

# Influence of Different Semi Active Control Strategies on the Performance of Macpherson Suspension Kinematic Parameters

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## ABSTRACT

Suspension kinematic parameters such as camber, caster and king-pin angles as well as track width are important in improving handling performance and stability of a vehicle. Using a new model of the Macpherson suspension system, the effects of different hybrid semi-active control strategies on the performance of suspension kinematic parameters and on improvement of ride quality are investigated. The control strategies considered in this work comprise hybrid skyhook-groundhook, modified skyhook and, passive-skyhook controllers. It is shown that although contribution of these controllers on the improvement of ride quality of the vehicle is similar, they affect the performance of the Macpherson suspension kinematic parameters significantly different. Simulation results are presented and discussed.

## INTRODUCTION

The main performance requirements of a suspension system, such as delivering an effective ride comfort for passengers, keeping the tire in contact with the road and, enhancing the handling performance, are conflicting in nature. For instance, the large suspension deflection with low damping provides a better isolation against road roughness while a large damping yields a superior stability condition.

The structure of the suspensions can be categorized into three main types: passive, active and semi-active systems. Passive suspension systems, traditionally used in automotive applications, deliver an optimal trade off between the specifications in a limited range of the road frequencies. The alternative suspension systems are active suspensions. This type of suspensions offer a better road roughness isolation, stability and, handling performance compared to those of passive type in a wide range of disturbance frequencies. However, having complicated circuit components, expensive structure and heavy weight make implementation of this type of suspensions impractical. The third type includes semi-active suspension systems. These types of suspensions are more attractive in automotive industries because of low

energy requirement, simple structure and having a performance in between that of passive and fully active systems.

In order to reconcile the conflicting performance requirements, various control strategies for active and semi-active suspension systems have been developed [1-4]. The focus of most of the works is on the improvement of ride quality and stability condition of the vehicle whereas, concerns regarding the performance of the suspension kinematic parameters subjected to active or semi-active actuation force are rare. Main kinematic parameters which affect the handling performance of the vehicle are camber, caster and kingpin angles as well as track width alteration. Camber angle alterations are important in reducing tire life and increasing the temperature of the tire as well as in generating lateral forces which affect the stability of the vehicle. Caster and kingpin angles are two main geometry parameters in producing self aligning torques influencing the handling of the vehicle. Track changes cause the rolling tire to slip and may affect the steering and handling. Generally speaking, the proper performance of the kinematic suspension parameters delivers a better driving safety, superior steering, and longer tire life, as well as less fuel consumption and less strain in suspension components. It should be noted that each specific suspension structure has different kinematic performance compared to other types of suspensions. Hence, in order to investigate the performance of these parameters subjected to the actuation force which is designed for improvement of ride quality, an accurate model of the suspension, like Macpherson suspension system, is required.

Two well-known control strategies applicable to semi-active suspension systems are skyhook and groundhook controllers. The former, proposed by Karnopp et al. [5], is notable in isolation of the chassis from road disturbances while the latter, suggested by Novak et al. [6], is important in the attenuation of the peak resonance of the wheel. The skyhook controller has inverse effect on the attenuation of the peak resonance of the wheel and groundhook controller has undesirable effects on the chassis disturbance isolation. In order to take the advantages of both control policies,

different combinations of those strategies have been developed and investigated analytically and experimentally. For example, Ahmadian et al. proposed a hybrid control strategy using a combination of the groundhook and skyhook controllers and analyzed its performance in time and frequency domains [7-9]. Batterbee and Sims [10] modified the effect of the skyhook controller by considering the advantages of the passive damper force and suggested a semi-active control policy as a set-point force for a feedback linearization force tracking control of MR damper. Yi and Song [11] made another modification of the skyhook controller by including the benefits of the passive, skyhook and groundhook damper forces together. Hong et al [12] mixed the passive and skyhook damper forces to modify the skyhook controller and applied it on the Macpherson suspension. The effects of the three different semi-active control methods on the acceleration and relative displacement transmissibility of the suspension systems have been studied by Shen et al analytically and experimentally [13]. In this study, upon introducing a new model of Macpherson suspension system, the effects of three semi-active control strategies, widely used in literature, are investigated and it is shown that while their contribution in ride quality improvement and road holding is similar; their influences on the performance of the Macpherson kinematic parameters are significantly different.

