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ACTIVE SUSPENSION CONTROL OF EIGHT DEGREES OF FREEDOM VEHICLE MODEL

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Abstract - In this study, the dynamic behavior of eight degrees of freedom vehicle model having active suspensions and a PID controlled passenger seat is examined. The suspensions considered are Mc Pherson strut type of independent suspensions. Three cases of control strategies are taken into account. First, only the passenger seat is controlled. Second, only the vehicle body is controlled. Third, both the vehicle body and the passenger seat are controlled at the same time. PID controller is applied as a conventional method. Since this control method can be applied easily and is widely well known, it has an important place in control applications. The time and frequency responses of the vehicle model due to road disturbance are obtained for each control strategy. At the end of the study, performances of these strategies have been compared and discussed.

Keywords - PID Control, Vehicle Model, Active Suspensions

1. INTRODUCTION

A systematic treatment of the vehicle as a dynamic system best starts with the basic properties of a vehicle on its suspensions. The response of the vehicle to various inputs and disturbances can be simulated. They isolate the chassis and the passengers from roughness on the road to have a better ride. Due to the developments in the control technology, electronically controlled suspensions have gained more interest. These suspensions have active components, which are controlled by a microprocessor. By using this arrangement, significant achievements in vehicle response can be obtained. The control method being used is also important during the design process. In this study PID controllers parallel to Mc Pherson strut type independent suspensions are used. The major advantages of this control method are its relative simplicity and widely well known. For simplification of model, a number of researchers assumed vehicle models as linear.

In the last decade many researchers applied some linear and non-linear control methods to the vehicle models. Because of simplicity, quarter car models were mostly preferred. Redfield and Karnopp [1] examined the optimal performance comparisons of variable component suspensions on a quarter car model. Yue *et al.* [2] applied some linear control methods on a quarter car model. Stein and Ballo [3] designed a passenger seat for off-road vehicles having active suspensions. Hac [4] applied optimal linear preview control on the active suspensions of a quarter car model. Rakheja *et al.* [5] added a passenger seat in their analysis. Passenger seat suspension system was described by a generalized two degrees of freedom model and with non-linearities such as shock absorber damping, linkage friction and bump stops. Since the quarter car model is insufficient to give information about the angular motions of the vehicle, some of the researchers used more complex models like half and full car models. These models give information about pitch, roll and bounce motions of the vehicle body. Crolla and Abdel-Hady [6] compared some active suspension control methods on a full car model. Integrated or filtered white noise was taken as the road input. The same researchers applied linear optimal control law to a similar model in 1992. Hrovat [7] compared the performances of active and passive suspension systems on quarter, half and full car models using optimal control laws. Alleyne *et al.* [8] compared sliding mode controlled active suspensions with PID controlled active suspensions. Yagiz *et al.* [9] applied sliding mode controlled active suspensions for a linear seven degrees of freedom vehicle model. In this study, the aim is to compare some control strategies applied on a full car model in order to obtain ride comfort using PID control. A passenger seat is added to the vehicle model so that the response of the passenger due to a road disturbance can be observed. There are three strategies. The first strategy includes conventional suspensions and controlled passenger seat. In the second strategy, the model has active suspensions and a normal passenger seat. The last one is fully controlled. In other words both the suspensions and passenger seat have controllers.

2. VEHICLE MODEL

The full car model used in this study is shown in Fig. 1. It includes all of the possible control strategies. This full car model has 8 degrees of freedom, namely x_1, x_2, x_3, x_4, x_5, x_6, $x_7=\theta$, $x_8=\alpha$. These are the motion of the right front axle, the motion of the left front axle, the motion of the right rear axle, the motion of the left rear axle, the bounce motion of the passenger seat, the bounce motion of the vehicle body, the pitch motion of the vehicle body and the roll motion of the vehicle body, respectively. The aim is to improve the ride comfort of the passengers.

In general, the state-space form of a linear dynamic system can be written as follows:

$$\underline{\dot{x}}=\underline{f}(x)+[B]\underline{u} \tag{1}$$

Here $\underline{x} = [x_1\ x_2\ x_3\ \ldots.\ x_{16}]^T$ where $x_9=\dot{x}_1$, $x_{10}=\dot{x}_2$ and so on. $\underline{f}(x)$ is vector functions composed of first order differential equations, [B] is the controller coefficient matrix and $\underline{u} = [u_1\ u_2\ u_3\ u_4\ u_5]^T$ is the control input vector written for the most general case in this study. $\underline{f}(x)$ and [B] are given in the Appendix with nomenclature of vehicle parameters. Mathematically, u_1, u_2, u_3 and u_4 do not have to exist together. In order to control the vehicle body motions, three controller forces are enough since the body has three degrees of freedom in this study. These are bounce, pitch and roll motions. But for practical reasons, four controllers parallel to suspensions are introduced. The yaw motion is neglected.

