

Skid Steering based Driving Control of a Robotic Vehicle with Six In-Wheel Drives

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ABSTRACT

This paper describes a driving control algorithm based on a skid steering for a Robotic Vehicle with Articulated Suspension (RVAS). The RVAS is a kind of unmanned ground vehicle based on a skid steering using independent in-wheel drive at each wheel. The driving control algorithm consists of four parts: a speed controller for following a desired speed, a lateral motion controller that computes a yaw moment input to track a desired yaw rate or a desired trajectory according to the control mode, a longitudinal tire force distribution algorithm that determines an optimal desired longitudinal tire force and a wheel torque controller that determines a wheel torque command at each wheel in order to keep the slip ratio at each wheel below a limit value as well as to track the desired tire force. The longitudinal and vertical tire force estimators are required for the optimal tire force distribution and wheel slip control. A dynamic model of the RVAS is developed for simulation study and validated using the vehicle test data. Simulation and vehicle tests are conducted in order to evaluate the proposed driving controller. It is found from simulation and vehicle test results that the proposed driving controller provides satisfactory motion control performance according to the control mode.

INTRODUCTION

Recently, diverse unmanned ground vehicles are developed in order to conduct multi-tasks such as logistics supports, surveillance and light combat operation. In this paper, as a part of autonomous vehicle control for military or robotic

vehicle, skid steering based autonomous driving control algorithm is investigated. Robotic Vehicle with Articulated Suspension (RVAS), as shown in [Figure.1](#), is a kind of unmanned ground vehicle based on a skid steering using independent in-wheel drive at each wheel. The RVAS, unlike the conventional wheeled vehicles, is not equipped with steering linkages. Instead, it is steered through differential traction force which is occurred from in-wheel motor at each wheel. Steering in this fashion requires much more power consumption than in kinematic steering using Ackerman's linkages [1, 2]. However, this offers simple structure and more room in the vehicle for mission equipment installment. In the aspect of mobility, the RVAS benefits from its in-wheel drives and articulated suspensions, which provide an independent wheel traction control capability and a great improvement of obstacle negotiation ability, respectively. This paper focuses on the control strategy based on skid steering of the RVAS in order to track a desired yaw rate or a desired trajectory according to the control mode.

A dynamic model for the RVAS is developed and validated using test data. Simulation studies and vehicle tests are conducted in order to evaluate the proposed driving controller. It is found from the simulation and test results that the proposed driving controller based on a skid steering provides good maneuverability.



Figure.1. Robotic Vehicle with Articulated Suspension (RVAS)

2. DYNAMIC MODEL OF A VEHICLE WITH INDEPENDENT IN-WHEEL DRIVE

2.1. DEVELOPMENT OF VEHICLE DYNAMIC MODEL

A full dynamic model of the RVAS is developed to conduct numerical simulation studies. The full dynamic model of the RVAS is designed as three parts; driving system, arm dynamic model and vehicle body dynamic model as shown in [Figure.2](#). The driving system contains an in-wheel motor model, a wheel dynamic model and a tire model. The arm dynamic model determines the dynamic behaviors of i -th arm rod. The vehicle body dynamic model is designed to represent the dynamic behavior of a sprung mass.

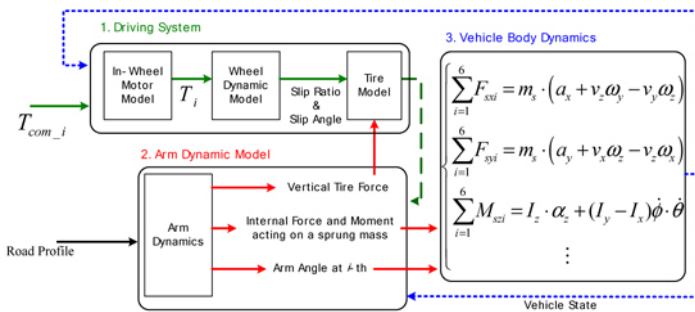


Figure.2. The schematic diagram of Dynamic Model

The driving system in the dynamic model consists of in-wheel motor model, wheel dynamic model and tire model. The in-wheel motor is modeled using a first order transfer function. The torque/speed curve as shown in [Figure.3](#) is used to consider the maximum power of the in-wheel motor in the dynamic model of the RVAS.