Generally, in the present study, the effects of three different hybrid semi-active control policies on the Macpherson suspension kinematics are investigated. In the following sections, the nonlinear relations of the Macpherson kinematic parameters are formulated and are verified by ADAMS software. The hybrid control strategies are explained. Later, the focus will be on the comparison of the controllers using simulation results. Finally, the results are discussed and summarized drawing valid conclusions.

## MODEL DESCRIPTION

A schematic of the Macpherson suspension, invented by Earl S. Macpherson in 1949, is shown in Figure 1. The main components forming the structure consist of control arm, spindle and, strut including helical spring and damper. The two-degree-freedom model of the Macpherson suspension which is employed in the current study is represented in Figure 2. The vertical displacement of the chassis (sprung mass),  $z_s$ , and the rotational motion of the control arm,  $\theta$ , are considered as two generalized coordinates of the system. In addition,  $K_s$  and  $K_t$  stand for the suspension and tire stiffness coefficients,  $f_{sa}$  and  $f_d$  represent the semi-active and disturbance forces, respectively,  $m_s$ ,  $m_u$  and  $m_{ca}$  are the sprung, unsprung and control arm masses.  $z_r$  is the road disturbance. The origin of the coordinate system, O, is at the equilibrium position  $B_1$ . Considering the kinematic constraints of the system arising from constant length of the control arm and cylindrical motion of the strut, one can formulate the system of kinematic equations as the following.

