# Optimal Control with Adaptive Weighting Coefficients for Integrated Vehicle Dynamics Control

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**Abstract:** In this work, an optimal control scheme with adaptive weighting coefficients is presented which coordinates different vehicle dynamics control objectives, thus ruling out possible conflicts among them. In a new approach, the weighting coefficients in optimal control are adjusted according to the vehicle state in the phase plane in such a way that a priority is given to each objective of handling and stability in each region. The optimal control acts as a high-level control for the vehicle body, which determines the body lateral force and yaw moment for stable vehicle motion. The body lateral force and yaw moment provide the inputs to the mid-level force (control) distribution module, which works out the desired lateral and longitudinal forces at each wheel. Therefore, the high-level control objectives are allocated to individual tire forces in an optimal manner with the assumption of a 4-wheel-independent car. A low-level control uses the desired individual tire forces to compute the steering angle and applied torque at each wheel. Simulation tests with a nonlinear vehicle model are conducted and comparison with the well-recognized work in the literature is made to show the efficiency of the proposed method.

**Keywords:** Adaptive Weighting Coefficients, Handling, Optimal control, Stability

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## 1 INTRODUCTION

A major part of advances in automotive technology in the past decade has been in the area of safety, and most modern passenger vehicles are now equipped with active safety systems. Vehicle dynamics control components are systems designed to provide safety by assisting driver in critical conditions, for which tires have gone saturated and corresponding nonlinear behaviors of tire forces preclude driver from a proper understanding of driving circumstances [1]. Handling and stability form the main objectives of vehicle dynamics control, which are achieved through following desired yaw rate and bounding side-slip angle within the stable region in the phase plane [2].
There are several ways of controlling the yaw dynamics as well as the side slip angle of a vehicle (see Ref. [3] and Refs. therein). Integrated control strategies have been introduced to control several aspects of vehicle dynamics, for different objectives, with avoiding possible conflicts [4-8]. In this regard, different subsystems are integrated in such a way that individual control objectives are achieved without negatively affecting each other. For example, Smakman [4] makes use of active suspension and braking subsystems, or Selby [5] and Burgio [7] integrate individual brakes and active steering for vehicle dynamics control. An integration of active steering, driveline, and braking subsystems is presented in Ref. [6].
In such works, vehicle dynamics control objectives are assigned to individual subsystems based on some fuzzy rules, without assuring the use of maximum capacity of the whole integrated system. Therefore, optimal control allocation (OCA) methods have been recently introduced into vehicle dynamics control to utilize the maximum capacity of available actuators in an integrated system [9-18]. In this regard, a set of actuators (individual tire forces) jointly produce body forces and moments commanded by some high-level controller, and in addition optimize an objective function for definite performance.
In this work, in a new approach, optimal control with adaptive weighting coefficients is presented for a vehicle stabilization scheme. Referring to Fig. 1, the stabilization strategy consists of a high-level module that deals with the vehicle dynamics control objectives (handling and stability), a low-level module that handles the steering angle and applied torque for each wheel, and a mid-level OCA module that generates tire longitudinal and lateral force references for the low-level control module. The high-level control design is based on Optimal Control with Adaptive Coefficients (OCAC), for which the weighting coefficients of the cost function are adapted based on the vehicle state in the phase plane such that the objectives of handling and stability are achieved with avoiding possible conflicts.

The mid-level OCA unit maps the high-level control objectives to desired individual tire forces. Therefore, the body lateral force and yaw moment of the high-level control are intended while minimizing a proper cost function. To this end, an optimization problem is defined and an analytical solution is derived such that a real-time implementation can be realized without the use of numeric optimization software. The low-level control uses the desired lateral and longitudinal forces of each tire to compute the corresponding wheel steering angles and applied torques.
The simulation cases show that the suggested control strategy stabilizes the vehicle in extreme maneuvers where the nonlinear vehicle dynamics otherwise (without active control) becomes unstable in the sense of over/under steering. In addition, comparison with the well-recognized work in the literature is made to demonstrate the efficiency of the proposed scheme in enhancing vehicle stability.