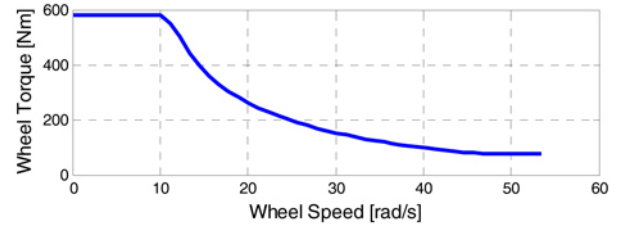


Figure.3. Torque/Speed Curve of each In-wheel Motor

Longitudinal and lateral tire model are modeled using a combined Pacejka tire model [2]. The tire forces are related to wheel slip ratio and slip angle. The wheel slip ratio and the slip angle are defined as in [Eq. \(1\)](#).

$$\lambda_i = \begin{cases} \frac{R_i \cdot \omega_i - v_{txi}}{R_i \cdot \omega_i} & \text{if } (\lambda_i > 0) \\ \frac{R_i \cdot \omega_i - v_{txi}}{v_{txi}} & \text{if } (\lambda_i \leq 0) \end{cases}, \quad \alpha_i = -\tan^{-1} \left(\frac{v_{tyi}}{v_{txi}} \right) \quad (1)$$

where, v_{txi} and v_{tyi} denote the longitudinal and lateral velocity at the tire center and can be calculated as follows:

$$\dot{r}_{wi} = \begin{bmatrix} v_{txi} \\ v_{tyi} \\ v_{zi} \end{bmatrix} = \dot{r}_{cg} + \omega_{ch} \times r_{i/cg} + \omega_{arm} \times r_{wi/i} = \begin{bmatrix} v_x \\ v_y \\ v_z \end{bmatrix} + \begin{bmatrix} \dot{\phi} \\ \dot{\theta} \\ \dot{\psi} \end{bmatrix} \times \begin{bmatrix} l_i \\ \pm l_w \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ \dot{\theta}_{arm-i} \\ 0 \end{bmatrix} \times \begin{bmatrix} -l_{arm} \cdot \sin(\theta_{arm-i}) \\ 0 \\ -l_{arm} \cdot \cos(\theta_{arm-i}) \end{bmatrix} \quad (2)$$

The arm dynamic model determines the arm behavior of each arm rod and internal forces and moments acting on a sprung mass. The arm behavior is occurred from longitudinal, lateral and vertical tire forces and arm spring and damping torques. [Figure.4](#) shows the dynamic equilibrium of i -th arm rod.

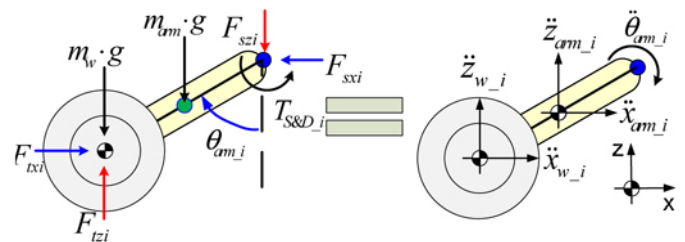


Figure.4. Dynamic Equilibrium of i -th Arm Rod

From [Figure.4](#), the dynamic equations for the arm dynamic model can be expressed as follows:

