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ACTIVE VIBRATION CONTROL OF A QUARTER CARS MODEL USING PID CONTROLLED AIR SPRING

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ABSTRACT

Air springs have been used to isolate vibrations and provide a comfortable ride to the passengers. It has been observed that high comfort can be provided by using air spring as secondary suspension system. In order to predict vibrations in the car body, a valid model of the air spring is needed. Since the experimental setup for dynamic analysis of air spring is time consuming, simulation via mathematical model is used to obtain the response behavior of the air spring. In this paper a linear quarter car model with air suspension system has been formulated. Here a comparative study among coil spring and linear air spring as suspension system has been done. A PID control algorithm has also been implemented for suspension system and responses have been obtained. Input provided to the quarter car model is band limited noise, given as road disturbance. The comparative study has depicted how a PID controlled linear air suspension model has been able to isolate the road disturbances to maximum extent.

Key words: *Air Spring, PID Control, Band Limited Noise, dynamic analysis*

1. Introduction

For vehicle designer the design of suspension system is of utmost concern, as it has direct influence on many parts of the car at the same time such as handling stability, ride comfort and ride performance [1]. Driving Comfort along with vehicle safety is very vital and can be improved by dynamic simulation of vehicle suspension system. Because of inherent advantages of ride performance, vehicle body control in electronically controlled air spring many manufacturers have started using them [2]. Using an air suspension in place of passive suspension system can reduce the vibrations to a great extent [3].Once we are into controlling part of the suspension system we have different types of controllers as PID controllers, fuzzy logic controllers, neural based controllers. A comparison between PDI controller and fuzzy logic controller has been dealt in this paper and has been shown how PID is better than other controlling strategies in case of electronically controlled suspension system.

2. Mathematical Model of Air Spring

An effective suspension design is as good as the mathematical model. Thus, in preceding sections focus will be on developing the mathematical model.

2.1 Free Body Diagram of System

Considering the free body diagram shown in figure(1), equation of motion is derived .It can be seen in equation (1), subsequently the spring rate was determined from polytropic relation. Equation also shows how spring rate is related to piston area, spring height, and the internal pressure.

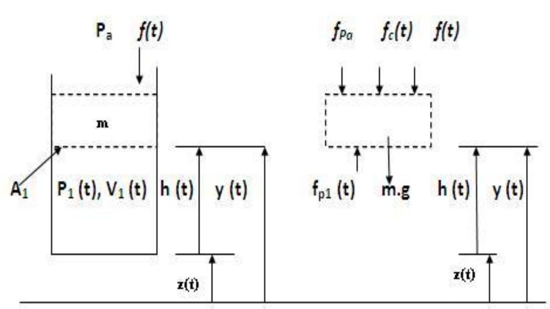

Fig. 1 Free body diagram

Having derived the equation for spring stiffness, the system was linearised about the equilibrium height using the simplified version of spring stiffness:

$$k_{equ} = \frac{nA_1P_1}{h_0} \qquad (1)$$

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Furthermore, the effective area of the spring was calculated. The area is determined by the following equation

$$A_1 = \frac{\pi}{8}(D_1^2 + D_2^2) \quad (2)$$

For keeping the system linear and calculation simple the linear air spring has been designed. To form linear air spring, effect of atmospheric pressure and damping force due to thermodynamic behavior are not considered.

The state equations are presented below. They are the equation of motion and the equivalent spring stiffness as derived for the linear spring.

$$\dot{y} = \int \ddot{y}(t)dt \quad (3)$$

$$\ddot{y}(t) = \frac{1}{m}\begin{Bmatrix} k_{equ}\left[\left(y(t) - \ddot{z}(t)\right) - (y_0 - z_0)\right] - \\ c\left(\dot{y}(t) - \dot{z}(t)\right) - f(t) \end{Bmatrix} \quad (4)$$

$$k_{equ} = \frac{nA_1P_0}{h_0} \quad (5)$$

States: $y(t), \dot{y}(t), \ddot{y}(t)$

3. Calculation for Air Spring Frequency

For ride comfort performance natural frequency of the vehicle vibration is a crucial factor. The vertical vibration frequency ranges between 1 ~1.6 HZ while walking up and down [6]. The vehicle body natural frequency as a system should be near to vertical vibration frequency range.

In case of mechanical analysis, vehicle treated as a single degree of freedom elastic system and its natural frequency is given by:

$$n = \frac{1}{2\pi}\sqrt{\frac{k_{equ}}{m}} \quad (6)$$

The natural frequency of the air spring designed for this caseis n=1.53 Hz.

4. Road Excitation

When approximately regarded the vehicle as a linear system, it can be calculated physical quantities power spectrum by the input power spectrum of road roughness and vehicle system frequency response function. They are used to analyze the vibration response of the system parameters on the impact of physical and evaluation of ride comfort performance.

