



MARMARA UNIVERSITY
FACULTY OF ENGINEERING



OPTIMAL CONTROL OF ELECTROMAGNETIC VEHICLE SUSPENSION SYSTEMS

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FACULTY OF ENGINEERING**



Optimal Control Of Electromagnetic Vehicle Suspension Systems

by

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YASİN GÜNERHAN, KERİM AYDIN

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ABSTRACT

OPTIMAL CONTROL OF ELECTROMAGNETIC VEHICLE SUSPENSION SYSTEMS

In most vehicles, suspension systems are hydraulic or pneumatic. However, in all developed countries people are encouraged to switch to electric vehicles to reduce carbon emissions. Luxury vehicles and electrical vehicles have an electromagnetic suspension system that changes the characteristics of the suspension according to road conditions. With the movement of the suspension system, both electricity is produced and some equipment such as hydraulic pumps is not needed.

In order to make this project, we have learned the working principles of active, passive and semi-active suspension systems through various books and articles. We modeled the electromagnetic suspension system using Matlab Simulink.

In addition to this, we have added dynamic output feedback control system.

SYMBOLS

ω : Angular velocity
 \dot{z} : Stroke speed
 J_m : Moment of inertia of motor rotor
 J_b : Moment of inertia of ball screw
 l : Lead of the ball screw
 I_d : Equivalent inertial inertia
 k_t : Torque constant
 Φ : Motor constant

i : Current
 V : Variable Voltage of Power Supply
 L : Inductance
 R : Resistance
 k_e : Induced Voltage Constant (k_t)

m_s : Sprung Mass
 m_r : Mass of rod (a component of EMD)
 m_u : Unsprung mass
 I_d : Equivalent inertial inertia
 k_1 : Spring constant of tire
 k_2 : Spring constant of suspension
 k_3 : Spring constant of mount
 c_3 : Damping coefficient of mount
 x_s : Displacement of sprung mass
 x_r : Displacement of rod
 x_u : Displacement of unsprung mass
 x_0 : Displacement of road profile
 u_{ref} : required output force

ABBREVIATIONS

DOF : degree of freedom
EMD : electromagnetic damper
EMS : electromagnetic suspension
DC : direct current
 L : Laplace transform
 s : Laplacian operator
DOF : Dynamic Output Feedback

1)INTRODUCTION

The importance of electromagnetic suspension systems is increasing day by day. That's why our job is to design a controller for electromagnetic suspension systems. We have reached a certain stage in design through Matlab/Simulink. We held regular meetings with our lecturer many times. Our lecturer sent us various articles and books. We determined the strategy we will follow in the first meeting. First of all, we learned about active, semi-active and passive suspension systems. Then, we did the state space model, transfer function and simulation for the quarter car model. We designed the passive suspension system, active suspension system, dynamic output feedback system for bump road model and random road model and got the desired results.

Our aim is to reduce the peak of the resonance frequencies of the wheel and the car formed in the bode plot, that is, to keep the oscillation of the vehicle and wheel to a minimum. Our simulation will become increasingly complicated.

2)Background, Modeling and Problem Statements of Active Suspensions

This section shows the background information on suspension systems. We mainly introduce three categories of suspension systems: two degrees of freedom (DOF) quarter car model, four DOF half car model and seven DOF full vehicle models with mathematical models expressed. The path excitement model is also given in time domain and spectrum form. In addition, the control objectives of active suspension systems are shown.

Suspension systems transmit all forces between the vehicle body and the road and thus mainly determine the driving comfort, handling and driving safety. Driving abilities are significantly affected by the dynamic behavior of the suspension system, so the performance increase of suspension systems does not have a positive effect on driver comfort but prevent the driver's physical fatigue and reduce the number of deaths in traffic accidents. Roughly speaking, vehicle suspensions can be divided into three types: passive, semi-active and active suspensions. Passive suspension systems consist of springs and shock absorbers placed between the vehicle body and the wheel axle assembly. Passive suspensions have the advantages of simple mechanism, easy application, and high reliability, but they are inadequate to improve ride comfort or handling due to the unchanging spring and damper characteristics inability to cope with different road conditions and conflicting criteria.

2.1)Control objectives of Active Suspension systems

The main function of the vehicle suspension is to connect the vehicle body to the wheels. Thus, it is possible to carry the body along the drive path and transmit the forces in the horizontal plane. The suspension gives the wheel a vertically aligned primary movement. As a result, the wheel will follow a route with uneven road surfaces to a certain extent. By using spring and damping elements, the resulting body movements are reduced and driving safety and comfort is provided.

In addition, vehicle suspension affects the geometry of the wheel and its position relative to the road with its spring and damping speed. This allows a systematic effect on the vehicle's dynamic driving characteristics. The adjustment of these features provides a compromise, because the requirements for good driving behavior and high comfort are most inconsistent with each other. Therefore, when designing the law of control for a suspension system, we usually need to consider the following aspects:

- Driving comfort: it is well known that driving comfort is an important performance for vehicle design, which is generally evaluated by body acceleration in vertical, longitudinal and lateral directions.

Road holding capability: to ensure the wheels keep in contact with the road without interruption, the dynamic tire load should not exceed static ones.

- Maximum suspension deflection: Due to the mechanical construction constraint, the maximum allowable suspension impacts should be taken into account to avoid excessive suspension drops, which could possibly cause deterioration in ride comfort and even structural damage.
- Saturation effect of the actuator: given the limited power of the actuator, the control force for the suspension system must be limited to a certain range.
- Reliability of closed loop systems: closed loop systems must be reliable when encountering non-ideal situations caused by actuators such as actuator input lag issues, sampled data and fault locating for unknown actuator failures.

3)Modeling of Electromagnetic Suspension System

The modeling of the electromagnetic damper (EMD) is developed as an electro-mechanical system.

3.1)Electromagnetic Damper

In this section, electromagnetic dampers (EMD) are represented by mechanical and electrical equations.

To formalize the EMD system in active suspensions, Figure 1 is used in a quarter car model with the parameters of a passenger car. Also, the formulation is confirmed by experiments for an automobile suspension equipped with EMD.

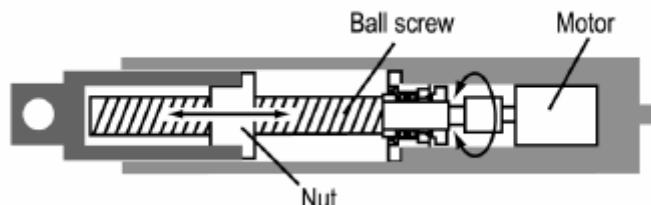


FIGURE 1. Concept of electromagnetic damper

The output force of the electromagnetic damper is expressed by the equation of rotary motion. The ball screw converts the input axial stroke into a rotary motion, and then the motor and the ball screw rotate dynamically with the moment of inertia.

In other words, in the conversion of the ball screw, the following equation is considered correct:

$$\omega = \frac{2\pi}{l} \dot{z} \quad (1)$$

where ω = angular velocity, and \dot{z} = input stroke speed. The equation of the rotor dynamics is described as follows:

$$(J_m + J_b)\dot{\omega} = \tau_m - \tau_d \quad (2)$$

where τ_m = output torque of motor, and τ_d = total output torque. The output torque is proportional to the motor current i , resulting in the following equation:

$$\tau_m = k_t i \quad (3)$$

Using the equilibrium of force, the output force in the axial direction, f_d , is written as

$$f_d = \frac{2\pi}{l} \tau_d \quad (4)$$

Therefore, using Eqs. 1 to 4, the output force of the electromagnetic damper is obtained as follows:

$$f_d = \Phi i - I_d \ddot{z} \quad (5)$$

where

$$\Phi = \frac{2\pi}{l} k_t \quad (6)$$

$$I_d = \left(\frac{2\pi}{l}\right)^2 (J_m + J_b) \quad (7)$$

For the application to automobile suspensions, an inertial term in Eq. 5 is so important that an equivalent inertia, I_d , cannot be ignored as compared to the unsprung mass. In practice, the mechanical friction of EMD is larger than that of a conventional oil damper. The output force, f_d , is considered as follows:

$$f_d = \Phi i - I_d \ddot{z} - f_r sgn(\dot{z}) \quad (8)$$

where f_r = dynamic friction. Equation 8 is used as the EMD output force in simulation. In this report, f_r zero was adopted to reduce some possible bad consequences.

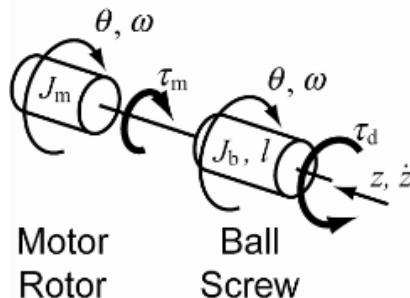


FIGURE 2. Rotor dynamics model of electromagnetic damper

Table 1. Parameters of rotor Dynamics model.

Symbol	Description
ω	Angular velocity
\dot{z}	Stroke speed
J_m	Moment of inertia of motor rotor
J_b	Moment of inertia of ball screw
l	Lead of the ball screw
I_d	Equivalent inertial inertia
k_t	Torque contant
Φ	Motor constant

3.2)Circuit Equation of Motor

The EMD motor circuit is modeled as an equivalent DC motor circuit. The motor circuit connected to the power supply is shown in Figure 3. The parameters of the circuit model are shown in Table 2. Since electromotive force is proportional to the angular velocity and stroke speed (Eq. 1), it is obtained as $\Phi\dot{z}$ and written as follows:

$$k_e \omega = k_e \frac{2\pi}{l} \dot{z} = \Phi\dot{z} \quad (9)$$

Therefore, the circuit equation is as follows:

$$v = L \frac{di}{dt} + Ri + \Phi\dot{z} \quad (10)$$

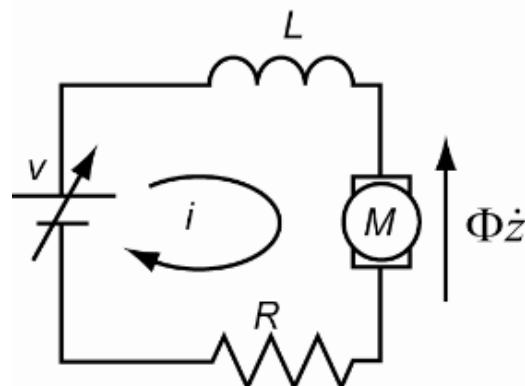


FIGURE 3. Circuit Diagram of Motor

Table 2. Parameters of circuit model

Symbol	Description
i	Current
V	Variable Voltage of Power Supply
L	Inductance
R	Resistance
k_e	Induced Voltage Constant (k_t)

4)Quarter-Car Suspension Model

If the movements of all four wheels should be separated and the suspension dynamics are only considered for the frequency range vertical vehicle dynamics (0-25 Hz), the quarter-car model has been extensively used in the literature, and the more detailed models represent the appropriate modeling framework, which captures most important features . It consists of the dynamic behavior of the unsprung mass (representing the mass of a tire, wheel, brake, wheel carrier and parts of the suspension system) and the spring mass (mainly determined by a quarter of the chassis mass, including the passengers and vehicle load) connected with the suspension system. Also, the tire in this model can be represented by a parallel spring and damper configuration.

EMS consists of EMD and coil spring. Although the EMD is used as an actuator independently, only that much energy is required to support the vehicle body through the EMD, so the EMS system has a coil spring.

