Research on Air Suspension System Based on Genetic LQG and PID Control

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Abstract. An active air suspension system for vehicles using the Linear Quadratic Regulator vibration controller and Proportional Integral Derivative control strategy is presented. A quarter car with two degrees of freedom air suspension vibration model is described. Compared with the conventional passive suspension system, the analysis is done to the system control performance. The road profile is simulated according to ISO8608. The analysis of the system response is obtained through a lot of simulation work. The simulation results indicate that the vibration of the vehicle can be reduced and the ride comfort is improved by the two proposed suspension system.

Introduction

A suspension system is one of the most important components in a vehicle. It has two main functionalities. One is increasing driving comfort, by isolating the vehicle body from external disturbances, which mainly come from irregular road surfaces. The other one is to maintain a firm contact between road and tires to provide guidance along the track. This is called handling performance. In a conventional suspension system, which is composed of passive springs and dampers, a trade-off is required to resolve the conflicting requirements of ride comfort and good handling performance. The reason is that a hard suspension is required for handling and to support the mass of the vehicle, on the other hand, a soft suspension is needed to isolate the disturbances from the road.

A semi-active system is able to alter some of its characteristics, e.g. the amount of damping. The active system is able to change the amount of dissipative power, but it can also supply power to the system, hence it can work in the four quadrants of the force-velocity diagram. This allows for significantly improving the performance of the suspension system with an active suspension. Many research work has been done to the control of the suspension system. An experimental verification on a quarter car setup of a robust controller in combination with a hydraulic actuator is conducted [1]. The robust controller was designed to reduce the body vibrations at 1.5 Hz without amplifying the body acceleration and tire force at other frequencies. To describe comfort, only the body acceleration between 1 and 5 Hz is taken into account. This resulted in a reduction of 50%. An $H_\infty$ robust controller was tested on a test vehicle with a semi-active suspension [2].

The Cowey Motor Works of Great Britain introduced air suspension that Lincoln ballyhooed for some models introduced in 1984 in 1909. It did not work well because it leaked. The first practical air suspension was developed by Firestone in 1933 for an experimental vehicle termed the Stout-Scarab. This was a rear-engine vehicle that used four rubberized bellows in place of conventional springs. Small M-P compressors attached to each bellow supplied air. As people might imagine, the air bag suspension was an expensive setup—still is, in fact. Air suspensions are designed to cushion the ride and keep the car, bus or truck level fore and aft and at a constant height regardless of load. Air suspension was introduced on many luxury vehicles in the late 1950s, but it was dropped after one or two model years. Recently, however, new leveling systems have been researched and developed for passenger automotive vehicle use, including air-adjustable rear shock absorber. A typical air
The suspension system consists of an ECE- or ICE-driven M-P compressor, air supply pneumatic tank, filter or condenser, pneumatic valves, piping, controls and air springs or bellows.

**Vehicle Model**

**Air Spring Model**

Mathematical progression theory with small deviation linearization method is adopted in the paper [3], to analyze the influence factors of air spring dynamic stiffness and the influence law, linear model is as presented as follows

\[
K_{s}^*(j\omega) = \frac{F_s}{h} = \left[ \frac{\beta (P_0 - P_i) - kp_0 A_{s0}}{V_{s0}} \right] \omega^2 + \gamma^2 A_s \left[ \frac{\alpha k^2 A_{s0}}{\rho_0 V_{s0}^2} \left( 1 + \frac{1}{n} \frac{f k P_0 (P_{c0} - P_0)}{\rho_0 V_{s0}^2} \right)^2 \right]^{1/2} + \left[ \frac{1 + \frac{m_{s0}}{m_{s0}} \frac{k P_0}{\rho_0 V_{s0}^2}}{n^2} \right] \omega^2 \]

(1)

where \( m_{s0}/m_{s0} = n \) is the initial static when additional air chamber and the main air chamber air mass ratio; At the same time, due to the initial state when the master and the density of the air chamber air is the same, so \( n = m_{s0}/m_{s0} = V_{s0}/V_{s0} \); \( n \) is the v of volume; \( m_{s0} = \rho_0 V_{s0} \), \( m_{s0} = \rho_0 V_{a0} \), \( \rho_0 \) is the initial gas indoor air density.