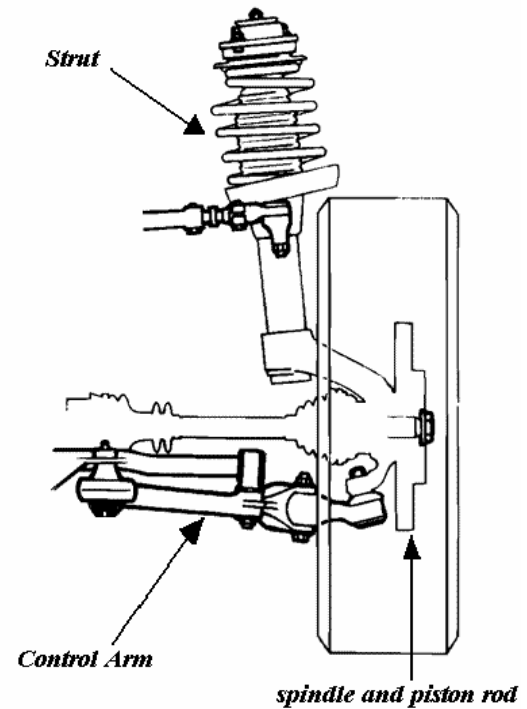


Figure 1 Schematic of the Macpherson suspension

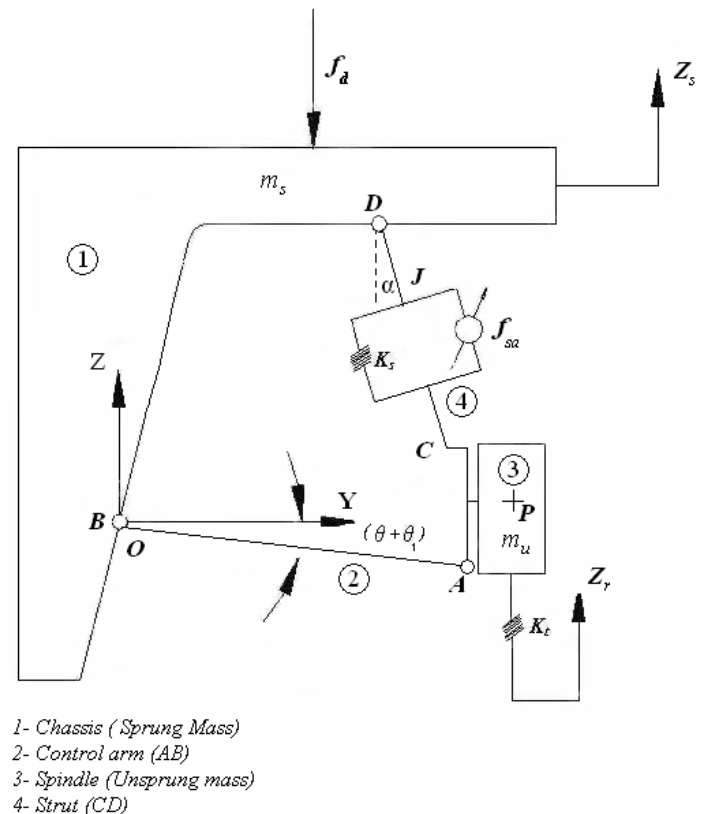


Figure 2 Model of the Macpherson suspension

$$\begin{aligned}
y_C &= (y_{C_1} - y_{A_1})a_{11} + (z_{A_1} - z_{C_1})a_{21} + y_A \\
z_C &= (y_{C_1} - y_{A_1})a_{21} + (z_{C_1} - z_{A_1})a_{11} + z_A \\
y_J &= (y_{J_1} - y_{A_1})a_{11} + (z_{A_1} - z_{J_1})a_{21} + y_A \\
z_J &= (y_{J_1} - y_{A_1})a_{21} + (z_{J_1} - z_{A_1})a_{11} + z_A \\
y_P &= (y_{P_1} - y_{A_1})a_{11} + (z_{A_1} - z_{P_1})a_{21} + y_A \\
z_P &= (y_{P_1} - y_{A_1})a_{21} + (z_{P_1} - z_{A_1})a_{11} + z_A \\
z_C &= \alpha(y_C - y_D) + z_D \\
z_J &= \alpha(y_J - y_D) + z_D \\
y_A &= L_A \cos(\theta + \theta_1) \\
z_A &= L_A \sin(\theta + \theta_1) + z_s
\end{aligned} \tag{1}$$

In Equation (1),  $(y_{A1}, z_{A1})$ ,  $(y_{B1}, z_{B1})$ ,  $(y_{C1}, z_{C1})$ ,  $(y_{D1}, z_{D1})$ ,  $(y_{J1}, z_{J1})$ ,  $(y_{P1}, z_{P1})$  are the initial positions of the points A, B, C, D, J and P, respectively, under static equilibrium conditions.  $\alpha$  is the slope of the strut,  $L_A$  is the length of the control arm and  $\theta_1$  is the initial angle of the control arm resulting from the static deflection and structure design. In addition,

$$\begin{aligned}
a_{11} &= a_{22} = \cos \varphi \\
a_{12} &= -a_{21} = \sin \varphi
\end{aligned} \tag{2}$$

where  $\varphi$  is the rotation angle of the wheel.

By differentiating system of Equation (1), the velocity of the main points and the angular velocity of the wheel,  $\dot{\varphi}$ , can be found. Using Lagrange's approach along the generalized coordinates, the acceleration of the generalized coordinates are obtained as

$$(m_s + m_u + m_{ca})\ddot{z}_s + m_u L_A [\cos(\theta + \theta_1) + \{\cos(\theta + \theta_1) + \alpha \sin(\theta + \theta_1)\} \frac{(y_C - y_J)(y_P - y_A)}{h}] \ddot{\theta} = \frac{\partial T}{\partial z_s} - \frac{\partial V}{\partial z_s} - \frac{\partial D}{\partial \dot{z}_s} \tag{3}$$

$$\begin{aligned}
&m_u \left[ L_A \cos(\theta + \theta_1) + (y_P - z_A) \frac{\partial \dot{\varphi}}{\partial \dot{\theta}} \right] \ddot{z}_s + \\
&\left[ m_u L_A \left[ \{(\cos(\theta + \theta_1) + \alpha \sin(\theta + \theta_1))\} \frac{(y_C - y_J)(z_A - z_P)}{h} \right] \times \right. \\
&\left. - L_A \sin(\theta + \theta_1) + (z_A - z_P) \frac{\partial \dot{\varphi}}{\partial \dot{\theta}} \right] + \\
&m_u L_A \left[ L_A \cos(\theta + \theta_1) + (y_P - z_A) \frac{\partial \dot{\varphi}}{\partial \dot{\theta}} \right] \times \\
&\left[ \{(\cos(\theta + \theta_1) + \alpha \sin(\theta + \theta_1))\} \frac{(y_C - y_J)(y_P - y_A)}{h} \right] + \\
&[\cos(\theta + \theta_1)]
\end{aligned}$$

$$I_u \ddot{\theta} \frac{\partial \dot{\varphi}}{\partial \dot{\theta}} \ddot{\theta} = \frac{\partial T}{\partial \theta} - \frac{\partial V}{\partial \theta} - \frac{\partial D}{\partial \dot{\theta}} \tag{4}$$

Where  $T$ ,  $V$  and  $D$  are the kinetic, potential and damping functions, respectively. Rearranging the equations of motion in compact form along with the conservative and external forces, one has

$$\begin{aligned}
\ddot{z}_s &= g_1(z_s, \dot{z}_s, \theta, \dot{\theta}, f_{sa}, f_d, z_r) \\
\ddot{\theta} &= g_2(z_s, \dot{z}_s, \theta, \dot{\theta}, f_{sa}, f_d, z_r)
\end{aligned} \tag{5}$$

Introducing the state variables as  $[x_1, x_2, x_3, x_4]^T = [z_s, \dot{z}_s, \theta, \dot{\theta}]^T$  and linearizing the Equation (5), the state space representation of the equations of motion is obtained as

$$\begin{aligned}
\dot{x} &= Ax(t) + B_1 f_{sa}(t) + B_2 z_r(t) + B_3 f_d(t) \\
x(0) &= x_e
\end{aligned} \tag{6}$$

where

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ \frac{\partial g_1}{\partial x_1} & \frac{\partial g_1}{\partial x_2} & \frac{\partial g_1}{\partial x_3} & \frac{\partial g_1}{\partial x_4} \\ 0 & 0 & 0 & 1 \\ \frac{\partial g_2}{\partial x_1} & \frac{\partial g_2}{\partial x_2} & \frac{\partial g_2}{\partial x_3} & \frac{\partial g_2}{\partial x_4} \end{bmatrix}_{x_e}$$

$$B_1 = \begin{bmatrix} 0 \\ \frac{\partial g_1}{\partial f_{sa}} \\ 0 \\ \frac{\partial g_2}{\partial f_{sa}} \end{bmatrix}_{f_{sa}=0}, \quad B_2 = \begin{bmatrix} 0 \\ \frac{\partial g_1}{\partial z_r} \\ 0 \\ \frac{\partial g_2}{\partial z_r} \end{bmatrix}_{z_r=0}, \quad B_3 = \begin{bmatrix} 0 \\ \frac{\partial g_1}{\partial f_d} \\ 0 \\ \frac{\partial g_2}{\partial f_d} \end{bmatrix}_{f_d=0}$$

More details on the dynamical relations of the model can be found in [14].

## SUSPENSION KINEMATIC PARAMETERS

The kinematic parameters of a suspension system are defined based on the geometry of the structure and are important in the suspension performance. The main parameters which affect the handling performance and steering of the vehicle as well as the life of the tire are camber angle, caster angle, kingpin inclination angle and track width. Camber angle is the angle between the vertical line to the road and center plane of the wheel. The angle,  $\varphi$ , in Equation (2) shows the variation of this parameter. Caster and kingpin angles are the cosine directions of the steering axis of the Macpherson suspension system in the x-z and y-z planes, respectively. Track width is defined as the lateral distance between the centers of the front wheels. The mathematical descriptions of these parameters are