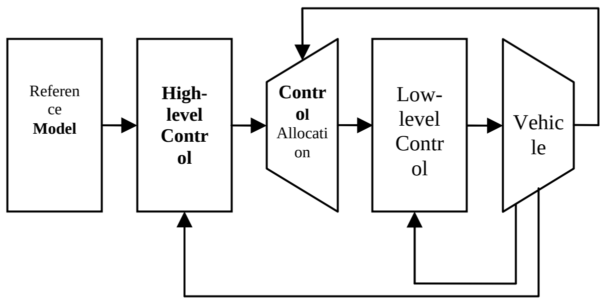


**Fig. 1** Integrated optimal control scheme

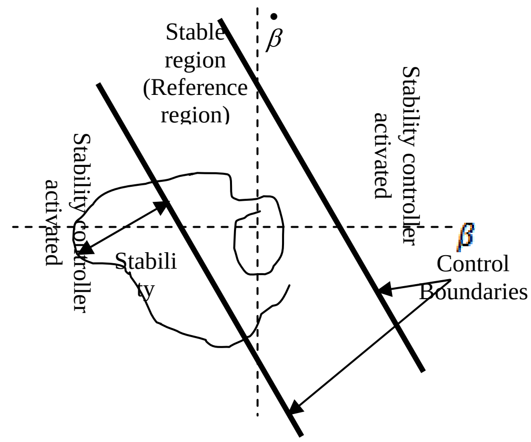


**Fig. 2** Different regions in $\beta - \dot{\beta}$ phase-plane [18]

## 2 HIGH-LEVEL CONTROL DESIGN

In general, vehicle handling and stability are achieved through following the desired yaw rate and limiting side-slip angle respectively. For this purpose, according to the vehicle lateral dynamics, there are two available virtual control inputs of total body lateral force ($Y$) and yaw moment ($M$). To design these virtual control inputs, an optimal control scheme with adaptive weighting coefficients is considered, for which the coefficients of the cost function are adjusted according to the phase-plane methodology. A description of stable and unstable regions in the phase-plane of the side-slip angle ($\beta$) has been shown in Fig. 2. Inside the stable region the side slip angle is small where the lateral tire forces possess linear characteristics.

Thus, the body lateral force increases with the vehicle side slip, resulting in a stable vehicle motion in the reference region. The lateral tire forces, however, saturate with the increase of the vehicle side slip beyond the stable region. In such conditions, further increase in the vehicle body lateral force is not viable, thus rendering the vehicle lateral dynamics unstable. So, the stability controller must be activated to control the vehicle outside the stable regime. The control design is based on a 2DOF linear vehicle model, with small angle and constant speed assumptions [19]. The basic equations of motion for this model are:

$$mV(\dot{\beta}+r)=Y \tag{1}$$

$$I_z\dot{r}=M \tag{2}$$

Where $m$ and $I_z$ denote the total mass and yaw inertia of the vehicle, $V$ is the vehicle velocity, and $r$ represents the vehicle yaw rate. Equations (1) and (2) are put into matrix form as follows:

$$\dot{x}=Ax+Bv \tag{3}$$

Where

$$x=\begin{bmatrix} r \\ \beta \end{bmatrix} \qquad v=\begin{bmatrix} M \\ Y \end{bmatrix} \qquad A=\begin{bmatrix} 0 & 0 \\ -\dfrac{1}{mV} & 0 \end{bmatrix}$$
$$B=\begin{bmatrix} \dfrac{1}{I_z} & 0 \\ 0 & \dfrac{1}{mV} \end{bmatrix} \tag{4}$$

For the high-level control design, the side slip angle $\beta$ and the yaw rate $r$ are considered as the two state variables while the body yaw moment $M$ and the body lateral force $Y$ constitute the set of control inputs, which must be determined from the control law. In what follows, first the design procedure based on optimal control is presented; then optimal control with adaptive weighting coefficients is demonstrated.

### 2.1. Optimal control

The target of the controller is to make the side slip angle $\beta$ and the yaw rate $r$ track the corresponding desired values of $\beta_d$=0 and $r_d$. In stationary turn, a definite relationship exists between the steering angle, the vehicle velocity and the yaw rate. This relation is used to derive the desired yaw rate ($r_d$) [1]. The integrated performance index of the optimal control is defined as:

$$J=\frac{1}{2}\int_0^{\infty}[R_M M^2+R_Y Y^2+Q_r(r-r_d)^2+Q_\beta\beta^2]dt \tag{5}$$