In this subsection, EMS is illustrated as a quarter-car model as shown in Fig. 4. In order to identify the effect of an equivalent inertial inertia, dI , and heavier weight of EMD due to a mechanical damper, a suspension mount and mass of a component of EMD, are considered. Equations of 3 DOF quarter-car model are written as follows:

$$m_s \ddot{x}_s + k_2(x_s - x_u) + k_3(x_s - x_r) + c_3(\dot{x}_s - \dot{x}_r) = 0 \quad (11)$$

$$m_r \ddot{x}_r - k_3(x_s - x_r) - c_3(\dot{x}_s - \dot{x}_r) = u - I_d(\ddot{x}_s - \ddot{x}_u) \quad (12)$$

$$m_u \ddot{x}_u - k_2(x_s - x_u) + k_1(x_u - x_0) = -(u - I_d(\ddot{x}_s - \ddot{x}_u)) \quad (13)$$

$$f_d = u - I_d \ddot{z} = u - I_d(\ddot{x}_s - \ddot{x}_u) \quad (14)$$

$$u = \emptyset i \quad (15)$$

$$z = x_r - x_u \quad (16)$$

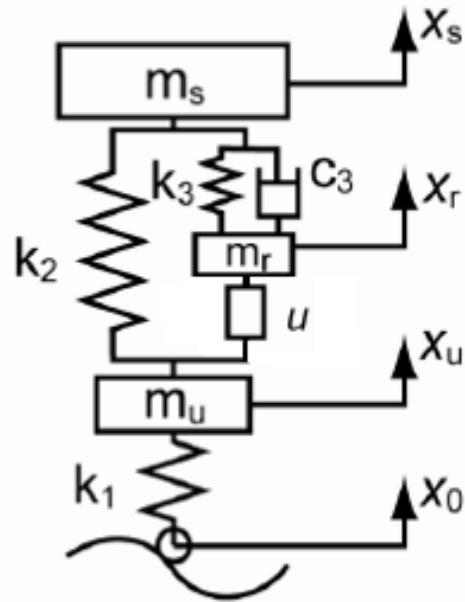


FIGURE 4. 3 DOF quarter-car model

Table 3. Parameters of 3DOF quarter-car model

Symbol	Description
m_s	Sprung Mass
m_r	Mass of rod (a component of EMD)
m_u	Unsprung mass
I_d	Equivalent inertial inertia
k_1	Spring constant of tire
k_2	Spring constant of suspension
k_3	Spring constant of mount
c_3	Damping coefficient of mount
x_s	Displacement of sprung mass
x_r	Displacement of rod
x_u	Displacement of unsprung mass
x_0	Displacement of road profile

5) Design of Quarter Car Suspension System Model in Matlab/Simulink

We have designed the simulinks in Figures 5, 9, 14, 18, 22 and 26 the screenshots of which are shown below, and we get the outputs. Our simulinks worked flawlessly. Our graphics turned out as desired.

5.1) Passive Suspension Systems

Passive suspension system consist of damping elements and spring. Damping elements does not change with time. Engineers design the passive system to be optimal, taking into account many parameters. Engineers choose the damping parameters and stiffness on the basis of a compromise between the road holding, ride comfort and road conditions. Therefore, vehicles with good road holding may have bad driving comfort, while vehicles with bad road holding may have good driving comfort.

Figure 4. (without k_3 , m_r , m_u) is an example for passive suspension system.

5.1.1) Passive suspension system for random road model

By using Figure 6. road model input we obtained Figure 7. displacement-time and Figure 8. acceleration-time graphs.