**Modeling of the Air Spring Active Suspension System**

A quarter-vehicle model, shown in Fig. 2, is used to develop the air spring active suspension system. A sensor, an air spring, a damper, an actuator and a control unit constitute an active suspension system. The controller generates the control input for the actuator, based on a control algorithm from the driving conditions of the vehicle. The vehicle body and wheel are represented by the sprung mass and the sprung mass, respectively, and the tire is represented by a linear spring [4]. The basic assumptions are that the tire always contacts the road surface and the sprung mass of the vehicle body is a rigid body. From Fig. 2, the following differential equations of motion for this quarter-vehicle air spring active suspension system can be derived as follows:

\[
m_s \ddot{x}_s = -c(\dot{x}_s - \dot{x}_u) - f_{ks} + f; m_u \ddot{x}_u = c(\dot{x}_s - \dot{x}_u) + f_{ks} - k_i (x_u - x_s) - f
\]

(2)

where \( m_s \) is the sprung mass; \( m_u \) is the sprung mass; \( c \) is the damping coefficient of the suspension; \( f_{ks} = f_{as} - P \) is the air spring force, in which \( P \) is the static load on the air spring; \( k_i \) is the tire stiffness; \( x_s - x_u \) and \( x_u - x_s \) are the suspension stroke and tire deflection, respectively. \( x_s \) is the vertical ground displacement caused by the road unevenness; and \( f \) is the active force.
The state vector, $X$, the control input vector, $U$, and the external disturbance vector, $W$, are defined as follows:

$$X = \begin{bmatrix} x \  \dot{x} \ x_u \ \dot{x_u} \end{bmatrix}^T = \begin{bmatrix} x_1 \ x_2 \ x_3 \ x_4 \end{bmatrix}^T$$

$$U = \begin{bmatrix} f \end{bmatrix}, W = \begin{bmatrix} w \end{bmatrix}$$

The equations can be written in the state variable form as follows:

$$\dot{X} = F(X,t) + G(X,t) + D(X,t)W$$

### Simulation of Road Profile

To compare simulations with measurements, the road profile must be similar for both simulation and measurements. However, it is not possible to measure the exact road profile of each measurement, hence the road profile used in the simulations has to be estimated. The road profile used in the simulation is therefore filtered white noise. This filter is a combination of a notch filter and two times a first order low-pass filter

$$W_{\text{road}} = G_{\text{road}} \left( \frac{1}{\omega_1^2 s^2 + 2 \beta_1 \omega_1 s + 1} \right) \left( \frac{1}{\omega_2^2 s^2 + 2 \beta_2 \omega_2 s + 1} \right)$$

where $\omega_1$ and $\omega_2$ are the frequencies of the notch filter; $\omega_3$ is the cut-off frequency of the low-pass filter; $\beta_1$ and $\beta_2$ are the damping coefficients of the notch filter; $G_{\text{road}}$ is the gain to achieve the correct road amplitude. In general, a road profile is classified by means of the ISO8608[5], the following expression is proposed to represent a road profile: $S_r(n) = S_r(n_0) \left( \frac{n}{n_0} \right)^{-w}$ where $S_r(n)$ is the PSD of the road profile; $n$ is the spatial frequency; The reference spatial frequency $n_0$ is defined as $n_0 = 0.1 \text{m}^{-1}$; $S_r(n_0)$ is the road roughness coefficient, which is the value of the PSD at the reference spatial frequency $n_0$, and represents different grades of road; $w$ is called waviness and indicates whether the road has more long wavelengths or short wavelengths.

### Design of Controller

#### LQG Controller

The vibration controller is designed based on the linear quadratic gaussian optimal control theory using full state feedback gains and considering perfect measurements. The following performance index is selected to minimize the most important ride comfort performance criteria such as vertical body acceleration, pitch acceleration, front and rear suspension working spaces, front and rear dynamic tire deflections. The performance target for LQG controller design scheme is the integral value of $T$ in the weighted sum of squares in the time domain for tyre, suspension travel and dynamic displacement body vertical vibration acceleration. The expression is showed as following:

$$J = \lim_{T \to \infty} \frac{1}{T} \int_0^T \left[ p_1[z_1 - q]^2 + p_2[z_2 - z]^2 + \rho z_2 \right] dt$$

where, $p_1$ is the weighted coefficient of tire dynamic displacement; $p_2$ is the weighted coefficient of suspension dynamic schedule; $\rho$ is the weighted coefficient of vehicle body acceleration.

