Active and Passive Suspension System Performance under Random Road Profile Excitations

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Passive suspensions are designed to satisfy the conflicting criteria of riding comfort and vehicle handling. An active suspension system attempts to overcome these compromises to provide the best performance for vehicle control. Different types of mathematical models have been used to study the suspension system of a vehicle. The quarter vehicle model is used for initial investigation. Later, the half vehicle and full vehicle models are used for the study, which is closer to the actual model of a vehicle suspension. In this paper, the behavior of a suspension system is analyzed using the full vehicle model. In the current work, the dynamic equation and their state-space formulation are presented for the full vehicle model to understand the system prior to the controller design. The open-loop response of the full vehicle suspension system, when subjected to various road excitations, is also studied. The procedure of modeling a SIMULINK model for passive suspensions system is discussed in detail. Design of the simple Proportional Integral Derivative (PID) feedback and feed-forward controller is presented for the active suspension system using transfer functions. Closed-loop transfer functions are also derived and their responses are plotted. To analyze the rollover behavior simulation for cornering is also performed in the current study.

NOMENCLATURE

T	Moment about center of contact patches of tires	F_C	Centripetal force
M	Mass of vehicle	V	Speed of the vehicle
g	Gravity acceleration	R	Radius of curvature
t_w	Track width	Θ	Pitch angle
a_y	Lateral acceleration	M_{roll}	Roll moment
h_0	Height of center of gravity	ω_n	Natural frequency
F_{zi}	Total normal load	λ	Wavelength
M_s	Sprung mass of the vehicle	t_d	Time delay between the front and rear wheel
M_u	Unsprung mass of the vehicle	K_p	Proportion gain
Z_s	Displacement of sprung mass	sf	Front suspension
Z_u	Displacement of unsprung mass	sr	Rear suspension
K_s	Spring constants of suspension	fl	front-left
K_u	Spring constant of tire	fr	front-right
C_s	Damping constant of shock absorber	rl	rear-left
F	Actuator control force	rr	rear-right
Z_r	Road Disturbance	ufl	front-left wheel unsprung mass
φ	Roll angle	ufr	front-right wheel unsprung mass
W	Width of sprung mass	url	rear-left wheel unsprung mass
I_{xx}	Roll axis moment of inertia	urr	rear-right wheel unsprung mass
I_{yy}	Pitch axis moment of inertia	rfl	front-left wheel road disturbance
a	Distance between front of vehicle and center of gravity of sprung mass	rfr	front-right wheel road disturbance
b	Distance between rear of vehicle and center of gravity of sprung mass	rrl	rear-left wheel road disturbance
h_{roll}	Distance between roll center and center of gravity	rrr	rear-right wheel road disturbance

1. INTRODUCTION

A vehicle has many functional subsystems: vehicle wheels, engine, and power transmission system, braking system, suspension, steering, and many embedded electronic components. All these subsystems are designed to make the vehicles work in the best way under existing conditions. Vehicle suspension is common to almost all vehicles. The function of a suspension is to insulate the vehicle body from road troubles for a comfortable ride and improve vehicle handling. The passive suspension system consists of a spring and a damper. A passive suspension system cannot satisfy the comfort requirements when subjected to different road profiles but a semi-active suspension system and active suspension system fully satisfy the comfort requirement when subjected to different road profiles. In an active suspension and semi-active suspension systems, control strategy becomes very important.¹ The strategies of active suspension systems and semi-active suspension systems have been offered to get more efficient, better performance, and remove the restrictions of passive suspensions. Nowadays many studies have focused on better ride comfort and vehicle handling of active suspension by selecting an appropriate control strategy. In the majority of research problems, it is observed that a linearized passive suspension, active suspension, and semi-active suspension models are used. In this study, a survey on how a Proportional Integral Derivative (PID) technique can be used to improve vehicle handling and improve ride comfort for a full vehicle model is undertaken.²

