Predictive Control for improving vehicle handling and stability

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Abstract

In the modern era active front steering (*AFS*) Control has been introduced in automotive industry due to increased vehicle maneuverability resulting risk of instability and the objective to reduce the occurrence of accidents. In this paper a novel control strategy of Model predictive controller (*MPC)* controller with robust optimal guaranteed cost feedback controller (OGCC) is introduced for *AFS* control. In the proposed scheme *MPC* is designed to compensate for the effect of the driver's steering input modeled as a known disturbance and giving the optimal solution by satisfy the control input constraint, while the OGCC controller is trying to render the system robust to the effect of system uncertainties whilst achieving acceptable performance for tracking the desired vehicle trajectory. The complete controller scheme achieves the best results with root mean square error of less then 10^{-4}, whilst guaranteeing the stability of vehicle in the presence of uncertain environment. Numerical simulation results are performed to the effectiveness of the proposed approach over other conventional techniques.

1. Introduction

As the emergence of automobile industry in 21st century and improvement in sensor technologies and improvement made in industry equipments the need of safety factor improvement become a very important issue in vehicle design and handling. We can divide the safety factor into two types the passive safety factor and the active safety factor. Passive safety factors come into action at the time of occurrence of accident and prevent the passengers from major injuries e.g. Seat belts, safety glass and air bags etc. while the active safety factors prevent vehicle from accident by automatically detecting the critical unstable situation and then by correcting the steer angle and by applying automated braking so that vehicle can maintain its stability that's why the active safety factors are the main interest of today's research.

In active safety factors various research articles has been published in which different control strategies has been employed for stabilizing and handling of vehicle such as active yaw stability control (AYC) with PID controller is discussed in [1]. Active differential control (ADC) was employed in [2]. Active steering control (ASC) with sliding mode controller was discussed in [3]. Active steering control with braking force distribution is discussed in [4]. The vehicle stability control (VSC) and the active front steer control (AFS) are also discussed in various research articles. The aim in all these control strategies is to stabilize the vehicle in different maneuvers and prevent it from the occurrence of accident.

The AFS deals with the handling issues of vehicle and compensate the steering angle that will be able to stabilize the vehicle in critical handling situations [5]. AFS control system provides more comfort and it reduces the steering effort and provides the enhanced dynamic behavior for the steering system [6]. Today's hydraulic power assisted steering system is commercially used for AFS system [7].

Many researchers have proposed controller for AFS system these controller uses feedback of yaw rate and try to control lateral motion and try to stabilize the vehicle in different maneuvers feedback H infinity control proposed by [8]. The aim of the proposed work is to stabilize vehicle by correcting the steering angle by controlling the yaw rate response of vehicle from falling into any unstable status.

In the proposed work we want to stabilize the vehicle by handling it in different critical situation using AFS control strategy with Model predictive controller. In MPC control strategy we control the plant on the basis of its model so if we do an accurate modeling MPC controller will give good result as in our case input steer given by driver acts as disturbance so if we model this disturbance information in our modeling we can easily reject its effect using MPC .The greater advantage of MPC control technique is that we can handle our input, state and output constraints easily and can avoid from saturation and we can guarantee that our system will work in safe zone in our case optimal superposition steering angle is input constraint we have to calculate the steering angle such that it does not exceed the maximum limit of steering angle. Steering angle is an input constraint and we solve FHOCP by RH strategy at each sampling instant MPC will solve the close loop problem and will use the current state value to predict the future outputs. An integrated control algorithm based on MPC for AFS and yaw moment control is proposed in [11]. MPC controller to control AFS and direct yaw moment is discussed in [12]. MPC controller to study the effect of roll dynamics is presented in [13]. In all these control strategies the uncertainties of systems are not considered and the Active steer disturbance rejection are not modeled in MPC but in proposed control strategy state feedback controller designed using optimal guaranteed cost controller (OGCC) [9] is used to reject the uncertainties of system and then combined with MPC to further reject the active front steer known disturbance and the combined controller will perform well in the presence of disturbance and uncertainties in system. The state feedback gain is obtained by solving LMI given in [9] the obtained feedback gain will guarantee the uncertainties nullification.