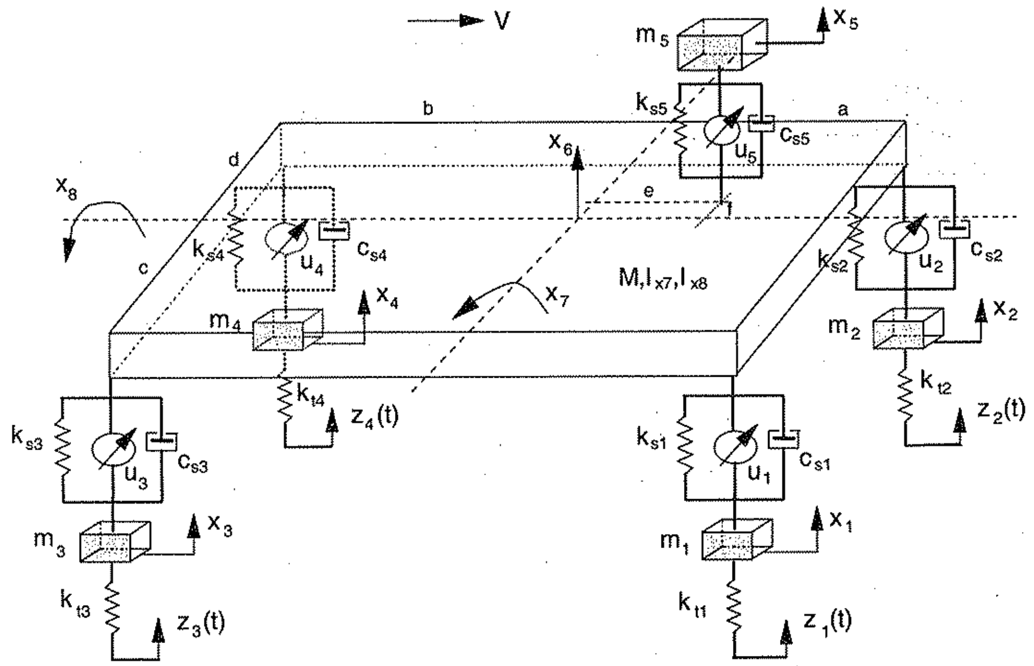

Figure 1. The full car model with a passenger seat

3. PID CONTROLLER DESIGN

In general the closed loop diagram of the feedback system is shown in Fig. 2.

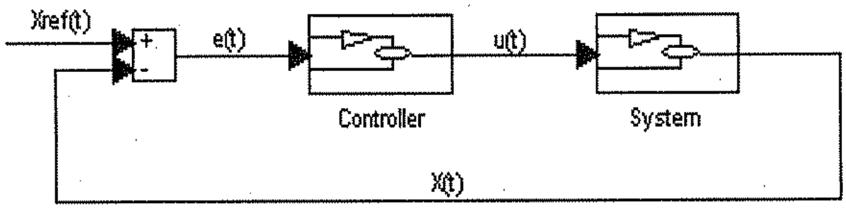

Figure 2. Closed loop block diagram with controller

Here $x_{ref}(t)$ is the desired value for the output of the system. x(t) is the output, e(t) is error and u(t) is the control signal. PID control has been used in industry widely and successfully. The control input u(t) is obtained as follows:

$$u(t) = K\,[e(t) + \frac{1}{\tau_i}\int_0^t e(t)dt + \tau_d \frac{de(t)}{dt}] \qquad (2)$$

K, τ_i and τ_d are proportionality constant, integral time and derivative time, respectively. These values are obtained using Ziegler-Nichols method [10].

4. SIMULATION

In the simulation stage, the time and frequency responses are obtained. The vehicle is assumed to travel over the bump shown in Fig. 3. The bump parameters are presented in the Appendix.