The use of an active control system and semi-active control system in a vehicle suspension system has been the subject of substantial research since the late 1960s, see for example^{3–8} and the references therein. Various theoretical and experimental studies have been carried out for semi-active vehicle models and active quarter-vehicle models form^{9,10} but only a few studies have dealt with full-vehicle models. The controller design for an active suspension system of a full vehicle model has first been investigated in,⁴ followed by work undertaken in.¹¹ Some simulation studies for the semi-active full-vehicle model have been undertaken in studies conducted from.^{12–15} Applications using a neural network controller for quarter, half, and full vehicle models have been discussed by researchers from.^{16–18} Four decoupled vehicle model using skyhook controllers has been investigated between the work from.^{19–21} A Nonlinear Filter based observer structure has been proposed in,²² which takes the nonlinearities of the damper into account. Magnetorheological dampers have been investigated through research work carried out from,^{23–26} which is a feedback control for a half and full-vehicle suspension system. An active suspension for the quarter, half, and full vehicle model was developed in the studies conducted from.^{27–29} The improved performance of the above models using a PID controller has been described in.³⁰ The full vehicle suspensions for passive and active models with PI controls have been discussed in³¹ and PID controls in³² were simulated using MATLAB/SIMULINK.

Table 1. Parameter values for full vehicle suspension.³³

Parameters	Value
Mass of vehicle body (sprung mass), M_s (kg)	1500
Roll axis moment of inertia of vehicle body, I_{xx} (kg-m ²)	460
Pitch axis moment of inertia of vehicle body, I_{yy} (kg-m ²)	2160
Mass of wheel (unsprung mass), M_u (kg)	59
Stiffness of vehicle body suspension spring for front, K_{sf} (N/m)	35000
Stiffness of vehicle body suspension spring for rear, K_{sr} (N/m)	38000
Tire spring stiffness, K_u (N/m)	190000
Front suspension damping, C_{sf} (N-s/m)	1000
Rear suspension damping, C_{sr} (N-s/m)	1100
Distance between front of vehicle and C.G. of sprung mass, a (m)	1.4
Distance between rear of vehicle and C.G. of sprung mass, b (m)	1.7
Width of sprung mass, w (m)	1.524
C.G. height, h_0 (m)	0.508
Distance between roll center and C.G. h_{roll} (m)	.3

Figure 1. Active suspension model for full-vehicle system.

2. MATHEMATICAL MODELING FOR FULL VEHICLE MODEL

The model of the full-vehicle suspension framework is given in Fig. 1. The full-vehicle suspension model was characterized as a linearized seven degree-of-freedom (7-DOF) system.³ It consisted of a single sprung mass (vehicle body) linked to four unsprung masses (front-left wheels, front-right wheels, rear-left wheels, and rear-right wheels) at each corner. The vehicle body mass or sprung mass was free to heave, pitch and roll while the unsprung masses were free to bounce vertically with respect to the sprung mass. The suspensions between sprung mass and unsprung masses were modeled as linear viscous dampers and spring elements, while the tires were considered as simple linear springs without viscous damping. For simplicity, all pitch and roll angles were assumed to be small. The mathematical modeling of the full vehicle suspension system was represented in the form of a dynamic equation followed by its space formulation. The parameters used in the full vehicle suspension system of this study are given in Table 1.

The dynamic equation of motion of the full-vehicle system was given by

$$M_s \ddot{Z} = -(2K_{sf} + 2K_{sr})Z - (2C_{sf} + 2C_{sr})\dot{Z} + (2aK_{sf} - 2bK_{sr})\theta + (2aC_{sf} - 2bC_{sr})\dot{\theta}$$

$$\begin{aligned}
& + K_{sf}Z_{ufl} + C_{sf}\dot{Z}_{ufl} + K_{sf}Z_{ufr} + C_{sf}\dot{Z}_{ufr} \\
& + K_{sr}Z_{url} + C_{sr}\dot{Z}_{url} + K_{sr}Z_{urr} + C_{sr}\dot{Z}_{urr} \\
& + F_{fl} + F_{fr} + F_{rl} + F_{rr}, \quad (1)
\end{aligned}$$

$$\begin{aligned}
I_{yy}\ddot{\theta} = & (2aK_{sf} - 2bK_{sr})Z + (2aC_{sf} - 2bC_{sr})\dot{Z} \\
& - (2a^2K_{sf} - 2b^2K_{sr})\theta - (2a^2C_{sf} - 2b^2C_{sr})\dot{\theta} \\
& - aK_{sf}Z_{ufl} - aC_{sf}\dot{Z}_{ufl}, \quad (2)
\end{aligned}$$