The rest of the paper is organized in the following sections. Section II discuss the vehicle model 2-DOF (Degree of Freedom) model is briefly discussed. Section III discusses the control system design first we discuss the desired model then Active front steer control is discussed briefly and then MPC problem formulation is discussed Section IV discusses the simulation results and Section V presents the concluding remarks.

2. Vehicle Model

For studying the handling stability of vehicle over taking different maneuvers the classical bicycle model is used extensively for controller design purpose because it covers the most important features of vehicle that are enough for stabilizing vehicle over different maneuvers. [10]. The 2 wheel bicycle model is linearized from vehicle model based on some assumptions first assumed that every tire force is operating in linear region, vehicle motion is on flat surface, On the front of vehicle left and right wheels and on the rare side of vehicle left and right wheels are lumped in a single wheel at the mid line of vehicle body, it is assumed the vehicle is moving with constant speed with zero acceleration on longitudinal axis (a_x=0), it is assumed that steering angle and sideslip angle are small, braking torque is not applied on any wheels, as the vehicle mass is changing so it is assumed that centre of mass does not shifts, the wheels on the front side have the same steering angle, the desired sideslip of vehicle is assumed to be zero in the steady state motion of vehicle.

2.1 2-DOF bicycle model

The handling of vehicle and its stability, the 2-DOF bicycle model is used. It is used due to the model simplicity and the capability to imitate the performance of stability and handling. In this model rear and front wheels are composed of a one wheel. The movement of the wheels relative to the body of the vehicle, vertical roll motion and pitch are ignored. Yaw motion and lateral motion are used to represent the 2 DOF models [14].

$$m_t V_x(\dot{\beta} + \dot{\psi}) = F_{yf} + F_{yr} - \dot{\psi} \tag{1}$$

$$I_z \ddot{\psi} = l_f F_{yf} - l_r F_{yr} \tag{2}$$

Where

$$F_{yf} = 2C_f(\delta_{fd} - \beta - (l_f \dot{\psi}/V_x))$$

$$F_{yr} = 2C_r(-\beta + (l_r \dot{\psi}/V_x))$$

$$\beta = {}^{V_y}\!/_{V_x}$$

The notations used in the above equations are
C_f, C_r=cornering stiffness of the front and rear tyre respectively.
F_{yf}, F_{yr} = lateral force for each tyre.
I_z = moment of inertia about z-axis.
l_f, l_r= distances from the vehicle centre point to the front axle and rear axle respectively.
V_x, V_y= vehicle longitudinal velocity and vehicle lateral velocity.
δ_{fd}= steer angle
β=sideslip angle
And
ψ= yaw rate
The state space model of 2-DOF bicycle model can be rewritten as [14],[15].

$$\dot{x} = A_0 x + B_0 u \tag{3}$$

Where

$$x = \begin{bmatrix} \beta \\ \dot{\psi} \end{bmatrix}$$

$$u = (\delta_c - \delta_{fd})$$

$$\delta_c = -Kx + v$$

In the above equation K is the state feedback control input calculated by OGCC discussed in [9] and v is the compensation control input calculated using MPC controller discussed in next section.

$$A_0 = \begin{bmatrix} -\dfrac{2C_f + 2C_r}{m_t V_x} & \dfrac{2C_r l_r - 2C_f l_f}{m_t V_x^2} - 1 \\ \dfrac{2C_r l_r - 2C_f l_f}{I_z} & \dfrac{2C_r l_r + 2C_f l_f}{I_z V_x} \end{bmatrix}$$

$$B_0 = \begin{bmatrix} \dfrac{2C_f}{m_t V_x} \\ \dfrac{2C_f l_f}{I_z} \end{bmatrix}$$

Here δ_{fd} represents the commanded steering angle (by the driver) and δ_c represents the corrective action by the controller. The 2 DOF vehicle bicycle model is shown in Fig. 1

