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Using Adaptive Pole Placement Control Strategy for Active Steering Safety System

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Abstract**—**This paper studies the design of an adaptive control strategy to tune an active steering system for better drivability and maneuverability. In the first step, adaptive control strategy is applied to estimate the uncertain parameters on-line (e.g. cornering stiffness), then the estimated parameters are fed into the pole placement controller to generate corrective feedback gain to improve the steering system dynamic's characteristics. The simulations are evaluated for three types of road conditions (dry, wet, and icy), and the performance of the adaptive pole placement control (APPC) are compared with pole placement control (PPC) and a passive system. The results show that the APPC strategy significantly improves the yaw rate and side slip angle of a bicycle plant model.

Keywords**—**Adaptive control, active steering, pole placement, vehicle dynamics.

I. Introduction

ACTIVE steering control has been used in passenger cars since 2003 [1]. The goal of designing an active steering system has been improving the vehicle's stability during cornering and spinning. In other words, when the vehicle is subjected to poor road conditions (e.g. icy road) it starts to slip or skid and the vehicle needs to keep its stability by applying some controllers to avoid over steering. To help the driver maintain a safe drive, it is important that a good stability control system is included into a vehicle. The driver faces a serious handling issue when he/she feels that the vehicle tends to be unstable. Whenever a driver loses some degree of control, the system will detect and stabilize the vehicle immediately, hence enabling the driver to regain the control of the vehicle [2]. As an example scenario, a sudden movement of the steering wheel may make a car skid dangerously and these could lead to a fatal accident. Unexpected child crossing a road may cost a driver to an evasive action.

Inexperienced and young drivers are more prone to overreact the car during uncertain situations. This will cause fatal accidents and must be corrected by control systems. A properly designed controller, e.g. automatic feedback system, can significantly assist the driver to reduce the risk of accidents. One of the common control strategies is controlling the plant with feedback gain. The feedback gain can be calculated using different types of control methods such as PPC, linear quadratic regulator (LQR), and model reference control (MRC).

The plant model in this paper is simplified steering model of a vehicle, which is called bicycle model, and APPC is implemented to improve steering dynamics characteristics (e.g. yaw rate and side slip angle). The system's parameters are assumed to be certain, except cornering stiffness of the front and rear wheels. The uncertain parameters are estimated online based on the available information (inputs and outputs) using gradient algorithm. Finally, the estimated parameters along with PPC strategy are combined to generate APPC gain at each time step.

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II. Model Development

In many studies on an active steering system [3]-[5], a simple but useful bicycle model is employed as a control-oriented model for control development purposes. In a bicycle model, the left and right front wheels are represented by one single wheel. Similarly, the two rear wheels are assumed as a single wheel. Fig. 1 shows the bicycle model used in this research for the control development. The input to the model is a steering angle (δ) that comes from either the driver's command or undesired disturbances.

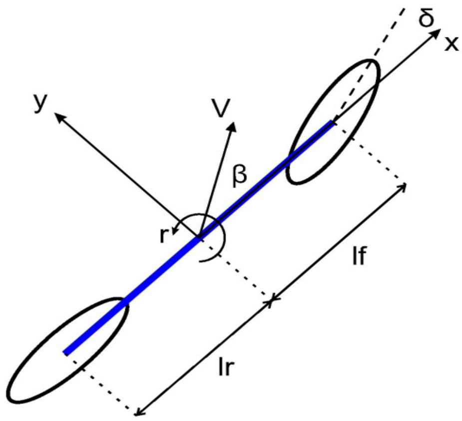

Fig. 1 Bicycle model of a vehicle steering system

The longitudinal and lateral motions of the system can be described by (1). The states of the system are side slip angle (β), forward velocity (V), and yaw rate (r) while f_x, f_y, and M_z represent longitudinal force, lateral force, and torque around the yaw axis respectively.

$$\begin{bmatrix} mV(\dot{\beta}+r) \\ m\dot{V} \\ I\dot{r} \end{bmatrix} = \begin{bmatrix} -sin\beta & cos\beta & 0 \\ cos\beta & sin\beta & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} f_x \\ f_y \\ M_z \end{bmatrix} \tag{1}$$