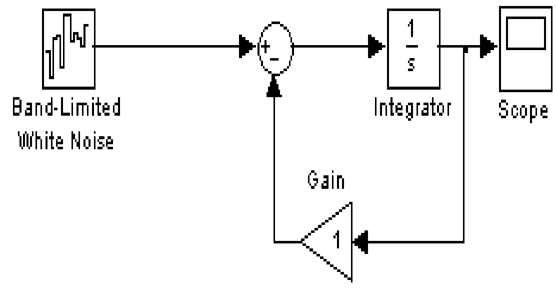

Fig. 2 White Noise in Simulink model

Using the SIMULINK of the B-level white noise produces a random road excitation as shown in figure (2), by modifying the initial value of which can be drawn from the results of different random excitation, as shown in figure (3).This is considering a pothole present in the road and its response to the system

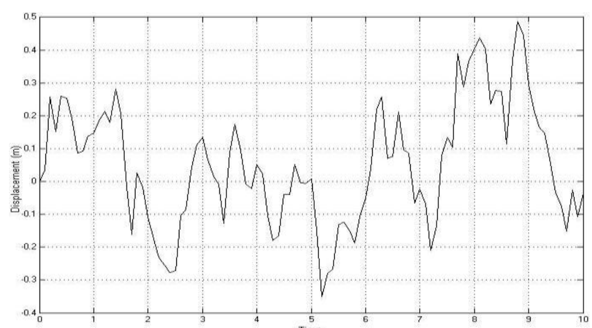

Fig. 3 White Noise Input

5. PID Control

Over the time quite a large number of researches have been done in the area of suspension control. Many algorithms are available as of now; some of them are PID control, sky hook control, fuzzy control, neural network control, optimal control, predictive control. Optimal control being widely used has some disadvantages. The non linear and uncertain behavior of vehicle poses problem for optimal control which requires accurate model of the system. Considering the PID control of air spring for a quarter car model Simulink model was developed and benchmark test were made. A clear coherence showed effectiveness of using PID as control strategy in controlling semi active air suspension [7].

Using adaptive and neural control will be time consuming and poor real time. So owing to its benefits of being easy to implement and control, PID is suitable choice for control algorithm for this case.

For PID control three parameters needs to be determined. For this trial and error method is used to obtain the required parameters.

$$u = K_p e + K_I \int edt + K_D \frac{de}{dt} \qquad (7)$$

In the equation (7) the signal (u) will be sent to the plant, and the new output will be obtained. This new output will be sent back to the sensor again to find the new error signal (e). The controller takes this new error signal and computes its derivative and it's integral again. This process goes on and on.

6. Simulink Model

Both differential equations and state equation can be used to establish the suspension model. In this work differential equations were made using MATLAB SIMULINK model of suspension system.

Following model as shown in figure (4) was developed using Simulink. This shows a linear suspension model for a quarter car model. Here the vertical displacement along with accelerations and velocity can be obtained in the scope as output.

6.1 Response of Coil Spring

When the vehicle with coil spring as secondary suspension system is given road disturbances it behaves as shown in figure (5).This shows how the passengers seated inside the vehicle are prone to continuous vertical displacements. It is not good from comfort point of view.

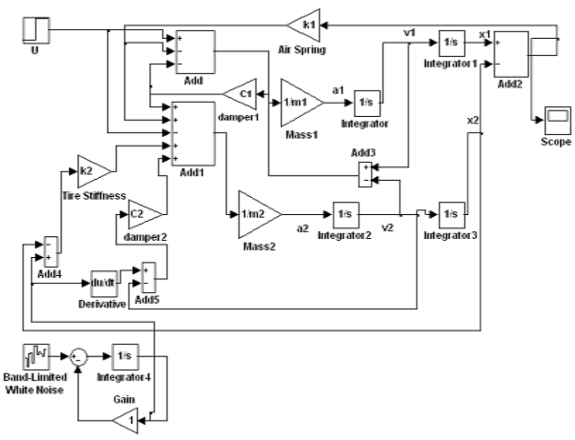

Fig. 4 Simulink Model of Quarter Car

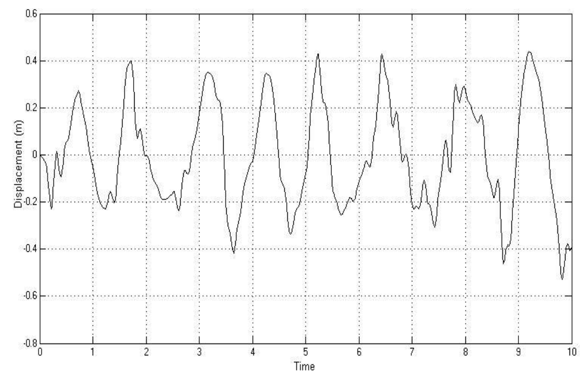

Fig. 5 Response of Coil Spring

6.2 Response of Air Spring

In comparison to coil spring as secondary suspension system the air spring can bring down the vertical displacements to almost half value as can be seen by figure (6). Thus the passengers sitting inside the vehicle will get lesser disturbances using air spring compared to coil spring as secondary suspension system.

