International Journal of Natural and Engineering Sciences
E-ISSN: 2146-0086 13 (2): 49-52, 2019

Suppression of Pitch Angle Vibrations of a 5-DOF Half Vehicle Model by System Identification Method

Levent MALGACA [1*] Şefika İpek LÖK [2] Mehmet UYAR [1]

[1]Department of Mechanical Engineering, Faculty of Engineering, Dokuz Eylül University, Izmir, Turkey
[2]Department of Mechatronics Engineering, The Graduate School of Natural and Applied Sciences, Dokuz Eylül University, Izmir, Turkey

*Corresponding Author
E-posta: levent.malgaca@deu.edu.tr

Received: 03 September 2019
Accepted: 14 November 2019

Abstract
A vehicle's suspension system helps to enhance the driver comfort, road handling, ride quality and steering stability. It is desirable to limit the angular displacements of the vehicles for comfort design. Pitch angle is an angular displacement of the body between the front and rear tyres. In this work, the active control of a 5-degree of freedom (DOF) half vehicle (HV) model with a driver is studied to reduce the pitch angle. A force based actuator between the body and front tyre together with a driver mass is considered. The mathematical model of the system is obtained by using the system identification (SI) method. Inputs of the system are the road profiles for the front and rear tyres and actuator. The output of the system is the pitch angle. Finite element vibration analyses are performed to obtain the output response to the given inputs. The transfer functions of the system are found by utilizing the inputs and output with the parametric SI models. Then, the closed-loop control is applied to the system to control the pitch angle. The values of the peak and settling time of the pitch angle response are successfully reduced with various PID gains.

Keyword: Active control, Half vehicle, Pitch angle, System identification.

INTRODUCTION

The suspension systems are aimed to improve the stability, road handling and ride comfort of the vehicles. There are three types of suspension system; passive, semi-active and active suspension systems. The passive suspension systems consist of the conventional springs and dampers. The semi-active systems consist of the conventional springs and controlled damper. The damping coefficients can be changed according to the vibration level of the system. In the active suspension system, a force actuator is used to suppress the vibration signals and a closed-loop control system is defined [1]. Literature is reviewed for the quarter/half/full vehicle models [2]. H2 & H∞ control methods are studied for the active suspension system [3]. The various control methods are used to improve the performance characteristics of the half vehicle such as the linear quadratic regulator [4], sliding mode control [5], fuzzy logic control [6], optimal control [7].

Ahmed et. Al (2015) designed an active suspension system for a 2-DOF quarter vehicle using a PID controller and compared the passive and active suspension systems [8]. Gandhi et. Al (2017) worked on the control of the half-car suspension system. They used and compared four different controllers; PID, LQR, FUZZY, and ANFIS [9].

The system identification (SI) is a method that estimating system parameters using input and output signals [10]. In the literature, the SI method is used to obtain the experimental and theoretical dynamic models of different mechanical systems. Sethi and Song (2008) used parametric and nonparametric SI methods to modeling a flexible beam, and then, designed closed-loop control with a linear pole placement controller [11, 12]. Saad et. al (2013) studied the identification and active vibration control of a flexible beam with the PID controller. ARX model from parametric SI models was used for the identification [13].

In this study, a 5-DOF HV model with a driver is considered. The SI method is used for the modeling of the HV. Input and output signals for the SI method are obtained by using ANSYS. The closed-loop control is performed with the PID controller. Then, the uncontrolled and controlled responses are obtained by using MATLAB.

MODELLING

The HV model considered in the study is shown in Fig. 1. The model of the HV is defined as 5-DOF.

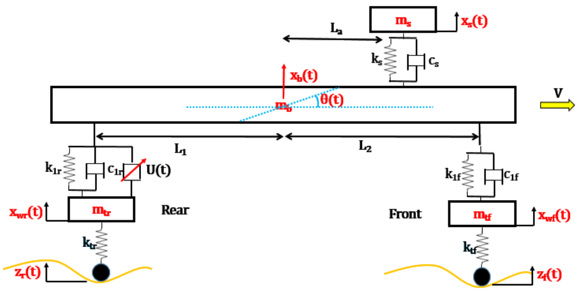


Figure1. 5-DOF Half vehicle model

The parameters m_s, m_b, m_{tf} and m_{tr} are the driver mass, body mass, front and rear tyre masses, respectively. k_s, c_s, k_{1f}, c_{1f}, k_{1r}, c_{1r}, k_{tf} and k_{tr} are the chair stiffness and damping, front suspension stiffness and damping, rear suspension stiffness and damping, front and rear tyre stiffness, respectively. Typical design parameters are taken from the reference study [14]. The numerical values of parameters are given in Table 1 below.

Table 1. The numerical values of parameters [14]

Parameters	Values [Units]
m_s, m_b	80, 505.1 [kg]
m_{tf}, m_{tr}	28.58, 28.58 [kg]
k_s, c_s	15000, 150 [N/m, N.s/m]
k_{1f}, c_{1f}	15000, 1828 [N/m, N.s/m]
k_{1r}, c_{1r}	15000, 1828 [N/m, N.s/m]
k_{tr}, k_{tf}	32000, 32000 [N/m, N/m]
I_{zz}	651 [kg.m^2]
L_1, L_2, L_a	1.098, 1.468, 0.70 [m]

$x_s(t)$, $x_b(t)$, $x_{wf}(t)$, $x_{wr}(t)$ are the displacements of driver mass, body masses, front and rear tyre masses, respectively. $\theta(t)$ is the pitch angle of the HV. $z_f(t)$ and $z_r(t)$ are the front and rear ground motions. U(t) is the actuator force. L_a is the distance between the driver and vehicle mass center. L_1 and L_2 are the distances from the vehicle mass center to the front and rear tyre axles, respectively. I_{zz} is the body mass moment of the inertia.

In this study, the HV shown in Fig. 2 is modelled as a lumped parameter system in ANSYS [15, 16]. The finite element (FE) model is created by using Mass21, Mpc184 and Combin14 elements. The system has 11 nodes. Mass21 elements are used for the driver, body and tyre masses while Combin14 elements are used for the stiffness and damping parts. Mpc184 is used in order to define the body mass as a rigid link.

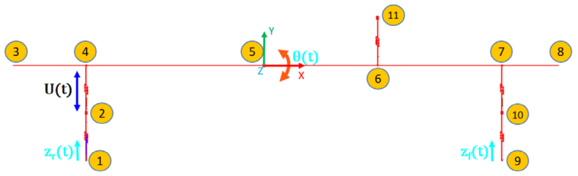


Figure 2. FE model

SYSTEM IDENTIFICATION

The SI method aims to directly describe the mathematical models of the system using the measurements of the system's input and output signals. The process of the SI method requires the system modelling, model selection, estimation and validation of system. Firstly, the input signals are defined to the FE model as the step input occurring from the road profile at the front and rear ground motions. U(t) is defined as the input signal between the rear tyre and body masses. The output of the system is chosen as $\theta(t)$. Then, the ARX model is selected to estimate the system. The equation of the ARX model is given as follow [8,17].

$$y_i(t)=\frac{B(q)}{A(q)}u_i(t)+\frac{1}{A(q)}e_i(t) \tag{1}$$

where, $y_i(t)$ is the output and $u_i(t)$ is the input of the system. $e_i(t)$ is a white noise error term. A(q), B(q) are the numerator and denominator polynomial of the transfer function, respectively.

Then, the orders of numerator and denominator for estimation of system are determined. Finally, the model validation is performed whether the obtained result is suitable.

The system has three input and an output signals $z_f(t)$, $z_r(t)$, U(t) and $\theta(t)$. The transfer functions of the system are given in Eq. 2.

$$\{\theta(s)\} = [G_{11}(s) \quad G_{12}(s) \quad G_{13}(s)]\begin{Bmatrix} Z_f(s) \\ Z_r(s) \\ U(s) \end{Bmatrix} \tag{2}$$

where, the transfer functions of the system $G_{11}(s)$, $G_{12}(s)$ and $G_{13}(s)$ represent the relationship from $Z_f(s)$ to $\theta(s)$, $Z_r(s)$ to $\theta(s)$ and U(s) to $\theta(s)$, respectively. $Z_r(s)$ is applied to $G_{12}(s)$ with a time delay t_d. t_d is calculated as given in Eq. 3. V is the velocity of the HV and taken as 50 km/h.

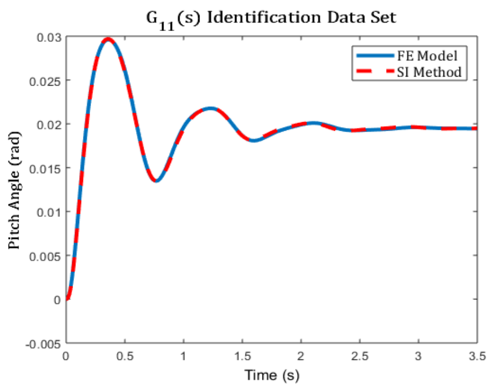
$$t_d=(L_1+L_2)*3.6/V \tag{3}$$

Two different data sets as the identification and validation data are used to model the system. After the transfer functions are estimated with the identification data set, the input signals are applied to the transfer functions and the output signal is compared with the validation data set.

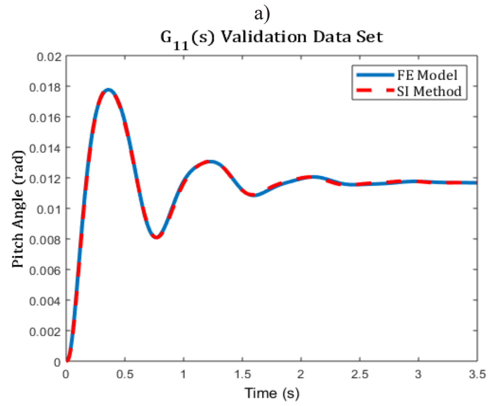
The success rates for $G_{11}(s)$, $G_{12}(s)$ and $G_{13}(s)$ are found as 97.44%, 97.42% and 96.86%, respectively. The transfer functions of the system are successfully obtained by using the SI method. The transfer functions are obtained and verified with the validation data set. For example, $G_{13}(s)$ is given as follow

$$G_{13}(s)=\frac{2.593s^7+13.62s^6+3476s^5+1.613x10^4s^4+7.254x10^5s^3+1.887x10^6s^2+2.099x10^7s+9.776x10^5}{s^{10}+1015s^9+8.49x10^4s^8+2.61x10^6s^7+4.52x10^7s^6+7.65x10^8s^5+5.83x10^9s^4+5.13x10^{10}s^3+1.7x10^{11}s^2+8.12x10^{11}s+3.763x10^{10}} \tag{4}$$

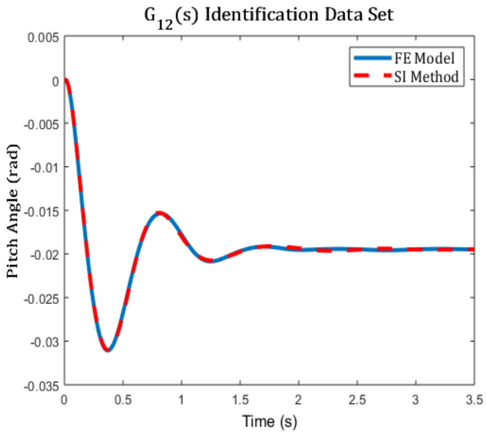
Results for the ARX model in the SI method are shown in Fig. 3.



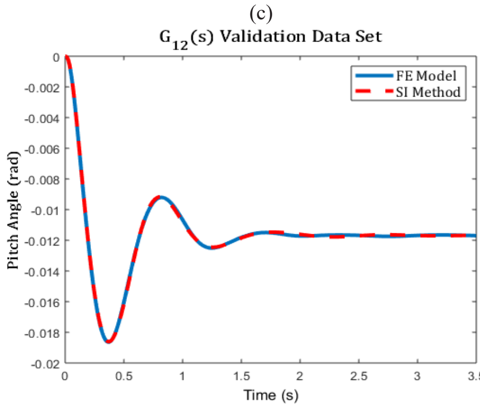
a)



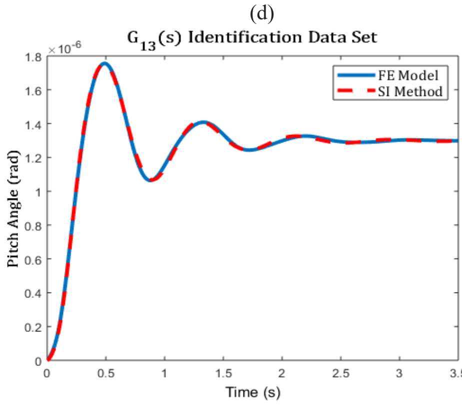
(b)



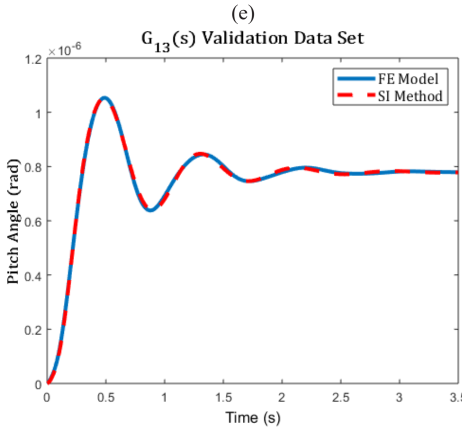
(c)



(d)



(e)



(f)

Figure 3. Identification (a, c, e) and validation data sets (b, d, f) for the ARX model

VIBRATION CONTROL

The natural frequencies of the system are important to perform the transient analysis. Modal analysis is done and the undamped natural frequencies are given in Table 2.

Table 2. Natural frequencies of the system

Order of frequency	Frequencies [Hz]
f_{n1}	0.917
f_{n2}	1.103
f_{n3}	2.456
f_{n4}	6.489
f_{n5}	6.507

The time step Δt and settling time t_{ss} for the transient analysis depends on the natural frequencies. Hence, the time step and settling time are determined as $\Delta t=1/f_{n1}/20=0.0545$s by considering the first vibration mode and $t_{ss}=3.40$ s, respectively [18].

After obtaining transfer functions, the closed-loop control of the system is studied in MATLAB. The block diagram for the closed-loop control is shown in Fig. 4.

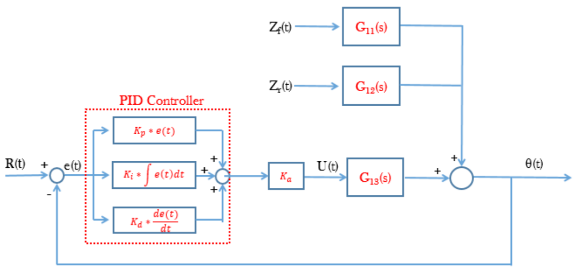


Figure 4. Closed-loop block diagram

R(t) is the reference input taken as zero to eliminate the vibrations. $Z_f(s)$ and $Z_r(s)$ are the disturbance inputs occurring from the ground motions. θ(t) is the feedback signal. The PID controller is used and K_p, K_i, K_d are proportional, integral, and derivative controller gains, respectively. K_a is the actuator gain. The value of K_a is chosen as 100.

The error value e(t) is calculated by subtracting the reference value from the feedback signal. The error value is the input of the controller. The actuating signal is generated by multiplying the controller output and K_a. Then, the actuating signal is applied to $G_{13}(s)$ to suppress the vibration of θ(t). $Z_f(s)$ and $Z_r(s)$ are applied to $G_{11}(s)$ and $G_{12}(s)$, respectively. The disturbance signal is obtained by summing signals occurring from $Z_f(s)$ and $Z_r(s)$. The output signal is calculated by summing U(s), $Z_f(s)$ and $Z_r(s)$. The uncontrolled and controlled responses for the step input of 0.05 m are obtained and shown in Fig. 5.

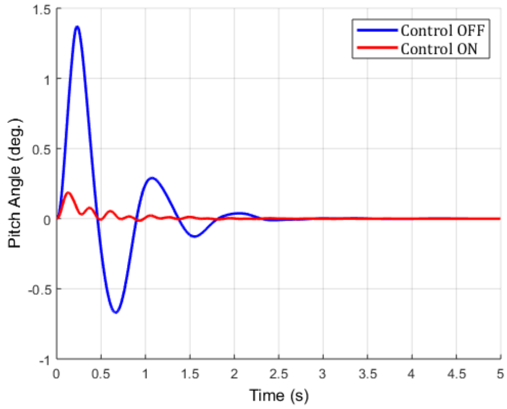


Figure 5. Uncontrolled and controlled responses with controller gains, K_p=125000, K_i=1250, K_d=62500.

A comparison of the uncontrolled and controlled responses for θ(t) is shown in Fig. 5. As seen from Fig. 5, the peak value (M_p) is reduced from 1.372 degree to 0.187 degree.

Various controller gains are studied. Controller gains are determined by the trial and error method. The performance of the controller can be evaluated with M_p and t_{ss}. The values of t_{ss} are calculated for error band of 5%. Numerical results are listed in Table 3.

Table 3. Numerical results for the performance of the controller

Control Gains			M_p(deg.)	t_{ss}(s)
K_p	K_i	K_d		
0	0	0	1.372	3.40
100	0	0	1.296	4.09
700	5	0	0.988	24.78
1250	12.5	100	0.532	3.81
1250	**12.5**	**625**	**0.187**	**3.35**
1250	12.5	825	0.149	3.79

Firstly, the P control is applied to the system. It is observed from Table 3 that t_{ss} increases and M_p decreases. Then, the PI control is applied for the active control. The same result as the P control occurs. Finally, the D control is added for decreasing t_{ss} and M_p.

CONCLUSIONS

In this study, the active control of θ(t) of the HV is studied. The SI method is used to obtain the transfer functions. The FE model of the system is created in ANSYS and modal analysis is done to find the undamped natural frequencies. The input signals are defined as the disturbance signals occurring the front and rear tyres ground motions due to the road profile and control force. The output signal is chosen as the pitch angle of the HV. ARX model is used to obtain the transfer functions. The active control of the pitch angle is performed. The uncontrolled and controlled time responses are presented.

REFERENCES

[1] A. R. Bhise, R. G. Desai, R. N. Yerrawar, A.C. Mitra, R. R. Arakerimath, Comparison Between Passive And Semi-Active Suspension System Using Matlab/Simulink, IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)

[2] N. M. Ghazaly, A. O. Moaaz. The future development and analysis of vehicle active suspension system, IOSR Journal of Mechanical and Civil Engineering, 11 (2014) 19-25.

[3] F. Jamshidi and A. Shaabany. Robust control of an active suspension system using h2 control methods. Journal of American Science, 7 (2011) 1-5.

[4] M. S. Yahaya and M. R. A. Ghani, Lqr controller for active car suspension. In Proceeding in TENCON, September 2000.

[5] I. Z. M. Darus, E. Akbari, M. Farsadi, R. Ghelichi. Observer design for active suspension system using sliding mode control. In Proceedings of IEEE Student Conference on Research and Development, December 2010.

[6] M.Hady H. Elbab, E. Allam, S. Abouel-Seoud. Performance of active suspension with fuzzy control. Technical report, SAE Technical Paper, May 2009.

[7] H. Zohoor J. Marzbanrad, Y. Hojjat, S.K. Nikravesh. Optimal preview control design of an active suspension based on a full car model. Scientia Iranica, 10 (2003) 23-36.

[8] A. E. S. Ahmed, A. S. Ali, N. M. Ghazaly, G. T. A. Jaber, PID controller of active suspension system for a quarter car model, International Journal of Advances in Engineering & Technology, Dec., 2015.

[9] P. Gandhia, S. Adarsh, K I Ramachandran, Performance Analysis of Half Car Suspension Model with 4 DOF using PID, LQR, FUZZY and ANFIS Controllers, 7th International Conference on Advances in Computing & Communications, ICACC-2017, 22-24 August 2017, Cochin, India.

[10] L. Ljung, System Identification: Theory for the User, 2nd Ed., Prentice-Hall, 1999.

[11] V. Sethı, G. Song, Multimodal vibration control of a flexible structure using piezoceramic sensor and actuator. Journal of Intelligent Material Systems and Structures, 19 (2008) 573–582.

[12] Sethı V., Song G. Multimodal vibration control of a flexible structure using piezoceramics. Proceedings of the 2005 IEEE/ASME International Conference on Advanced Intelligent Mechatronics Monterey, California, USA, 24-28 July, 2005.

[13] Saad M. S., Jamaluddin H.and Darus I. Z. M., Active vibration control of a flexible beam using system identification and controller tuning by evolutionary algorithm , Journal of Vibration and Control, vol. 21, issue 10, pp. 2027-2042, 2016 .

[14] W. Abbas, A. Emam, S. Badran, M. Shebl, O. Abouelatta, Optimal Seat and Suspension Design for a Half-Car with Driver Model Using Genetic Algorithm, Intelligent Control and Automation, vol:2013(4), pp:199-205, 2013.

[15] L. Malgaca, M. Uyar, Active Vibration Suppression of a Single-link Smart Flexible Manipulator, International Journal of Natural and Engineering Sciences (IJNES), vol:11(1),pp:13-19, 2017.

[16] ANSYS Software, 2018, ANSYS Inc. http://www.ansys.com (accessed: 01.10.2018).

[17] K. J. Keesman, System Identification an Introduction, Springer Publications, London, 2011.

[18] Newmark NM (1959) A method of computation for structural dynamics. Journal of Engineering Mechanics, ASCE 85: 67–94.