The above set of equations can be simplified further by assuming constant forward velocity ($\dot{V}=0$) and small side slip angle ($\beta \ll 1$). Equation (2) represents the simplified model with two state variables:

$$\begin{bmatrix} mV(\dot{\beta}+r) \\ I\dot{r} \end{bmatrix} = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} f_y \\ M_z \end{bmatrix} \Rightarrow \begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \end{bmatrix} \delta \quad (2)$$

The coefficients a_{ij} and b_i are functions of physical parameters of the system such as vehicle mass (m), inertia (I), front and rear wheels distance from the center of mass (l_f, l_r), front and rear cornering stiffness (C_f, C_r), forward velocity (V), and road adhesion coefficient (μ) as described in (3). The cornering stiffness is proportional coefficient which relates cornering (lateral) forces and slip angles.

$$\begin{aligned} a_{11} &= -\frac{(C_f+C_r)\mu}{mV} \\ a_{12} &= -1+\frac{(C_r l_r - C_f l_f)\mu}{mV^2} \\ a_{21} &= \frac{(C_r l_r - C_f l_f)\mu}{I} \\ a_{22} &= -\frac{(C_f l_f^2 + C_r l_r^2)\mu}{IV} \\ b_1 &= \frac{C_f \mu}{mV} \\ b_2 &= \frac{C_f l_f \mu}{I} \end{aligned} \quad (3)$$

III. Adaptive Pole Placement Control

This section presents the method used to estimate unknown parameters (e.g. cornering stiffness). Then, the PPC and APPC development are presented for an active steering system.

A. Parameter Estimation

The front and rear cornering stiffness of the bicycle model are assumed as uncertain parameters. The cornering stiffness of the front and rear tires are estimated by applying the gradient algorithm at each time step [6], [7]. A static parametric model (SPM) and the gradient algorithm are presented in (4) and (5). z, $\hat{z}$, and φ are available (or measurable) signals and θ represents unknown parameters. The adaptive gain (γ) is set to 1000 in (5). Moreover, the normalizing factor ($\sigma = 0.1$) is chosen to bound the error signal ($z - \hat{z}$).

$$\Sigma: \begin{cases} z = \theta^{*T}\varphi \\ \hat{z} = \theta^T \varphi \end{cases} \quad (4)$$

$$\dot{\theta} = \gamma\left(\frac{z-\hat{z}}{\sigma^2}\right)\varphi\ , \quad \theta(0) = \theta_0 \quad (5)$$

B. PPC Strategy

PPC method is somehow similar to the root-locus method; it means that in pole placement method, closed loop poles are forced to the desired poles (Fig. 2) via feedback gain [8]. The desired poles are defined by considering natural frequencies, damping ratio, and bandwidth of a system. In under damped systems, the eigenvalues or poles are in a complex conjugate form as shown in (6):

$$s^2 + 2\xi w_n + w_n^2 = 0,\ s_{1,2} = -\xi w_n \pm \left(w_n\sqrt{1-\xi^2}\right)j \quad (6)$$

The desired poles are defined based on optimizing both natural frequencies (w_n) and damping ratio (ξ) of a dynamic system to improve the system's overshooting and settling time. In this paper, the MATLAB function "*place*" is employed to move the system's poles to the desired poles by feedback gain.

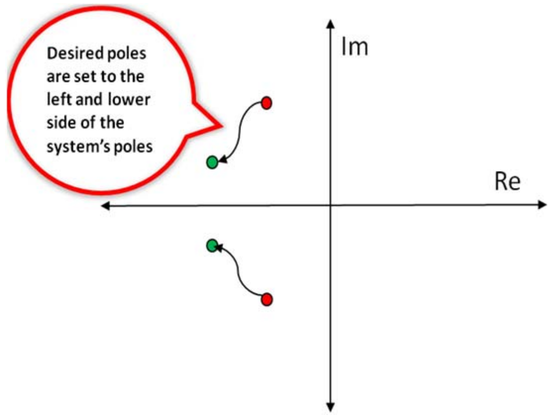

Fig. 2 Moving poles to the more stable zone

The function "*place*" calculates the feedback gain (K) based on the dynamic system (e.g. matrices A and B) as well as desired poles matrix (J). Equation (7) shows a typical state-space dynamic system, where the feedback signals is $u = -KX$. The feedback gain is calculated from (8):