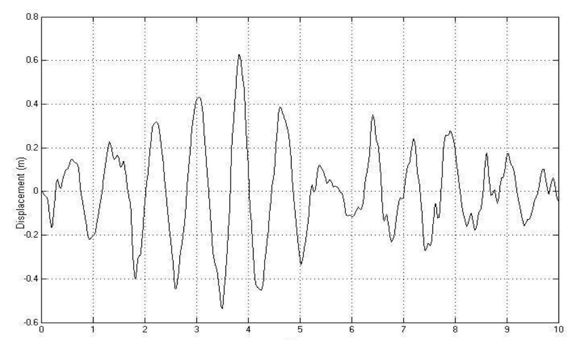

Fig. 6 Response of Air Spring

6.3 Response of PID Controlled Air Spring

After the responses in vertical direction were obtained, the PID controlled suspension was developed and vertical vibrations were measured using Simulink, as can be seen in figure (7). It shows how an active vibration control system suppresses the vertical disturbances effectively, almost ten times less as compared to convention secondary suspension system. It can be observed in figure (8).

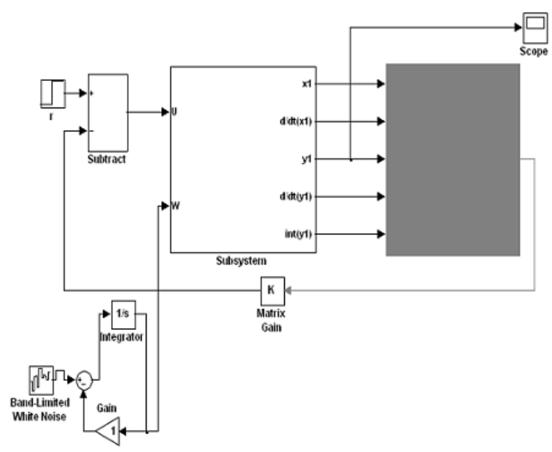

Fig. 7 Simulink Model of PID controlled Air Suspension

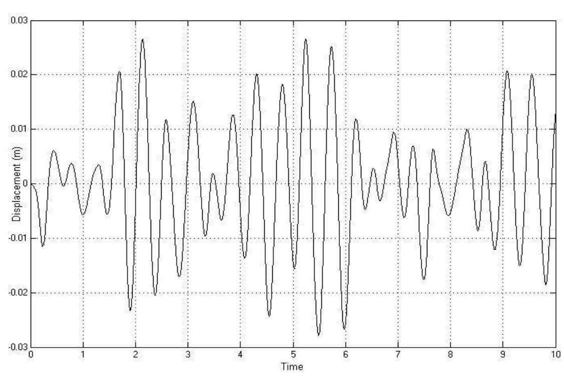

Fig. 8 Response of PID controlled Air Suspension System

7. Conclusion

As can be seen from the above discussion the control provides a considerable amount of reduction in vertical vibration. Comparison passive suspension with the air suspension, the dotted line for the passive suspension in the white noise inputunder vertical displacement response curve, solid line for the air suspension. It can be seen from the figures, when the same random input, vertical displacement change amplitude of air suspension is smaller than the passive suspension. Body vibration reduction will improve vehicle ride comfort performance. Air suspension stiffness is low; vehicles equipped with air suspension can get a lower natural frequency. Then the ride comfort performance is good. It can be extend the life of the vehicle and reduce damage on the road. It can be seen from results of the suspension system dynamic simulation that the suspension system dynamic simulation is an effective way. The method is intuitive and accurate. The first step of the dynamic simulation is to establish a dynamic model of suspension system, and then initialize the system parameters, and finally the dynamic simulation. Vibration contains two kinds of situations: low frequency vibration and high frequency vibration. The low frequency vibration is the wheels of the vibration and the high frequency vibration is the body of vibration. It can be observed the impact of the vehicle ride performance by changing the suspension of the relevant parameters.

8. Nomenclature

Y	:	Vertical Displacement
$\dot{y}$	:	Vertical Velocity
$\ddot{y}(t)$	:	Vertical Acceleration
Z	:	Final Displacement
z_0	:	Initial Displacement
A_1	:	Effective Area of Air Spring
D_1	:	Outer Diameter of Air Spring
D_2	:	Inner Diameter of Air Spring
C_1	:	Damping Coefficient of Air Spring
C_2	:	Damping Coefficient of Tire
N	:	Gas Constant
h_0	:	Air Spring Height
P_0	:	Gauge Pressure
m_1	:	Sprung Mass
m_2	:	Un sprung Mass
f(t)	:	Controlling force
P_a	:	Atmospheric Pressure
f_{Pa}	:	Force due to Atmospheric Pressure
f_c	:	Damping Force
f(t)	:	Controlling Force
f_1	:	Air Spring Force
V_1	:	Volume of Air Spring
K_p	:	Proportional Gain
K_I	:	Integral Gain
K_D	:	Differential Gain
E	:	Error
U	:	Input Signal

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