Equation (6) is expressed in form of matrix

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When the vehicle parameters and weighted coefficient values are determined, the optimal control feedback gain matrix \( K \) can be determined by the Riccati equation, the following form:

\[
PA + A^T P + PBR^{-1}B^T P - Q = 0
\]

where \( P \) is the solution of riccati equation \( K = R^{-1}B^T P \). The actuator optimal control force can be obtained from

\[
U = -KX_t
\]

With a lot of simulation, the actuator optimal control force is expressed as follows

\[
K = [1755.0, -1101.5, -15970.3, -667.3, 16677.2]
\]

It has already been carried out on the control system of vehicle active suspension systems design, for the performance of the inspection system shall be carried out in a computer simulation to verify, if found in the process of simulation to verify the performance of the system cannot meet the requirements, still need to redesign and modify the parameters of the controller.

**PID Controller**

The objective of this controller is to cancel out the deflections in both front and rear suspension systems due to the vehicle body load variations, thus maintaining the vehicle body at a constant height above the ground. The controller is designed by using the Proportional-Integral-Derivative compensator. This controller is widely used in process controllers to eliminate the steady state offsets and to improve the transient behavior. For both front and rear suspension systems, a low pass first order filter is used to improve the PID controller loop stability. The equation of motion for this filter is given by:

\[
y_f + \tau \frac{dy_f}{dt} = x_{BW}
\]

where \( y_f \) is the filter output; \( x_{BW} = x_{Bf} - x_{Wf} \) stands for the front suspension system; \( \tau \) is the filter time constant; \( x_{BW} = x_{Br} - x_{Wr} \) is the rear suspension system. The front and rear suspension deflections \( (x_{BW})_{front} \), \( (x_{BW})_{rear} \) are assumed to be measured by displacement transducers mounted on each wheel. The filter output \( y \) is used as input to the proportional integral derivative controller. The mathematical representation for this controller is given by:

\[
y_L = K_{LP} y_f + K_{IL} \int y_f dt + K_{LD} \frac{dy_f}{dt}
\]

where \( y_L \) is the PID demand signal; \( K_{LP}, K_{IL}, K_{LD} \) are the proportional, integral and derivative gains. The PID demand signal is used to drive the proportional directional control valve. In practice, care must be considered when using the integral and derivative actions in the PID compensator especially when there is a noise in the controlled signal. This is because the integral action has the property of resetting windup which causes a large overshoot due to the controller effort build up under certain controller deviation, and the derivative action amplifies the high frequency components which will cause violent motion in the actuator.

**Results and Discussion**

To verify the validity of the method, the paper includes a great deal of calculating process based on the simulation software “Matlab/Simulink”. The performance of the active control scheme is illustrated through a series of simulations.
Fig. 3 indicates the controlled system and the passive system sprung mass vertical acceleration response. For comparison, they are shown in one figure. The solid line represents the system before control, the dash line indicates the system after PID control, and the solid line indicates the system after LQG control. Fig. 4 shows the controlled suspension travel response and the passive system suspension travel response. Then Fig. 5 demonstrates the controlled dynamic tire displacement and the passive system dynamic tire displacement. From these figures, we can see that the vertical control of the sprung mass is decreased by the controlled systems, which indicates that the PID and LQG control system is effective in improving the system ride comfort, and for the model presented in the paper, the PQG controller is more effective in performance melioration than the PID controller. The response of suspension travels and the dynamic tire deflections emphasizes the fact that the controlled system can improve the system stability.

**Figure 3.** Body vertical acceleration. **Figure 4.** Suspension travel. **Figure 5.** Dynamic tire displacement.

Table 1 includes the performance analysis of the system. The body acceleration and pitching angular acceleration is greatly decreased, and the passenger acceleration is reduced dramatically, which indicates that the genetic algorithm and neural network controller is effective in improving the system riding comfort. The tire dynamic load is reduced, which indicates that the system stability is meliorated.

<table>
<thead>
<tr>
<th>r.m.s</th>
<th>unit</th>
<th>Passive system</th>
<th>PID controller</th>
<th>Performance melioration (%)</th>
<th>LQG controller</th>
<th>Performance melioration (%)</th>
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<tr>
<td>Body acceleration</td>
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<tr>
<td>dynamic tire displacement</td>
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<td>0.0102</td>
<td>5.6%</td>
<td>0.0092</td>
<td>14.81%</td>
</tr>
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</table>

**Conclusions**

In this work, using a two degrees of freedom active air suspension model, the Linear Quadratic Regulator vibration controller and Proportional Integral Derivative controller has been designed to isolate the body vibration from the road surface irregularities. It is obvious from the response plots that vehicle body vertical acceleration, the suspension travel response decreased compared with the passive suspension system, which naturally brings ride comfort. And the tire distortion is decreased, which indicates that the proposed controller proves to be effective in the comfort and stability improvement of the suspension system.

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References