$$\begin{aligned} \sum F_x &= F_{txi} - F_{sxi} = & m_{arm} \cdot \ddot{x}_{arm-i} + m_w \cdot \ddot{x}_{w-i} \\ \sum F_y &= F_{tyi} - F_{syi} = & m_{arm} \cdot \ddot{y}_{arm-i} + m_w \cdot \ddot{y}_{w-i} \\ \sum F_z &= F_{tzi} - F_{szi} - (m_w + m_{arm}) \cdot g = & m_{arm} \cdot \ddot{z}_{arm-i} + m_w \cdot \ddot{z}_{w-i} \end{aligned} \quad (3)$$



$$\begin{aligned}\sum M_x &= I_{arm_x} \cdot \alpha_{arm_x} + (I_{arm_z} - I_{arm_y}) \cdot \omega_{arm_y} \cdot \omega_{arm_z} + I_{arm} \cdot \cos(\theta_{arm_i}) \cdot \left(m_w \cdot \ddot{y}_{wi} + m_{arm} \cdot \frac{\ddot{y}_{arm_i}}{2} \right) \\ \sum M_y &= \left\{ \begin{aligned} &I_{arm_y} \cdot \alpha_{arm_y} + (I_{arm_z} - I_{arm_x}) \cdot \omega_{arm_x} \cdot \omega_{arm_z} \\ &- \left(m_w \cdot \ddot{x}_{wi} + m_{arm} \cdot \frac{\ddot{x}_{arm_i}}{2} \right) \cdot I_{arm} \cdot \cos(\theta_{arm_i}) + \left(m_w \cdot \ddot{z}_{wi} + m_{arm} \cdot \frac{\ddot{z}_{arm_i}}{2} \right) \cdot I_{arm} \cdot \sin(\theta_{arm_i}) \end{aligned} \right\} \\ \sum M_z &= I_{arm_z} \cdot \alpha_{arm_z} + (I_{arm_y} - I_{arm_x}) \cdot \omega_{arm_x} \cdot \omega_{arm_y} - I_{arm} \cdot \sin(\theta_{arm_i}) \cdot \left(m_w \cdot \ddot{y}_{wi} + m_{arm} \cdot \frac{\ddot{y}_{arm_i}}{2} \right)\end{aligned}\quad (4)$$

In Eqns. (3) ~ (4), the rotational accelerations of the arm rod and the translational accelerations of i-th wheel are given as shown in Eqns. (5) ~ (6) [18].

$$\alpha_{arm} = \begin{bmatrix} \alpha_{arm_x} \\ \alpha_{arm_y} \\ \alpha_{arm_z} \end{bmatrix} = \alpha_{ch} + \alpha_{arm/ch} + \omega_{ch} \times \omega_{arm/ch} = \begin{bmatrix} \ddot{\phi} \\ \ddot{\theta} \\ \ddot{\psi} \end{bmatrix} + \begin{bmatrix} 0 \\ \ddot{\theta}_{arm_i} \\ 0 \end{bmatrix} + \begin{bmatrix} \dot{\phi} \\ \dot{\theta} \\ \dot{\psi} \end{bmatrix} \times \begin{bmatrix} 0 \\ \dot{\theta}_{arm_i} \\ 0 \end{bmatrix} = \begin{bmatrix} \ddot{\phi} - \dot{\theta}_{arm_i} \cdot \dot{\phi} \\ \ddot{\theta} + \dot{\theta}_{arm_i} \\ \ddot{\psi} + \dot{\theta}_{arm_i} \cdot \dot{\psi} \end{bmatrix}\quad (5)$$

$$\begin{aligned}\ddot{r}_{wi} &= [\ddot{x}_{wi} \quad \ddot{y}_{wi} \quad \ddot{z}_{wi}]^T = \ddot{r}_i + \omega_{arm} \times [\omega_{arm} \times r_{wi/i}] + \alpha_{arm} \times r_{wi/i} \\ \ddot{r}_i &= [\ddot{x}_i \quad \ddot{y}_i \quad \ddot{z}_i]^T = \ddot{r}_{cg} + \omega_{ch} \times [\omega_{ch} \times r_{i/cg}] + \alpha_{ch} \times r_{i/cg}\end{aligned}\quad (6)$$