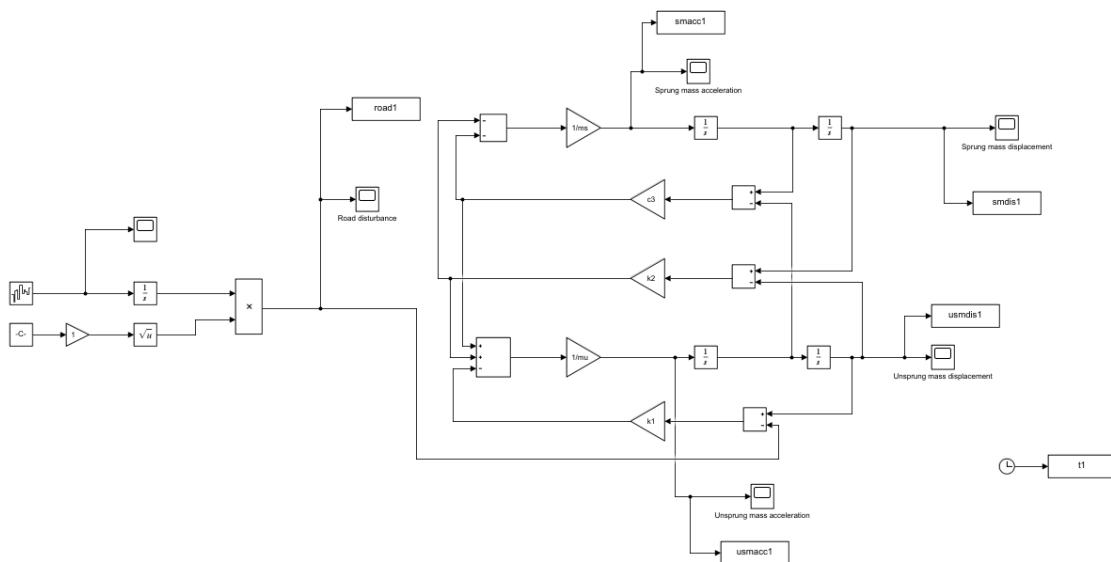


FIGURE 5. Passive suspension system for random road model design in simulink

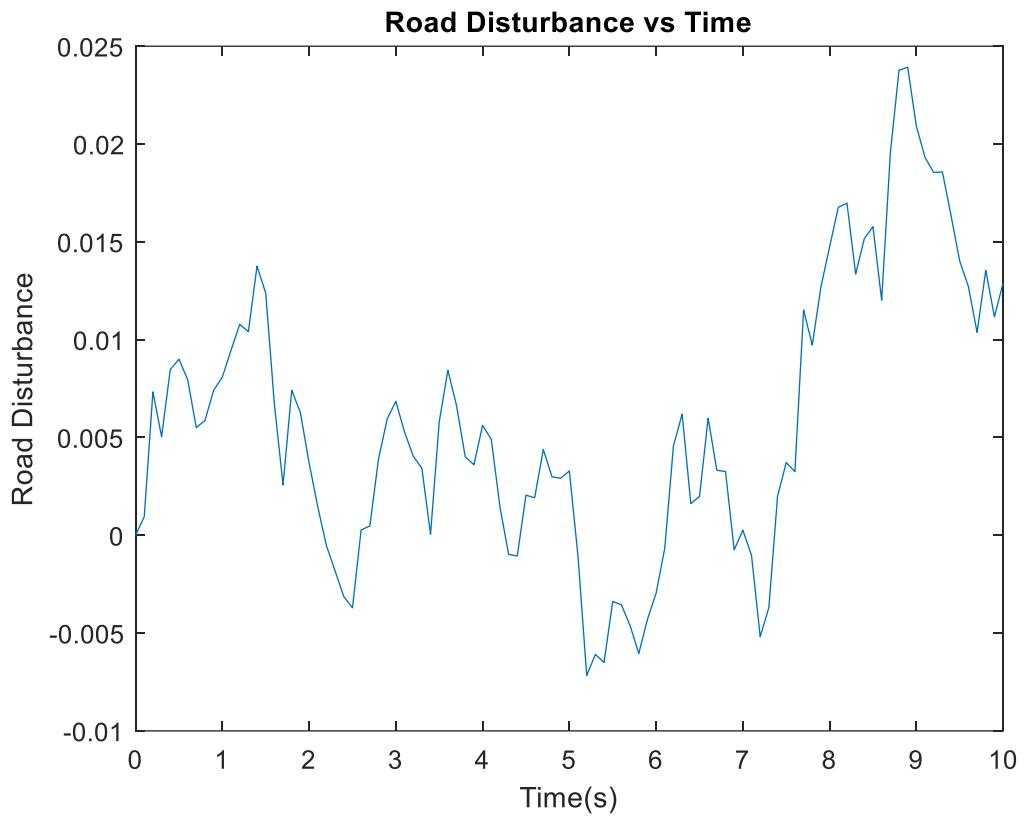


FIGURE 6. Road Profile Model

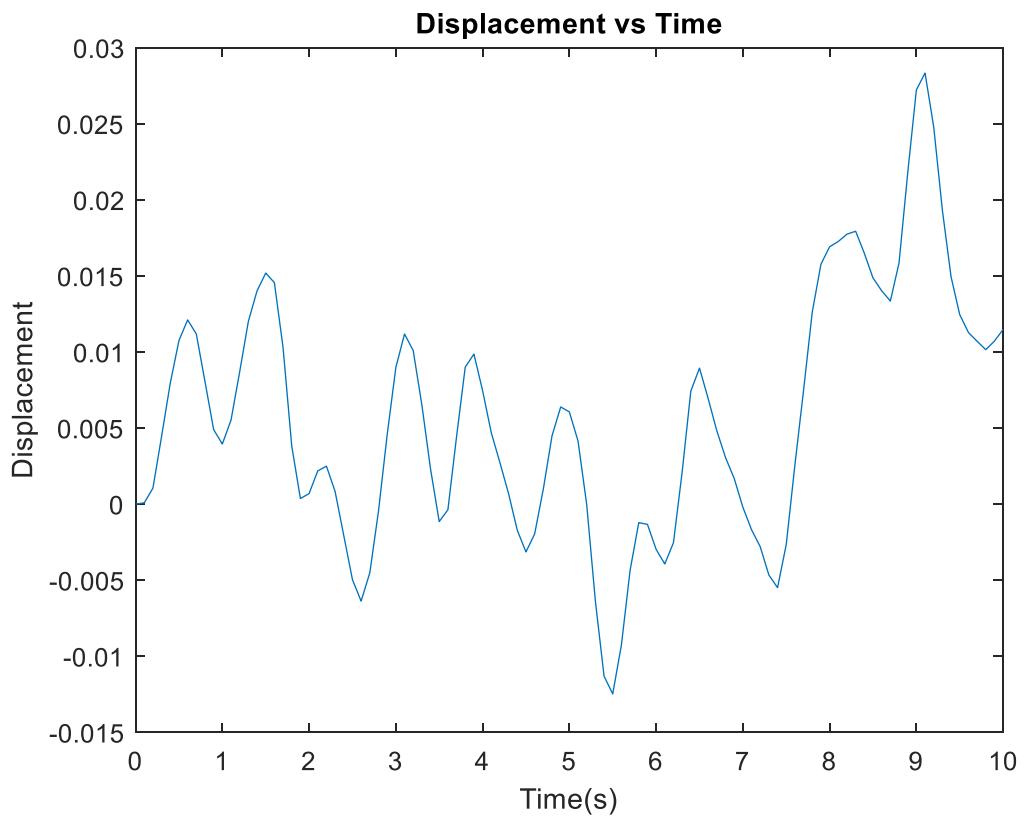


FIGURE 7. Displacement vs Time Plots

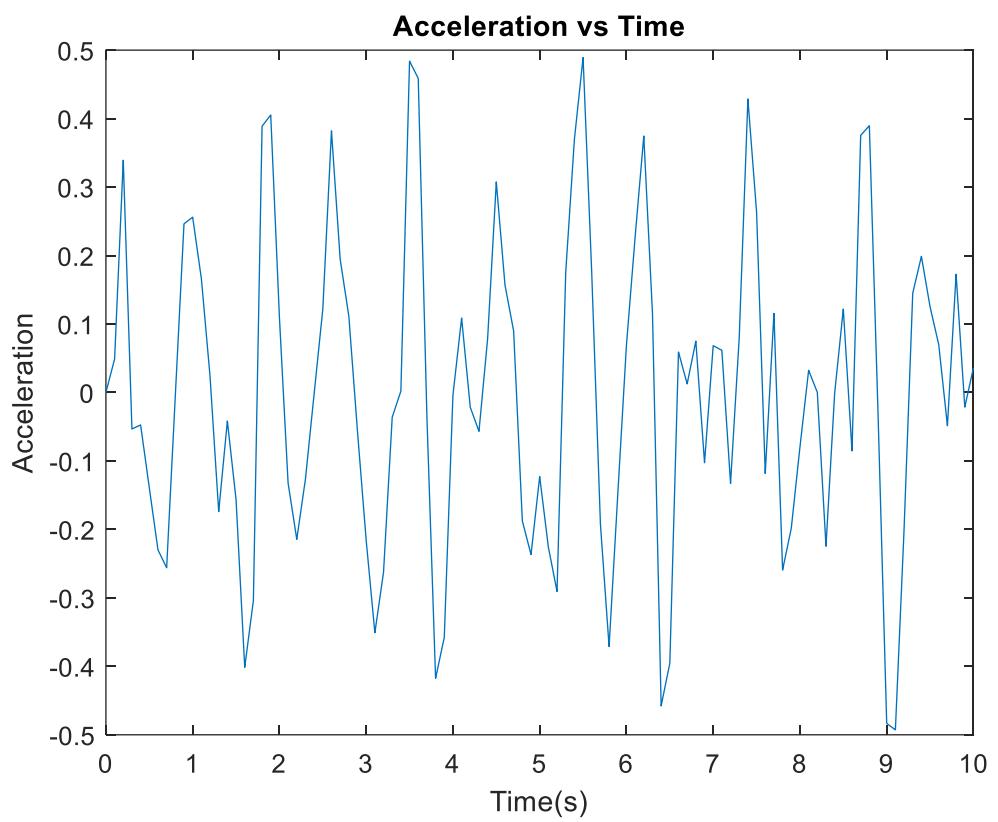


FIGURE 8. Acceleration vs Time Plots

5.1.2) Passive suspension system for bump road model

By using Figure 10. road model input we obtained Figure 11. displacement-time and Figure 12. acceleration-time graphs.

