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Modelling and PID Control of Quarter Car Suspension System

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Abstract:

The main functions of an automotive suspension system are to provide vehicle support, stability and directional control during handling maneuvers and to provide effective isolation from road disturbance. These different tasks result in conflicting design requirements. Directional control and stability requires a suspension that is neither very stiff nor very soft. Insensitivity to external loads requires a stiff suspension, whereas good ride comfort demands a soft suspension. In a conventional passive suspension system, the designer is faced with the problem of choosing the suspension stiffness and damping parameters, which inevitably involves a difficult compromise in view of the wide range of conditions over which a vehicle operates. This paper presents mathematical modelling and active control of a linear quarter car suspension system. A 2 DoF nonlinear quarter car model is developed in Matlab/Simulink® environment and implemented for control applications. PID control strategy is modelled and implemented for ride comfort applications. It is observed that PID controller significantly reduces RMS acceleration as compared to the passive suspension system. Result shows that PID control performs better as compared to the passive system. Thus one can say that PID controlled system provides better ride comfort.

Keywords **— PID, Active Control, Passive System.**

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I. INTRODUCTION

The main functions of an automotive suspension system are to provide vehicle support, stability and directional control during handling maneuvers and to provide effective isolation from road disturbance. These different tasks result in conflicting design requirements. Directional control and stability requires a suspension that is neither very stiff nor very soft. Insensitivity to external loads requires a stiff suspension, whereas good ride comfort demands a soft suspension.

The conventional system i.e. passive suspension system, which comes as is, is a system of springs, shock absorbers, bushings, rods, linkages and arms. Vehicles generally have two suspension systems – one for the front wheel and other for rear. These two systems work together to control driving and breaking forces to provide smooth ride for driver and passengers.

The primary thrust of the commercial research and development in active suspension has been to improve ride, handling and stability in the on-road environment while minimizing system cost and mean time between failures. Much of this work was pioneered by Lotus Engineering, located in Norwich, England and has been continued by several major automotive manufacturers, including U.S. based Ford and GM.

With recent advances in microelectronics and actuators, there has been an upsurge in the concept of active suspension control. Active suspension offers the potential of being adapted to the quality of the road surface, vehicle speed and different safety and comfort requirements, with the choice being selected either by the driver or by an adaptive control algorithm embodied in a microcontroller.

Metered and ELsawaf [1] had implemented particle swarm optimization (PSO) algorithm to tune the PID controller implemented on a semi-active quarter car model suspension system. A 2DoF vehicle model with MR damper is simulated in Matlab/Simulink environment. The PSO tuned PID controller is compared with conventional PID controller tuned using Ziegler-Nicholas method, passive suspension system and uncontrolled MR damper. Bump and random road inputs were used to test the system in time and frequency domain. It was observed that POS tuned PID controller improves ride comfort and vehicle stability.

Kesarkar and Selvaganesan [2] designed fractional order PID controller using artificial bee colony algorithm with objective functions such as integral absolute error, integral square error and integral time absolute error implemented to a multi-modal complex optimization problem. Author observed that the the results were promising as compared to the conventional PID method.

Nui [3] had implemented GA based optimization method to tune PID parameters of active suspension system. Absolute error control is used as objective function to tune the PID parameters. GA based optimized PID controller improves the dynamic performance of a active suspension system and improves stability.

Hamid and Hamid [4] analysed a fuzzy based PID controller for a half car active suspension system. In this analysis, suspension working space is the criterion under observation. Active control system using fuzzy logic, fuzzy PID an∞d aHre implemented and studied. It was observed that PID controller has lowest overshoot, mean square error and improved comfort and safety as compared to other control strategies.

Tammam, Aboelela, Moustafa and Seif [5] implemented Multi-objective GA based PID controller to control load frequency of a single area power system. It is observed that GA based PID controller is simple and easy to implement and improves system performance effectively.

Gad, Metered, Bassuiny and Ghany [6] implemented fractional order PID controller to a semi- active seat suspension. PID parameters are tuned using multi-objective GA for a seat suspension system using 6-DoF human body. It is observed that GA based PID controller improved objective function such as SEAT value, VDV ratio and crest factor as compared to passive system and classical PID controller. Results are obtained in time as well as frequency domain.

Hung-Cheng [8] studied optimal fuzzy based PID controller implemented to active magnetic bearing. Authors have studied GA to optimize fuzzy PID controller. The proposed algorithm is better in convergence of speed and stability, thus shows overall good performance.