The moment summations in Eq. (4) are shown in Eq. (7).

$$\begin{cases} \sum M_x = -M_{sxi} + I_{arm} \cdot \cos(\theta_{arm_i}) \cdot F_{y1} \\ \sum M_y = -T_{S\&D_i} - I_{arm} \cdot \sin(\theta_{arm_i}) \cdot (m_w + m_{arm}/2) \cdot g + I_{arm} \cdot \sin(\theta_{arm_i}) \cdot F_{zxi} - I_{arm} \cdot \cos(\theta_{arm_i}) \cdot F_{xzi} \\ \sum M_z = -M_{szi} - T_{szi} - I_{arm} \cdot \sin(\theta_{arm_i}) \cdot F_{y1} \end{cases}\quad (7)$$

where, $T_{S\&D_i}$ denotes a sum of spring and damping torques of i-th arm rod as follows:

$$T_{S\&D}(\theta_{arm_i}, \dot{\theta}_{arm_i}) = -k_s \cdot \theta_{arm_i} - c_d \cdot \dot{\theta}_{arm_i}\quad (8)$$

Substituting Eqns. (5) ~ (6) into Eq. (4), the dynamic equation for the arm motion can be obtained as follows:

$$\left\{ I_{arm_y} + \left(m_w + \frac{m_{arm}}{4} \right) l_{arm}^2 \right\} \ddot{\theta}_{arm_i} = \begin{cases} \sum M_y - \left\{ I_{arm_y} + \left(m_w + \frac{m_{arm}}{4} \right) l_{arm}^2 \right\} \ddot{\theta} - (I_{arm_x} - I_{arm_z}) \cdot \dot{\phi} \cdot \dot{\phi} \\ + \left(m_w + \frac{m_{arm}}{2} \right) l_{arm} \cdot [\cos(\theta_{arm_i}) \cdot \ddot{x} - \sin(\theta_{arm_i}) \cdot \ddot{z}] \\ - \left(m_w + \frac{m_{arm}}{4} \right) l_{arm}^2 \cdot [\sin(\theta_{arm_i}) \cdot \cos(\theta_{arm_i}) \cdot (\dot{\phi}^2 - \dot{\psi}^2) + \{\cos^2(\theta_{arm_i}) - \sin^2(\theta_{arm_i})\} \cdot \dot{\phi} \cdot \dot{\psi}] \end{cases}\quad (9)$$

Using the arm motion obtained from Eq. (9), the internal forces and moments acting on the sprung mass can be obtained from Eqns. (3) ~ (4). The vehicle body dynamic model determines dynamic behavior of the sprung mass using the internal forces and moments obtained from the arm dynamic model. The dynamic equations of the sprung mass can be obtained from the d'Alembert's principle [18].

2.2. VALIDATION OF DYNAMIC MODEL

Validation of the vehicle dynamic model has been conducted by comparison of simulation results to test data of the RVAS. For the validation of the dynamic model, the driving control algorithm which was implemented in the vehicle test is used identically in the dynamic model simulation. Also, the desired speed obtained from the vehicle test is used in the simulation. Figure.5 shows a comparison of test data and simulation results. From Figure.5, it is found that the simulation results are quite identical to the test data except the yaw rate as shown in Figure.5-(b). The difference between the test data and the simulation results is occurred from vertical road profile and an offset of mass center of the test platform. In the case of the vehicle test, the yaw rate is fluctuated by the road profile. Also, the offset of mass center causes the smaller left yaw motion than the right yaw motion in an identical yaw motion command. However, the simulation results agree generally with the test data, suggesting that the designed dynamic model is feasible as a test platform for development of the driving controller.