$$\begin{aligned}
I_{xx}\ddot{\varphi} = & -0.25w^2(2K_{sf} + 2K_{sr})\varphi - 0.25w^2(2C_{sf} + 2C_{sr})\dot{\varphi} \\
& + 0.5wK_{sf}Z_{ufl} + 0.5wC_{sf}\dot{Z}_{ufl} - 0.5wK_{sf}Z_{ufr} \\
& - 0.5wC_{sf}\dot{Z}_{ufr} + 0.5wK_{sr}Z_{url} + 0.5wC_{sr}\dot{Z}_{url} \\
& - 0.5wK_{sr}Z_{urr} - 0.5wC_{sr}\dot{Z}_{urr} + 0.5wF_{fl} \\
& - 0.5wF_{fr} + 0.5wF_{rl} - 0.5wF_{rr}, \quad (3)
\end{aligned}$$

$$\begin{aligned}
M_u\ddot{Z}_{ufl} = & K_{sf}Z + C_{sf}\dot{Z} - aK_{sf}\theta - aC_{sf}\dot{\theta} + 0.5wK_{sf}\varphi \\
& + 0.5wC_{sf}\dot{\varphi} - (K_{sf} + K_u)Z_{ufl} - C_{sf}\dot{Z}_{ufl} \\
& + K_uZ_{rfl} - F_{fl}, \quad (4)
\end{aligned}$$

$$\begin{aligned}
M_u\ddot{Z}_{ufr} = & K_{sf}Z + C_{sf}\dot{Z} - aK_{sf}\theta - aC_{sf}\dot{\theta} - 0.5wK_{sf}\varphi \\
& - 0.5wC_{sf}\dot{\varphi} - (K_{sf} + K_u)Z_{ufr} - C_{sf}\dot{Z}_{ufr} \\
& + K_uZ_{rfr} - F_{fr}, \quad (5)
\end{aligned}$$

$$\begin{aligned}
M_u\ddot{Z}_{url} = & K_{sr}Z + C_{sr}\dot{Z} + bK_{sr}\theta + bC_{sr}\dot{\theta} + 0.5wK_{sr}\varphi \\
& + 0.5wC_{sr}\dot{\varphi} - (K_{sr} + K_u)Z_{url} - C_{sr}\dot{Z}_{url} \\
& + K_uZ_{rrl} - F_{rl}, \quad (6)
\end{aligned}$$

$$\begin{aligned}
M_u\ddot{Z}_{urr} = & K_{sr}Z + C_{sr}\dot{Z} + bK_{sr}\theta + bC_{sr}\dot{\theta} + 0.5wK_{sr}\varphi \\
& - 0.5wC_{sr}\dot{\varphi} - (K_{sr} + K_u)Z_{urr} - C_{sr}\dot{Z}_{urr} \\
& + K_uZ_{rrr} - F_{rr}, \quad (7)
\end{aligned}$$

2.1. Mathematical Modeling for Rollover

When a vehicle is running on a curved road, centripetal force acts at the vehicle's center of gravity given by Eq. (8)

$$F_c = \frac{MV^2}{R}, \quad (8)$$

where F_c , M , and V were centripetal force, the total mass of the vehicle, and vehicle speed, respectively. R was the radius of curvature of the road. This centripetal force tends to roll the vehicle over, which should be avoided. In this study an attempt was made to analyze the rollover dynamics, thus an extra input to the vehicle was given in addition to four wheels road input disturbances (as the road is being made to move rather than the vehicle). For creating a situation where the vehicle is running on a curved road and to incorporate rollover, a centripetal force was applied as input to the system model. The roll moment acting on the vehicle due to this centripetal force was given by Eq. (9)