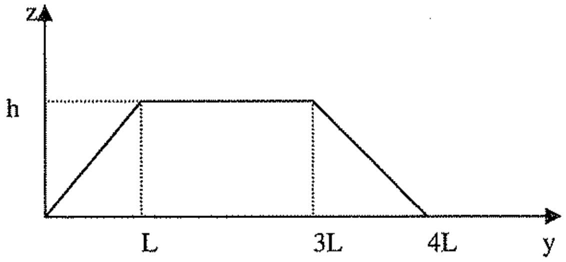

Figure 3. Road disturbance

There is a time delay between the front and rear wheel inputs. This time delay is as,

$$\delta t = (a+b)/V \quad (3)$$

(a+b) is the distance between front and rear axles and V is the velocity of the vehicle. Vehicle body acceleration and displacements for the road surface input is shown in Fig. 4. Figure 4 displays the responses of the passive and the active systems. It could be seen that maximum vertical acceleration and displacements of the vehicle body in the active system are less than those of the passive system and the active system returns rest faster. Sprung mass pitch and roll angular acceleration and displacements for controlled and uncontrolled cases are shown in Fig. 4, respectively. The success of the controller is obvious when both body acceleration and its displacements and angular accelerations and their displacements are considered.

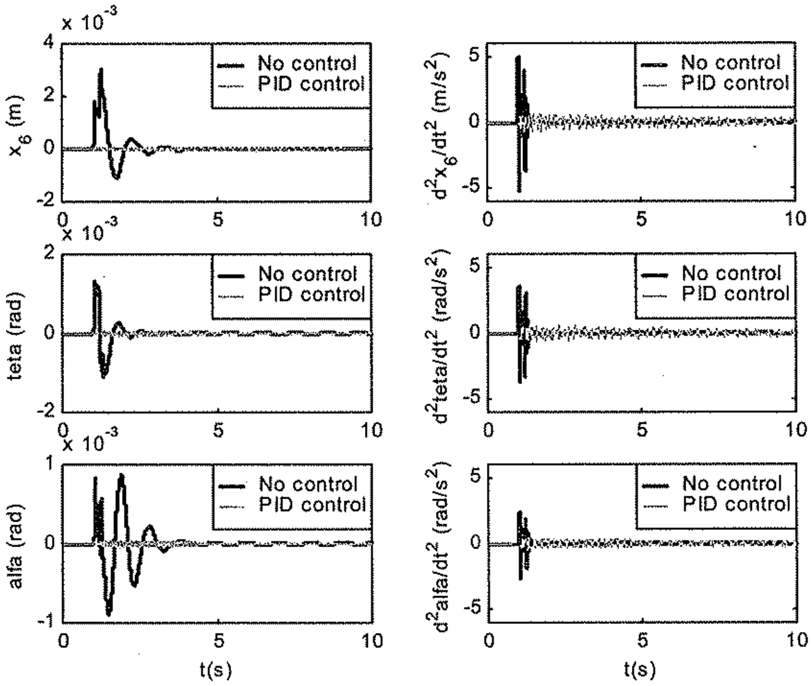

Figure 4. Time responses of vehicle body displacement and its accelerations, pitch and roll angular displacement and their accelerations for controlled and uncontrolled cases