<figure 5 here>

3. CONTROLLER DESIGN

The driving control algorithm based on the skid steering is designed to control vehicle motion according to the control mode. There are two control modes. One is the remote control maneuvering mode. In the case of the remote control maneuvering mode, the vehicle motion is controlled according to remote driver's command, desired speed and steering command. The other is the autonomous driving control mode for tracking the desired trajectory. The control mode is determined by the remote driver. Figure.6 shows a control strategy for the driving controller proposed in this paper. Since it is difficult to determine the wheel torque command directly, the driving control algorithm for the RVAS is designed with three parts as follows: a upper-level controller, a tire force distribution algorithm and a wheel torque controller.

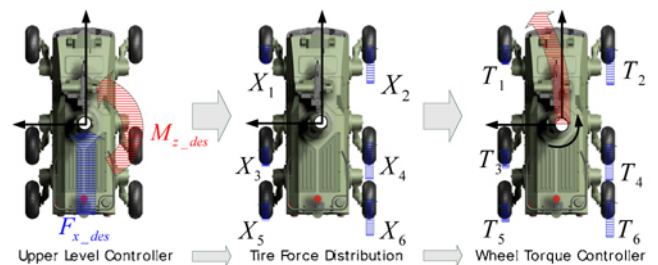


Figure.6. Control Strategy of the driving Controller based on skid steering

A schematic diagram of the proposed driving control algorithm is shown in Figure.7. The speed controller in the upper-level controller is designed to follow the desired speed using PI control method [5, 17]. The lateral motion controller determines the yaw moment input in order to track the desired yaw motion which is determined according to the control mode, the remote control mode and the autonomous driving control mode. The upper level control inputs, the total traction force and the yaw moment input, should be applied to the RVAS through longitudinal tire force distribution algorithm and wheel torque controller. Therefore, the objective of tire force distribution algorithm is to determine how much force is required at each wheel. From the concept of friction circle, the desired longitudinal tire force of each wheel is determined in proportion to a vertical tire force of each wheel. Finally, a wheel torque controller is designed to track the desired longitudinal tire force and at the same time to keep a wheel slip ratio of each wheel below a limit value. Additionally, longitudinal and vertical tire force estimators are required for optimal tire force distribution and wheel slip control [17, 19, 20].

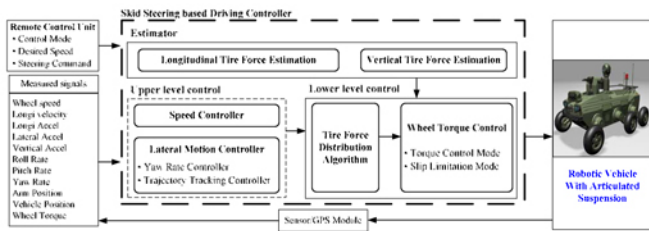


Figure.7. Block diagram of autonomous driving control algorithm based on a skid steering

3.1. UPPER LEVEL CONTROLLER

The upper level controller consists of the speed controller and the lateral motion. In the upper level controller, the speed controller computes a total traction force for following a desired speed. In this paper, the speed controller is designed based on PI control method [5, 17]. The lateral motion controller determines yaw moment input according to the control mode. The control mode is selected by remote driver's input and there exists two control modes. One is the remote control maneuvering mode and the other is the autonomous driving control mode. In the case of the remote control mode, yaw rate controller is activated and it computes the yaw moment input for following the desired yaw rate. In the remote control maneuvering mode, a desired yaw rate is generated based on remote driver's steering command. In the autonomous driving control mode, trajectory tracking controller is activated and it determines the yaw moment input to track the prescribed trajectory. The upper level control inputs are applied to the test platform through lower level controller.

3.1.1. REMOTE CONTROL MANEUVERING

In the remote control mode, the yaw rate controller is activated in order to force the vehicle to track a desired yaw rate generated by a remote driver's steering command, δ_{com} . The yaw rate controller determines the yaw moment input to reduce the yaw rate error between the desired and the actual yaw rate. The desired yaw rate, γ_d , is determined using the remote driver's steering command and the 1st order transfer function as follows [1]:

$$\gamma_d = \frac{k_{SR}}{1 + \tau_{SR} \cdot s} \delta_{com} \quad (10)$$

A 2-DOF bicycle model modified in view of skid steering vehicle is used to design the yaw rate controller as shown in Figure.8. Eq. (11) represents the dynamic equation of the modified bicycle model.