$$M_{roll} = F_ch_{roll}, \quad (9)$$

$$\begin{aligned}
I_{xx}\ddot{\varphi} = & -0.25w^2(2K_{sf} + 2K_{sr})\varphi - 0.25w^2(2C_{sf} + 2C_{sr})\dot{\varphi} \\
& + 0.5wK_{sf}Z_{ufl} + 0.5wC_{sf}\dot{Z}_{ufl} - 0.5wK_{sf}Z_{ufr} \\
& - 0.5wC_{sf}\dot{Z}_{ufr} + 0.5wK_{sr}Z_{url} + 0.5wC_{sr}\dot{Z}_{url} \\
& - 0.5wK_{sr}Z_{urr} - 0.5wC_{sr}\dot{Z}_{urr} + 0.5wF_{fl} \\
& - 0.5wF_{fr} + 0.5wF_{rl} - 0.5wF_{rr} + M_{roll}, \quad (10)
\end{aligned}$$

where M_{roll} was the rolling moment acting on the vehicle and h_{roll} was the distance between the center of gravity of the vehicle and roll center. The rolling moment input was applied to the Eq. (3) and the converted equation was given by Eq. (10).

2.2. State Space Formulation of the Full Vehicle Model

The state-space variables for the full vehicle suspension model are assigned as in.⁷

Nomenclature

$y_1 = \dot{Z}$;	velocity (payload speed of sprung mass)
$y_2 = \dot{\theta}$;	pitch angular velocity
$y_3 = \dot{\varphi}$;	roll angular velocity
$y_4 = \dot{Z}_{ufl}$;	left-front wheel unsprung mass speed
$y_5 = \dot{Z}_{ufr}$;	right-front wheel unsprung mass speed
$y_6 = \dot{Z}_{url}$;	left-rear wheel unsprung mass speed
$y_7 = \dot{Z}_{urr}$;	right-rear wheel unsprung mass speed
$y_8 = Z$;	heave position (ride height of sprung mass)
$y_9 = \theta$;	pitch angle
$y_{10} = \varphi$;	roll angle
$y_{11} = Z_{ufl}$;	left-front wheel unsprung mass displacement
$y_{12} = Z_{ufr}$;	right-front wheel unsprung mass displacement
$y_{13} = Z_{url}$;	left-rear wheel unsprung mass displacement
$y_{14} = Z_{urr}$;	right-rear wheel unsprung mass displacement

The state-space equation was

$$\{\ddot{y}\} = A\{y\} + B\{f\} + D\{r\}, \quad (11)$$

where $\{y\} = [y_1 \ y_2 \ y_3 \ y_4 \ y_5 \ y_6 \ y_7 \ y_8 \ y_9 \ y_{10} \ y_{11} \ y_{12} \ y_{13} \ y_{14}]^T$ was the state vector; $\{r\} = [Z_{rfl} \ Z_{rfr} \ Z_{rrl} \ Z_{rrr} \ M_{roll}]^T$ was the road disturbance input vector; $\{f\} = [F_{fl} \ F_{fr} \ F_{rl} \ F_{rr}]^T$ was the control force input vector; A, B, and D were invariant coefficient.

3. ROAD PROFILE

Road profile irregularities were categorised as being smooth, rough minor, or rough. A smooth road profile signifies a road disturbance with a single bump. The rough minor and rough road profiles were characterized by uniform bump height and non-uniform bump height, respectively.³

$$Z_{rf} = \begin{cases} \frac{a_0}{2} \left(1 - \cos \left(\frac{2\pi V t_f}{\lambda_0} \right) \right) & 1 \leq t_f \leq 1 + \frac{\lambda_0}{V} \\ 0 & \text{otherwise} \end{cases}, \quad (12)$$

$$Z_{rr} = \begin{cases} \frac{a_0}{2} \left(1 - \cos \left(\frac{2\pi V t_r}{\lambda_0} \right) \right) & t_{r0} \leq t_r \leq t_{r0} + \frac{\lambda_0}{V} \\ 0 & \text{otherwise} \end{cases}; \quad (13)$$

Figure 2. Sinusoidal bump road input disturbance time histories for the front and rear wheels.

Figure 3. Vertical bounce with a passive suspension system.

In this study, two types of road input excitation were given namely left wheel input and a speed breaker respectively to study the behavior. For the left wheel input, a single bump was given to the front-left and rear tires. For the speed breaker input, either the front or rear tires hit the bump. A graphical representation of the road profile is given by Fig. 2. The profile road bump inputs for the front wheels and rear wheels were given by the following equations.⁴ Where a_0 was the bump amplitude, V was the vehicle forward velocity, λ_0 was the disturbance wavelength, t was the simulation time and subscripts rf and rr denote the road-front and road-rear wheel inputs to the suspension respectively. $tr_0 = t + t_d$ where t_d was the time delay between the front and rear wheel, written as:

$$t_d = \frac{a + b}{V}, \quad (14)$$

where a was the distance between the front of the vehicle and center of gravity of sprung mass and b was the distance between the rear of the vehicle and center of gravity of sprung mass.

