

Influence of Active Suspension Preview Control on Vehicle Ride and Braking Performance

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ABSTRACT

The integrated control between the vehicle chassis subsystems (suspension, brake, and steering) became one of the most important aspects for current developments to improve the dynamics of the vehicles. Therefore, the aim of this study is to investigate the influence of the preview control of the active suspension on the vehicle ride and braking performance. The vehicle performance was examined theoretically using a longitudinal half vehicle model with four degrees of freedom considering the rotational motion of the tires. The active suspension system model, tire-road interface model and braking system model are included in the vehicle model.

In order to study the influence of the preview control on the vehicle ride and braking performance, an active suspension system control algorithm employing the look-ahead preview information and the wheel-base time delay based on the optimal control theory is derived. On the other hand, the ABS control algorithm is designed based on the slip-control strategy. The vehicle ride performance is evaluated in terms of discomfort and road holding, while the braking distance and time is considered as evaluation criteria for the vehicle brake performance. The results are generated in the time domain to simulate the vehicle response during braking, while wheels are subjected to vertical road input. Comparisons between passive and active suspension systems in terms of ride and braking performance are discussed as well as correlated and uncorrelated active suspension systems. The improvement of the preview control of the active suspension system on the ride and braking performance is shown.

CITATION: Kaldas, M. and Soliman, A., "Influence of Active Suspension Preview Control on Vehicle Ride and Braking Performance," *SAE Int. J. Passeng. Cars - Mech. Syst.* 7(2):2014, doi:10.4271/2014-01-0862.

INTRODUCTION

Recently, with the advance in microcontrollers various electronic control systems and controllers have been developed and manufactured extensively for automotive application, such as the active suspension system, anti-lock braking system (ABS), electronic brake force distribution system (EBD), traction control system (TC), and so on. These microcontrollers or subsystems were greatly developed and applied widely to improve the ride comfort, safety and stability of the vehicles. However, each subsystem often fulfills a single function. For example, the active suspension system is designed to improve the vehicle ride comfort performance, while the ABS is developed to improve the performance of the braking system. Therefore, different control theories and strategies of the ABS and active suspension system for different types of road vehicles have been proposed individually by several literatures for decades [1, 2, 3, 4, 5, 6, 7, 8, 9, 10].

In fact, the control system of the vehicle is complicated and the performance of the vehicle depends not only on every separated subsystem but also on the interaction between these subsystems. For example, the main idea of the integrated control between the active suspension system and the ABS is using the active suspension actuator to provide an excessive active force. This force can be considered as a normal force subjected onto wheel during braking for improving the vehicle stopping distance. Hence, the interaction between the suspension and braking systems is far too complex model adequately. Therefore, few studies on this topic are available [11, 12, 13, 14, 15, 16, 17]. A simple 2 degree of freedom quarter vehicle model with active suspension system integrated with ABS by employing the nonlinear back stepping design schemes is developed by [11]. In this study, the integrated control scheme has been used for improving the vehicle braking performance. However, this work can be considered as a preliminary investigation in integration of the subsystems of the vehicle, because the model ignored the

where

$$Z_{bf} = Z_b - L_f \theta_b, \quad Z_{br} = Z_b + L_r \theta_b$$

$$h_i = h + (Z_{bi} - Z_{wi}) + (Z_{wi} - Z_{oi}), \quad i = f, r$$

In the above equations, M_b , M_{wf} and M_{wr} represent the body, front wheel and rear wheel masses respectively. K_{sf} and K_{sr} represent the front and rear suspension springs stiffnesses. While K_{tf} and K_{tr} represent the front and rear tire stiffnesses. C_{sf} and C_{sr} stands the damping coefficient of the front and rear dampers and U_{sf} and U_{sr} are the active suspension forces generated from the linear actuators at front and rear. L_p , L_r and h are the distance from the front and rear axles to the vehicle body CG and the distance from the CG to road surface. The body, wheel, and road displacements are denoted as Z_b , Z_w , and Z_o respectively, and they are given with respect to the inertial frame. F_{xf} , F_{rf} , F_{xr} and F_{rr} represent the braking and rolling forces at the front and rear wheels respectively.