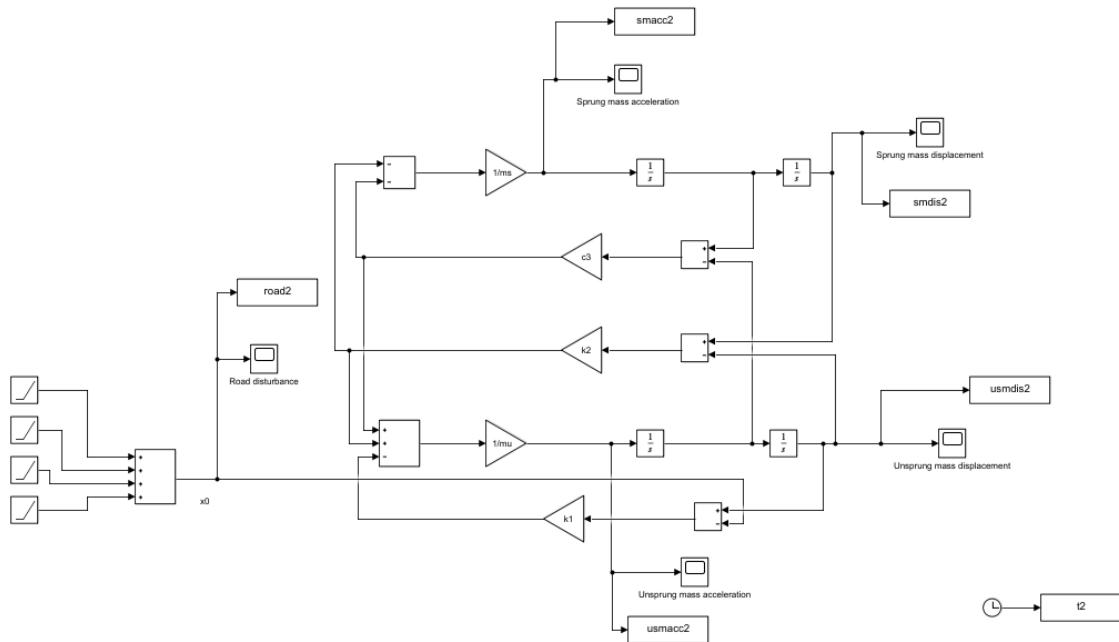


FIGURE 9. Passive suspension system for bump road model design in simulink

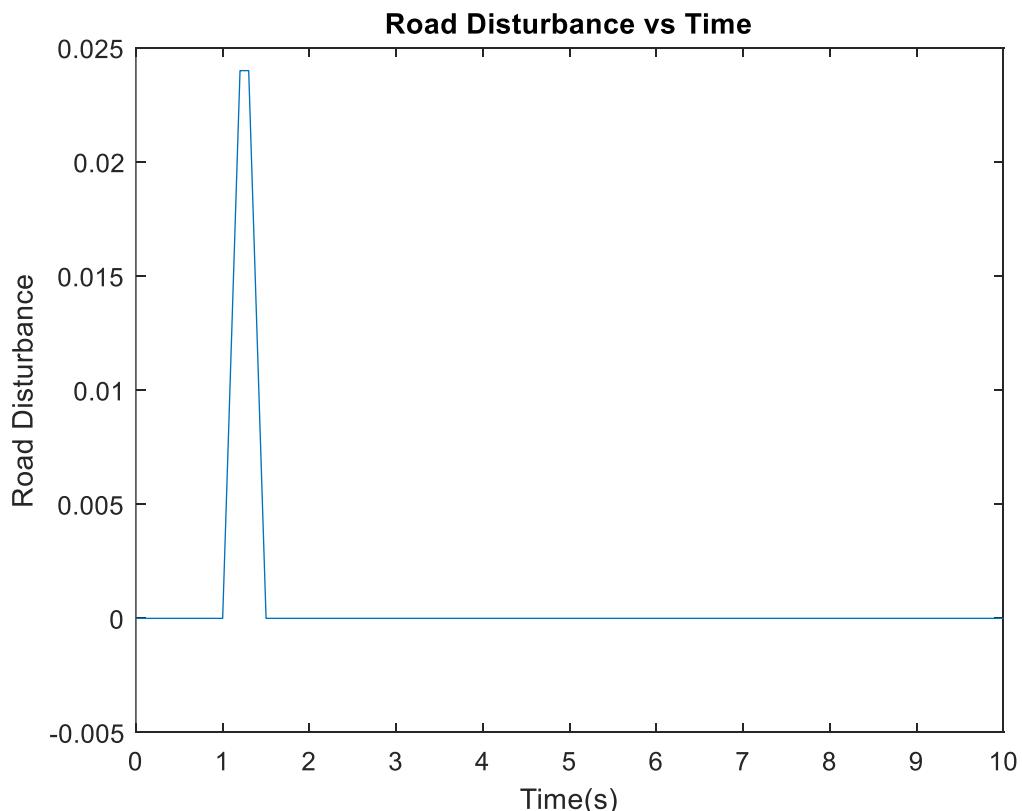


FIGURE 10. Road Profile Model

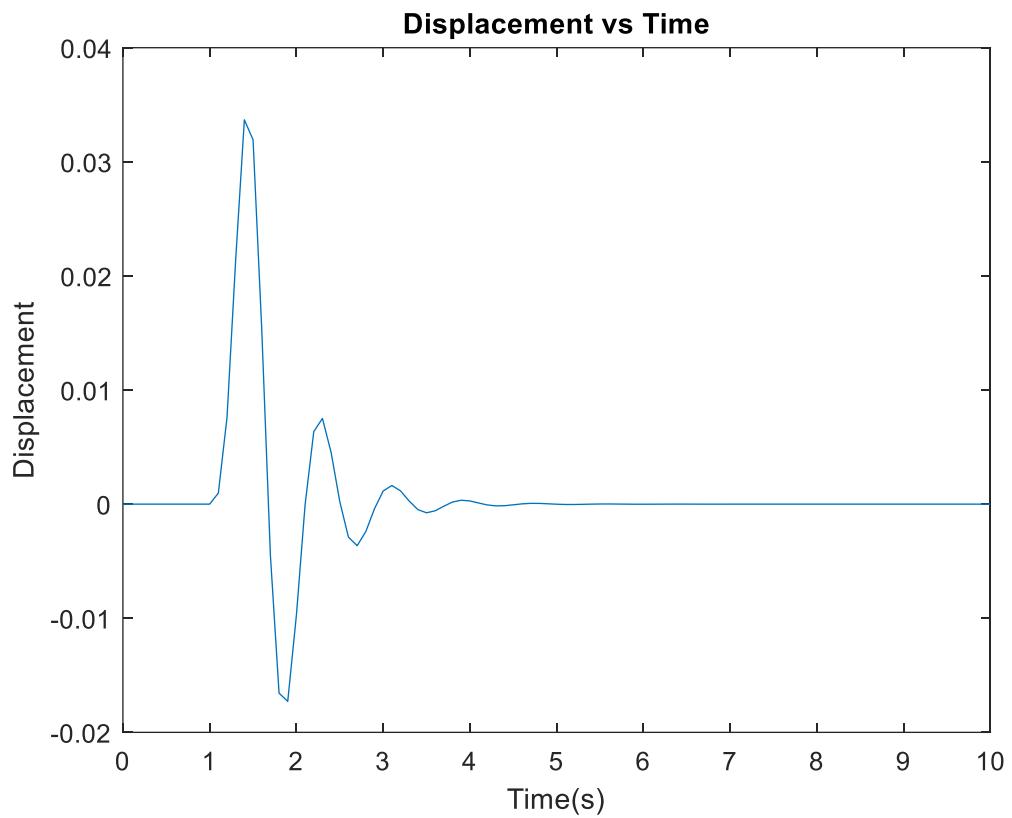


FIGURE 11. Displacement vs Time Plots

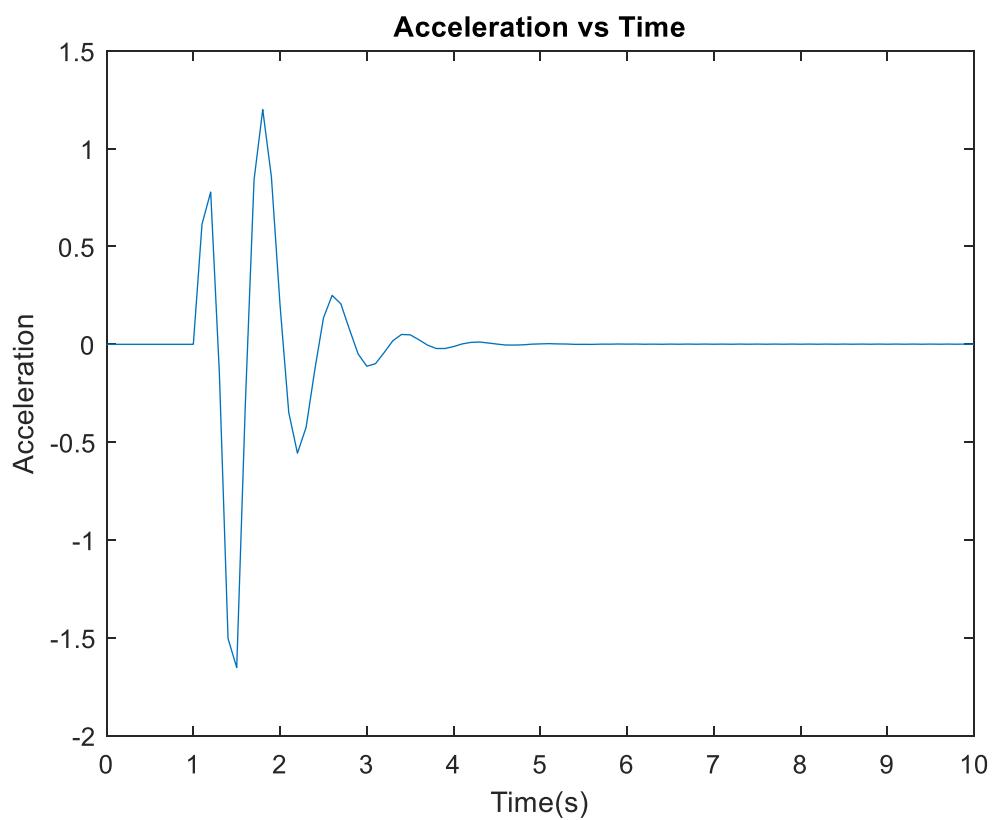


FIGURE 12. Acceleration vs Time Plots

5.2) Active (PI Controller) Suspension Systems

In active suspension system, actuator is used instead of spring and damper. High frequency response servo valve controlled the hydraulic actuator. Control bandwidth is enough to handle the natural frequency of sprung masses (1-3 Hz) and natural frequency of unsprung masses (10-15 Hz). Obtaining applicable bandwidth is so hard. If bandwidth increases, power consumption also increases. Noise transmission can be also serious problem. If important flexibility added in series with the strut, it might be solved. Some active suspension system generally powered by electro-hydraulic. These systems are so expensive and have a noise.

In our active suspension system, we added control voltage, motor current and rms commands to obtain power consumption, current and voltage. When we run the simulink model we get the desired results and graphs

Figure 4. is an example for passive suspension system.

5.2.1) Control System

The current of motor in the EMS system is controlled to follow the required output force, u_{ref} , by PI controller as shown in Fig. 5. The required output force, a reference signal for control output, is calculated using the following equation:

$$u_{ref} = -C_s \dot{x}_s - C_g \dot{x}_u \quad (17)$$

where C_s and C_g are feedback gains of velocity of sprung mass and unsprung mass, respectively. $C_s > 0$ and $C_g \leq 0$ are required to stabilize the EMS system. The positive feedback gain of the sprung mass velocity represents the sky hook damper for isolation, and the negative feedback gain of the sprung mass represents the ground hook damper for road maintenance.

The voltage of power supply, v , is defined by PI controller as follows:

$$\mathcal{L}\{v\} = \left(K_p + \frac{K_I}{s} \right) \mathcal{L}\{i_{ref} - i\} \quad (18)$$

where ‘L’ is Laplace transform, and ‘s’ Laplacian operator. The PI gain is adjusted for the EMS system to control the resonance mode of the unsprung mass.

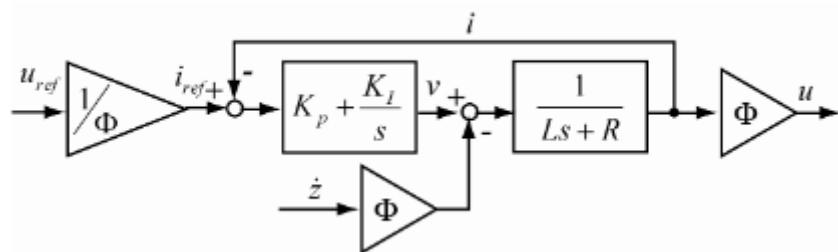


FIGURE 13. Block diagram of PI controller

5.2.2) Active suspension system for random road model

By using Figure 15. road model input we obtained Figure 16. displacement-time and Figure 17. acceleration-time graphs.