For this study $a_0 = 0.075\text{m}$; $\lambda_0 = 0.775\text{m}$ and $V = 15.5\text{m/s}$.

Various parameters have a strong effect on the performance of the active suspension system when this concept is brought into realization. In this study, to analyze the impact of the parameters like suspension stiffness, static stability factor, time delay and discrete-time sampling on the active suspension system were presented.

4. RESULT AND DISCUSSION

The responses of the passive vehicle suspension system are obtained and compared for the two inputs. From Fig. 3 and

Figure 4. Pitching response with a passive suspension system.

Figure 5. Rolling response with a passive suspension system.

Fig. 4, we can see that the amplitude of the response for the vertical bounce of the vehicle and the pitch motion are more in speed breaker input than for the left wheel input. This is because for the speed breaker input, both front-wheel tires meet the bump at the same time, hence the vertical altitude and pitch angle are greater. Fig. 5 shows that there is no roll motion when there is a speed breaker input because both the wheels of the front and rear suspension hit the bump at the same time, but when a bump is applied only to the left wheels, the left side of the vehicle is lifted and a rolling motion is seen as expected to occur.

Figure 6 displays the vertical movement of the front and rear wheel's response. The vertical amplitude of motion of the rear wheel is higher than that for the front wheel because the rear suspension is harder. The wheel displacement plot is used to analyze the tire-to-road surface contact. Figs. 7 and 8 show the difference between the wheel's vertical motion and the road irregularities for the front and rear suspension when the road bump input is given only to left tires.

A positive value indicates that there is a gap between the tire

Figure 6. Vertical displacement of front and rear wheels.

Figure 7. Vertical displacement of front and rear wheels.

Figure 8. Vertical displacement of front and rear wheels.

and the road, a negative sign indicates that the tire is 'bitten' by the road surface. For good vehicle handling the contact should be continuous and the difference must be closest to zero.

4.1. Design of PID Controller for Active Suspension System

In the current work, the Ziegler-Nichols method is used to design the PID controllers. Transfer functions which are used for obtaining the critical gain (K_{cr}) and the critical time period (P_{cr}), and using these, the parameters of the PID controller are calculated.

4.2. Single Input Single Output System

The transfer function for the system between the road disturbance applied to the front left tire and the body bounce is given as

$$G_{11}(s) = \frac{Z_s(s)}{Z_{rfl}(s)} = \frac{2147s^{11} + 2.145e5s^{10} + 3.344e7s^9 + 2.2e8s^8 + 1.61e11s^7 + 7.07e12s^6 + 2.69e14s^5 + 7.495e4s^4 + 1e17s^3 + 1.707e18s^2 + 9.291e18s + 1.022e20}{s^{14} + 84.05s^{13} + 1.844e12s^{12} + 1.073e6s^{11} + 1.18e8s^{10} + 4.7e9s^9 + 3.27e11s^8 + 8.06e12s^7 + 3.63e14s^6 + 4.314e15s^5 + 1.075e17s^4 + 7.074e3s^3 + 1.13e19s^2 + 3.389e19s + 3.727e20}$$

(15)

Figure 9. Bode plot for open loop transfer function $G_{11}(s)$.

Figure 10. Close loop unit step input ($K_p = 25.409$).

The Bode plot of this open-loop transfer function $G_{11}(s)$ is given in Fig. 9. The critical gain and frequency are given as

$$GM = -20\log_{10} K_{cr} \text{ dB}; \quad K_{cr} = 25.409$$

$$\omega = 56.7 \text{ rad/s}; \quad P_{cr} = 0.1108 \text{ sec};$$

Where gain margin (GM) is vertical distance in the magnitude Bode plot in dB (decibel) from the point where the gain crosses 0dB to the point where it crosses the phase crossover frequency (frequency at which the locus in phase Bode plot intersects the -180° axis).