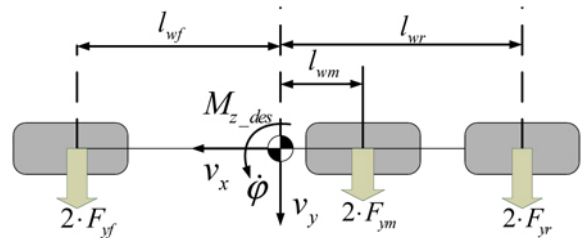


Figure.8. Modified 2-DOF bicycle model

$$\begin{aligned} m \cdot (\dot{v}_y + v_x \cdot \dot{\phi}) &= 2 \cdot F_{yf} + 2 \cdot F_{ym} + 2 \cdot F_{yr} \\ I_z \cdot \ddot{\phi} &= l_{wf} \cdot F_{yf} - l_{wm} \cdot F_{ym} - l_{wr} \cdot F_{yr} + M_{z_des} \end{aligned} \quad (11)$$

Sliding mode control method is used for the design of the yaw rate controller, and sliding surface is defined using the difference between the desired yaw rate and the actual yaw rate as follows:

$$s = \dot{\phi} - \gamma_d \quad (12)$$

The control objective is to keep the sliding surface at zero. This can be achieved by choosing the control law such that

$$\frac{1}{2} \frac{d}{dt} s^2 = s \dot{s} < -\eta \cdot |s| \leq -K \cdot |s| \quad (13)$$

Using Eqns. (11),(12),(13), the yaw moment input for tracking the desired yaw rate can be obtained as follows:

$$M_{z_des} = -\{2 \cdot l_{wf} \cdot F_{yf} - 2 \cdot l_{wm} \cdot F_{ym} - 2 \cdot l_{wr} \cdot F_{yr}\} - I_z \cdot K \cdot \text{sgn}(s) \quad (14)$$

In Eq. (14), the lateral tire forces can be expressed using the linear tire model as follows:

$$F_{yf} = C_f \cdot \left(-\frac{v_y + l_{wf} \cdot \dot{\phi}}{v_x}\right), \quad F_{ym} = C_m \cdot \left(-\frac{v_y - l_{wm} \cdot \dot{\phi}}{v_x}\right), \quad F_{yr} = C_r \cdot \left(-\frac{v_y - l_{wr} \cdot \dot{\phi}}{v_x}\right) \quad (15)$$

Substituting Eq. (15) into Eq. (14), the yaw moment input can be obtained as follows:

$$M_{z_des} = -A_3 \cdot I_z \cdot \beta - \frac{A_4}{v_x} \cdot I_z \cdot \gamma - I_z \cdot K \cdot \text{sgn}(s) \quad (16)$$

where,

$$A_3 = -\frac{2 \cdot (l_{wf} \cdot C_f - l_{wm} \cdot C_m - l_{wr} \cdot C_r)}{I_z}, \quad A_4 = -\frac{2 \cdot (l_{wf}^2 \cdot C_f + l_{wm}^2 \cdot C_m + l_{wr}^2 \cdot C_r)}{I_z}$$

Finally, the saturation function with boundary, Φ , is used to cope with chattering phenomenon as follows:

$$M_{z_des} = \begin{cases} -A_3 \cdot I_z \cdot \beta - \frac{A_4}{v_x} \cdot I_z \cdot \gamma - I_z \cdot K \cdot \text{sgn}\left(\frac{s}{\Phi}\right) & \text{if } (|s| \geq \Phi) \\ -A_3 \cdot I_z \cdot \beta - \frac{A_4}{v_x} \cdot I_z \cdot \gamma - I_z \cdot K \cdot \frac{s}{\Phi} & \text{if } (|s| < \Phi) \end{cases} \quad (17)$$