$$\dot{X} = AX + Bu \quad (7)$$

$$K = place\ (A, B, J) \quad (8)$$

The APPC is developed by integrating the PPC strategy and parameter estimation (at each time step) to calculate an adaptive pole placement gain (9). The adaptive system matrices are represented by $\hat{A}$ and $\hat{B}$, where the cornering stiffness of the front and rear wheels are updated at each time step in the system's matrices. Therefore, the feedback gain ($\hat{K}$) is also adaptive, and is updated at each time step.

$$\hat{K} = place\ (\hat{A}, \hat{B}, J) \quad (9)$$

IV. Simulation Results and Discussion

The developed model and control strategies are evaluated for three types of road condition such as dry, wet, and icy. The input to the system is a sinusoidal steering angle as shown in Fig. 3. The parameters and values of the presented bicycle model are given in Table I.

The simulation results compare the performance of the passive system (without controller) with the active system (with pole placement feedback gain). In the APPC strategy, the feedback gain is integrated with an estimation of front and rear cornering stiffness at each time step. The estimation of the front and rear cornering stiffness at each time step is shown in Figs. 4 and 5 for dry road condition, Figs. 8 and 9 for wet road condition, and Figs. 12 and 13 for icy road condition.

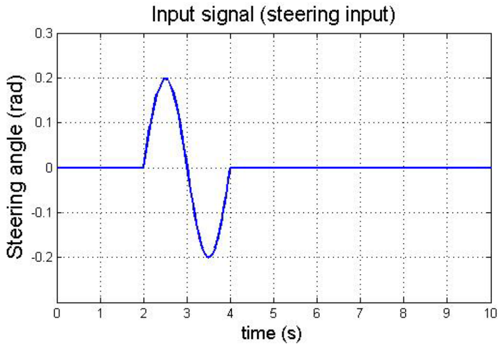

Fig. 3 Input steering angle to the system

TABLE I
PARAMETERS AND VALUES OF A SEDAN CAR

Parameter	Values
M (vehicle mass)	1170 [kg]
l_f (distance from front wheels to center of mass)	1.4 [m]
l_r (distance from rear wheels to center of mass)	1.8 [m]
V (vehicle forward velocity)	30 [m/s]
I (yaw moment of inertia)	1550 [kg.m^2]
Dry Road	
μ (road adhesion coefficient)	1
C_f (front cornering stiffness)	6000 [N/rad]
C_r (rear cornering stiffness)	10000 [N/rad]
Wet Road	
μ (road adhesion coefficient)	0.7
C_f (front cornering stiffness)	25000 [N/rad]
C_r (rear cornering stiffness)	25000 [N/rad]
Icy Road	
μ (road adhesion coefficient)	0.3
C_f (front cornering stiffness)	25000 [N/rad]
C_r (rear cornering stiffness)	25000 [N/rad]

A. Dry Road Condition

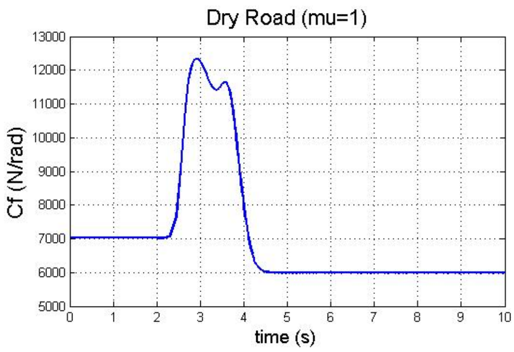

Fig. 4 Estimated front cornering stiffness (dry road)

