Modeling, Design and Simulation of Active Suspension System Frequency Response Controller using Automated Tuning Technique

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Abstract
Car suspension system is a mechanism that separates the car from the tire. The objective of the suspension system is to improve a ride quality by ensuring passenger's comfort and good car road handling when the car is subjected to an input excitation. The dynamics of the suspension system was mathematically modeled. The system transfer function determined. The suspension system performance characteristics of overshoot and settling time set at not more than 5% and 5seconds respectively. Frequency response controller was designed by using the MATLAB function sisotool in the automated tuning technique. The result of the simulation indicated that the overshoot and settling time 2.97% and 3.95seconds respectively. The design requirement satisfied.
Keywords: Suspension, active, Frequency response.

1. Introduction

Suspension system can be classified into; passive, semi-active and active suspension system. Traditional suspension consists springs and dampers are referred to as passive suspension, then if the suspension is externally controlled it is known as a semi active or active suspension. An early design for automobile suspension systems was focused on unconstrained optimizations for passive suspension system which indicate the desirability of low suspension stiffness, reduced unsprung mass, and an optimum damping ratio for the best controllability Alleyen and Hedrick, (1995). Thus the passive suspension system, which approach optimal characteristics had offered an attractive choice for a vehicle suspension system and had been widely used for passengers. However, the suspension spring and damper do not provide energy to the suspension system and control only the motion of the car body and wheel by limiting the suspension velocity according to the rate determined by the designer. To overcome the above problem, active suspension systems have been proposed by various researchers. Active suspension systems dynamically respond to changes in the road profile because of their ability to supply energy that can be used to produce relative motion between the body and wheel. Typically, the active suspension systems include sensors to measure suspension variables such as body velocity, suspension displacement, and wheel velocity and wheel and body acceleration Esmailzadeh and Taghirad, (1997). An active suspension is one in which the passive components are augmented by actuators that supply additional forces. These additional forces are determined by a feedback control law using data from sensors attached to the vehicle. The focus of this thesis is on active suspension system controller design. "The process of selecting controller parameters to meet given performance specifications is known as controller tuning" (Ogata, 2002, p. 682). A variety of theoretical approaches have been used to produce PID-tuning formulas for a first-order plant with time delay. A heuristic time-domain analysis (Hang et al., 1991) used set-point weighting to improve Ziegler and Nichols' (1942) original PID-tuning formulas, which were also determined empirically. "Repeated optimizations using a third-order Padé approximation of time delay produced tuning formulas for discrete values of normalized dead time" (Zhuang and Atherton, 1993). Barnes et al., (1993) used open-loop frequency response to design PID controllers by finding the least – squares fit between the desired Nyquist curve and the actual curve. In reviews of the performance and robustness of both PI- and PID-tuning formulas, tuning algorithms optimized for set point change response were found to have a gain margin of around 6 dB, and those that optimized for load disturbance had margins of around 3.5 dB (Ho et al., 1995; Ho et al., 1996). PID-tuning formulas were derived by identifying closed-loop pole positions on the imaginary axis, yielding the system's ultimate gain and period. Dynamics are said to suffer, however, for processes where time delay dominates "due to the existence of many closed-loop poles near the imaginary axis, where the effect of zero addition by the derivative term is insignificant to change the response characteristics" (Mann et al., 2001). Bode diagram or the Nichols chart are used to design the compensator in the frequency domain. The Bode diagram are more advantageous than the Nichols chart since the contribution from the poles and zeros of the compensator becomes additive in the log scale.

2. Mathematical Modelling of Automobile Active Suspension for Quarter Car Model

Designing an automobile suspension system to meet performance specification of ride comforts and good road handling is an interesting and challenging control problem. For ease of design, analysis and simulation, quarter automobile suspension system model is used to simplify the problem. This model represent an active suspension where an actuator is included that is able to generate the required control force to control the automobile dynamics. From the quarter car model, the design can be extended into full car model.

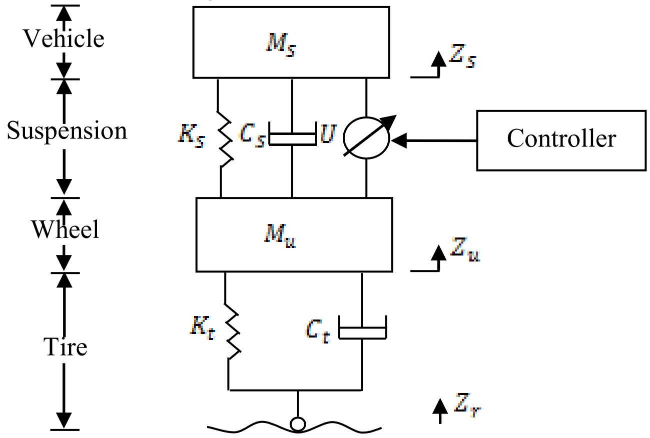

Figure 1. Quarter Car Model

Figure 1 above shows a basic two – degree – of freedom system representing the model of a quarter car, where the quarter mass of the automobile is represented with M_S, referred to as sprung mass in control and dynamics literature. M_U is the mass of the wheel (unsprung mass). The spring constant of the suspension is K_S, K_t is spring constant of the wheel and tire respectively. The damping constant of the suspension system is C_S while C_t is the damping constant of the wheel and tire respectively. U is the control force of the actuator. From the figure 1 above and using the Newton's law, we can obtain the system equations of motion below:

For M_S,

F = Ma,

$$M_S\ddot{Z}_S(t) = C_S[\dot{Z}_u(t) - \dot{Z}_S(t)] + K_S[Z_u(t) - Z_s(t)] + U(t) \tag{1}$$

For M_U,

F = Ma,

$$M_u\ddot{Z}_u(t) = C_S[\dot{Z}_S(t) - \dot{Z}_u(t)] + K_S[Z_S(t) - Z_u(t)] + K_t[Z_r(t) - Z_u(t)] + C_t[\dot{Z}_r(t) - \dot{Z}_u(t)] + U(t) \tag{2}$$

where

$Z_S - Z_U$ = Suspension travel

$\dot{Z}_S$ = Car body Velocity

$\ddot{Z}_S$ = Car Body Acceleration

$Z_U - Z_r$ = Wheel Deflection

$\dot{Z}_U$ = Wheel Velocity

Z_U = Wheel Acceleration

In control theory, the transfer function of a system is defined in terms of an output to input ratio, but the use of a transfer function in system dynamics and vibration testing implies certain physical properties, depending on whether position, velocity, or acceleration is considered as the response (output). From equation (1) and (2); taking Z_S and Z_U as output variables and taking U and Z_r as input variable respectively. The system transfer function is shown below.

$$G_1(s) = \frac{Z_S(s) - Z_u(s)}{U(s)} = \frac{(M_S + M_u)S^2 - C_tS + K_t}{\Delta} \tag{3}$$

$$G_2(s) = \frac{Z_S(s) - Z_u(s)}{Z_r(s)} = \frac{M_S C_t S^3 - M_u K_t S^2}{\Delta} \tag{4}$$

where

$$\Delta = (M_S S^2 + C_S S + K_S)\cdot(M_u S^2 + (C_S + C_t)S + (K_S + K_t) - (C_S S + K_S)\cdot \quad (C_S S + K_S) \tag{5}$$

The quarter – vehicle model parameter are listed in Table 1, (Du *et al.,* 2008) for the following controller design.

Table1. Parameter for Quarter Car Model

M_s	M_u	K_s	K_t	C_s	C_t
973 kg	114kg	42720N/m	101115N/m	1095Ns/m	14.6Ns/m

3. Design Requirements

The key suspension performance requirements are that the automobile should have a satisfactory road holding ability and still able to provide ride comfort when riding over undulating road surfaces, that is if the vehicle experience road disturbance such as falling into pot holes, uneven pavement and cracks, the oscillation that follows should not be too large and must dissipate quickly.

The focus of this project is to design a controller that will control the oscillation such that the output response overshoot will be less than 5% with a settling time shorter than 5seconds such that, when the automobile runs into a pothole of say 100cm (0.1m) the automobile body will oscillate within $\pm$ 5mm and becomes steady in less than 5seconds.

4. Frequency Response Controller Design

The main idea of frequency – based design is to use the Bode plot of the open – loop transfer function to estimate the closed – loop response. Adding a controller to the system changes the open – loop Bode plot so that the closed – loop response will also change.

The MATLAB function “sisotool” uses a Graphical User Interface (GUI) window where the Control Architecture and system data are selected and imported respectively then the graphical tuning plot (Open – loop Bode plot) for tuning/adjusting the value of the compensator and parameters of the system, an analysis plot (Step Response plot) for displaying the characteristics of the system. Integrator, lead network and notch filter are added to the compensator. Moving the locations of the notch filter (notch zeros and notch poles), lead network (lead zero and lead pole), and adjusting the gain of the compensator results in achieving the design specification.

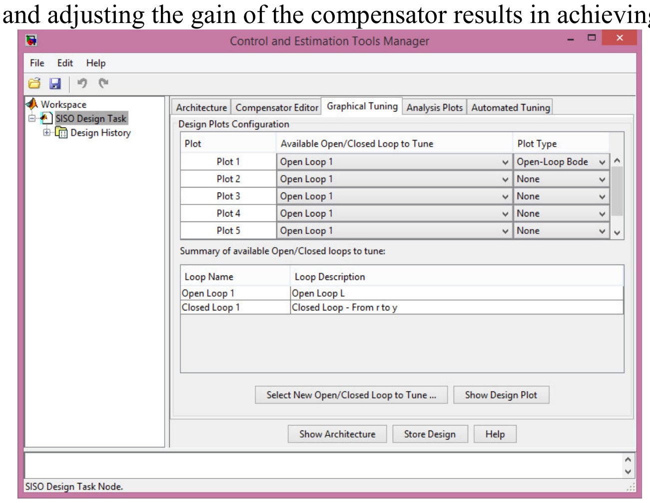

Figure 2. Graphical Tuning tab

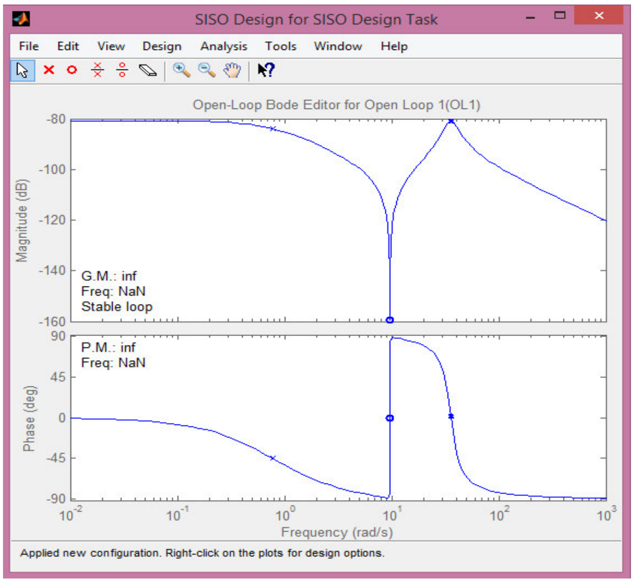

Figure 3. Graphical tuning plot (Tunable Open – Loop Bode plot)

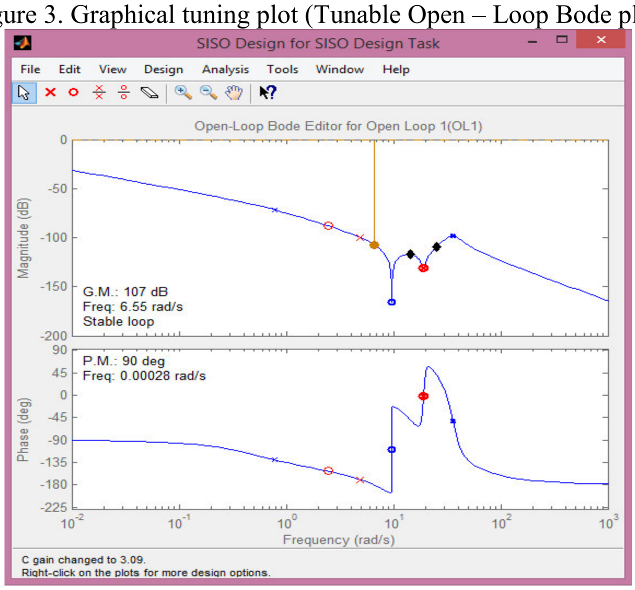

Figure 4. Open – Loop Bode plot further adjusting the gain of the compensator and moving the location of the Lead Pole and Lead Zero

5. Results and discussion

Table 2. Closed – Loop Step response plot characteristics with 0.1-m high step.

Properties	Values
Peak amplitude	-0.419
Overshoot	2.97%
Rise time	>6.99seconds
Settling time	3.95seconds
Final value	1.44e-12

The overshoot and the settling time meets the design requirement i.e. 2.97% and 3.95seconds which is less than 5% and 5seconds respectively

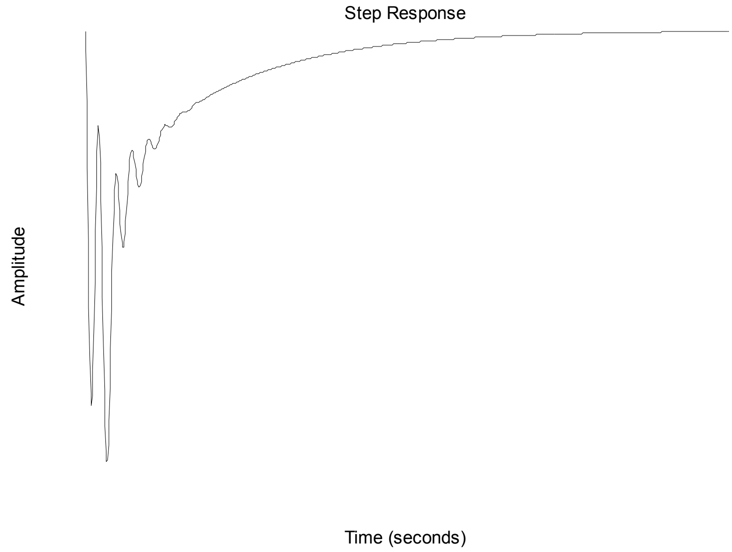

Figure 3. Closed – Loop Response to 0.1-m High Step

The simulation result shown in the above table are values that satisfied the design requirements of overshoot and settling time.

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