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

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Citation

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

Husiyono 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. Guida 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
- \( c_1 \) and \( c_2 \) 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.m²
- \( I_1 \) and \( I_0 \) Mass moments of inertia of the main body about roll and pitch axes, respectively, kg.m²
- \( k_1 \), \( c_1 \) 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
- \( \gamma_i \) and \( \gamma_{2,i} \) Roll motions of the front and rear axles
- \( \lambda_1 \) and \( \lambda_2 \) Pitch motions of the front and rear axles
- \( y \) and \( \lambda \) 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, \((x_i, y_i, z_i)\) to represent vertical linear motion and roll motion of the front axle, \((x_i, y_i, z_i)\) to represent vertical linear motion and roll motion of the rear axle, and \((x, y, \lambda)\) 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

\[
\begin{align*}
    m_1 \ddot{x}_1 + 2(k_1 + k_2)x_1 - (2k_2)x - (2k_2L_1)\lambda + 2(c_1 + c_2)x_1 - (2c_2)x - (2c_2L_1)\dot{\lambda} & = k_1(y_1 + y_2) + c_1(y_1 + y_2) \\
    l_1 \ddot{y}_1 + 2(k_1 + k_2)B^2 y_1 - (2k_2B^2)\gamma & = k_2B(y_1 - y_2) - c_2B(\dot{y}_1 - \dot{y}_2) \\
    m_2 \ddot{x}_2 + 2(k_1 + k_3)x_2 - (2k_3)x + (2k_3L_2)\lambda + 2(c_1 + c_3)x_2 - (2c_3)x + (2c_3L_2)\dot{\lambda} & = k_1(y_3 + y_4) + c_1(y_3 + y_4) \\
    l_2 \ddot{y}_2 + 2(k_1 + k_3)B^2 y_2 - (2k_3B^2)\gamma & = k_2B(y_3 - y_4) - c_2B(\dot{y}_3 - \dot{y}_4) \\
    m \ddot{x} - (2k_2)x_1 - (2k_2)x_2 + 2(k_2 + k_3)x + 2(k_2L_1 - k_3L_2)\lambda - (2c_2)x_1 - (2c_3)x_2 + 2(c_2 + c_3)x + 2(c_2L_1 - c_3L_2)\dot{\lambda} & = 0 \\
    l_1 \ddot{y} - (2k_2B^2) y_1 - (2k_2B^2) y_2 + 2(k_2 + k_3)B^2 \gamma - (2c_2B^2) \dot{y}_1 - (2c_3B^2) \dot{y}_2 + 2(c_2 + c_3)B^2 \dot{\gamma} & = 0 \\
    l_2 \ddot{y} - (2k_3L_1) x_1 + (2k_3L_2)x_2 + 2(k_3L_1 - k_3L_2)x + 2(k_3L_1^2 + k_3L_2^2)\lambda - (2c_2L_1)x_1 + (2c_3L_2)x_2 + 2(c_2L_1 - c_3L_2)x + 2(c_2L_1^2 + c_3L_2^2)\dot{\lambda} & = 0
\end{align*}
\]

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

\[
z^2 = \sum_{i=1}^{4} z_i^2
\]

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

\[
z_1 = x - By + L_1\lambda
\]

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.

\[ m_1 = 40 \text{ kg} \quad m_2 = 100 \text{ kg} \quad m = 1000 \text{ kg} \]

\[ I_z = 20 \text{ kg.m}^2 \quad I_r = 500 \text{ kg.m}^2 \quad I_i = 1000 \text{ kg.m}^2 \]

\[ k_1 = 2 \times 10^5 \text{ N/m} \quad k_2 = 0.5 \times 10^5 \text{ N/m} \quad k_3 = 0.5 \times 10^5 \text{ N/m} \]

\[ c_1 = 1000 \text{ N.s/m} \quad L_1 = 1.0 \text{ m} \quad L_2 = 2.5 \text{ m} \]

\[ B = 0.75 \text{ m} \]

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 \((z^2)_{\text{max}}\) with \( c \) for the cases 1,2,3 and 4 of the Set 1 described in Table 1. Similarly, Figure 2b shows the variation of \((z^2)_{\text{max}}\) with \( c \) for the cases 5,6,7 and 8 of the Set 2. These two Figures reveal that for a value of \( c = 1000 \text{ N.s/m} \), the values of \((z^2)_{\text{max}}\) are minimum. Therefore, \( c_2 = c_3 = 1000 \text{ N.s/m} \) may be considered as optimum damping.

**Table 1. Different disturbances chosen**

<table>
<thead>
<tr>
<th>Case</th>
<th>( y_1(0) )</th>
<th>( y_2(0) )</th>
<th>( y_3(0) )</th>
<th>( y_4(0) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.1</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>0.1</td>
<td>0.2</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>3</td>
<td>0.1</td>
<td>0</td>
<td>0.2</td>
<td>0</td>
</tr>
<tr>
<td>4</td>
<td>0.1</td>
<td>0</td>
<td>0</td>
<td>0.2</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Case</th>
<th>( \dot{y}_1(0) )</th>
<th>( \dot{y}_2(0) )</th>
<th>( \dot{y}_3(0) )</th>
<th>( \dot{y}_4(0) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>500</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>6</td>
<td>500</td>
<td>1000</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>7</td>
<td>500</td>
<td>0</td>
<td>1000</td>
<td>0</td>
</tr>
<tr>
<td>8</td>
<td>500</td>
<td>0</td>
<td>0</td>
<td>1000</td>
</tr>
</tbody>
</table>

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 \((z^2)_{\text{max}}\) for all the disturbances.
References