Braking System Dynamics Model

The equation of motion for vehicle brake dynamics in the longitudinal direction can be written as [18]

$$M_{tot} \dot{v} = -F_{xf} - F_{xr} - F_{rf} - F_{rr} - F_d \quad (5)$$

$$M_{tot} = M_b + M_{wf} + M_{wr} \quad (6)$$

where M_{tot} is the total mass of the vehicle, F_d , F_r , and F_x are the aerodynamic drag force, rolling resistance force and tire-road friction force respectively. These forces can be calculated from the following formulas according to operating conditions [18].

$$F_d = 0.5 * C_d * \rho * A_f * v^2 \quad (7)$$

where C_d , ρ , A_f and v are the vehicle aerodynamic coefficient, air density, frontal projected area, and vehicle speed respectively. At the same time, for vehicle speed up to 150 km/hr the rolling resistance forces can be described using the following equation [18].

$$F_{ri} = (0.0136 + 0.4 * 10^{-7} * v^2) F_{zi} \quad (8)$$

where F_{zi} is the normal forces due to vehicle weight and suspension system. The dynamic modes of the vehicle wheels due to braking process can be calculated as the following:

$$J_{wi} = (F_{xi} + F_{ri}) r_{di} - T_{bi} \quad (9)$$

where J_{wi} , r_{di} and T_{bi} are the wheel inertia including any drive train effect, effective dynamic tire radius and braking torque at each wheel respectively.

Tire-Road Interface Model

The tire-road friction force is the main problem in the ABS design. However, the relation between longitudinal wheel slip ratio and tire-road friction coefficient is physically unknown to day knowledge. Therefore, this relation is usually determined from empirical models as the Dugoff and Magic Formula tire models [6, 9, 19, 20]. In this study, a widely used semi empirical model is used to calculate the relation between tire-road friction coefficient and longitudinal slip ratio at different operating condition [20]. The relationship between the tire-road friction coefficient and the wheel slip ratio used in this study is shown in Figure 2 for different road condition.

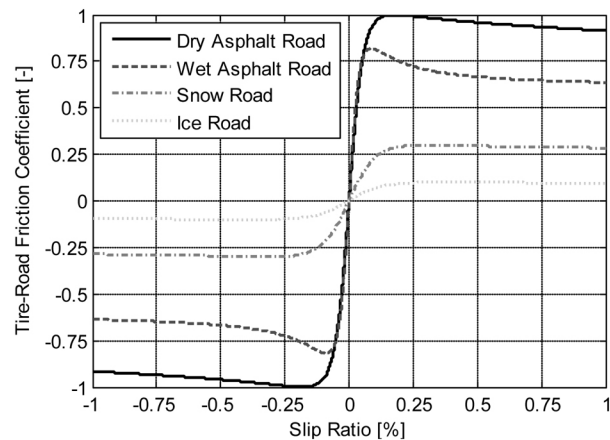


Figure 2. The relationship between tire-road friction coefficients and slip ratio.

$$\mu_i(\lambda_i) = D \sin \left(C \cdot \arctan \left(B \lambda_i - E (B \lambda_i - \arctan(B \lambda_i)) \right) \right) \quad (10)$$

where μ_i and λ_i are the tire-road friction coefficient and the wheel slip ratio. Which $\lambda_i = 1 - [(\omega \cdot r_{di})/V]$ and B, C, D, and E are the Magic formula tire model coefficients, which shown in Table 1.

The tire-road friction force can be calculated using the following equation:

$$F_{xi} = \mu_i(\lambda_i) F_{zi} \quad (11)$$

Table 1. Tire-road interface model coefficients.

Road condition	B	C	D	E
Dry Asphalt	10	1.9	1	0.97
Wet Asphalt	12	2.3	0.82	1
Snow	5	2	0.3	1
Ice	4	2	0.1	1

Road Input Model

In this work, a time domain random road profile is generated as a filtered white noise signal considering the road roughness and the change in vehicle speed [17]. The random road profile is shown in Figure 3.

$$\dot{Z}_o(t) = -2\pi \cdot f_o \cdot Z_o(t) + \sqrt{G \cdot G_o(t) \cdot v(t)} w(t) \quad (12)$$

where Z_o is the road input displacement, G_o is the road roughness coefficient, v is the vehicle speed and $w(t)$ is zero-mean Gaussian white noise with intensity 1. f_o is the low cut-off frequency, which is assumed to be 0.01 Hz in this study.

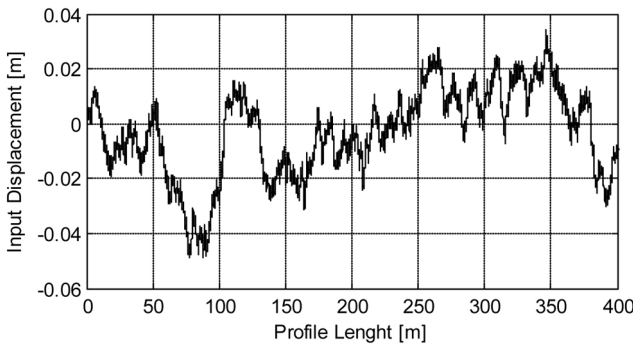


Figure 3. The random road profile.