3.1.2. AUTONOMOUS DRIVING CONTROL

The trajectory tracking controller is activated in the autonomous driving control mode. The controller computes the yaw moment input in order to track the desired trajectory. In this study, the trajectory tracking controller is designed based on the optimal finite preview control method as shown in Figure.9. [5, 6]

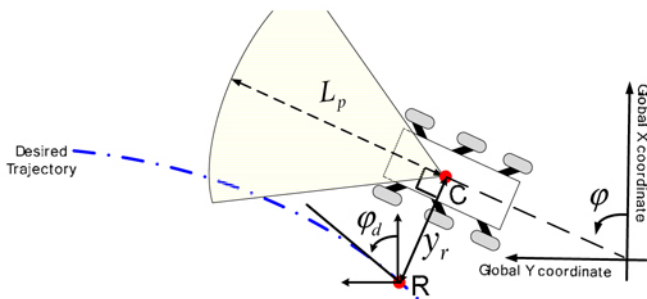


Figure.9. Trajectory Tracking Controller based on optimal finite preview control

In Figure.9, lateral position error, y_r , is defined as the lateral distance between the vehicle C.G (C) and the centerline of the desired trajectory (R). Yaw angle error, $\phi - \phi_d$, is defined using the yaw angle of vehicle and the desired yaw angle as dictated by the desired trajectory. The rate of change of lateral position error and yaw angle error are defined as follows:

$$\Delta y_r \approx v_y \cdot \Delta t + v_x \cdot \Delta t \cdot (\phi - \phi_d) \Rightarrow \dot{y}_r = v_y + v_x \cdot (\phi - \phi_d) \quad (18)$$

$$\Delta \phi_d \approx \frac{v_x \cdot \Delta t}{\rho} \Rightarrow \dot{\phi}_d = \frac{v_x}{\rho} \quad (19)$$

where, ρ denotes the curvature radius of the desired trajectory.

The trajectory tracking controller is designed to eliminate the lateral position error and the yaw angle error through a combination of feedback and feed-forward control. The feedback control input of the trajectory tracking controller is computed using the lateral position error and the yaw angle error. The feed-forward control input is determined using the road information within the preview distance. To develop the trajectory tracking controller based on the optimal finite preview control theory, preview distance, L_p , is transformed into preview time, T_p , as follows:

$$T_p = \frac{L_p}{v_x} \quad (20)$$

A modified 2-DOF bicycle model described in the previous chapter is used similarly to design the trajectory tracking controller. Using the linear tire model of Eq. (15) and Eqns. (18) ~ (19), the lateral tire forces can be represented as follows:

$$\begin{aligned} F_{yf} &= C_f \cdot \left\{ -\frac{\dot{y}_r + l_{wf} \cdot (\dot{\phi} - \dot{\phi}_d)}{v_x} + (\phi - \phi_d) - \frac{l_{wf}}{v_x} \cdot \dot{\phi}_d \right\} \\ F_{ym} &= C_m \cdot \left\{ -\frac{\dot{y}_r - l_{wm} \cdot (\dot{\phi} - \dot{\phi}_d)}{v_x} + (\phi - \phi_d) + \frac{l_{wm}}{v_x} \cdot \dot{\phi}_d \right\} \\ F_{yr} &= C_r \cdot \left\{ -\frac{\dot{y}_r - l_{wr} \cdot (\dot{\phi} - \dot{\phi}_d)}{v_x} + (\phi - \phi_d) + \frac{l_{wr}}{v_x} \cdot \dot{\phi}_d \right\} \end{aligned} \quad (21)$$

Substituting Eq. (21) into Eq. (11), the state equations for design of trajectory tracking controller can be obtained as follows: