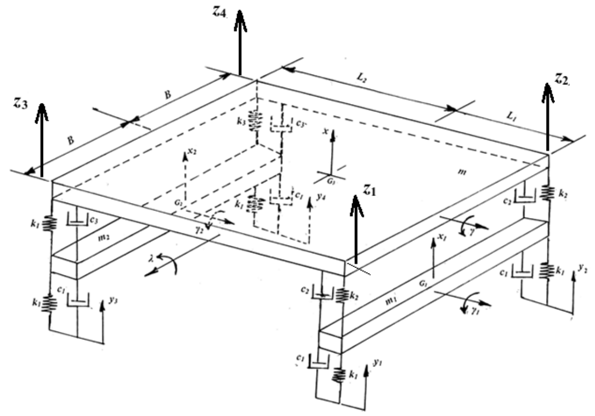

Figure 1. *Schematic arrangement of full car model of fully conventional suspension system*

$$\begin{aligned} &m_1\ddot{x}_1 + 2(k_1+k_2)x_1 - (2k_2)x - (2k_2L_1)\lambda \\ &+2(c_1+c_2)\dot{x}_1 - (2c_2)\dot{x} - (2c_2L_1)\dot{\lambda} = k_1(y_1+y_2) + c_1(\dot{y}_1+\dot{y}_2) \end{aligned} \quad (1a)$$

$$\begin{aligned} &I_1\ddot{\gamma}_1 + 2(k_1+k_2)B^2\gamma_1 - (2k_2B^2)\gamma \\ &+ 2(c_1+c_2)B^2\dot{\gamma}_1 - (2c_2B^2)\dot{\gamma} = -k_1B(y_1-y_2) - c_1B(\dot{y}_1-\dot{y}_2) \end{aligned} \quad (1b)$$

$$\begin{aligned} &m_2\ddot{x}_2 + 2(k_1+k_3)x_2 - (2k_3)x + (2k_3L_2)\lambda \\ &+ 2(c_1+c_3)\dot{x}_2 - (2c_3)\dot{x} + (2c_3L_2)\dot{\lambda} = k_1(y_3+y_4) + c_1(\dot{y}_3+\dot{y}_4) \end{aligned} \quad (1c)$$

$$\begin{aligned} &I_2\ddot{\gamma}_2 + 2(k_1+k_3)B^2\gamma_2 - (2k_3B^2)\gamma \\ &+ 2(c_1+c_3)B^2\dot{\gamma}_2 - (2c_3B^2)\dot{\gamma} = -k_1B(y_3-y_4) - c_1B(\dot{y}_3-\dot{y}_4) \end{aligned} \quad (1d)$$

$$\begin{aligned} &m\ddot{x} - (2k_2)x_1 - (2k_3)x_2 + 2(k_2+k_3)x + 2(k_2L_1-k_3L_2)\lambda \\ &-(2c_2)\dot{x}_1 - (2c_3)\dot{x}_2 + 2(c_2+c_3)\dot{x} + 2(c_2L_1-c_3L_2)\dot{\lambda} = 0 \end{aligned} \quad (1e)$$

$$\begin{aligned} &I_r\ddot{\gamma} - (2k_2B^2)\gamma_1 - (2k_3B^2)\gamma_2 + 2(k_2+k_3)B^2\gamma \\ &- (2c_2B^2)\dot{\gamma}_1 - (2c_3B^2)\dot{\gamma}_2 + 2(c_2+c_3)B^2\dot{\gamma} = 0 \end{aligned} \quad (1f)$$

$$\begin{aligned} &I_p\ddot{\lambda} - (2k_2L_1)x_1 + (2k_3L_2)x_2 + 2(k_2L_1-k_3L_2)x + 2(k_2L_1^2+k_3L_2^2)\lambda \\ &- (2c_2L_1)\dot{x}_1 + (2c_3L_2)\dot{x}_2 + 2(c_2L_1-c_3L_2)\dot{x} + 2(c_2L_1^2+c_3L_2^2)\dot{\lambda}=0 \end{aligned} \quad (1g)$$

The Equations (1) is forming a set of seven second order, linear, nonhomogeneous ordinary differential equations.

3. Analysis for Optimum Damping of the Suspension System

In order to study the comfort level of the passengers, a quantity z^2 defined as the sum of the squares of vertical displacements at each corner of the vehicle, is considered. It may be expressed mathematically as

$$z^2 = \sum_{i=1}^{4} z_i^2 \quad (2)$$

where, the displacements z_i, i = 1 to 4 may be expressed as

$$z_1 = x - B\gamma + L_1\lambda \quad (3a)$$

$$z_2 = x + B\gamma + L_1\lambda \quad (3b)$$

$$z_3 = x - B\gamma - L_2\lambda \quad (3c)$$

$$z_4 = x + B\gamma - L_2\lambda \quad (3d)$$

As it is defined, z^2 is a positive definite quantity, and hence the minimum value is zero. This can happen only when z_i , i = 1 to 4 are all zeroes, which implies that all the four corners of the main body are having zero absolute displacements, implying that the main body is not moving at all. Therefore, the best comfort may be achieved when the value of z^2 is zero. However, this is an ideal situation. Therefore, one attempts to look for achieving a possible minimum value of z^2 .

Based on a practical road vehicle *Santro Xing*, numerical values are assigned to various parameters involved for the suspension system as follows.

m_1= 40 kg	m_2 = 100 kg	m= 1000 kg
I_2=20 $kg.m^2$	I_r = 500 $kg.m^2$	I_p= 1000 $kg.m^2$
k_1=2×10^5N/m	k_2=0.5×10^5N/m	k_3=0.5×10^5N/m
c_1=1000$N.s/m$	L_1 = 1.0 m	L_2= 2.5 m
B= 0.75 m		

In the present analysis the damping parameters c_2 and c_3 are taken same as, $c_2 = c_3 = c$. Some typical disturbances are chosen as listed in Table 1. The equations of motion expressed in Equation (1) are integrated for each of the disturbance chosen, for different values of c in the range 500 to 6000 $N.s/m$. For each value of c, the maximum value of z^2 is noted. Figure 2a shows the variation of (z^2) $_{max}$ with c, for the cases 1,2,3 and 4 of the Set 1 described in Table1. Similarly, Figure 2b shows the variation of (z^2) $_{max}$ with c for the cases 5,6,7 and 8 of the Set 2. These two Figures reveal that for a value of c = 1000 $N.s/m$, the values of (z^2) $_{max}$ are minimum. Therefore, $c_2 = c_3$ = 1000 $N.s/m$ may be considered as optimum damping.

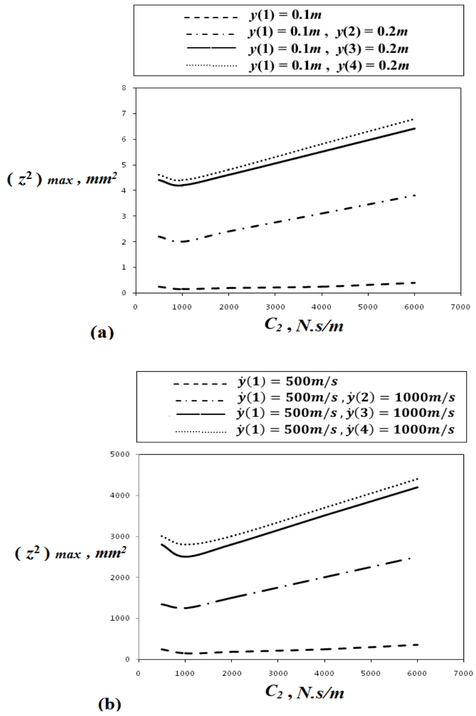

Figure 2. *Effect of damping on (z^2)$_{max}$ for fully conventional suspension system*

Table 1. *Different disturbances chosen*

Set 1. *Displacement disturbances (all are in m).*

Case 1	$y_1(0) = 0.1$	$y_2(0) = 0$	$y_3(0) = 0$	$y_4(0) = 0$
Case 2	$y_1(0) = 0.1$	$y_2(0) = 0.2$	$y_3(0) = 0$	$y_4(0) = 0$
Case 3	$y_1(0) = 0.1$	$y_2(0) = 0$	$y_3(0) = 0.2$	$y_4(0) = 0$
Case 4	$y_1(0) = 0.1$	$y_2(0) = 0$	$y_3(0) = 0$	$y_4(0) = 0.2$

Set 2. *Velocities disturbances (all are in m/s).*

Case 5	$\dot{y}_1(0) = 500$	$\dot{y}_2(0) = 0$	$\dot{y}_3(0) = 0$	$\dot{y}_4(0) = 0$
Case 6	$\dot{y}_1(0) = 500$	$\dot{y}_2(0) = 1000$	$\dot{y}_3(0) = 0$	$\dot{y}_4(0) = 0$
Case 7	$\dot{y}_1(0) = 500$	$\dot{y}_2(0) = 0$	$\dot{y}_3(0) = 1000$	$\dot{y}_4(0) = 0$
Case 8	$\dot{y}_1(0) = 500$	$\dot{y}_2(0) = 0$	$\dot{y}_3(0) = 0$	$\dot{y}_4(0) = 1000$

The existence of optimum value of damping may be explained as follows. When damping is nearly zero, the system would be an undamped system. Therefore, one can expect large amplitudes of forced vibrations. When damping is very large, it may be realised that the damper is providing a rigid connection of chassis to the axles. Hence, the entire disturbance is directly transferred to the main body. One should allow free movement of the plunger of the damper for pumping out the energy of the system effectively. Therefore, there must exist an optimum value of damping.

4. Concluding Remarks

The work presented in this paper and significant conclusions that may be drawn based on the present work may be summarized as follows.

(i) A full car model of fully conventional suspension system is studied for optimum damping parameters.
(ii) For the purpose of study, the values of various other parameters are taken which correspond to a real practical automobile.
(iii) A quantity z^2 is defined to indicate the comfort level for the passengers.
(iv) The damping of the shock absorber is varied from 500 to 6000 $N.s/m$ in steps of 500 $N.s/m$. For each damping value the time response is observed for some typical disturbances and the maximum value of z^2 is noted.
(v) The reason for the existence of an optimum value of damping is discussed.
(vi) A damping value of 1000 $N.s/m$ is found to offer the lowest value for (z^2)$_{max}$ for all the disturbances.

References

[1] Hedrick, J. K., *Nonlinear control of a Quarter Car active suspension*, Proceedings of the American Control Conference, Chicago, June 1992.

[2] Majjad, R., *Estimation of suspension parameters*, Proceedings of the IEEE International Conference on Control Application, Hartford, October 1997.

[3] Gobbi, M. and Mastinu, G., *Analytical description and optimization of the dynamic behaviour of passively suspended road vehicles*, Journal of Sound and Vibration, Vol. 245, No.3, pp. 457 – 481, 2001.

[4] Rajalingam, C. and Rakheja, S., *Influence of suspension damper asymmetry on vehicle vibration response to ground excitation*, Journal of Sound and Vibration, Vol. 266, pp.1117 – 1129, 2003.

[5] Ahmad Faheem, Firoz Alam, and Varikan Thomas, *The suspension dynamic analysis for a quarter car model and half car model*, 3rd BSME-ASME International Conference on Thermal Engineering, Dhaka, December 2006.

[6] Jacquelien, M.A., Scherpen, Dimitri Jeltsema, and Francois Maulny, *Parallel damping injection for the quarter car suspension system*, Proceedings of the 17th International Symposium on mathematical theory of networks and Systems, Japan, July 2006.

[7] Wei Gao, Nong Zhang, *Dynamic analysis of vehicles with uncertain parameter*, 14th International Congress on Sound and Vibration, Australia, July 2007.

[8] Wei Gao, Nong Zhang, and Jun Dai, *A stochastic quarter car model for dynamic analysis of vehicle with uncertain parameters*, International Journal of Vehicle Mechanics and Mobility, Vol. 46, No.12, pp. 1159 – 1169, 2008.

[9] Kamalakannan, K., Elaya Perumal, A., Mangalaramanan, S., Arunachalam, K., *Performance analysis of behaviour characteristics of CVD (semi active) in quarter car model*, Jordan Journal of Mechanical and industrial engineering, Vol. 5, pp. 261 – 265, June 2011.

[10] Sawant, S.H., Belwakar,V., Kamble, A., Pushpa, B., and Patel, D., *Vibrational analysis of quarter car vehicle dynamic system subjected to harmonic excitation by road surface*, Undergraduate Academic Research Journal ,Vol. 1, pp. 46 – 49, 2012.

[11] Thite, A.N., *Development of a refined quarter car model for the analysis of discomfort due to vibration*, Advances in Acoustics and Vibration, Hindawi Publishing Corporation, 2012.

[12] Gadhia Utsav, D., Sumanth, P., *Quarter model analysis of Wagon-R Car's rear suspension using ADAMS*, International Journal of Engineering Research and Technology, Vol. 1, pp. 1 – 5, 2012.

[13] Gao, W., Zhang, N., and Du, H.P., *A half-car model for dynamic analysis of vehicles with random parameters*, 5th Australasian Congress on Applied Mechanics, Brisbane, Australia, December 2007.

[14] Lin, Y., and Kortum, W., *Identification of system physical parameters for vehicle systems with nonlinear components*, International Journal of Vehicle Mechanics and Mobility, Vol. 46, No.12, pp. 354 – 365, 2007.

[15] Husenyin Akcay, and Semiha Turkay, *Influence of tire on mixed H_2 / H_∞ synthesis of half car active suspensions*, Journal of Sound and Vibration, Vol. 322, pp.15 – 28, 2009.

[16] Li-Xing Gao, and Li-Ping-Zing., *Vehicle vibration analysis in changed speeds solved by pseudoexcitation method*, Mathematical Problems in Engineering, Hindawi Publishing Corporation, 2010.

[17] Thite, A.N., Banvidi, S., Ibicek, T., and Bennett, L., *Suspension parameter estimation in the frequency domain using a matrix inversion approach*, International Journal of Vehicle Mechanics and Mobility, Vol. 49, No.12, pp. 1803 – 1822, 2011.

[18] Roberto Spinola Barbosa, *Vehicle dynamic response due to pavement irregularity*, Journal of Brazilian Society of Mechanical Sciences and Engineering, Vol. 33, No. 23, pp. 302 – 307, 2011.

[19] Roberto Spinola Barbosa, *Vehicle vibration response subjected to long wave measured pavement roughness*, Journal of Mechanical Engineering and Automation, Vol. 2, pp. 17 – 24, No. 2, 2012.

[20] Libin Li, and Qiang Li., *Vibration analysis based on full multi-body model for the commercial vehicle suspension system*, Proceedings of the 6th International Conference on Signal Processing, Robotics and Automation, Greece, pp. 203 – 207, February 2007.

[21] Pater Gaspar, and Laszlo Nadai., *Estimation of dynamical parameters of road vehicles on freeways*, 5th International Symposium on Intelligent Systems and Informatics, Serbia, August 2007.

[22] Anil Shirahatt, Prasad, P.S.S., Pravin Panzade, and Kulkarni, M.M., *Optimal design of passenger car suspension for ride and road holding*, Journal of Brazilian Society of Mechanical Sciences and Engineering, Vol. 30, No.1, pp.66 – 74, 2008.

[23] Hajkurami, H., Samandari, H., Ziaei-Rad, S., *Analysis of chaotic vibration of nonlinear seven degree of freedom full car model*, 3rd International Conference on Integrity, Reliability and Failure, Portugal, July 2009.

[24] Ikbal Eski, and Sahin Yildirim, *Vibration control of vehicle active suspension system using a new robust neural network control system*, Journal of Simulation Modelling Practice and Theory, Vol. 17, pp. 778 – 793, 2009.

[25] Guidaa, D., Nilvetti,F., and Pappalardo, C.M., *Parameter identification of full-car model for active suspension design*, Journal of Achievements in Materials and Manufacturing Engineering, Vol. 40, No. 2, pp. 138 – 148, 2010.

[26] Balaraju, A. and Venkatachalam, R., *Analysis of vibrations of automobile suspension system using full car model*, International Journal of Scientific and Engineering Research, Vol. 4, Issue 9, September 2013, pp. 2305-2311.

[27] Balaraju, A. and Venkatachalam, R., *A study on conventional versus independent suspension system of an automobile*, Applied Mechanics and Materials, Vol. 541-542 (2014), pp. 827-831.