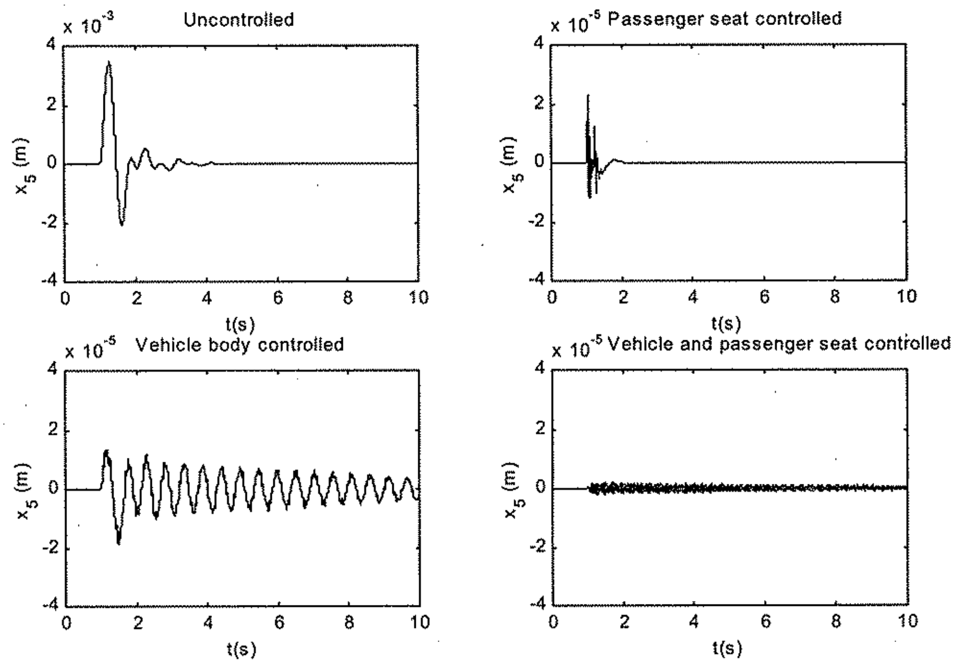

Figure 5. Time responses of the passenger displacement

As shown in Fig. 5, the improvement obtained as a result of application of active suspensions is clearly seen. For the third case, in other words when vehicle body and passenger seat are controlled at the same time, the passenger is almost insensitive to the disturbance. This method with selected strategy is seen to be very effective.

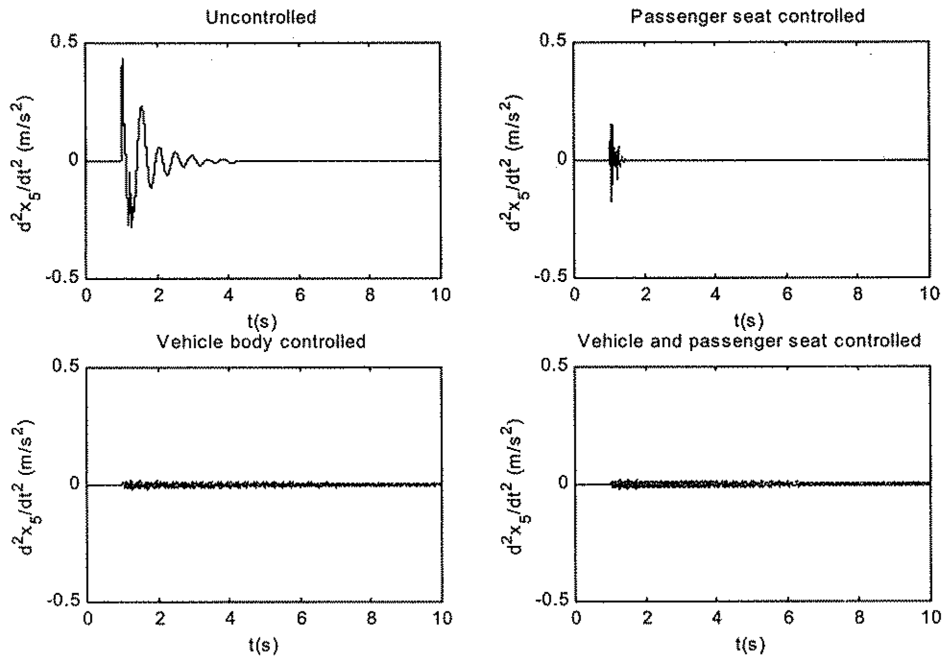

Figure 6. Time responses of the passenger vertical acceleration

The vertical acceleration of the passenger is also an important criterion, which mainly effects ride comfort since the force generated by the inertia of the passenger is disturbing. In other words, minimizing the vertical displacement may not mean an improvement only by itself, but an improvement in the acceleration should also be obtained. In Fig. 6, the acceleration of the passenger for the different cases has been shown. It is seen that the PID controller decreases the amplitude of the acceleration in all three cases when compared with the uncontrolled one. The most suitable strategy is seen to be the third one.

Another criterion is the control forces used since it is directly related with the cost of the controller. Fig. 7 shows the controller force inputs for the selected strategies. In the first case, using a maximum control force of 60 N can damp the passenger's vertical displacement. But no comment can be made for the angular displacement of the passenger in this case. In the third case, the front and rear suspensions apply about a maximum force of 5000 N. The amount of the force applied to the passenger seat decreases since the body is controlled and the passenger seat is a little isolated already. A 3 N maximum force is sufficient to bring the passenger to the reference value of zero displacement.

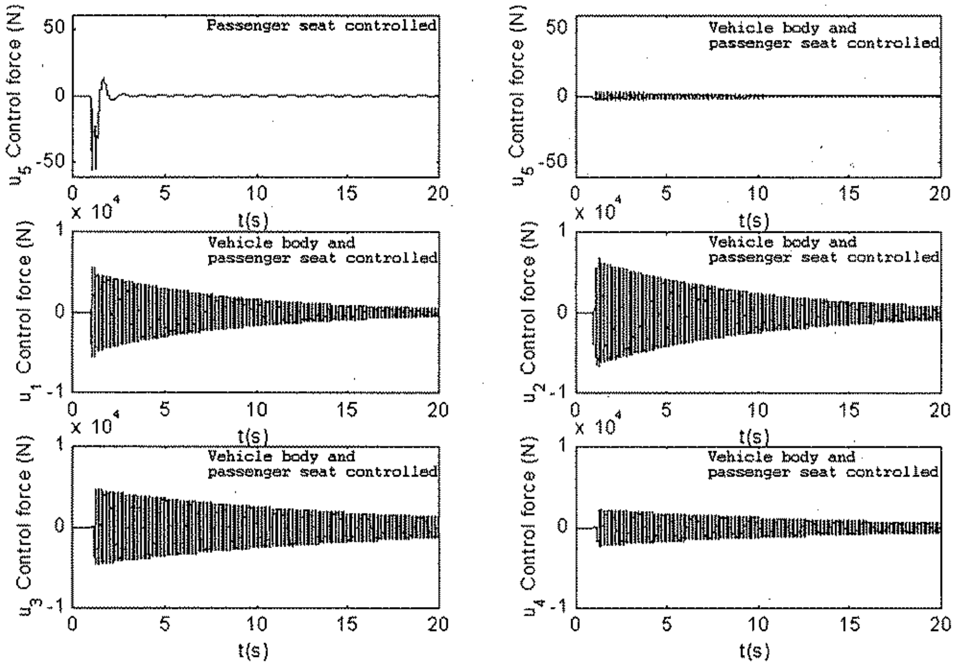

Figure 7. Control force inputs for the strategies

5. FREQUENCY RESPONSE OF THE VEHICLE MODEL

Frequency response is another key to understand the behaviour of a dynamic system. In Fig. 8, frequency response plots of the passenger seat displacements and vertical accelerations for all strategies are considered.

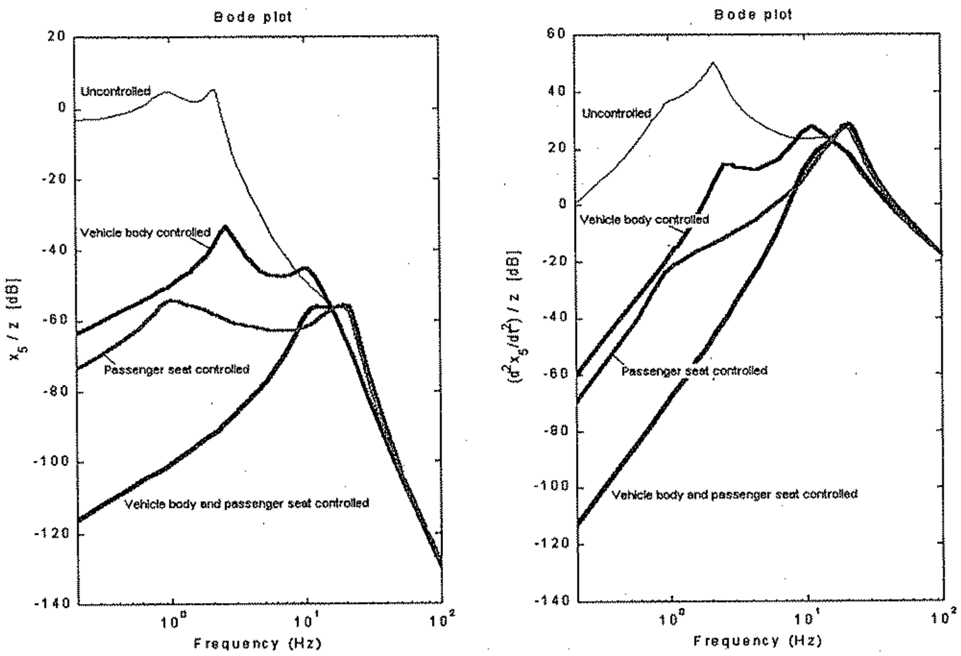

Figure 8. Frequency response plots of passenger displacements and accelerations

Three resonance frequencies are clearly observed in the uncontrolled case. These belong to vehicle body motions, passenger and unsprung masses, respectively. In case of all control strategies, amplitudes of resonance frequencies of almost all degrees of freedom decrease, the only exception being in unsprung mass resonances. The overall magnitudes also decrease for the controlled cases. The improvement observed in the unsprung mass resonance frequencies is not very effective as the ones of the others. The reason is, the controllers only work on the vehicle body and the passenger seat. The third case gives the maximum displacement and acceleration isolation for the passenger as shown in Fig. 8. The best improvement in terms of ride comfort is obtained when both vehicle body and passenger seat are under control action.

6. CONCLUSION

The main idea behind proposing this controller is ability of using these types of controllers on vehicles with developing technology. PID control which is easy to design and good in performance has been applied. The simulation results prove that, among the three control strategies considered, using controllers under vehicle body and passenger seat will provide the best ride comfort.

The first case where only having controller under passenger seat can not guarantee the ride comfort since it does not have control over angular motions of the vehicle and passenger. The second case where only vehicle body is under control, the logic controls the vehicle motions sufficiently but could not supply a good control for the ride comfort of the passenger as the third strategy does in which both vehicle body and passenger seat have controllers. Using the last strategy, the bounce motion of the passenger almost vanishes with an extra controller that applies very small force input. A successful improvement has also been obtained in the isolation of the vertical displacement and acceleration of the passenger. Frequency response plots of the passenger for these alternatives support the results obtained. As a result, adding a controller under the passenger seat improves the ride comfort greatly.

REFERENCES

1. R. C. Redfield and D. C. Karnopp, Optimal Performance of Variable Component Suspensions, *Vehicle System Dynamics* **17**, 231-253, 1998.
2. C. Yue, T. Butsuen and J. K. Hedrick, Alternative Control Laws for Automotive Active Suspensions, *Journal of Dynamic Systems, Measurement, and Control* **111**, 286-291, 1989.
3. G. J. Stein and I. Ballo, Active Vibration Control System for the Driver's Seat for Off-Road Vehicles, *Vehicle System Dynamics* **20**, 57-78, 1991.
4. A. Hac, Optimal Linear Preview Control of Active Vehicle Suspension, *Vehicle System Dynamics* **21**, 167-195, 1992.
5. S. Rakheja, Y. Afework and S. Sankar, An Analytical and Experimental Investigation of the Driver-Seat-Suspension System, *Vehicle System Dynamics* **23**, 501-524, 1994.
6. D. A. Crolla and M. B. A. Abdel Hady, Active Suspension Control; Performance Comparisons Using Control Laws Applied to a Full Car Model, *Vehicle System Dynamics* **20**, 107-120, 1991.
7. D. Hrovat, Applications of Optimal Control to Advanced Automotive Suspension Design, *Journal of Dynamic Systems, Measurement, and Control* **115**, 328-342, 1993.
8. A. Alleyne, P. D. Neuhaus and J. K. Hedrick, Application of Non-linear Control Theory to Electronically Controlled Suspensions, *Vehicle System Dynamics* **22**, 309-320, 1993.
9. N. Yagiz, I. Yuksek and S. Sivrioglu, Robust Control of Active Suspensions for a Full Vehicle Model Using Sliding Mode Control, *JSME International Journal C* **43**, No.2, 253-258, 2000.
10. K. Ogata, *Modern Control Engineering*, Prentice-Hall, New Jersey, 595-605, 1990.

LIST OF SYMBOLS

M : mass of the vehicle body.
I_{x7} : mass moment of inertia of the vehicle body for pitch motion.
I_{x8} : mass moment of inertia of the vehicle body for roll motion.
m_1 : mass of the right front axle.
m_2 : mass of the left front axle.
m_3 : mass of the right rear axle.
m_4 : mass of the left rear axle.
m_5 : mass of the passenger.

c_{s1} : damping coefficient of right front suspension.
c_{s2} : damping coefficient of left front suspension.
c_{s3} : damping coefficient of right rear suspension.
c_{s4} : damping coefficient of left rear suspension.
c_{s5} : damping coefficient of passenger seat.
k_{s1} : spring constant of right front suspension.
k_{s2} : spring constant of left front suspension.
k_{s3} : spring constant of right rear suspension.
k_{s4} : spring constant of left rear suspension.
k_{s5} : spring constant of passenger seat.
$z_i\,(t)$: i^{th} road excitation.
k_{t1} : stiffness coefficient of right front tire.
k_{t2} : stiffness coefficient of left front tire.
k_{t3} : stiffness coefficient of right rear tire.
k_{t4} : stiffness coefficient of left rear tire.
a , b : distances of axle to the center of gravity of the vehicle body.
c , d : distances of unsprung masses to the center of gravity of the axles.
e , f : distances of passenger seat to the center of gravity of the vehicle body.
x_i : i^{th} state variable.

