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Reconfigurable Fault-Tolerant Control of In-Wheel Electric Vehicles with Steering System Failure ⋆

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Abstract:
The paper deals with the robust and fault tolerant control of an in-wheel electric vehicle, operated by four independently actuated in-wheel motors and steer-by-wire steering system. The goal of the design is to realize velocity and road trajectory tracking of the in-wheel vehicle, even during a fault event in the steer-by-wire steering system. When such a fault is detected, the control system is reconfigured and the torque allocation of the in-wheel engines is recalculated to substitute for the yaw moment originally generated by the steering system. Thus, the vehicle is able to follow the designed path and velocity solely with torque vectoring generated by the independently-controlled in-wheel motors. The high-level motion control of the vehicle is realized founded on the LPV framework, while the reconfiguration is based on constrained optimization techniques. The operation of the designed control system is demonstrated in a CarSim simulation environment.

Keywords: in-wheel vehicle, 4WIA vehicle, integrated control, reconfigurable control, trajectory tracking.

1. INTRODUCTION

As economical and environmentally friendly transportation means, in-wheel electric vehicles are in the focus of automotive companies and researchers as well. These vehicles with hub motors integrated in two or four wheels have several advantages and very few drawbacks compared to conventional vehicles with internal combustion engines. The compact size of in-wheel motors and the absence of regular drive train components (gearbox, differential, etc...) enable space-efficient passenger cabin design, also resulting in reduced overall weight. On the other hand, the increased unsprung mass of the vehicle may lead to adverse vertical vibrations, which may affect the ride comfort and stability of the in-wheel vehicle. This issue can been addressed by designing an active suspension, as proposed by Wang et al. (2014b). From a vehicle control point of view the most appealing properties of a four-wheel independently-actuated (4WIA) vehicle are the fast and accurate torque generation, enabling the design of very effective stabilizing and anti-slip control systems. Moreover, the efficiency of regenerative braking can be increased, see Wang et al. (2014a). With the precise knowledge of engine torque, the estimation of road conditions becomes much simpler.

Recent studies of in-wheel vehicles focus on exploiting the specific properties of the hub motors. High performance wheel slip control has been introduced by several authors, see Castro et al. (2012); Ringdorfer and Horn (2011). Rollover avoidance methods have been proposed by Kawashima et al. (2009). Several articles focus on the lateral control of in-wheel vehicles, see Wu et al. (2013); Shuai et al. (2013); Xiong et al. (2012).

The complexity of in-wheel vehicles and the increased potential dangers of a fault event makes fault tolerant control design a necessity, see Johansen and Fossen (2013). In-wheel motors were studied by Ifedi et al. (2013), focusing on achieving a high-torque density and the capability of sustaining an adequate level of performance following a failure. Wang and Wang (2012) proposed a fault-tolerant control system designed to accommodate hub motor faults by automatically reallocating the control effort among other healthy wheels. Hu et al. (2011) designed a fault-tolerant control based on the estimation of the maximum transmissible torque for the wheels.

This paper deals with the fault tolerant control of a 4WIA electric vehicle with a steer-by-wire steering system. A reconfigurable and robust control method is designed in order to sustain the desired yaw rate and velocity tracking performance of the in-wheel vehicle in case the steer-by-wire steering system is affected by total failure or performance degradation. The reconfiguration of the faulty steering system is based on additional yaw torque

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generation by the in-wheel motors, considering actual vehicle dynamics during the optimization process of the control reallocation. The main novelty of the paper lies in the high-level LPV control design and the fault tolerant reconfiguration method for steering system failure.

The paper is organized as follows: Section 2 presents the control-oriented vehicle model. Section 3 deals with the robust trajectory tracking controller based on the LPV framework. Section 4 presents the control implementation issues. Section 5 describes the reconfiguration method during a steering system failure. Section 6 shows the operation of the proposed control method in a high-fidelity simulation environment.

2. VEHICLE MODEL FOR TRAJECTORY TRACKING

The aim of the control design is to guarantee both velocity and road trajectory tracking for the in-wheel electric vehicle. Thus, the longitudinal and lateral dynamics of the vehicle are both considered, while the vertical motion is neglected. The motion equation of the vehicle is based on the two-wheeled bicycle model, see Figure 1. The differential equations of the planar motion are the following:

$$J\ddot{\psi} = c_1 l_1 \alpha_1 - c_2 l_2 \alpha_2 + M_z \tag{1a}$$

$$m\dot{\xi}(\dot{\psi} + \dot{\beta}) = c_1\alpha_1 + c_2\alpha_2 \tag{1b}$$

$$m\ddot{\xi} = F_l - F_d \tag{1c}$$

where m is the total mass, J is the yaw inertia of the vehicle, l_1 and l_2 are geometric parameters, c_1 and c_2 are cornering stiffness of the tires on the front and rear axle. The following notations are used for the side slip angles of the wheels: $\alpha_1 = \delta - \beta - \dot{\psi} l_1/\dot{\xi}$ and $\alpha_2 = -\beta + \dot{\psi} l_2/\dot{\xi}$. The vehicle yaw rate is denoted with $\dot{\psi}$, β is the vehicle side-slip angle, while ξ is the longitudinal displacement. The control

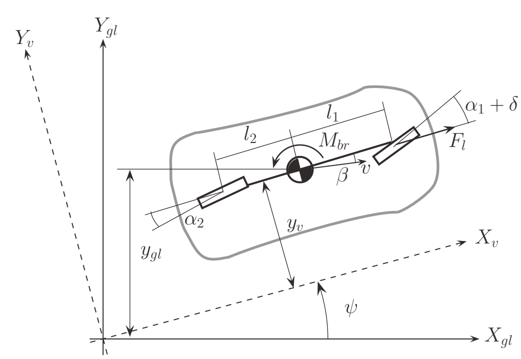

Fig. 1. Single track bicycle model

inputs of the vehicle system are the following: F_l is the longitudinal control force, δ is the front steer angle, while M_z is the generated yaw moment. The disturbance force F_d consists of the road slope, the drag disturbance and the rolling resistance. The system nonlinearity is caused by the longitudinal velocity $\dot{\xi}$. By applying a scheduling variable

$$\rho = \dot{\xi} \tag{2}$$

the nonlinear model is converted into an LPV model.

Thus, the state-space form is given as follows:

$$\dot{x} = A(\rho)x + B_1 w + B_2 u \tag{3}$$

where the state vector of the system is $x = \left[\dot{\xi}\ \xi\ \dot{\psi}\ \beta\right]^T$. The control inputs are given in the input vector $u = [F_l\ \delta\ M_z]^T$. The measured outputs are the vehicle velocity and yaw rate, i.e. $y = \left[\dot{\xi}\ \dot{\psi}\right]^T$. The disturbance of the system is expressed by $w = [F_d]^T$.

3. LPV CONTROL DESIGN

Since trajectory and velocity tracking of the in-wheel vehicle are both required, it is necessary to define the corresponding reference signals. Considering the longitudinal vehicle motion a predefined velocity (speed limit) must be tracked. The lateral motion of the 4WIA vehicle is given by the road curvature, which can be calculated. Knowing the curvature of the road and the vehicle velocity, the reference yaw rate can be calculated as follows:

$$\dot{\psi}_{ref} = \frac{\dot{\xi}}{r} \tag{4}$$

where r is the curve radius. The desired velocity $\dot{\xi}_{ref}$ representing the speed limit is also defined for the vehicle, thus the reference signals are given in a reference vector $R = \left[\dot{\xi}_{ref}\ \dot{\psi}_{ref}\right]^T$.

The reference signals given in reference vector R must be tracked, thus the velocity error $z_{\dot{\xi}} = |\dot{\xi}_{ref} - \dot{\xi}|$ and yaw rate error $z_{\dot{\psi}} = |\dot{\psi}_{ref} - \dot{\psi}|$ has to be minimized. For this purpose, the optimization criterion $z_{\dot{\xi}} \to 0$ and $z_{\dot{\psi}} \to 0$ must be fulfilled. Hence, a performance vector $z_1 = \left[z_{\dot{\xi}}\ z_{\dot{\psi}}\right]^T$ is introduced. Actuator saturation also has to be avoided by considering the maximal outputs of the in-wheel motors and the physical construction limits of the steer-by-wire steering system. Thus, a performance vector is also formulated for the input signals as $z_2 = [F_l\ \delta\ M_z]^T$.

The control framework is based on the closed-loop interconnection structure shown in Figure 2, where the control objectives are represented with appropriately selected weighting functions. The weighting functions denoted with

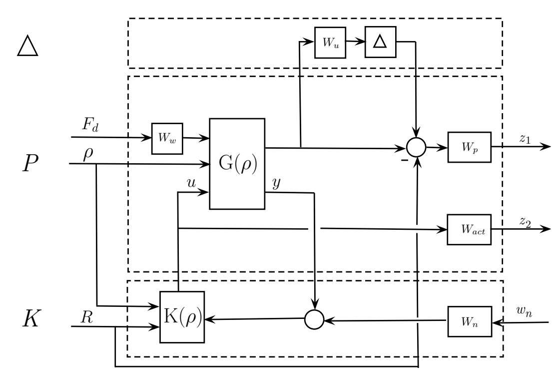

Fig. 2. Closed-loop interconnection structure

W_p guarantee an optimal trade-off between the above listed control performances. They can be considered as penalty functions, and are applied in a second-order proportional form. The goal of the weighting functions W_w and W_n is to represent the disturbance and sensor noises, while the neglected dynamics of the vehicle are handled by the weighting function W_u.