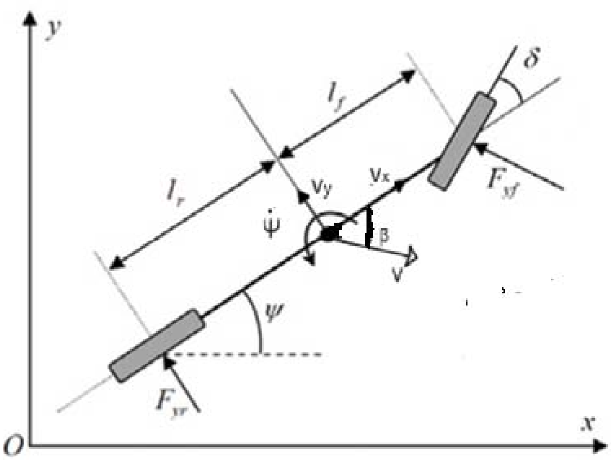

Fig. 1 Vehicle lateral dynamic
B. Mashadi (2014)

3. CONTROL SYSTEM DESIGN

3.1 Desired response model

The stability and handling performance is represented by slide slip angle response and yaw rate response respectively. They are also treated as desired responses and tracked by actual vehicle. Slide slip angle of the vehicle is equal to zero, [16]. By using longitudinal velocity and steering angle we can calculate the yaw response. A first order yaw rate response model is used. The desired model equations are

$$\beta_d = 0 \tag{4}$$

$$\ddot{\psi}_d = -\frac{1}{\tau_\gamma}\dot{\psi}_d + \frac{k_\gamma}{\tau_\gamma}\delta_{fd} \tag{5}$$

Where

$$\tau_\gamma = \frac{V_x}{(2C_f l_f(l_f + l_r) + m_t l_r l_f V_x^2)}$$

$$k_\gamma = \frac{2C_f l_f V_x(l_f + l_r)}{(l_f + m_t l_r V_x^2)}$$

Desired state space equation is

$$\dot{x}_d = A_d x_d + B_d w \tag{6}$$

$$x_d = \begin{bmatrix}\beta_d \\ \dot{\psi}_d\end{bmatrix}$$

$$A_d = \begin{bmatrix}0 & 0 \\ 0 & -\dfrac{1}{\tau_\gamma}\end{bmatrix}$$

$$B_d = \begin{bmatrix}0 \\ \dfrac{k_\gamma}{\tau_\gamma}\end{bmatrix}$$

The variables with subscript d represent the desired responses. δ_{fd} represents the steering command given by the driver. As mentioned in [17] cornering stiffness is not constant but varies with road adhesion coefficient.

3.2 Active Steering Control

Active steering control can be used in many ways for improving handling and stability. Types of active steering control are active front steering control (AFS), active rear steering control system (ARS), four wheel steering system (4WS). Most of the vehicles have front wheel steering system so the AFS control is more effective.

Fig. 2 shows front wheel angle is equal to the steering angle input given by the driver and superposition angle generated by the controller.

The AFS state space model is given bellow.

$$\begin{bmatrix}\dot{\beta} \\ \ddot{\psi}\end{bmatrix} = \begin{bmatrix}-\frac{2C_f+2C_r}{m_t V_x} & \frac{2C_r l_r - 2C_f l_f}{m_t V_x^2} - 1 \\ \frac{2C_r l_r - 2C_f l_f}{I_z} & \frac{2C_r l_r + 2C_f l_f}{I_z V_x}\end{bmatrix}\begin{bmatrix}\beta \\ \dot{\psi}\end{bmatrix} +$$

$$\begin{bmatrix}\frac{2C_f}{m_t V_x} \\ \frac{2C_f l_f}{I_z}\end{bmatrix}(\delta_c - \delta_{fd}) \tag{7}$$