The closed-loop response when the PID parameters are $K_p = 25.409$; $K_i = 0$; $K_d = 0$.

Sustained oscillations occur in response to a step input as expected using the Ziegler-Nichols criterion. The PID parameters according to the Ziegler-Nichols tuning rule and the transfer function of the PID controller are $K_p = 15.245$; $K_i = 275.19$; $K_d = 0.21$;

$$U_c(s) = \frac{0.21s^2 + 15.245s + 275.19}{s} \quad (16)$$

The closed-loop transfer function is

$$CL_{11}(s) = \frac{G_{11}(s)}{1 + U_c(s)G_{11}(s)} \quad (17)$$

The (SISO) system $G_{11}(s)$ is simulated for a passive system and with a controller and the responses are plotted in Fig. 11. The road profile shown in Section 3 is given as input to the

Figure 11. Vertical amplitude vs time for the SISO system $G_{11}(s)$.

Figure 13. Vertical displacement vs. time.

Figure 12. Bode plot for closed loop transfer function $CL_{11}(s)$.

system. The values of the PID parameters that are calculated by the Ziegler-Nichols method are only an intelligent guess. A better response can be obtained by varying the parameters.

The closed loop transfer function is plotted in Fig. 12. After applying the PID controller, the gain margin decreases employing that the system becomes more stable.

4.3. Multi Input Multi Output System

For tuning the PID controller for a MIMO system, the MATLAB-SIMULINK tuner is used and PID parameters are adjusted to get the desired close loop (active) response of the system. In PID tuning, the parameters are chosen based on the optimum combination of rise time, response time, settling time and overshoot.

4.4. Response of the Active Suspension System

An active full car suspension system model simulated the same left wheel bump road disturbance input used as shown Section 3. The simulated response of the active suspension

Figure 14. Pitch angle vs. time.

system with a PID controller is compared with that of the passive system. For example, when we deal with systems where we are not bothered with the actual dynamics of how the steady state is reached, but only care about the steady state itself, a good measure will be the steady state error of the system defined by Eq. (17).

$$e = Z_{final} - Z_{ref} \quad (18)$$

For dynamic systems where the transient behavior is important, it is important to consider several other criteria other than steady-state error.

4.5. Body Attitude

The three main attitudes of an automobile body which is important for ride comfort and safety are body bounce, pitch and roll. The reduction in vertical displacement (body bounce) of the vehicle translates into an increase in the ride comfort. The body bounce response is shown in Fig. 13; we can see that there is almost a 75% reduction in vertical bounce and 68% reduction in settling time of the passive response when the controller is applied.

Pitching is one of the unpleasant body attitudes of a vehicle and, to obtain good comfort, it should be minimized. The pitching motion response of the suspension model is shown in Fig. 14. It is seen that the pitch angle of the passive suspension system is reduced by almost 35% and the settling time is reduced by 55% with the controller.

The rolling motion response is shown in Fig. 15. Roll is an extremely dangerous attitude and causes most of the acci-

Figure 15. Roll angle vs. time.

Figure 17. Vertical amplitude vs. time during cornering.

Figure 16. Vertical displacement of wheels vs. time.

dents that occur during cornering or maneuvering. To increase safety, rollover should be prevented by reducing the roll angle to the minimum possible. From an analysis of the roll response, we see that there is almost a 45% reduction in roll and almost the same amount of reduction in settling time. From the above analysis, it can be concluded that the use of an active suspension system can greatly improve comfort and safety at high speeds.

4.6. Vertical Wheel Displacement

The vertical displacement of the wheel is used to analyze contact between the tire-to-road surfaces. The decrease of overshoot in Fig. 16 shows that the vehicle has improved tire to road surface contact when an active system is used. The active suspension system model should be designed in such a way that it prevents the vehicle from skidding or drifting when

Figure 18. (a) Pitch angle vs time, (b) Roll angle vs. time, respectively during cornering.

it hits a bump or when cornering. In Fig. 16, a positive sign indicates that there is a gap between tire and road; a negative sign shows that the tire is bitten by the road surface. Vertical wheel displacement for the left wheel is shown in Fig. 16 (a) and, for the right wheel, in Fig. 16 (b). The figures indicate that surface contact between the tire and road is improved with the active suspension system, consequently ride is also improved.