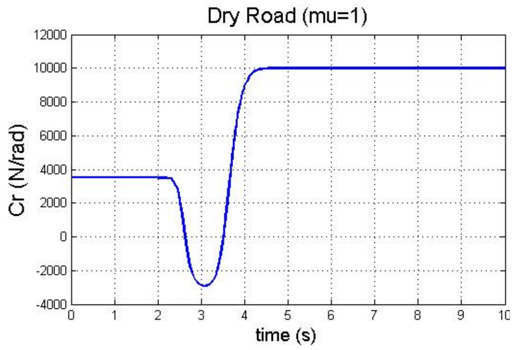

Fig. 5 Estimated rear cornering stiffness (dry road)

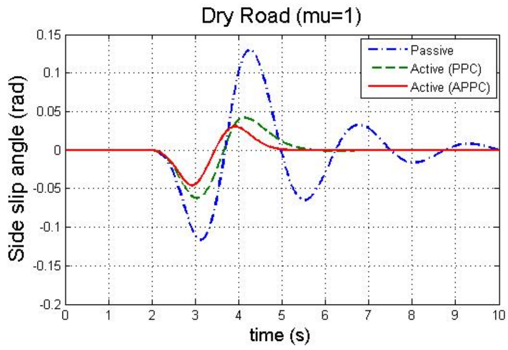

Fig. 6 Side slip angle (dry road)

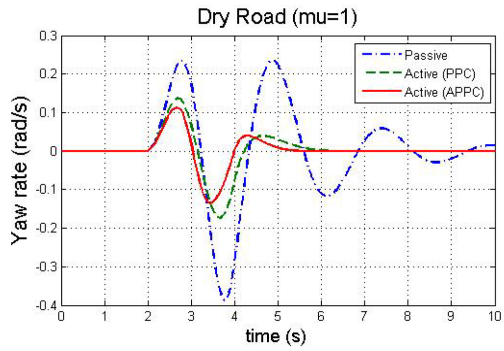

Fig. 7 Yaw rate (dry road)

B. Wet Road Condition

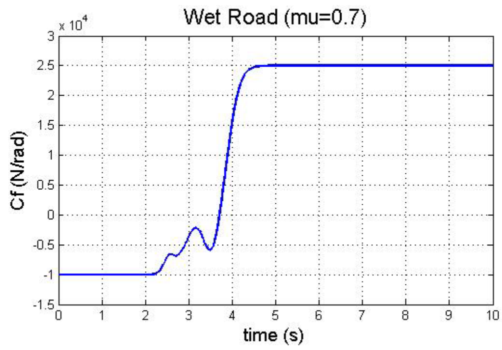

Fig. 8 Estimated front cornering stiffness (wet road)

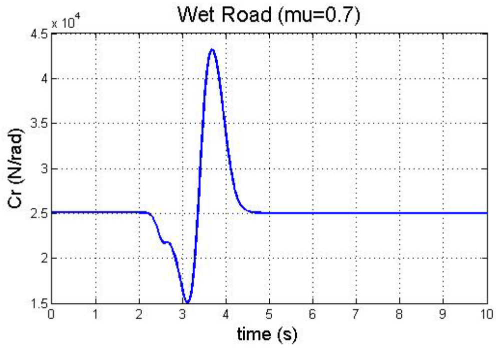

Fig. 9 Estimated rear cornering stiffness (wet road)

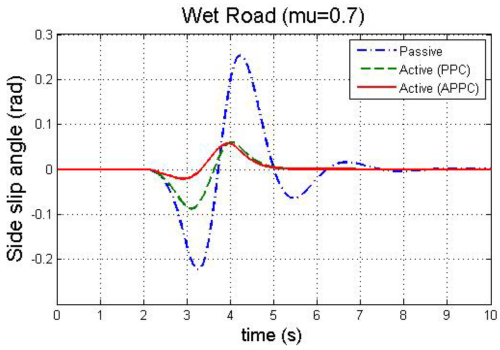

Fig. 10 Side slip angle (wet road)

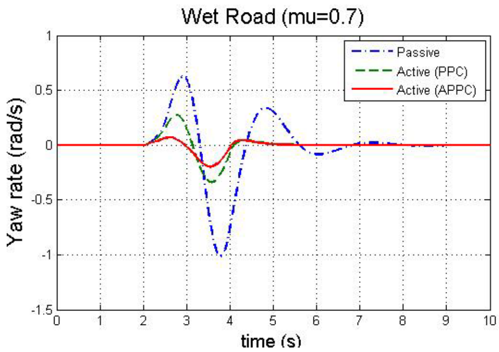

Fig. 11 Yaw rate (wet road)