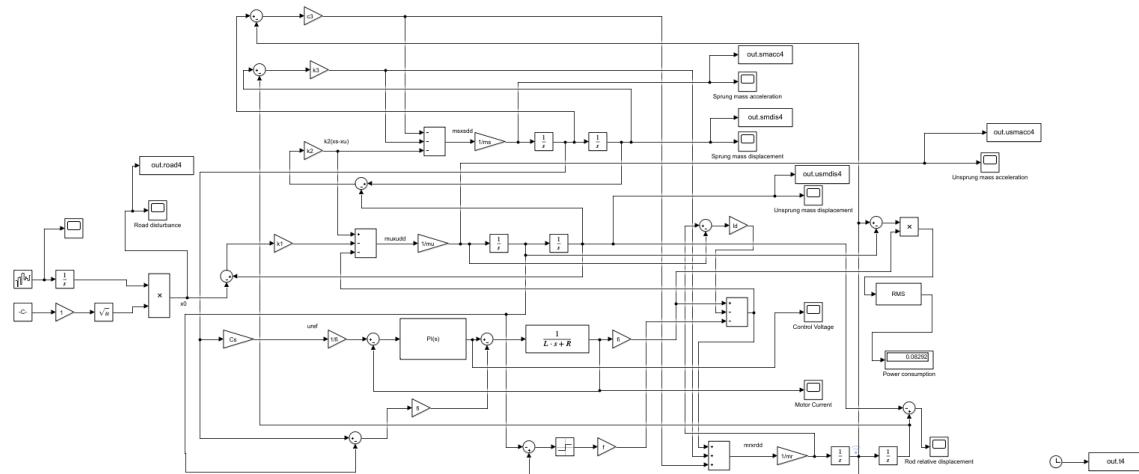


FIGURE 14. Active suspension system for random road model design in simulink

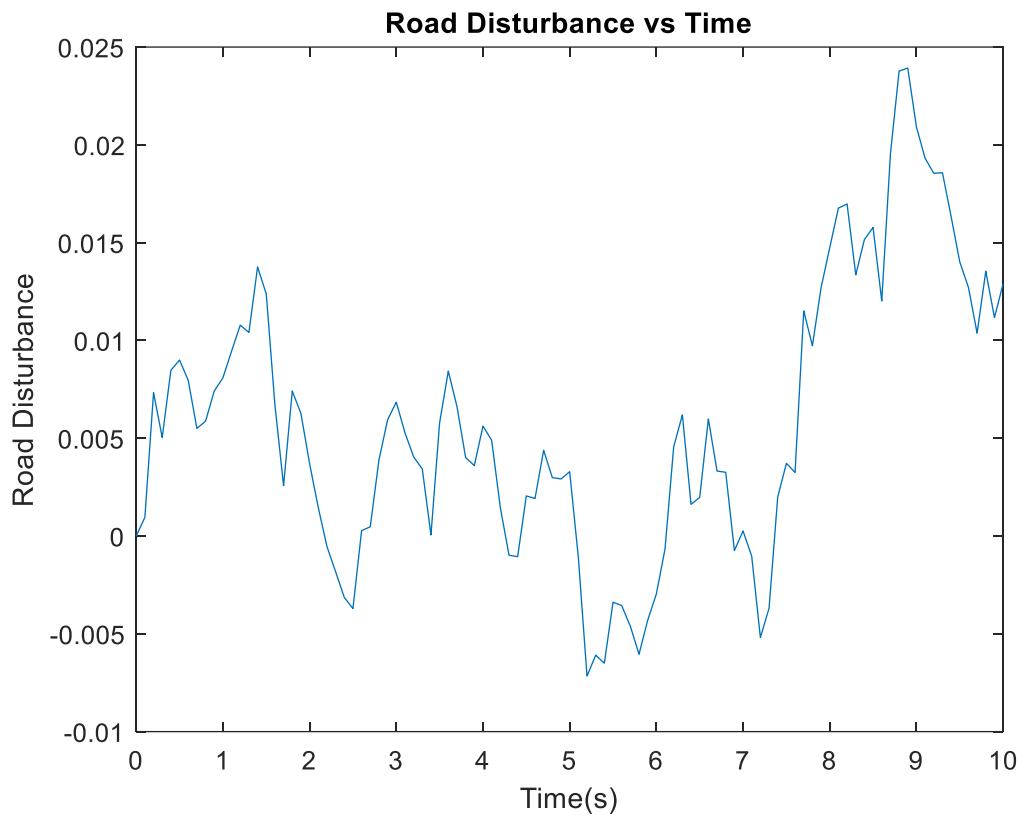


FIGURE 15. Road Profile Model

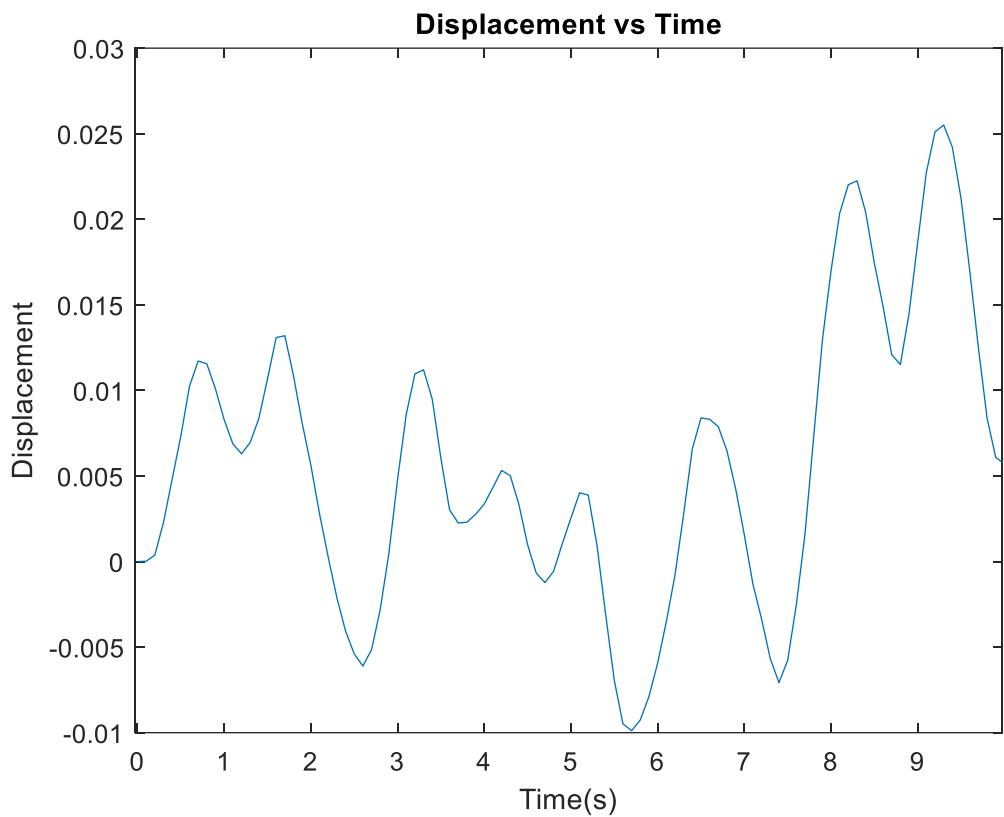


FIGURE 16. Displacement vs Time Plots

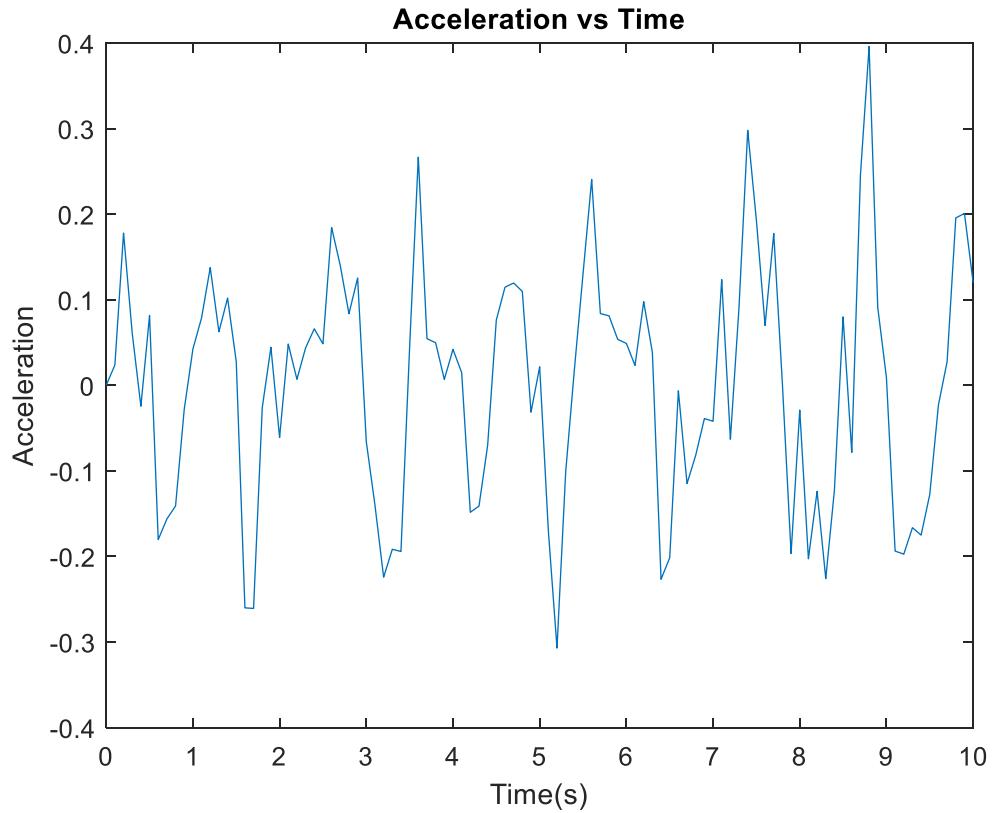


FIGURE 17. Acceleration vs Time Plots

5.2.3) Active suspension system for bump road model

By using Figure 19. road model input we obtained Figure 20. displacement-time and Figure 21. acceleration-time graphs.

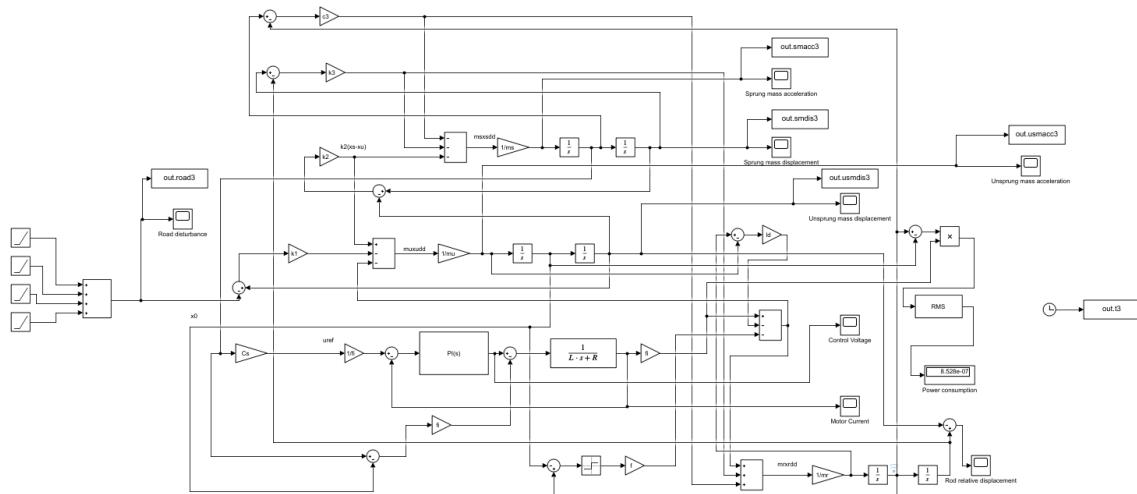


FIGURE 18. Active suspension system for bump road model design in simulink

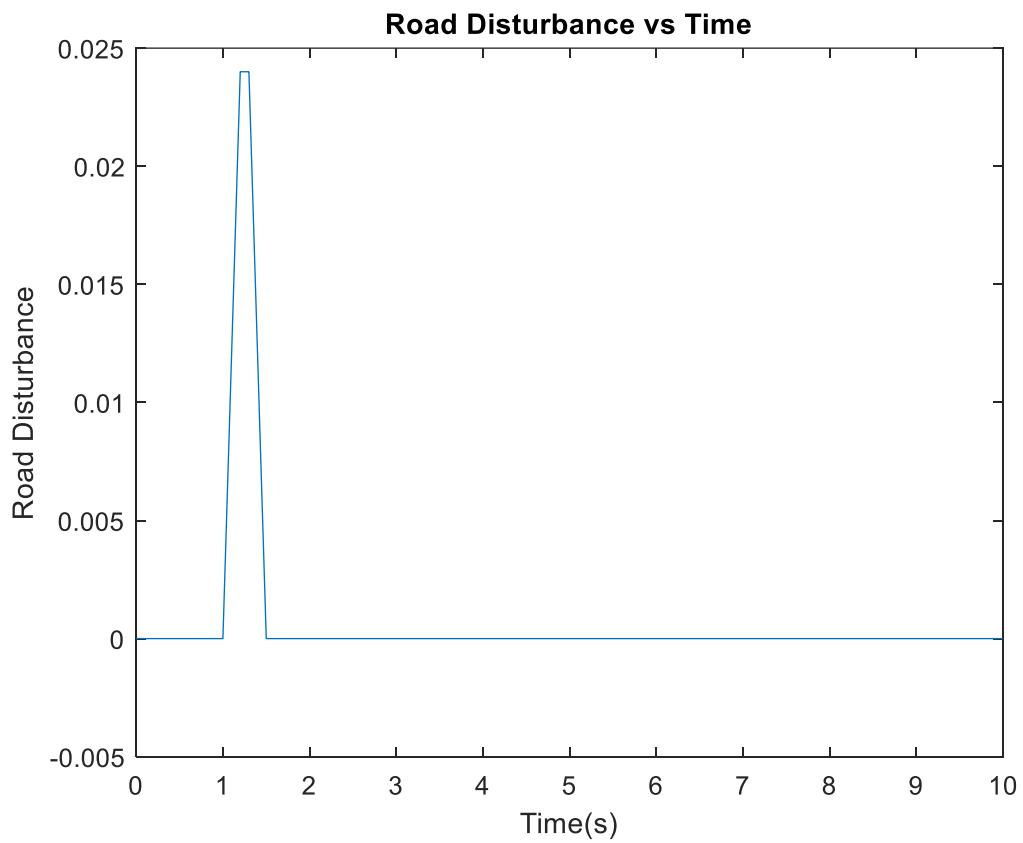


FIGURE 19. Road Profile Model

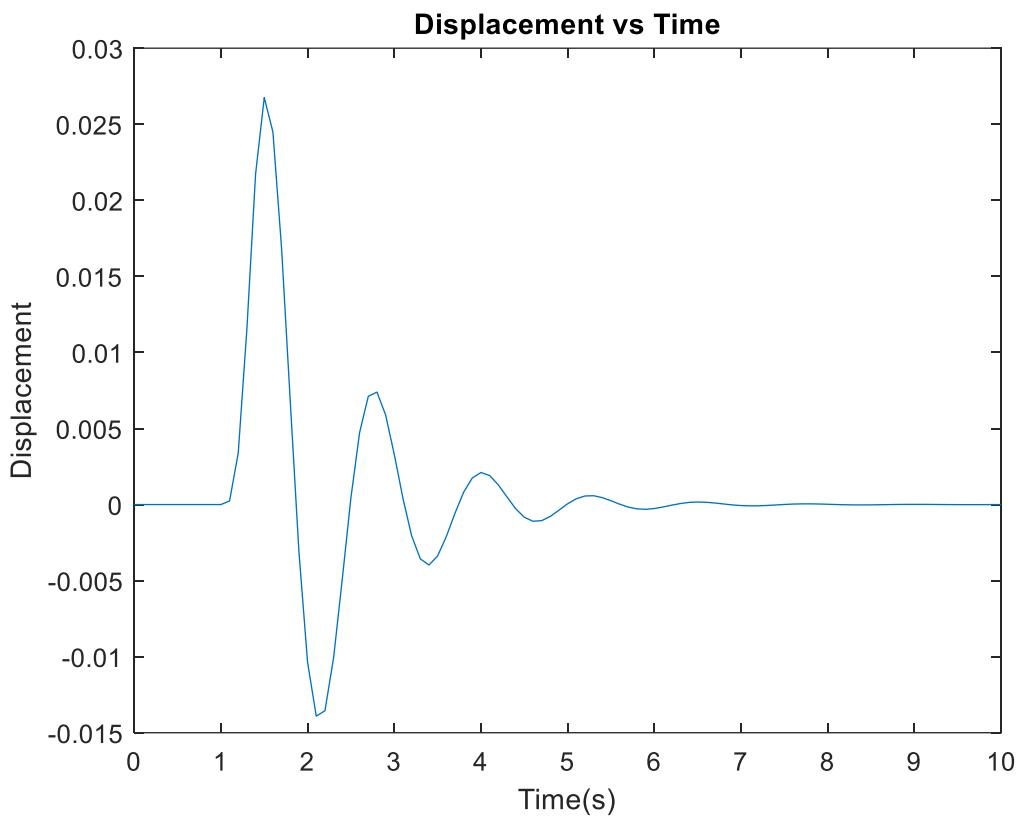


FIGURE 20. Displacement vs Time Plots

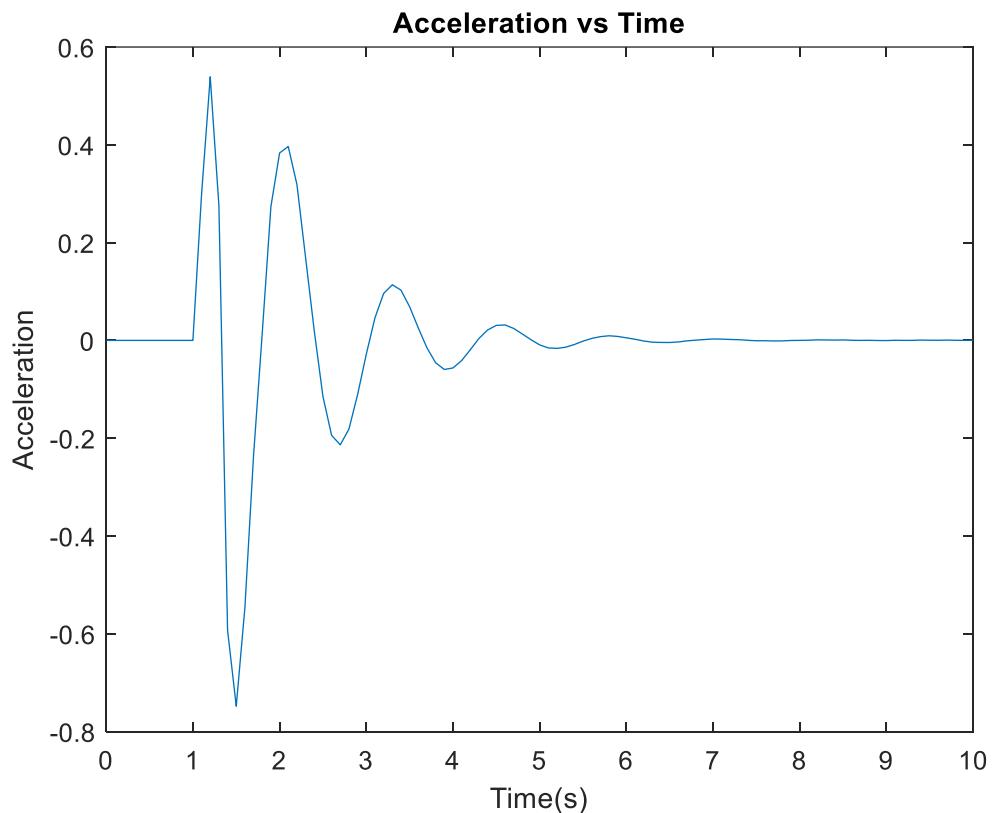


FIGURE 21. Acceleration vs Time Plots

5.3) Dynamic Output Feedback Control Systems

An observer-based controller is a dynamic feedback controller with a two-stage structure. First, the controller generates an estimate of the state variable of the system to be controlled, using the measured output and known input of the system. This estimate is generated by a state observer for the system. Next, the state estimate is treated as if it were equal to the exact state of the system, and it is used by a static state feedback controller. Dynamic feedback controllers with this two-stage structure appear in various control synthesis problems for linear systems.

$$\begin{aligned}\dot{z}_0 + avz_0 &= w \\ \dot{x} &= Ax + B_1w + B_2u\end{aligned}$$

where w equal to road state

Dynamic output feedback (DOF) controllers are designed to stabilize the system. Dynamic output feedback controllers created are indicated by y as input and u as output.

$$\begin{aligned}\dot{x}_c &= A_c x_c + B_c y \\ u &= C_c x_c + D_c y\end{aligned}$$

The main purpose of observer design is to approximate the state vector x by using the measured output y, and then use the approximate state vector to realize the full-state feedback control law. It was first described by Xu. Then the control law with $u = K\hat{x}$ was used. Here the eigenvalues of K, A+BK are selected so that they are in the desired places. The same operations were applied for eigenvalues of $A^T + C^T L^T$.

As a result of these operations, the general control law was formed as follows:

$$\begin{aligned}\dot{\hat{x}} &= (A + BK + LC + LDK)\hat{x} - Ly \\ u &= K\hat{x}\end{aligned}$$

Finally, the following are the states written inside the state-space block used for the controller created in Simulink:

$$\begin{aligned}x_c &= \hat{x} \\ A_c &= A + BK + LC + LDK \\ B_c &= -L \\ C_c &= K \\ D_c &= 0\end{aligned}$$

5.3.1) Dynamic output feedback control system for random road model

By using Figure 23. road model input we obtained Figure 24. displacement-time and Figure 25. acceleration-time graphs.

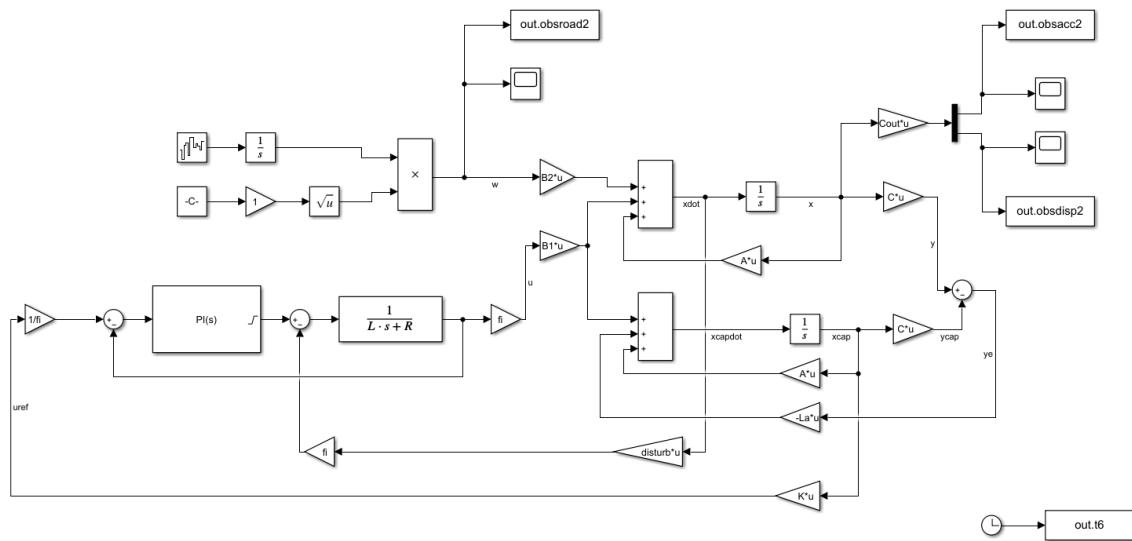


FIGURE 22. Dynamic output feedback control system for random road model design in simulink

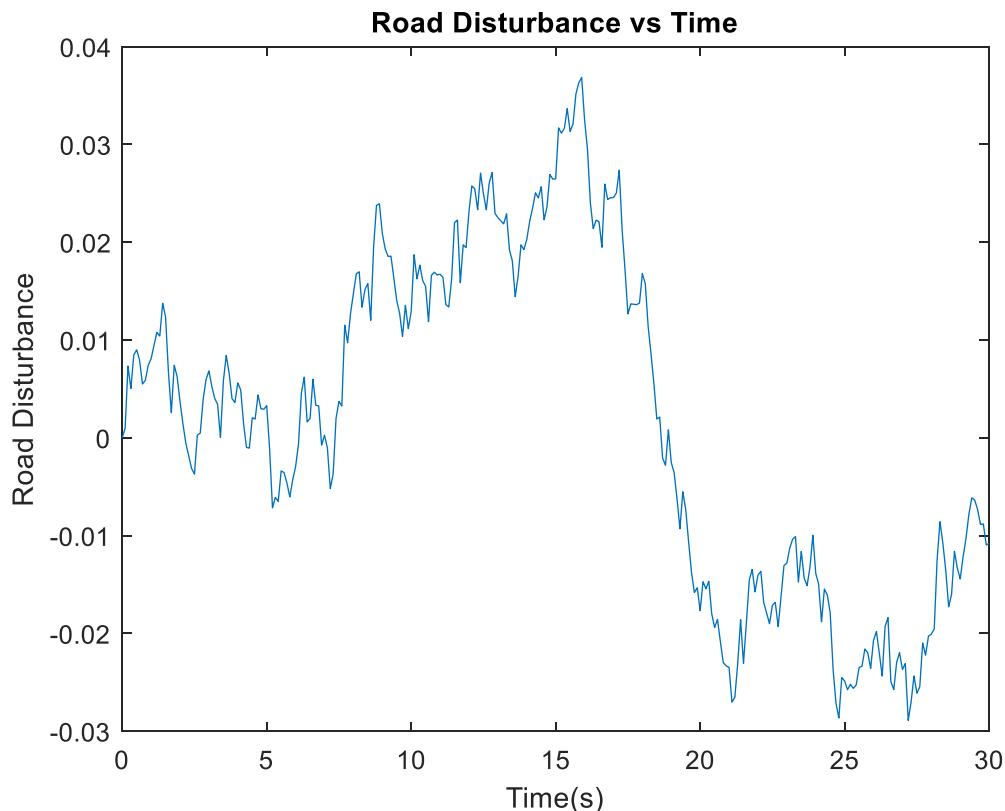


FIGURE 23. Road Profile Model

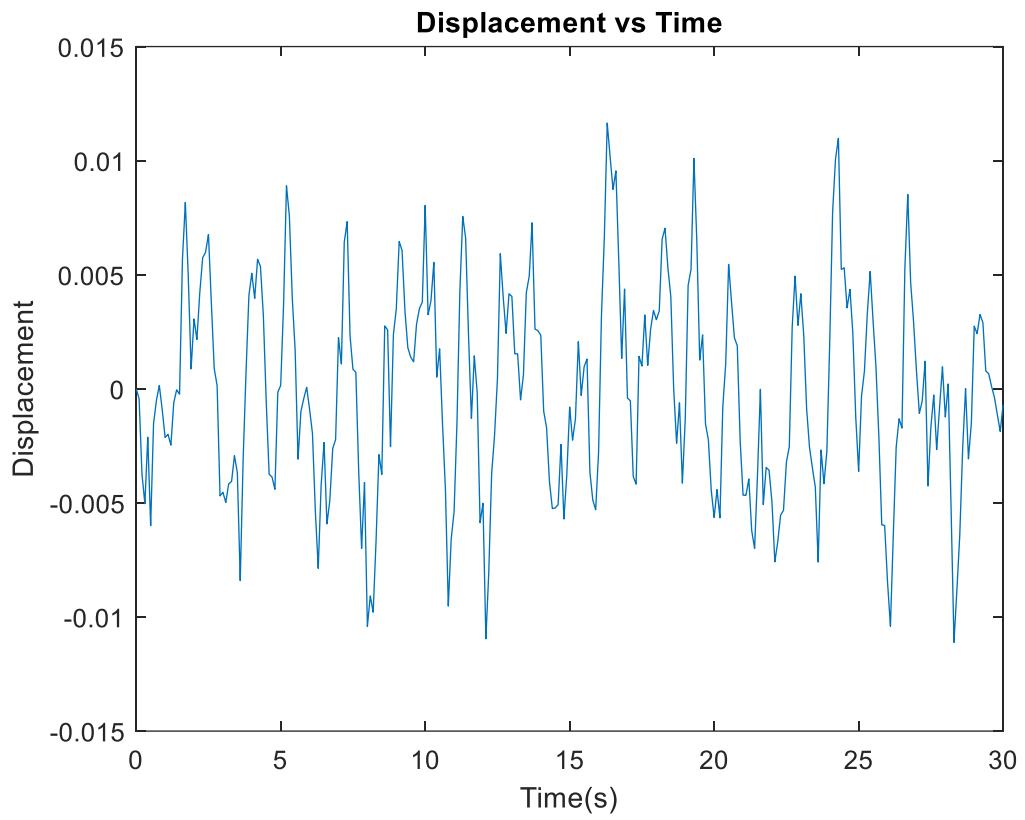


FIGURE 24. Displacement vs Time Plots

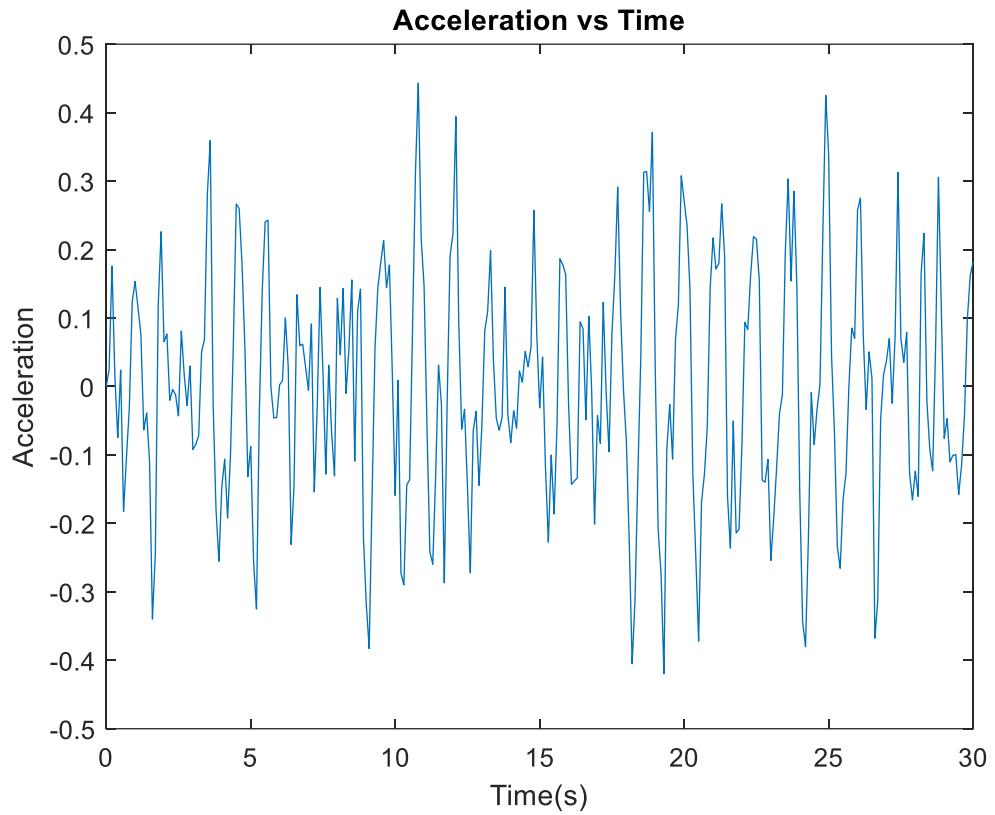


FIGURE 25. Acceleration vs Time Plots

5.3.2) Dynamic output feedback control system for bump road model

By using Figure 27. road model input we obtained Figure 28. displacement-time and Figure 29. acceleration-time graphs.

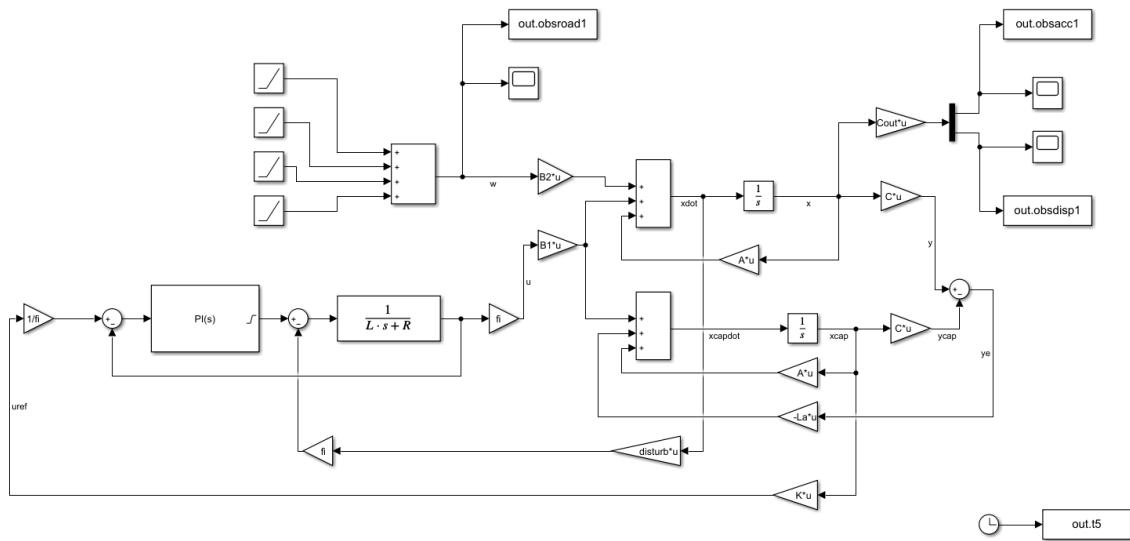


FIGURE 26. Dynamic output feedback control system for bump road model design in simulink

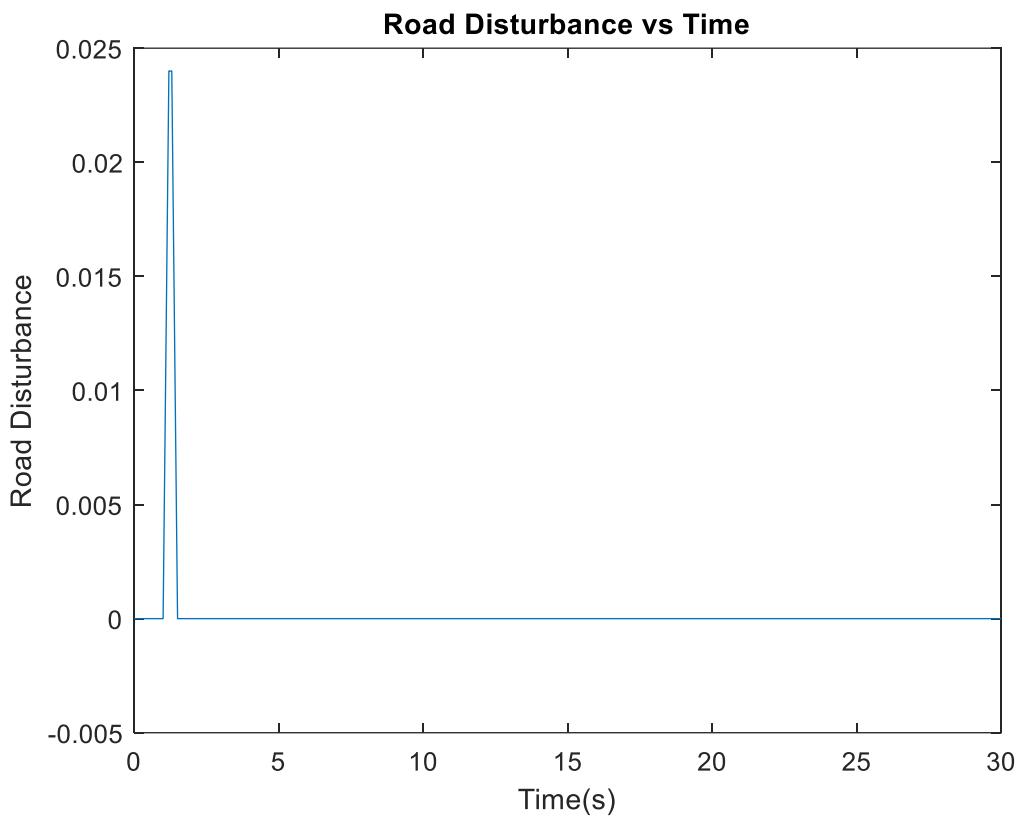


FIGURE 27. Road Profile Model

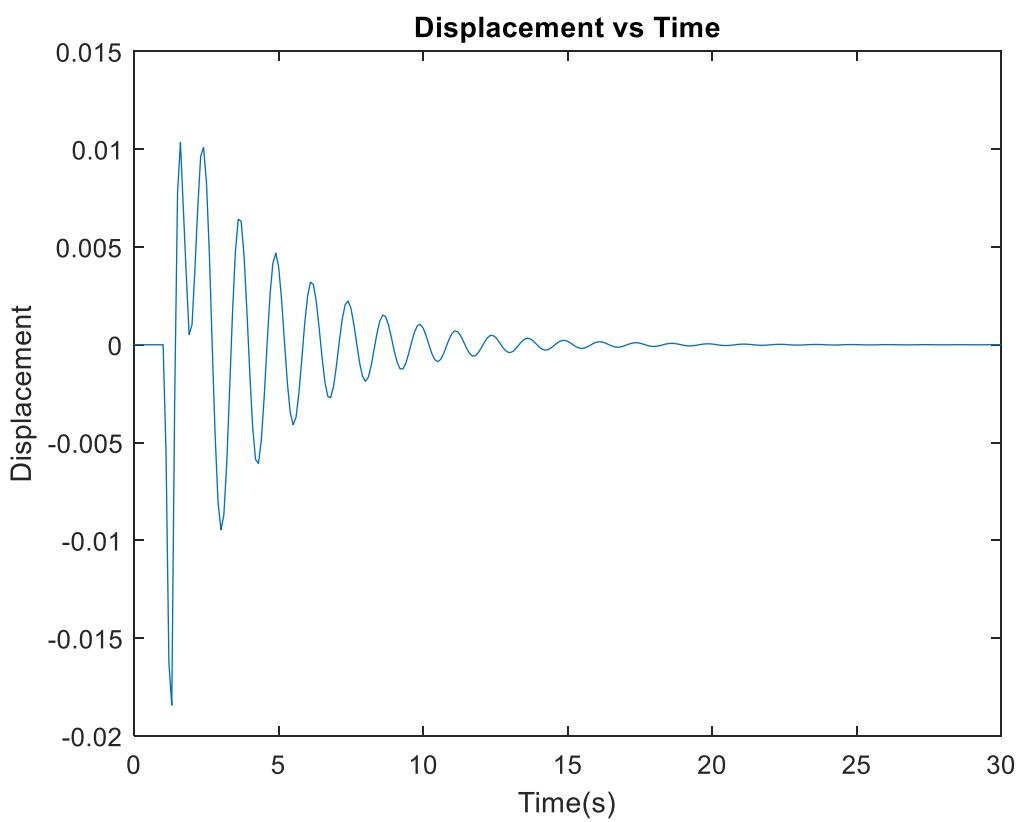


FIGURE 28. Displacement vs Time Plots

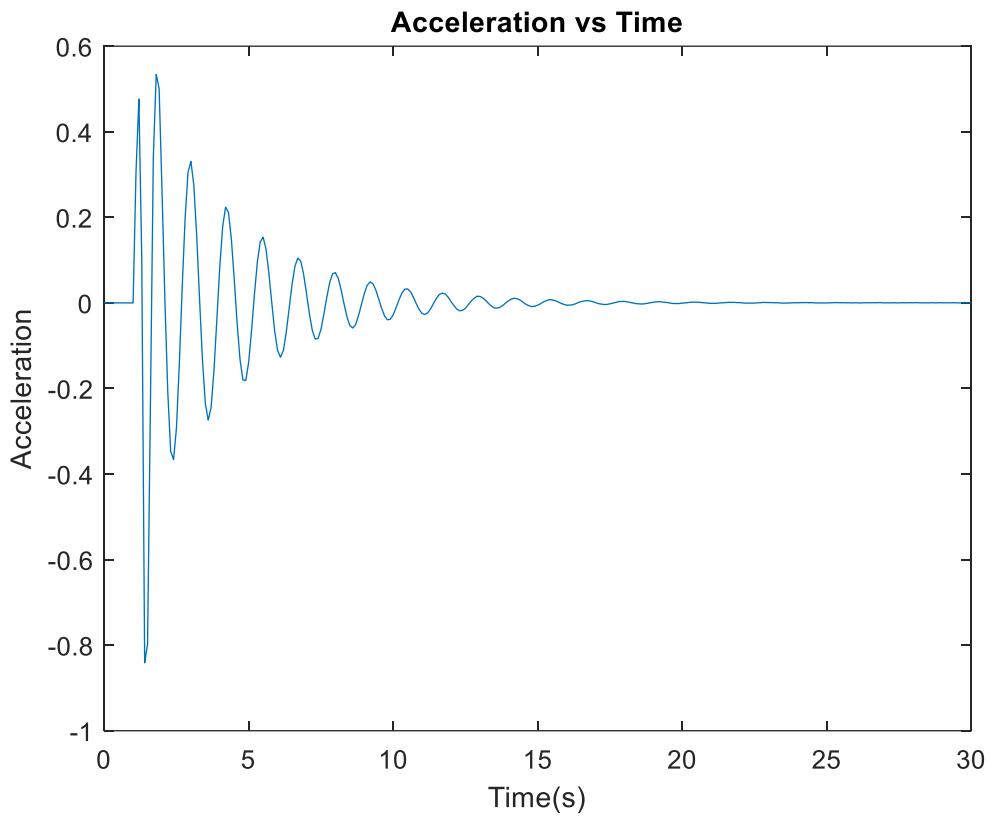


FIGURE 29. Acceleration vs Time Plots

6) CONCLUSION

To design the controller, we did long-term research from many sources sent by the instructor and other sources. While modeling in Matlab/Simulink, we encountered many errors and tried many different ways to fix these errors. While trying to fix these errors, we learned many commands from Matlab/Simulink.

In this project, we deal with suspension control system design for automobiles. Unlike conventional suspension system with non-variable feature, adjustment of practical suspension system improve riding comfort and road holding. To evaluate the performance of suspension system, main criterions are car features, road conditions, speed and steering maneuver. To solve the problems related with the suspension control design, free body diagram of quarter-car and quarter car analysis is so informative. Nevertheless, we cannot describe the road holding by one variable. Because of road inconsistencies, the impact of the variation of the tire load on the cornering behaviour of an automobile by a steady-state tire relations in combination with quarter car model.

So as to increase the suspension system performance, we have tried so many control methods. We have applied optimal control method to active suspension system. We can

apply optimal control theory to both limited state feedback system and full state feedback system. In quarter car analysis, we understand that although there is an active force generator and freedom to generate an arbitrary force, there is still conflict between road holding and riding comfort. The main reason of this situation is single actuator with a two degree of freedom model applies actuating forces on vehicle body and axle.

It is an unusual thing maximum ride comfort and maximum road holding for even well-adjusted passive suspension system. In active control suspension system, system parameters can be changed in some cases road holding might be improved at the cost of ride comfort. Limited state feedback system can compete with full state feedback control system and limited state feedback configuration is much logical because in full state feedback system we need to determine the much more states.

In quarter car model, there are some limitation due to contradiction between road holding and ride comfort because of invariant points in the transfer functions. We can find a particular area in frequency domain to good riding comfort without making road holding worse. By the way, in some cases we cannot improve the ride comfort to prevent deteriorate to road holding.

We should remember this cost of active suspension system and power consumption of active suspension system is so high. That's why engineers discussed on semi active damping and different strategies to improve road holding and ride comfort. According to simulation result, to increase the ride comfort, we may use the skyhook control system.

In semiactive suspension systems, range of damper is so important. It affects the ride comfort directly. It should be the larger the range between soft and firm, the larger gain in ride comfort but road holding deteriorate.

Comment About Plots

As a result, Displacement-time and Acceleration-time plots were created using two different road models for passive suspension system, active suspension system and dynamic output feedback control system. These road models have been called the random road model and the bump road model.

First, let's look at the results that occur when using the random path model. Here, the Displacement-time plot in the passive suspension system and the Displacement-time plot in the active suspension system are quite similar to each other. But the Displacement-time plot in the dynamic output feedback control system is quite different. There is quite a lot of oscillating motion in the Acceleration-time plots in all three systems. But these plots are quite different from each other.

Second, let's look at the results that occur when using the bump road model. Here, the Displacement-time plot and the Acceleration-time plot become stable in the fifth second in the passive suspension system. But with the , Displacement-time plot and Acceleration-time plot become stable in the seventh second in the active suspension system.Unlike passive and active suspension systems, Displacement-time plot and Acceleration-time plot become stable twentieth second in the dynamic output feedback control system.

Reasons why plots differ from each other in the random road model and reasons why systems have different stable times in the bump road model:

- Each system has a different structure,
- Adding an active suspension system,
- From the design of the structure of the dynamic output feedback system,
- Some desired locations do not match the actual poles.

7) REFERENCES

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8)APPENDICES

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Chapter 12 Design via State Space

Now that we have designed controllers and observers for transient response and steady-state error, we summarize the chapter with a case study demonstrating the design process.

Case Study

Design
D

Antenna Control: Design of Controller and Observer

In this case study, we use our ongoing antenna azimuth position control system to demonstrate the combined design of a controller and an observer. We will assume that the states are not available and must be estimated from the output. The block diagram of the original system is shown on the front endpapers, Configuration 1. Arbitrarily setting the preamplifier gain to 200 and removing the existing feedback, the forward transfer function is simplified to that shown in Figure 12.22.

$$\frac{U(s) = E(s)}{s(s + 1.71)(s + 100)} \rightarrow Y(s) = \theta_o(s)$$

FIGURE 12.22 Simplified block diagram of antenna control system shown on the front endpapers (Configuration 1) with $K = 200$

The case study will specify a transient response for the system and a faster transient response for the observer. The final design configuration will consist of the plant, the observer, and the controller, as shown conceptually in Figure 12.23. The design of the observer and the controller will be separate.

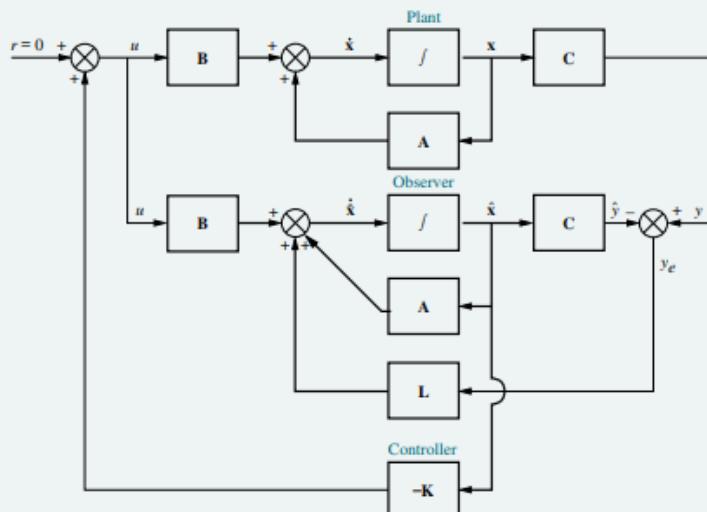


FIGURE 12.23 Conceptual state-space design configuration, showing plant, observer, and controller

FIGURE 30.

We used above figure while designing dynamic output feedback control systems.

MATLAB CODES

```

%%%
clc
clear
ms=200;
mr=10;
mu=40;
k1=17000;
k2=8000;
k3=3000;
c3=3000;
KP=5;
KI=0;
L=0.1;
R=20;
Cs=-2500;
fi=70;
Id=0.0003;
f=-0.0000003;

%%%
a1=0.15; % rad/m
v=40; % m/s

%syms k1 k2 k3 ms c3 Id mu mr u
a=[0 0 0 0 1 0 0];
b=[0 0 0 0 0 1 0];
c=[0 0 0 0 0 0 1];
e=[(-k2-k3)/ms k3/ms k2/ms 0 -c3/ms c3/ms 0];
f=[(((((-k2-k3)/ms)*Id)+k2)/(mu+Id)*-1+((-k2-k3)/ms))*-Id+k3]/mr ((((-Id*k3/ms)/(mu+Id))+(k3/ms))*-Id-k3)/mr (((Id*k2/ms+(-k2-k1))/(mu+Id))*-1+k2/ms)*-Id)/mr ((Id*k1)/(mu+Id))/mr (c3-Id*((-c3/ms)-((Id*c3/ms)/(mu+Id))))/mr 0];
g=[((((-k2-k3)/ms)*Id)+k2)/(mu+Id) ((Id*k3/ms)/(mu+Id)) (Id*k2/ms+(-k2-k1))/(mu+Id) k1/(mu+Id) (-c3*Id/ms)/(mu+Id) (c3*Id/ms)/(mu+Id) 0];
k=[a;b;c;e;f;g]
B2=k(:,4);
A=[k(:,1) k(:,2) k(:,3) k(:,5) k(:,6) k(:,7)];
B1=[0;0;0;0;mu/(mr*(mu+Id));-1/(mu+Id)]
B=[B1,B2];

Cout=[(-k2-k3)/ms k3/ms k2/ms -c3/ms c3/ms 0;1 0 -1 0 0 0];
C=[1 -1 0 0 0 0;0 1 -1 0 0 0]

disturb=[0 1 -1 0 0 0];
D=zeros(2,2)

Controllability=ctrb(A,B);
rank(Controllability);
Observability=obsv(A,C);
rank(Observability);

%%%
format longG

p=[-1 -5 -21 -13 -15 -19];

```

```

K=place(A,B1,p)
K=-K;
La=place(A',C',p)
La=-La';

```

```

%%%
Ac=A+B1*K+La*C
Bc=-La
Cc=K;
Dc=[0 0]
eig(A+B1*K)
eig(A+La*C)

```

The states of systems

$$\begin{bmatrix} \dot{Z}_s \\ \dot{Z}_r \\ \dot{Z}_u \\ \ddot{Z}_s \\ \ddot{Z}_r \\ \ddot{Z}_u \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & -a_1 v & 0 & 0 & 0 \\ \frac{-k_2-k_3}{ms} & \frac{k_3}{ms} & \frac{k_2}{ms} & 0 & -\frac{c_3}{ms} & \frac{c_3}{ms} & 0 \\ \frac{k_3-l_d \left(\frac{-k_2-k_3}{m_s} - \frac{k_2+l_d(-k_2-k_3)}{(m_u+l_d)} \right)}{m_s} & \frac{-l_d \frac{k_3}{m_s}}{m_s - (m_u+l_d)} & \frac{-l_d \left(\frac{k_3}{m_s} - \frac{(-k_2-k_1)+l_d \frac{k_2}{m_s}}{(m_u+l_d)} \right)}{m_r} & \frac{\left(l_d \frac{k_1}{(m_u+l_d)} \right)}{m_r} & \frac{c_3-l_d \left(\frac{-c_3}{m_s} - \frac{l_d \frac{c_3}{m_s}}{(m_u+l_d)} \right)}{m_r} & \frac{-c_3-l_d \left(\frac{c_3}{m_s} - \frac{l_d \frac{c_3}{m_s}}{(m_u+l_d)} \right)}{m_r} & 0 \\ \frac{m_r}{(m_u+l_d)} & \frac{l_d \frac{k_3}{m_s}}{(m_u+l_d)} & \frac{m_r}{(m_u+l_d)} & \frac{k_1}{(m_u+l_d)} & \frac{l_d \frac{-c_3}{m_s}}{(m_u+l_d)} & \frac{l_d \frac{c_3}{m_s}}{(m_u+l_d)} & 0 \end{bmatrix} * \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \\ x_5 \\ x_6 \\ x_7 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 1 \\ 0 \\ 0 \\ 0 \end{bmatrix} * w + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ m_u \\ \frac{m_u}{m_r(m_u+l_d)} \\ -1 \end{bmatrix} * u$$

$$y = \begin{bmatrix} -k_2 - k_3 & \frac{k_3}{m_s} & \frac{k_2}{m_s} & 0 & \frac{-c_3}{m_s} & \frac{c_3}{m_s} & 0 \\ \frac{m_s}{1} & 0 & -1 & 0 & 0 & 0 & 0 \end{bmatrix} * \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \\ x_5 \\ x_6 \\ x_7 \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix} * \begin{bmatrix} w \\ u \end{bmatrix}$$