

Development of Driving Control System Based on Optimal Distribution for a 6WD/6WS Vehicle

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ABSTRACT

This paper describes a driving controller to improve vehicle lateral stability and maneuverability for a six wheel driving / six wheel steering (6WD/6WS) vehicle. The driving controller consists of upper and lower level controller. The upper level controller based on sliding control theory determines front, middle steering angle, additional net yaw moment and longitudinal net force according to reference velocity and steering of a manual driving, remote control and autonomous controller. The lower level controller takes desired longitudinal net force, yaw moment and tire force information as an input and determines additional front steering angle and distributed longitudinal tire force on each wheel. This controller is based on optimal distribution control and has considered the friction circle related to vertical tire force and friction coefficient acting on the road and tire. Distributed longitudinal/lateral tire forces are determined in proportional to friction circle according to the changes of a driving condition. The response of the 6WD/6WS vehicle with the driving controller has been evaluated via computer simulations conducted using the Matlab/Simulink dynamic model. Computer simulations of a closed-loop driver model subjected to double lane change have been conducted to prove the improved performance of the proposed driving controller over a conventional direct yaw moment control (DYC). Adjustment weighting factors applied for the optimal distribution controller enhance performance by comparing controller applied fixed weighting factors.

INTRODUCTION

The Army's Future Combat Systems (FCS) has been developed to improve military's performance and power continuously. Among the FCS, researches about unmanned ground vehicle and manned vehicle system are very important and essential to achieve successful military

operations. Recently, military ground vehicle system has been developed using commercial vehicle technique such as hybrid, stability and autonomous control system. By combining conventional power with an electric drive, advanced hybrid electric systems and rechargeable energy storage, hybrid electric vehicles (HEVs) add a flexible new dimension to military operations. With smaller engines than traditional vehicles, hybrid combat vehicles are lighter to transport by air, can operate silently, have better fuel economy and can provide 'exportable' power for other equipment. They also have a smaller heat signature, making them harder to locate and attack. Most HEVs use a hybrid propulsion system consisting of a diesel engine to generate the electric power for mobility and all the vehicle subsystems. The electric system gives a number of advantages, including volume efficiency, fuel efficiency, reduced life cycle costs, reduced environmental impact and increased stealth characteristics. Batteries are integrated into the electric drive system, which allow the vehicle to be driven silently for several hours with the engines shut down. An independent 6WD/6WS (6 wheel drive / 6 wheel steering) mechanism is adopted in a special purposed, military armored vehicle to enhance its steering performance and its drive capability as an off-road. The six-wheeled vehicle is believed to have better performance than a four-wheeled vehicle in terms of its capability of crossing obstacles, off-road maneuvering and fail-safe handling when one or two of their tires are punctured. In order to achieve the best maneuverability during cornering for a six-wheeled vehicle, the middle and rear wheel steering angles need to be controlled according to the steering angle of the front wheels and velocity of the six-wheeled vehicle. Many methods have been studied and actively developed to improve a four-wheeled vehicle's lateral stability actively. However, there have only been a few studies on the lateral stability of a six-wheeled vehicle. [Huh et al.\(2000\)](#) set the middle wheel steering angle to half of the front wheel steering angle and controlled the rear wheel

steering to minimize the side slip angle of their six-wheeled vehicle. Jackson and Crolla (2002) proposed the yaw rate control method using the direct yaw moment control (DYC) to improve the stability of their six-wheeled vehicle during cornering. Chen et al. (2006) controlled the middle and rear wheel steering angle using the LQR technique with integral control. An et al. (2006) controlled the front, middle and rear wheel steering angle and velocity. However, they used the desired yaw rate of a four-wheeled vehicle's maneuvering during the cornering. The middle wheel's effect to influence the control target was not considered in the study. In this paper, a driving controller for the 6WD/6WS vehicle is proposed. This control system enhances performance of the vehicle stability and maneuverability using normal force and road friction coefficient information.

VEHICLE DYNAMIC MODEL

BODY DYNAMICS

In this paper, a six-wheeled vehicle is modelled as an object with 18 degree-of-freedom (DOF). The full vehicle model is composed of 6 DOF translational and rotational dynamic models of the sprung mass, 6 suspension models and 6 wheel dynamics models as shown in Figure 1.

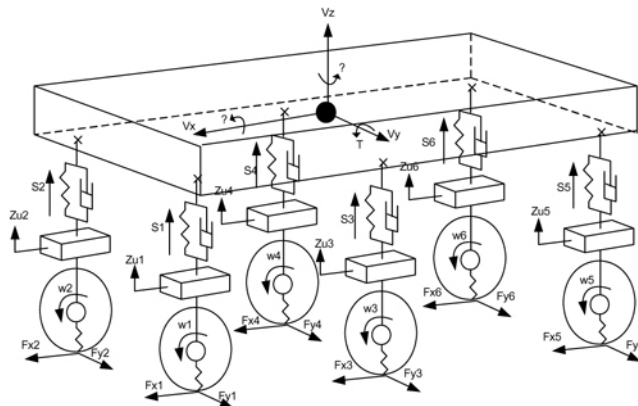


Figure 1. 3-dimensional Dynamics Model for the 6WD/6WS Vehicle

Table 1 shows specifications of a 6WD/6WS vehicle such as mass, tread, front wheelbase, rear wheelbase, wheel radius and z-axis moment of inertia.

Table 1. Specifications for the 6WD/6WS Vehicle

Vehicle Parameters	specifications	Vehicle Parameters	specifications
Mass	7200 [kg]	Rear wheelbase	1.50 [m]
Tread	2.0 [m]	Wheel radius	0.45 [m]
Front wheelbase	1.74 [m]	Z-axis moment of inertia	32434.2 [kgm ²]

MOTOR DYNAMICS

6WD/6WS vehicle equipped with 6 in-wheel-motors is able to operate differential tracking and braking drive modes. Figure 2 represents the structure of a wheel-in-motors and torque-angular velocity diagram. The maximum torque depends on capacity of the wheel-in-motor and is determined according to operating wheel angular velocity.

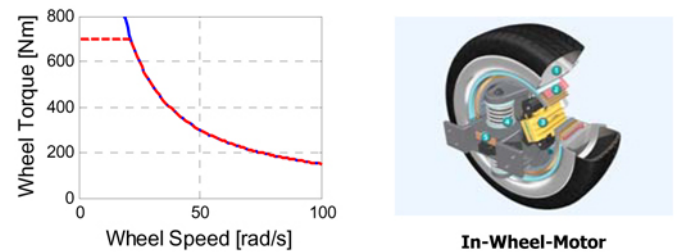


Figure 2. Motor Specification Curve and In-wheel-motors

In-wheel-motor is modeled using first order transfer function. This model operates in the following manner; control input is the desired output torque distributed by lower level controller and output of the motor model through this control input is the generating torque.

BRAKE FORCE LIMITATION

Longitudinal tire forces that can be generated in the individual wheels are constrained with physical limit as shown in equation (1). Therefore, when distributing brake force to generate required yaw moment, it should be considered that each braking force at the individual wheels cannot exceed longitudinal limitation force.

$$F_{xi} \leq \sqrt{(\mu F_{zi})^2 - F_{yi}^2} = F_{limit,i}, \quad i = fl, fr, ml, mr, rl, rr \quad (1)$$

DRIVING CONTROL ALGORITHM DEVELOPMENT

A driving controller consists of lateral stability based on optimal distribution control method and longitudinal velocity control. 6WD/6WS vehicle often drives on off-road driving condition where mass distribution is changing severely and contact between road and tire is unstable. Therefore, driving control algorithm must be considered according to the force acting on the road and tire. Figure 3 shows driving controller block diagram. The driving control consists of upper and lower level controller.

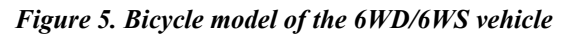


Upper level controller takes steering angle determined through manual steering, remote controller and autonomous driving controller and reference velocity as input and determines desired longitudinal tire net force and yaw moment. Figure 4 represents the upper level control block diagram



Reference Yaw Rate Design

In many studies for 4WD, desired lateral yaw rate model is used to define a linearization bicycle model. As well, 6WD/6WS vehicle model is defined as bicycle model. [Figure 5](#) represents bicycle model of the 6WD/6WS vehicle.



In case of low speed driving condition where tire slip hardly occurs, the center of vehicle steering motion is determined as point 'A' dominantly. On the other hand, due to tire slip generated on high speed driving condition, the center of vehicle steering motion shifts point 'A' to point 'B'. It is necessary that the desired model contain the slip effect which is generated on high speed condition. Dynamics of the 6WD/6WS bicycle model is shown in [equation \(2\)](#). Because middle steering angle depends on front steering angle, input of a state equation of the bicycle model dynamics can be written as only front steering angle.

$$\begin{aligned} \begin{bmatrix} \beta \\ \dot{\gamma} \end{bmatrix} &= \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} \beta \\ \gamma \end{bmatrix} + \begin{bmatrix} b_{11} \\ b_{21} \end{bmatrix} \delta_f + \begin{bmatrix} b_{12} \\ b_{22} \end{bmatrix} \delta_m \\ &= \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \begin{bmatrix} \beta \\ \gamma \end{bmatrix} + \begin{bmatrix} b_{11} + b_{12} \left(\frac{l_m + l_f}{L} \right) \\ b_{21} + b_{22} \left(\frac{l_m + l_f}{L} \right) \end{bmatrix} \delta_f \\ a_{11} &= \frac{-2(C_f + C_m + C_r)}{mv_x}, \quad a_{12} = \frac{-2(l_f C_f + l_m C_m - l_r C_r)}{mv_x^2} - 1 \\ a_{21} &= \frac{-2(l_f C_f + l_m C_m - l_r C_r)}{I_z}, \quad a_{22} = \frac{-2(l_f^2 C_f + l_m^2 C_m - l_r^2 C_r)}{I_z v_x} \\ b_{11} &= \frac{2C_f}{mv_x}, \quad b_{12} = \frac{2C_m}{mv_x}, \quad b_{21} = \frac{2l_f C_f}{I_z}, \quad b_{22} = \frac{2l_m C_m}{I_z} \end{aligned} \quad (2)$$

Steady state yaw rate gain can be obtained by excluding time-transient terms from bicycle dynamic model as [equation \(2\)](#) and substituting side slip angle for yaw rate. It is written as [equation \(3\)](#).

$$\frac{\gamma_{ss}}{\delta_f} = k = \frac{a_{22} \left(b_{21} + b_{22} \frac{l_m + l_r}{L} \right) - a_{21} \cdot \left(b_{11} + b_{12} \frac{l_m + l_r}{L} \right)}{(a_{12} a_{21} - a_{11} a_{22})} \quad (3)$$

A desired yaw rate is determined by steady state yaw rate gain and first-order transfer function expressed as time constant.

$$\gamma_{des} = \frac{k}{1 + \tau_{sync}s} \delta_f$$

Assuming constant lateral velocity, [equation \(5\)](#) provides a theoretical limit to the yaw rate that is achievable at the current friction condition between tire and road. Therefore, the desired reference yaw rate is reasonably constrained by the physical limit of tire-road friction, as shown in [\(6\)](#).

$$|ma_y| = |mV_x\gamma| \leq m\mu g \quad (5)$$

$$|\gamma_{des}| \leq \frac{\mu g}{V_x} \quad (6)$$

Yaw Rate Controller Design Using Sliding Control Method

To track the proposed single target yaw rate, a yaw rate controller is designed based on 2 D.O.F bicycle model. Designed yaw rate controller calculates required yaw moment to stabilize that dynamic behavior of a vehicle. In order to generate required yaw moment, it needs to be decided how braking forces should be distributed to the individual wheels. In linear models, lateral forces are regarded as linear functions of the slip angle assuming that the slip angle is very small. However, the assumption that the lateral forces are linear to the tire slip angle is valid only when the tire slip angle is smaller than 3 degrees. Therefore, lateral forces treated as nonlinear functions, and the bicycle model turns to be a nonlinear system. In order to track the reference yaw rate, a yaw rate controller is designed based on sliding mode control theory. The bicycle model is considered as the first order single input dynamic system as shown in [equation \(7\)](#).

$$\dot{\gamma} = 2\frac{l_f}{I_z}F_{yf}\cos\delta_f + 2\frac{l_m}{I_z}F_{ym}\cos\delta_m - 2\frac{l_r}{I_z}F_{yr} + \frac{M_z}{I_z} \quad (7)$$

Sliding surface is defined as yaw rate error, which is the difference between actual yaw rate and reference yaw rate, as follows:

$$s = \gamma - \gamma_{des} \quad (8)$$

The control objective is to keep the scalar s at zero and it can be achieved by choosing the control law satisfying the sliding condition states a

$$\frac{1}{2}\frac{d}{dt}s^2 \leq -\eta|s| \quad (9)$$

where η is a strictly positive constant.

[Equation \(9\)](#) states that the squared distance to the surface, as represented by s^2 decreases along all system state

trajectories. Thus, it constrains trajectories to point toward zero value of surface and once on the surface, the system trajectories remain on the surface. Equivalent control $M_{z,eq}$, which is the continuous control law that would maintain zero value of differential surface if the dynamics were exactly known, is given as follows:

$$M_{z,eq} = I_z \cdot \dot{\gamma}_d - 2l_f F_{yf} \cos\delta_f - 2l_m F_{ym} \cos\delta_m + 2l_r F_{yr} \quad (10)$$

Therefore, from [\(7\)](#) and [\(10\)](#), the sliding control law is obtained as a required yaw moment. Chattering must be eliminated for the controller to perform properly, and therefore, smoothing out the control discontinuity has been carried out by replacing the sign function with a saturation function. Accordingly, the control law is given as shown in [equation \(11\)](#).

$$\Delta M_z = -2l_f F_{yf} \cos\delta_f - 2l_m F_{ym} \cos\delta_m + 2l_r F_{yr} + I_z \cdot \dot{\gamma}_d + I_z k \text{sat}\left(\frac{s}{\Phi}\right) \quad (11)$$

Vehicle Speed Controller Design

A speed controller is designed to calculate desired longitudinal net force in order to satisfy a desired velocity using PID control theories as shown in [equation \(12\)](#). Assuming that the desired longitudinal velocity is determined by manual driver, remote controller and autonomous controller, a desired acceleration is computed from multiplying longitudinal velocity error by proportional gain P . Then, speed controller decides desired longitudinal net force using Newton's law of motions.

$$F_{x_des} = m \cdot K(V_{des} - V_x) \quad (12)$$

LOWER LEVEL CONTROLLER DESIGN

Proposed controller based on optimal control theory is suitable for independent driving and steering vehicle equipped with 6 in-wheel-motors and 6 steering motors. Lower level controller is designed to distribute front lateral tire force and longitudinal tire forces at each wheel in order to satisfy desired longitudinal net force and yaw moment calculated by upper level controller. Distributed tire forces are determined in proportion to friction circle according to changing driving condition on the assumption that friction circle, longitudinal and lateral tire force are known precisely.