APPENDIX

The parameters of the vehicle:
M=1100 kg, I_{x7}=1848 kg.m^2, I_{x8}=550 kg.m^2, m_1=m_2=25 kg, m_3=m_4= 45 kg, m_5= 90 kg
$k_{s1} = k_{s2}$ = 15000 N/m, $k_{s3} = k_{s4}$ = 17000 N/m, k_{s5} = 15000 N/m, $k_{t1} = k_{t2} = k_{t3} = k_{t4}$ = 250000 N/m
c_{s1}=c_{s2}=c_{s3}=c_{s4}=2500 N.s/m, c_{s5}=150 N.s/m, a=1.2 m, b=1.4 m, c=0.5 m, d=1.0 m, e=0.3 m, f=0.25 m

The parameters of the road bump:
h = 0.035 m, L = 0.025 m

The controller force matrix:

$$[B]=\begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ -\frac{bd}{(a+b)(c+d)m_1} & -\frac{d}{(a+b)(c+d)m_1} & \frac{1}{(c+d)m_1} & \frac{-d(b+e)+f(a+b)}{(a+b)(c+d)m_1} \\ -\frac{bc}{(a+b)(c+d)m_2} & -\frac{c}{(a+b)(c+d)m_2} & -\frac{1}{(c+d)m_2} & \frac{-c(b+e)-f(a+b)}{(a+b)(c+d)m_2} \\ -\frac{ad}{(a+b)(c+d)m_3} & \frac{d}{(a+b)(c+d)m_3} & 0 & -\frac{d(a-e)}{(a+b)(c+d)m_3} \\ -\frac{ac}{(a+b)(c+d)m_4} & \frac{c}{(a+b)(c+d)m_4} & 0 & -\frac{c(a-e)}{(a+b)(c+d)m_4} \\ 0 & 0 & 0 & \frac{1}{m_5} \\ \frac{1}{M} & 0 & 0 & 0 \\ 0 & \frac{1}{I_{x7}} & 0 & 0 \\ 0 & 0 & \frac{1}{I_{x8}} & 0 \end{bmatrix}_{16\times 4}$$

The $\underline{f}(x)$ of the linear state space model:

$$
\underline{f}(x)=\begin{bmatrix}
x_9 \\
x_{10} \\
x_{11} \\
x_{12} \\
x_{13} \\
x_{14} \\
x_{15} \\
x_{16} \\
\left(\begin{array}{l} -(k_{s1}+k_{t1})x_1+k_{s1}x_6+ak_{s1}\mathrm{Sin}x_7-ck_{s1}\mathrm{Sin}x_8-c_{s1}x_9+c_{s1}x_{14} \\ +ac_{s1}\mathrm{Cos}x_7\,x_{15}-cc_{s1}\mathrm{Cos}x_8\,x_{16}+k_{t1}z_1 \end{array}\right)/m_1 \\
\left(\begin{array}{l} -(k_{s2}+k_{t2})x_2+k_{s2}x_6+ak_{s2}\mathrm{Sin}x_7+dk_{s2}\mathrm{Sin}x_8-c_{s2}x_{10}+c_{s2}x_{14} \\ +ac_{s2}\mathrm{Cos}x_7\,x_{15}+dc_{s2}\mathrm{Cos}x_8\,x_{16}+k_{t2}z_2 \end{array}\right)/m_2 \\
\left(\begin{array}{l} -(k_{s3}+k_{t3})x_3+k_{s3}x_6-bk_{s3}\mathrm{Sin}x_7-ck_{s3}\mathrm{Sin}x_8-c_{s3}x_{11}+c_{s3}x_{14} \\ -bc_{s3}\mathrm{Cos}x_7\,x_{15}-cc_{s3}\mathrm{Cos}x_8\,x_{16}+k_{t3}z_3 \end{array}\right)/m_3 \\
\left(\begin{array}{l} -(k_{s4}+k_{t4})x_4+k_{s4}x_6-bk_{s4}\mathrm{Sin}x_7+dk_{s4}\mathrm{Sin}x_8-c_{s4}x_{12}+c_{s4}x_{14} \\ -bc_{s4}\mathrm{Cos}x_7\,x_{15}+dc_{s4}\mathrm{Cos}x_8\,x_{16}+k_{t4}z_4 \end{array}\right)/m_4 \\
\left(\begin{array}{l} -k_{s5}x_5+k_{s5}x_6+ek_{s5}\mathrm{Sin}x_7+fk_{s5}\mathrm{Sin}x_8-c_{s5}x_{14}+c_{s5}x_{14} \\ +ec_{s5}\mathrm{Cos}x_7\,x_{15}+fc_{s5}\mathrm{Cos}x_8\,x_{16} \end{array}\right)/m_5 \\
\left(\begin{array}{l} k_{s1}x_1+k_{s2}x_2+k_{s3}x_3+k_{s4}x_4+k_{s5}x_5-(k_{s1}+k_{s2}+k_{s3}+k_{s4}+k_{s5}) \\ x_6-(a(k_{s1}+k_{s2})-b(k_{s3}+k_{s4})+ek_{s5})\mathrm{Sin}x_7-(d(k_{s2}+k_{s4})- \\ c(k_{s1}+k_{s3})+fk_{s5})\mathrm{Sin}x_8+c_{s1}x_9+c_{s2}x_{10}+c_{s3}x_{11}+c_{s4}x_{12}+c_{s5}x_{13} \\ -(c_{s1}+c_{s2}+c_{s3}+c_{s4}+c_{s5})x_{14}-(a(c_{s1}+c_{s2})-b(c_{s3}+c_{s4})+ec_{s5}) \\ \mathrm{Cos}x_7x_{15}-(d(c_{s2}+c_{s4})-c(c_{s1}+c_{s3})+fc_{s5})\mathrm{Cos}x_8\,x_{16} \end{array}\right)/M \\
\left(\begin{array}{l} ak_{s1}x_1+ak_{s2}x_2-bk_{s3}x_3-bk_{s4}x_4+ek_{s5}x_5-(a(k_{s1}+k_{s2})-b(k_{s3}+k_{s4})+ek_{s5}) \\ x_6-(a^2(k_{s1}+k_{s2})+b^2(k_{s3}+k_{s4})+e^2k_{s5})\mathrm{Sin}x_7-(d(ak_{s2}-bk_{s4})- \\ c(ak_{s1}-bk_{s3})+efk_{s5})\mathrm{Sin}x_8+ac_{s1}x_9+ac_{s2}x_{10}-bc_{s3}x_{11}-bc_{s4}x_{12}+ec_{s5}x_{13} \\ -(a(c_{s1}+c_{s2})-b(c_{s3}+c_{s4})+ec_{s5})x_{14}-(a^2(c_{s1}+c_{s2})+b^2(c_{s3}+c_{s4})+e^2c_{s5}) \\ \mathrm{Cos}x_7x_{15}-(d(ac_{s2}-bc_{s4})-c(ac_{s1}-bc_{s3})+efc_{s5})\mathrm{Cos}x_8\,x_{16} \end{array}\right)\mathrm{Cos}x_7/I_{x7} \\
\left(\begin{array}{l} -ck_{s1}x_1+dk_{s2}x_2-ck_{s3}x_3+dk_{s4}x_4+fk_{s5}x_5-(d(k_{s2}+k_{s4})-c(k_{s1}+k_{s3})+fk_{s5}) \\ x_6-(d(ak_{s2}-bk_{s4})-c(ak_{s1}-bk_{s3})+efk_{s5})\mathrm{Sin}x_7-(d^2(k_{s2}+k_{s4})+c^2(k_{s1}+k_{s3}) \\ +f^2k_{s5})\mathrm{Sin}x_8-cc_{s1}x_9+dc_{s2}x_{10}-cc_{s3}x_{11}+dc_{s4}x_{12}+fc_{s5}x_{13} \\ -(d(c_{s2}+c_{s4})-c(c_{s1}+c_{s3})+fc_{s5})x_{14}-(d(ac_{s2}-bc_{s4})-c(ac_{s1}-bc_{s3})+efc_{s5}) \\ \mathrm{Cos}x_8x_{15}-(d^2(c_{s2}+c_{s4})+c^2(c_{s1}+c_{s3})+f^2c_{s5})\mathrm{Cos}x_8\,x_{16} \end{array}\right)\mathrm{Cos}x_8/I_{x8}
\end{bmatrix}
$$