Roshdy, Yosra and Mohamed [9] studied performance of PV system based on GA optimized PID controller. Here, PID controller is implemented to enhance PV system output with minimum overshoot and minimum rise time in output voltage with better response.

Hung-Cheng and Sheng-Hsiung [10] studied GA based PID tuning for active magnetic bearing. The proposed PID controller shows that the active magnetic bearing had good static and dynamic performance and showed better performance and effectiveness.

This paper presents mathematical modeling of linear quarter car model. PID control is implemented by various researchers with control objective to minimize frequency weighted RMS sprung mass acceleration (hereafter called as RMS sprung mass acceleration), RMS suspension space requirement, tyre dynamic force and road holding along with RMS optimal control force. The constraints during optimization are maximum control force, RMS sprung mass acceleration, maximum sprung mass acceleration, maximum suspension space requirement, maximum tyre deflection and maximum unsprung mass displacement.

II. SYSTEM MODELING

The system mathematical model is a part of descriptive and functional model category and usually used in engineering. Nowadays, one of the great interests in the vehicle research field is improving the car ride quality and handling performance. These performance aspects can be analysed and improved with design and analyses of the car suspension system. The suspension system designs and analysis is done based on the suspension mathematical model. The model must capture realistic dynamic behaviour of the car passive suspension system.

Modeling of suspension system is done in the vertical plane. Longitudinal or transverse deflection of the suspension components is considered negligible in comparison to vertical deflections. The complete vehicle mass is divided into two masses i.e the sprung mass and the unsprung mass respectively. Springs and dampers are connected between the sprung and unsprung masses and unsprung mass and ground respectively.

B: Quarter Car Model

A linear 2-DOF system is used as a model for the road vehicle. The two masses mus and ms, of the vehicle model represent the wheel (and axle if there is any) and the vehicle body (often called the unsprung and the sprung mass, respectively). These masses (representing one half or a quarter of a real car). The spring stiffness are ks, kt and the damper stiffness is cs.

To analyze the effectiveness of a two degree-of-freedom isolation system we consider the base supporting the two degree-of-freedom system to be a moving base as shown in Fig.1. In analyses of such systems, one usually assumes that the masses are initially at rest and that there are no applied forces directly on the inertial elements and xr(t) is given. Eqs. to account for the moving base become,

$$m_s\ddot{x}_s = k_s(x_{us} - x_s) + c_s(\dot{x}_{us} - \dot{x}_s)$$

$$m_{us}\,\ddot{x}_{us} = -k(x_{us} - x_s) - c_s(\dot{x}_{us} - \dot{x}_s) - k_t\,(x_{us} - x_r) \qquad (1)$$

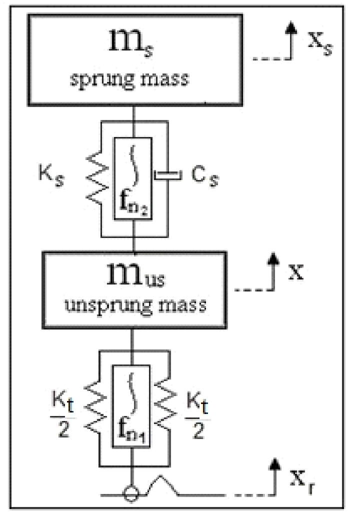

Fig. 1: Quarter Car Model

Quarter Car Parameters are –

m_1 =40 kg, c_1=414 Ns/m, k_1=124660 N/m

m_2=243 kg, c_2=370 Ns/m, k_2=14671 N/m

Modelling of automotive suspension is of great interest for automotive and vibration engineers. Vehicles ride quality is prime concern for the engineers when a vehicle passes over the speed bump. For our analysis 2 DOF quarter car model has been developed. Refer Fig. 3.3.

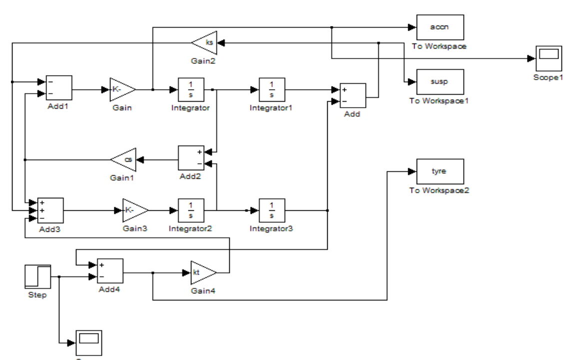

Fig. 3: Simulink Model – Quarter Car

C. PID controller

PID controller involves P+I+D control actions. This is the most powerful but complex mode ofcontrol. This control action can be used for virtually any process condition. PID controller is presented by equation (2). Refer Fig. 4.

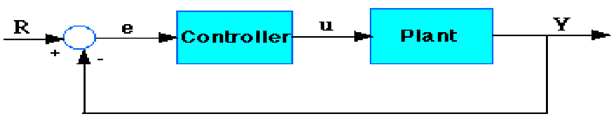

Fig. 4 : Close Loop system

The PID controller is a three-term controller:

$$K_p + \frac{K_I}{s} + K_D s = \frac{K_D s^2 + K_p s + K_I}{s} \tag{2}$$

The variable (e) represents the tracking error, the difference between the desired input value (R) and the actual output (Y).

The signal (u) will be sent to the plant, and the new output (Y) will be obtained. This new output (Y) 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 its integral again. This process goes on and on. Refer equation (3).

$$u = K_p e + K_I \int e dt + K_D \frac{de}{dt} \tag{3}$$

III. RESULTS AND DISCUSIIONS

It is observed that, RMS acceleration for PID controller is 0.6565 m/s^2, which is a little uncomfortable. In PID controlled system, RMS acceleration is reduced by 30% (passive suspension system has RMS acceleration 0.9322 m/s^2 which is uncomfortable) Also RMS suspension space and RMS tyre deflection is less as compared to the passive suspension system. The results of RMS optimal control force, RMS sprung mass acceleration, RMS suspension space, dynamic tyre force, road holding, maximum sprung mass acceleration, are tabulated in Table 1 and refer Fig. 5 for time domain results.

TABLE I
RESULTS – PID CONTROLLER

Parameter	PID Control	Passive System
Control Force (N)	36.0332	--
RMS Acceleration (m/s^2)	0.6565	0.9322
RMS Suspension Space Deflection (m)	0.0033	0.009311
RMS Tyre Deflection (m)	0.0024	0.004473
Max Acceleration (m/s^2)	2.3003	3.1672
Max Suspension Space Deflection (m)	0.0115	0.0305
Max Tyre Deflection (m)	0.0101	0.0251

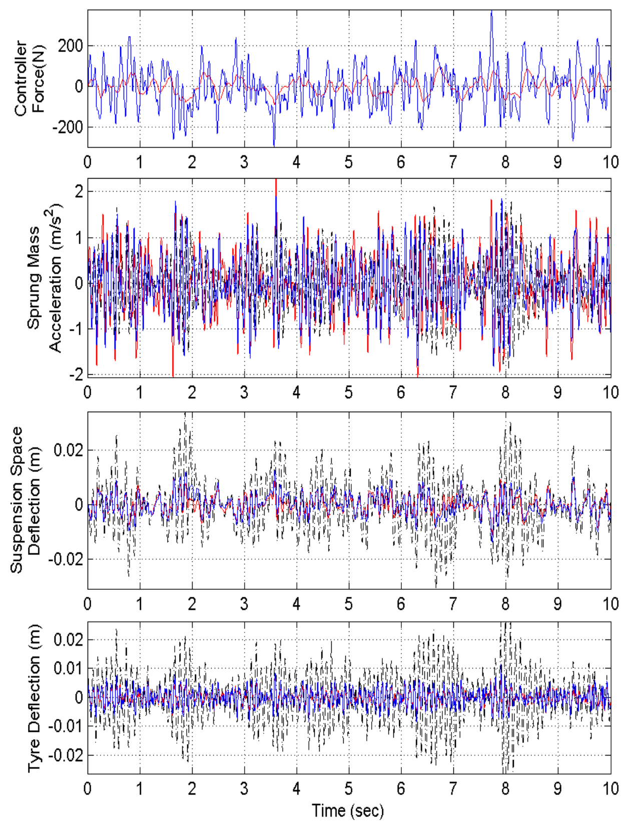

Fig. 5: PID Control Results – Time domain.

IV. CONCLUSIONS

The A PID control active control system is simulated in Matlab/Simulink environment. It is observed that RMS sprung mass acceleration is very less for PID control system as compared to the passive suspension system. It is observed that PID control system reduces RMS sprung mass acceleration by 30% as compared to the passive suspension system. Thus PID controlled suspension system enhances ride as compared to the passive suspension system.

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