ACTIVE SUSPENSION SYSTEM CONTROLLERS

LQR Active Suspension Algorithm

In this investigation, the optimal control theory is used to derive a Linear Quadratic Regulator (LQR) for the active suspension system. The active suspension control algorithm is based on the limited state feedback concept [21]. Therefore, the body and wheels velocity and displacement are used as controller inputs. The control algorithm is derived as shown in Equation 13 and 14. The Genetic Algorithm (GA) optimization routine is used to optimize the feedback gains of the active suspension control algorithm. The optimization goal is to minimize the vertical and pitch accelerations of the vehicle together with the dynamic tire loads whilst at the same time maintains the suspension working space in the allowable limits. The performance index used in the optimization process is presented in Equation 15.

$$U_{sf} = -[K_{11}Z_{bf} + K_{12}Z_{wf} + K_{13}\dot{Z}_{bf} + K_{14}\dot{Z}_{wf}] \quad (13)$$

$$U_{sr} = -[K_{21}Z_{br} + K_{22}Z_{wr} + K_{23}\dot{Z}_{br} + K_{24}\dot{Z}_{wr}] \quad (14)$$

$$PI = 0.5 \int_{t_0}^{t_f} q_1 \ddot{Z}_b^2 + q_2 \ddot{\theta}_b^2 + q_3 (Z_{bf} - Z_{wf})^2 + q_4 (Z_{wf} - Z_{of})^2 + q_5 (Z_{br} - Z_{wr})^2 + q_6 (Z_{wr} - Z_{or})^2 + \rho_1 U_{sf}^2 + \rho_2 U_{sr}^2 \quad (15)$$

where $K_{11}, K_{12}, \dots, K_{24}$ are the feedback gains of the control algorithm and $q_1, q_2, \dots, q_6, \rho_1$ and ρ_2 are the weighting parameters of the performance index.

LQR Active Suspension Algorithm with Preview

In the preview case it is assumed that, the road excitation input Z_r for $[t, t+t_p]$ is measured by the preview sensor at the front bumper. That is, the preview information about the ground excitation $Z_r(t)$ up to (t_p) ahead of current time (t) is available. In addition, for rear wheel the input Z_r is known for $[t, t+t_p+t_l]$, where t_p and t_l equals L_p/V and L_p+L/V respectively. The method for developing continuous time optimal preview control law for systems with multiple inputs were described in [22, 23, 24]

Since the road input information is available, the feed forward gain can be considered in the control algorithm and the Genetic Algorithm (GA) optimization routine is used to optimize the feedback and feed forward gains of the active control algorithm. The control algorithm with preview information is shown in Equations 16 and 17.

$$U_{sf} = -[K_{11}Z_{bf} + K_{12}Z_{wf} + K_{13}\dot{Z}_{bf} + K_{14}\dot{Z}_{wf}] + [K_{15}Z_{pr} + K_{16}Z_{of}] \quad (16)$$

$$U_{sr} = -[K_{21}Z_{br} + K_{22}Z_{wr} + K_{23}\dot{Z}_{br} + K_{24}\dot{Z}_{wr}] + [K_{25}Z_{pr} + K_{26}Z_{or}] \quad (17)$$

where K_{15}, K_{16}, K_{25} and K_{26} are the feed forward gains of the front and rear control algorithm.

ABS CONTROLLER

In the ABS controller the wheel slip ratio is used as a wheel look criterion. In the ABS mode the brake pressure is increased and decreased to prevent wheel lockup depending on the calculated signal of the wheel slip ratio. In the control strategy the brake pressure is triggered to maintain the wheel slip ratio around 20 percent. The ABS controller uses input data stemming from the existing sensors of the ABS, such as measured wheel angular velocities at each wheel and measured or estimated vehicle speed. The calculated slip ratio is then compared with the desired value of the slip to produce error signal, for each value of the error, the controller returns 1 if the error value is greater than zero and hence the brake torque is increased, returns zero if the error value is equals to zero and hence the brake torque remains unchanged, and -1 if it less than zero and hence brake torque is reduced. Figure 4 shows a schematic drawing of the ABS system used in this study.