$$\delta_c = -Kx + v \tag{8}$$

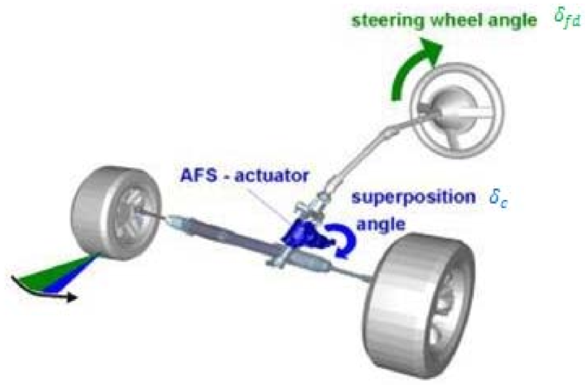

Fig. 2 AFS Control scheme
Xiang Dan (2011)

3.3 MPC Controller Design

First we have to obtain the discrete time equivalent model of the system given in Eq. 3, for that the state variable are sampled by using ZOH after measuring them with the sampling rate of T_s Hence the discrete time equivalence can be obtained by using Euler approximation

$$\dot{x}(t) \cong \frac{x(t+Ts) - x(t)}{Ts} \tag{9}$$

Then

$$A_d = (I + T_s A) \tag{10}$$

$$B_d = T_s B \tag{11}$$

Hence the discrete time equivalent system will be

$$x(k+1) = A_d x(k) + B_d u(k) \tag{12}$$

where

$$u(k) = -Kx(k) - \delta_{fd} + v(k) \tag{13}$$

Then

$$x(k+1) = A_d x(k) + B_d(-Kx(k) - \delta_{fd} + v(k))$$

$$x(k+1) = (A_d - KB_d)x(k) + B_d(v(k) - \delta_{fd}) \quad (14)$$

if

$$A_c = A_d - KB_d \quad (15)$$

$$v_c(k) = v(k) - \delta_{fd} \quad (16)$$

Then

$$x(k+1) = A_c x(k) + B_d v_c(k) \quad (17)$$

As in MPC we predict the future states by using the current state information so the formation of MPC can be done in following way

$$x(k+1) = A_c x(k) + B_d v_c(k)$$

$$x(k+2) = A_c^2 x(k) + A_c B_d v_c(k) + B_d v_c(k+1)$$

$$x(k+3) = A_c^3 x(k) + A_c^2 B_d v_c(k) + A_d B_d v_c(k) + B_d v_c(k+1)$$

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$$x(k+N) = A_c^N x(k) + A_c^{N-1} B_d x(k) + \cdots + B_d v_c(k+N-1)$$

In the above equation N represents the horizon length. In matrix form we can write it down as given below.

$$X = Hx(k) + \emptyset V \quad (18)$$

Where

$$H = \begin{bmatrix} A_c \\ A_c^2 \\ A_c^3 \\ \vdots \\ A_c^N \end{bmatrix}$$

And

$$\emptyset = \begin{bmatrix} B_d & \cdots & 0 \\ \vdots & \ddots & \vdots \\ A_c^{N-1} B_d & \cdots & B_d \end{bmatrix}$$

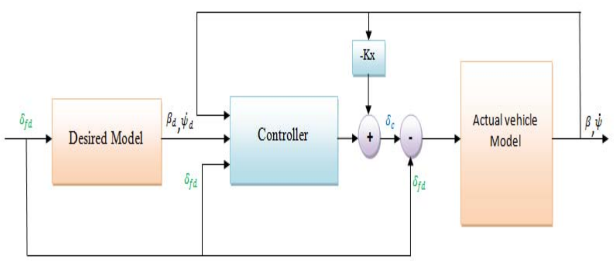

Fig.3 Block diagram of controller scheme

As the desire system calculates the desired sideslip and desire yaw rate so we can generate a vector containing the desired response information .

Let

$$x_d = [\beta_d \;\; \dot\psi_d]'$$

$$R_d = \begin{bmatrix} x_d \\ x_d \\ \vdots \\ x_d \end{bmatrix}$$

Its dimention will depend on the horizon length if N=5 then its dimention will be 10 rows and 1 column or $m * N \times 1$ where m are the number of states. Now we are in the position to write the cost function

$$cost = (X - R_d)'Q(X - R_d) + V'RV$$

In the above equation Q and R are weighting matrixes they can be defined as

$$Q = \begin{bmatrix} 1 & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & 1 \end{bmatrix} * 10000$$

$$R = \begin{bmatrix} 1 & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & 1 \end{bmatrix} * .1$$

The above cost function can be solved by using yalmip optimization tool. Figure 3 shows the complete block diagram of control scheme

4. SIMULATION RESULTS

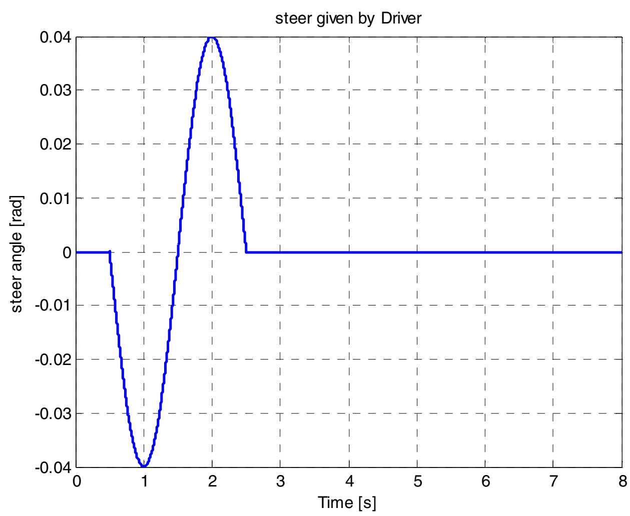

Fig. 4 Steering input signal

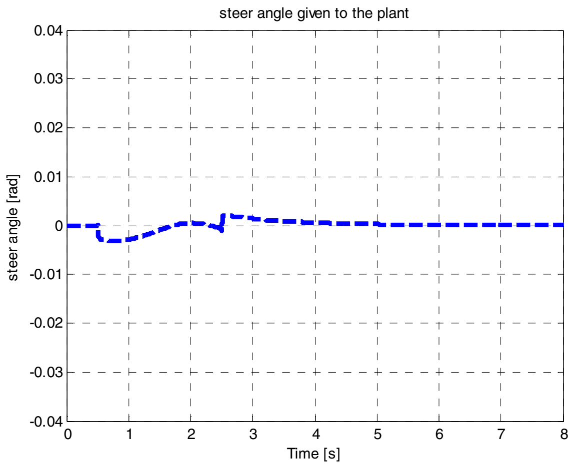

Fig. 5 Steer angle given to the vehicle

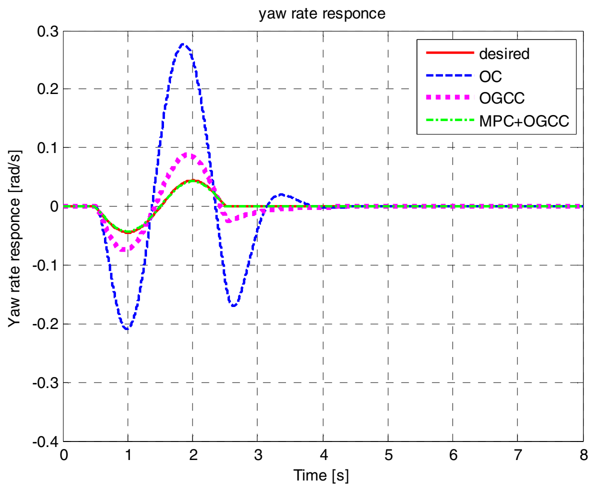

Fig. 6 Desired and actual yaw rate response (μ=0.2)

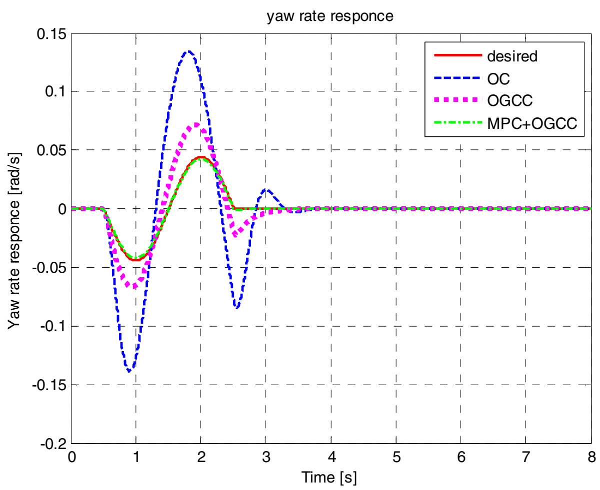

Fig. 7 Desired and actual yaw rate response (μ=0.8)

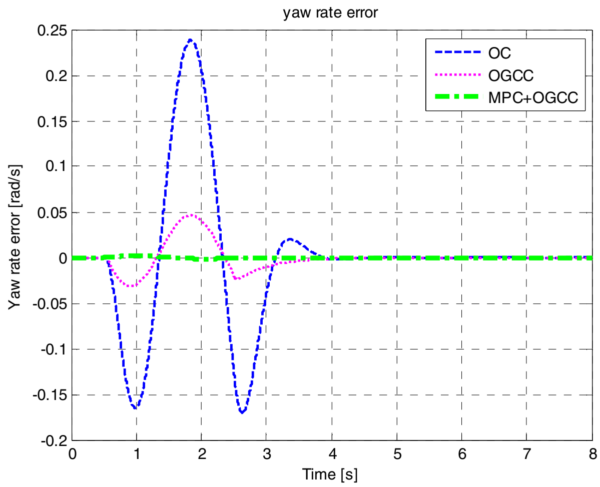

Fig. 8 Yaw rate error (μ=0.2)

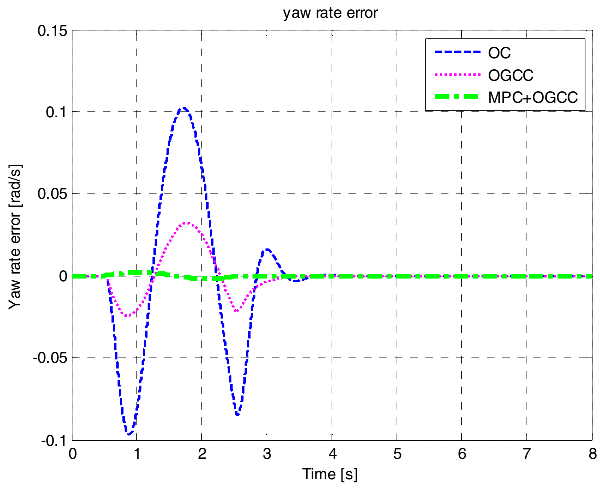

Fig. 9 Yaw rate error (μ=0.8)

Fig. 4 shows the steering angle input given by the driver from this information desired yaw rate response is produced and we want to track it on different road adhesion coefficients (μ=0.2, 0.8) as shown in the Fig. 6 and 7 respectively. Form the above plots we can see that the response of proposed controller is better than the other existing controllers.

Fig. 8 and 9 shows the yaw rate error of different controllers at different road adhesion coefficients and from the plots we can see that the suggested controller is very better than the optimal OC and OGCC controllers we can see that its response is so close to the actual response and have approximately zero error.

μ = road adhesion coefficient	OC	OGCC	MPC+OGCC
0.8	0.0357	0.0104	7.5057e-04
0.2	0.077	0.0145	6.3720e-04

Table 1. RMSE of Yaw Rate

6. Conclusions

In the given work we try to improve the handling and stability of vehicle in the presence of model uncertainties for dealing the model uncertainties we used an OGCC feedback controller and MPC controller is applied to further reduce the error between the actual and desired response by compensating the steer angle keeping the driver steer angle in mind so the total control strategy is able to nullify the model uncertainties and able to reach the desired response with approximately zero error.

7. References

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