C. Icy Road Condition

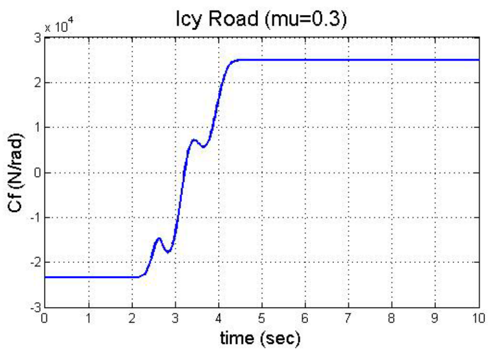

Fig. 12 Estimated front cornering stiffness (icy road)

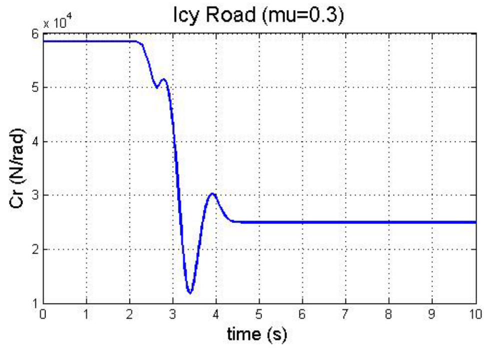

Fig. 13 Estimated rear cornering stiffness (icy road)

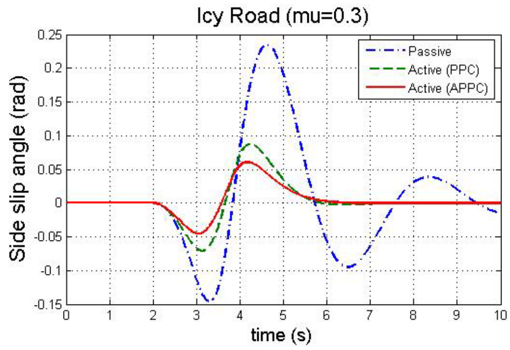

Fig. 14 Side slip angle (icy road)

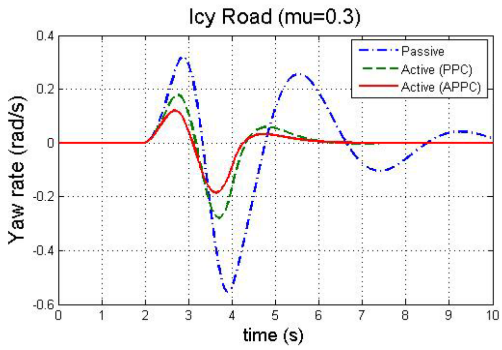

Fig. 15 Yaw rate (icy road)

In all cases (dry, wet, and icy road conditions), the performance of the active system is significantly higher than passive system to damp oscillations due to the sinusoidal input (disturbance) to the system. Moreover, the adaptive active system (APPC) works better than the active system (PPC). The side slip angle plot (Figs. 6, 10, and 14) and the yaw rate plot (Figs. 7, 11, and 15) of the vehicle model prove the advantages of APPC strategy to enhance active safety of a vehicle when subjected to the disturbances (e.g. sliding or spinning).

V. Conclusion

In this paper, the dynamic model of an active steering system has been formulated and derived based on the well-known bicycle model. The uncertain parameters (front and rear cornering stiffness) have been estimated by employing the gradient algorithm. PPC strategy is used to generate feedback gain to improve the dynamics' performance of the proposed model (e.g. reduce yaw rate and side slip angle). In the proposed APPC, the estimated parameters are fed into the PPCbox to generate feedback gain at each time step.

The simulations were generated in MATLAB/Simulink [9], and the results were shown for a sinusoidal steering angle, as an input to the system, for three types of road conditions (dry, wet, and icy). In all cases, the performance of the system with the APPC is better than the PPC, and significantly better than passive system (without any controller).

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