4.7. Response to Cornering

Till now the study was restricted to straight road drive only but as discussed, that rolling is a very dangerous body attitude for a vehicle while cornering or maneuvering and many road accidents are caused by rollover. Hence from the safety point of view, this should be eliminated or minimized. What follows is a model of the full vehicle suspension system incorporating the effect of rollover.

In the analysis, a curved road with a radius of 40 m curvature is given as the road input and with the vehicle speed of 25 m/s.

Figure 19. Vertical wheel Displacement vs. time during cornering.

The length of the curved section is such that the vehicle takes 2 sec. to enter into a curved path, covers a distance of 250 m, and comes out on the straight road in 2 sec. Figure 17 and Fig. 18 (a) shows the vertical displacement and the pitching response of the vehicle respectively. The active and passive responses are coincident and the response is nearly zero, as is expected because the vehicle is running on a flat curved road. Fig. 18 (b) depicts the roll response of the active and passive suspension systems during cornering. The passive system has a maximum roll angle of 0.1019 rad (5.84°) while the active system has 0.001696 rad (0.09°), which is almost negligible. There is almost a 98% reduction in roll angle, implying that rollover can be prevented. Figures 17 and 18 (a) show the vertical displacement and the pitching response of the vehicle respectively. The active and passive responses are coincident and the response is nearly zero, as is expected because the vehicle is running on a flat curved road. Figure 18 (b) depicts the roll response of the active and passive suspension systems during cornering. The passive system has a maximum roll angle of 0.1019 rad (5.84°) while the active system has 0.001696 rad (0.09°), which is almost negligible. There is almost a 98% reduction in roll angle, implying that the rollover can be prevented.

Figures 19 (a) and (b) show the vertical displacement of the left and right wheel respectively while using the passive and active suspension models. When the vehicle is turning toward its left; the left wheel is lifted by a few millimeters and there is no longer any contact between the left tires and the road, which is undesirable and should be eliminated for safety. The response of the active suspension shows that this problem is

Figure 20. Response with feed-forward system.

overcome and a better tire to road grip is achieved. The results demonstrate that the active suspension system is safer against rollover during cornering and maneuvering, and desirable for high-speed vehicles.

4.8. Feed-Forward Control System

The methodology behind the feed-forward control system is that if the excitation is known in advance, the control signal can be sent earlier to perform the required corrective action. In the case of the vehicle suspension system model, the road profile that is experienced by the front wheel will be the same as the disturbance to the rear wheels but with a time delay that depends on the speed of the vehicle. To control the rear suspension's disturbances, the control signals, which are applied to the front suspension, are given after a time delay, which is decided by the response time of the controller and the speed of the vehicle.

The response obtained while using the feed-forward control is presented in Fig. 20, where the feed-forward response has a smaller amplitude but a longer settling time than the feedback response. This is undesirable; this is because the front and rear suspensions are connected by the body mass i.e. spring-mass

when the front suspension is disturbed it also affect the rear suspension and the bodyweight shifts on the rear suspension when rear suspension gets disturbed the bodyweight acts on the front suspension but the applied to control which is taken from the front suspension still assumes the bodyweight is acting on the rear suspension.

5. CONCLUSION

This work involves the designing of both a Simple PID and a feed-forward controller. The response is studied for disturbances such as road excitation and those caused due cornering of the vehicle. From the results obtained while investigating the performance of the vehicle running on a straight road and that subjected to bump disturbances, it has been found that when an active suspension system is used a 75% reduction in vertical amplitude is obtained while the settling time of the vertical disturbances due to the bump is reduced by 68%. Furthermore, it is evident from the results that the pitch angle is reduced by 35% followed by a reduction of 55% in settling time during pitching. A 45% reduction in roll angle as well as settling time during rolling motion is achieved by the use of an active suspension system. The results obtained also help to conclude that a 98% reduction in roll angle is also achieved during the cornering of the vehicle. The current study shows that, during the simulation of an active suspension system with sensing and actuating constraints, the performances of components of the system have a significant impact on the results obtained. The controller should be tuned carefully, failing in which the overshoot of the system rises rapidly and affects the results.

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