Where $R_M$, $R_Y$, $Q_r$, and $Q_\beta$ are the weighting coefficients that indicate the relative importance of the corresponding term. Considering Eq. (4), the cost function in Eq. (5) is written as:

$$J=\frac{1}{2}\int_0^{\infty}[v^T Rv+(x-x_d)^T Q(x-x_d)]dt \tag{6}$$

In which

$$R=diag(R_M,R_Y); \qquad Q=diag(Q_r,Q_\beta);$$
$$x_d=[r_d \quad \beta_d]^T \tag{7}$$

Optimal control problem is defined to find the control inputs $M$ and $Y$ that minimize the cost function Eq. (6) subjected to the system Eq. (3). The solution to this problem is provided in the form of the linear feedback/feed forward control law [20].

$$v=-R^{-1}B^T P \tag{8}$$

In which

$$P=Kx+S \tag{9}$$

Where the feedback gain $\boldsymbol{K}$ is the solution of the algebraic equation

$$KA+A^T K+Q-KBR^{-1}B^T K=0 \tag{10}$$

And the feed forward term $\boldsymbol{S}$ satisfy

$$(A^T-KBR^{-1}B^T)S-Qx_d=0 \tag{11}$$

### 2.2. Optimal control with adaptive weighting coefficients

Optimal control with adaptive weighting coefficients is used to make consistency between various control objectives and to rule out possible conflicts. Vehicle dynamics control objectives include handling and stability. Inside the stable region of the phase plane (Fig. 2), handling improvement is the main control objective that is achieved through tracking the desired yaw rate. On the other hand, vehicle stability is related directly to the side slip motion, and this motion should be restricted to the stable region. Inside the stable regime, where vehicle has small side slip angle and tires remain in linear region, vehicle stability is not a major concern and it is guaranteed by linear characteristics of lateral tire forces [4]. However, as the vehicle state leaves the reference (stable) boundaries, with the growth of the side slip angle, linear tire force property turns into saturation, and vehicle stability will be in danger.

In such conditions, stability dominates handling and the high level control should focus on returning the vehicle state into the reference region for stable vehicle motion. It is shown in Ref. [4] that vehicle handling improvement and stability cannot be fulfilled simultaneously outside the stable region due to the tires saturation in this regime, where the control objectives need to be put in order of priority. To this end, optimal control with adaptive weighting coefficients is suggested for integrated vehicle dynamics control. Consider the boundaries of the reference region in Fig. 2 defined through Ref. [5].

$$\left|\frac{1}{24}\dot{\beta}+\frac{4}{24}\beta\right|<1 \tag{12}$$

Then, the weighting coefficients of $Q_r$ and $Q_\beta$ are updated according to the vehicle state in the phase plane, as shown in Fig. 3. In this regard, when the vehicle state lies inside the stable regime the weighting coefficient corresponding to yaw rate tracking has a larger value than that of the side slip angle. Therefore, the performance index of handling in the cost function Eq. (5) dominates vehicle stability and vehicle handling is given priority inside the stable region. Conversely, beyond reference boundaries, vehicle stability is prioritized by choosing larger value for the weighting coefficient $Q_\beta$ (as shown in Fig. 3). In such conditions, the integrated vehicle dynamics control focuses to draw the phase plane trajectory back into the stable region and utilizes the control inputs in an optimal manner for this purpose. The considered adaptation mechanism in Fig. 3 transits the control task from one to another instead of switching between several types of control objectives. As a result, abrupt system responses that can be induced by sudden hard switching actions are excluded.

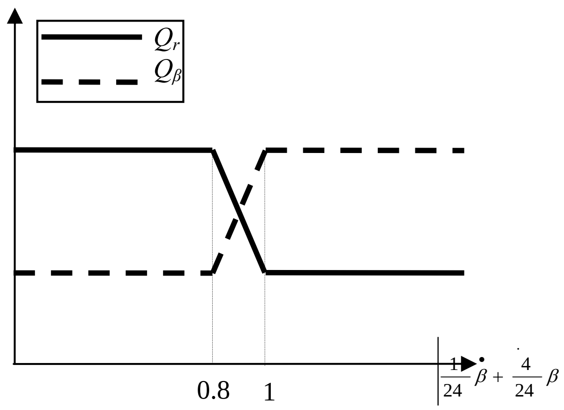


**Fig. 3** Adaptation of weighting coefficients based on phase plane notion

## 3 ALLOCATOIN OF HIGH-LEVEL CONTROL TO INDIVIDUAL TIRE FORCES

To achieve control objectives, the longitudinal and lateral forces of each tire must be determined. In this paper, a vehicle system in which each tire can be braked/derived and steered independently is considered. Thus, the overall control system contains eight constrained actuators for only three control objectives, making the integrated vehicle system an over-actuated control scheme. A general approach to resolve redundancy is to define an optimization problem where a cost function, for specific performance, is minimized. The well-accepted cost function for OCA in vehicle systems is the sum of work load of four wheels, which is written as:

$$f=\sum_{i=1}^{4}A_i\frac{X_i^2+Y_i^2}{(\mu_i Z_i)^2} \tag{13}$$

Where $i$ denotes the wheel number, $X_i$ and $Y_i$ stand for the desired longitudinal and lateral tire forces, respectively, $Z_i$ is the vertical load, $A_i$ is the weighting coefficient, and $\mu_i$ is friction coefficient of the $i^{th}$ tire. All the variables are defined in the vehicle body fixed coordinate system, as shown in Fig. 4. Defining the actuators vector, ***u***, an 8×1 vector which contains the desired tire forces, is given as:

$$u=[X_1 \quad X_2 \quad X_3 \quad X_4 \quad Y_1 \quad Y_2 \quad Y_3 \quad Y_4] \tag{14}$$

The cost function in matrix form is written as:

$$f(u)=u^T W u \tag{15}$$

In which $W_{8\times 8}$ is the diagonal weighting matrix for control allocation problem. To satisfy the high-level control objectives, all the variables in the cost function must satisfy the two equality constraints given by (Fig. 4):

$$Y = \sum_{i=1}^{4} Y_i \quad (16)$$

$$M = \sum_{i=1}^{2}(L_f Y_i - L_r Y_{(i+2)}) + \frac{d}{2}\sum_{i=1}^{2}(X_{2i} - X_{(2i-1)}) \quad (17)$$

Also, the longitudinal forces, $X_i$, should satisfy the demanded total longitudinal acceleration, $a_x$, by driver, i.e.

$$X = ma_x = \sum_{i=1}^{4} X_i \quad (18)$$

As observed in Eqs. (16)-(18), these equality constraints are linear in terms of actuator inputs and can be expressed in matrix form as :

$$Au = v \quad (19)$$

Where $A \in R^{3\times 8}$ is the constant matrix.

$$A = \begin{bmatrix} 1 & 1 & 1 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 1 & 1 & 1 \\ -\frac{d}{2} & \frac{d}{2} & -\frac{d}{2} & \frac{d}{2} & L_f & L_f & -L_r & -L_r \end{bmatrix} \quad (20)$$

And the vector of generalized forces/moment, $\boldsymbol{v}$, is given by:

$$v = \begin{bmatrix} X & Y & M \end{bmatrix}^T \quad (21)$$

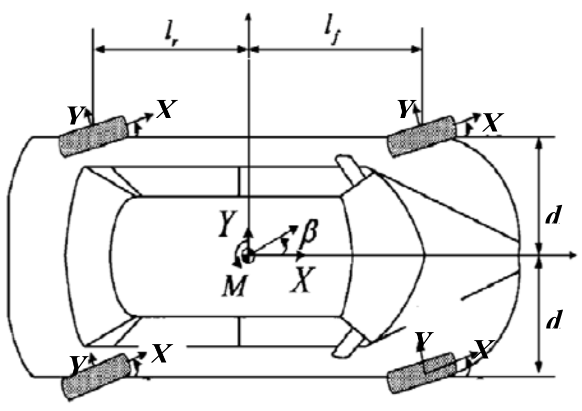


**Fig. 4** Vehicle plane motion parameters

Now, the optimization problem is defined as follows:

$$\begin{cases} Minimize & f = (1/2)u^T Wu \\ Subject-to & g = Au - v = 0 \end{cases} \quad (22)$$

To realize $\boldsymbol{u}$, first order optimality conditions can be written as:

$$\begin{cases} \frac{\partial f}{\partial u} + \sum_{i=1}^{3} \lambda_i \frac{\partial g_i}{\partial u} = 0 \Rightarrow \quad Wu + A^T\lambda = 0 \\ \qquad\qquad Au = v \end{cases} \quad (23)$$

In which $\lambda$ denotes the vector of Lagrange multipliers for equality constraints. To solve the set of linear equations in Eq. (23), first $u$ is written in terms of $\lambda$ as:

$$u = -W^{-1}A^T\lambda \quad (24)$$

Next Eq. (24) is inserted into second part of Eq. (23) and solves the corresponding equation for $\boldsymbol{\lambda}$.

$$\lambda = -(AW^{-1}A^T)^{-1}v \quad (25)$$

Using Eq. (25) in first part of Eq. (23), $u$ is attained to be:

$$u = W^{-1}A^T(AW^{-1}A^T)^{-1}v \quad (26)$$

Subsequently the desired longitudinal and lateral forces of each tire are determined by the use of the vector of control input $u$ . Then, using the inverse of a simple tire model [9], the active steering angle of each wheel ($\delta_i$) can be determined. In addition, the individual longitudinal forces are fed into a low-level slip ratio controller to realize the applied braking torque at each wheel.

## 4 LOW-LEVEL SLIP RATIO CONTROL DESIGN

The longitudinal force of each tire is related to the corresponding longitudinal slip ratio and is adjusted through Slip Ratio Control (SRC). The slip ratio of the $i^{th}$ tire, $\sigma_i$ , is defined as :

$$\sigma_i = \frac{R\omega_i - V}{V} \qquad during\ braking \quad (27)$$

$$\sigma_i = \frac{R\omega_i - V}{R\omega_i} \quad during\ Acceleration \qquad (28)$$

Where $R$ denotes the radius and $\omega_i$ is the angular velocity of the $i^{th}$ wheel. In the case where longitudinal slip ratio is small, the longitudinal tire force is found to be proportional to the slip ratio. Then, it gains its maximum value at a typical value of $\sigma^*$, after which it starts to lessen. Experimental studies have established that the tire lateral force decreases with increasing slip ratios greater than $|\sigma^*|$ as well [21]. In this paper, the SRC scheme presented in Ref. [17] is employed. In this regard, when the tire slip ratio is smaller than $\sigma^*$, by neglecting the wheel rotational inertia [9], the applied braking/traction torque, $T_i$, at wheel $i$ is obtained as:

$$T_i = RX_i \qquad (29)$$

In this case, the SRC works for Desired Longitudinal Force Generation (DLFG). However, when the demanded $X_i$ is too high, applying (29) would increase the slip ratio beyond $\sigma^*$, inevitably leading to wheel lock and lateral tire force drop. In such conditions, the idea of Anti-lock Braking System (ABS) is employed to keep the slip ratio of tires at $\sigma^*$. This idea is utilized during both braking and traction. When traction torque applies the proposed slip ratio control is in the Traction Control System (TCS) mode. The SRC scheme is shown in Fig. 5.

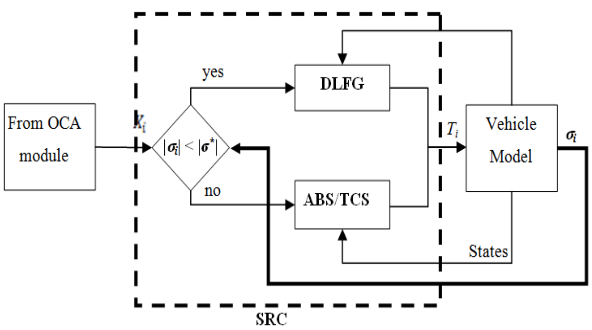


**Fig. 5** The SRC scheme applied to the vehicle system [17]

## 5 SIMULATION RESULTS

To evaluate the proposed vehicle dynamics control scheme, computer simulations are performed. A 9DOF nonlinear vehicle model with Dug off's tire model and a driver model [18] is utilized for this purpose. The vehicle behavior is tested during a single lane change maneuver with driver's braking. The vehicle is assumed to move with an initial velocity of 130 km/h on a slippery road with the coefficient of friction 0.3. To show the efficiency of the presented method, comparisons are made with the results of integrated vehicle control including conventional Optimal Control (OC) as the high-level unit (Section 2.1).

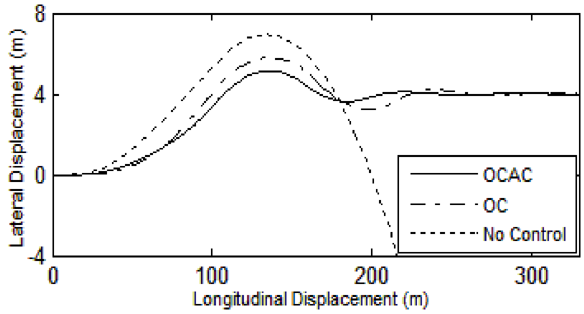


**Fig. 6** Vehicle path

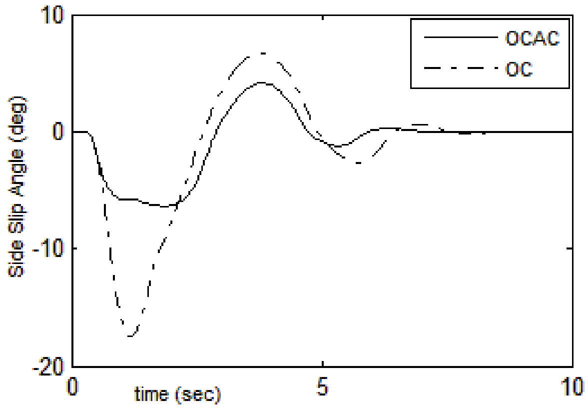


**Fig. 7** Vehicle side slip angle

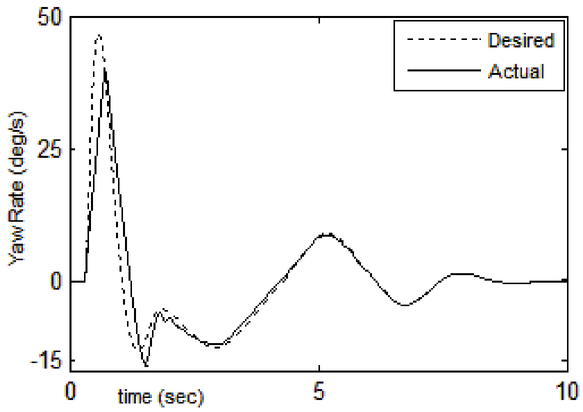


**Fig. 8** Vehicle yaw rate by OC

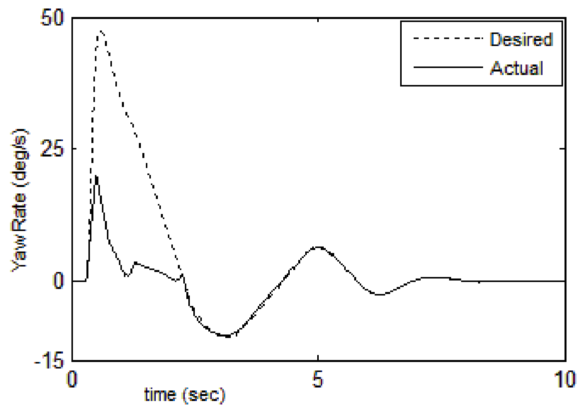


**Fig. 9** Vehicle yaw rate by OCAC

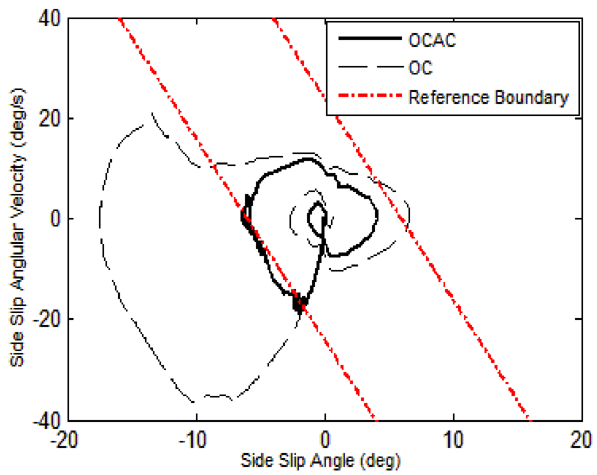


**Fig. 10** Phase plane trajectory

The simulation results for this scenario are shown in figures 6-10. As shown by Fig. 6, the vehicle with no active control and only guided by driver has an unstable response, representing the adverse conditions of the considered maneuver. However, the integrated control system with either OCAC or OC as the high-level control makes the vehicle converge properly to the driver's intended path. Furthermore, when compared to the system with OC, the vehicle under the presented method (OCAC) has a faster convergence and less deviation from the desired path.

Vehicle side slip angle for both methods is presented in Fig. 7, which shows the superiority of the proposed method in reducing the side slip motion, in that OCAC has much less side slip angle than OC does. Moreover, yaw rate time histories in figures 8 and 9 establish that, actual yaw rate under OCAC has faster convergence to the desired value and is smoother than that of OC.

In Fig. 10, after phase plane trajectory has left the stable region, it has been attracted again to this area immediately due to efficient function of the considered adaptation law. In the unstable regime, maintaining the vehicle stability dominates the task of improving vehicle handling, thus deteriorating the quality of yaw rate tracking in some parts (Fig. 9). At the same time, the integrated system with high-level OC pulls the state trajectory further out of the stable region, due to control objective conflicts at the handling limit by this approach.

## 6 CONCLUSION

Four-wheel independent steering/driving vehicle dynamics control system can be described as a redundantly actuated system, for which input redundancy may be used to achieve optimized control performances. In this way, the overall vehicle control scheme is split into three layers, where generalized forces/moment for body control are determined by a high-level optimal control as the first layer. To organize different control objectives to rule out possible conflicts, an optimal control scheme with adaptive weighting coefficients is presented where contribution of each control objective in the cost function is adjusted based on the vehicle state in the phase plane. To produce body forces/moment, a mid-level force distribution module allocates the high-level control outputs to individual longitudinal and lateral forces. To achieve high-level objectives in an optimal way an optimization problem is defined and solved analytically. The applied torque and steering angle at each wheel are determined through a low-level unit, comprising a slip ratio control scheme. Simulation testing is performed using a nonlinear vehicle model to implement and compare the presented vehicle dynamics approach with other vehicle control approaches. Testing conducted under various adverse driving conditions showed that the integrated vehicle control developed in this work significantly enhance vehicle handling and stability.

## 7 NOMENCLATURE

| | |
|---|---|
| $\boldsymbol{A}$ | state matrix |
| $a_x$ | driver's braking acceleration |
| $A_i$ | weighting coefficients of *i*th tire |
| $\boldsymbol{B}$ | control input matrix |
| $D$ | vehicle tread |
| $F$ | objective function in OCA |
| $I_z$ | yaw moment of inertia of vehicle |
| $J$ | objective function in optimal control |
| $\boldsymbol{K}$ | optimal control gains matrix |
| $L_{f,r}$ | distance between mass center and axle |
| $m$ | vehicle mass |

| | |
|---|---|
| $M$ | body yaw moment |
| $\boldsymbol{P}$ | matrix in Ricatti equation |
| $\boldsymbol{Q}$ | matrix of weighting coefficients for vehicle state |
| $Q_r$ | weighting coefficient of $r$ in optimal control |
| $Q_\beta$ | weighting coefficient of $\beta$ in optimal control |
| $R$ | wheel radius |
| $\boldsymbol{R}$ | matrix of weighting coefficients for virtual control input |
| $R_M$ | weighting coefficient of $M$ in optimal control |
| $R_Y$ | weighting coefficient of $Y$ in optimal control |
| $r$ | yaw angle velocity |
| $\boldsymbol{S}$ | matrix in Ricatti equation |
| $T_i$ | applied torque at the $i$th wheel |
| $\boldsymbol{u}$ | vector of allocated tire forces |
| $V$ | vehicle velocity |
| $v_{xi}$ | longitudinal velocity of the $i$th tire |
| $\boldsymbol{v}$ | vector of virtual control input |
| $\boldsymbol{W}$ | matrix of weighting coefficients in OCA |
| $\boldsymbol{x}$ | state vector |
| $X$ | Body longitudinal force |
| $Y$ | Body lateral force |
| $X_i$ | allocated longitudinal force to the $i$th tire |
| $Y_i$ | allocated lateral force to the $i$th tire |
| $Z_i$ | vertical load of $i$th tire |

**Greek letters**

| | |
|---|---|
| $\beta$ | vehicle side slip angle |
| $\delta_i$ | steering angle of $i$th tire |
| $\sigma_i$ | slip ratio of the $i$th tire |
| $\sigma^*$ | reference value of slip ratio |
| $\lambda_i$ | vector of Lagrange multipliers |
| $\mu_i$ | friction coefficient of the $i$th tire |
| $\omega_i$ | angular velocity of the $i$th tire |

**Subscripts/Superscripts**

| | |
|---|---|
| $f$ | front |
| $r$ | rear |
| $d$ | desired value |

**Coordinate system**

| | |
|---|---|
| $(x,y,z)$ | moving coordinate attached to vehicle centre |

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