

Allocation-Based Fault Tolerant Control for Electric Vehicles with X-by-Wire

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ABSTRACT

This paper proposed a novel fault-tolerant control method based on control allocation via dynamic constrained optimization for electric vehicles with XBW systems. The total vehicle control command is first derived based on interpretation on driver's intention as a set of desired vehicle body forces, which is further dynamically distributed to the control command of each actuator among vehicle four corners. A dynamic constrained optimization method is proposed with the cost function set to be a linear combination of multiple control objectives, such that the control allocation problem is transformed into a linear programming formulation. An analytical yet explicit solution is then derived, which not only provides a systematic approach in handling the actuation faults, but also is efficient and real-time feasible for in-vehicle implementation. The simulation results show that the proposed method is valid and effective in maintaining vehicle operation as expected even with faults.

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INTRODUCTION

As electric vehicle becomes a new trend for automotive industry to tackle various challenges, it also enables the x-by-wire (XBW) technology such that all wheels can be electrically operated individually and independently. This offers great advantages for more controllable degrees of freedom, faster and more precise control responses to improve vehicle performance in handling and stability compared to conventional vehicles [1]. However, the potential faults of XBW systems have become a growing concern as the system complexity increases with more electric actuators used. The fault of an XBW system is one of the leading causes to the degradation of vehicle performance, and more critically, to the failure of vehicle operation. Fault tolerant control (FTC) thus becomes essential for electric vehicles with XBW systems to ensure the required reliability and safety.

Actuator redundancy has been widely adopted as a means to improve overall system reliability, which often results in system over-actuation when the number of actuators exceeds the number of controlled variables. Control allocation has been regarded as an effective approach in handling system over-actuation [2]. Through control re-allocation when one or more actuators fail, the system may remain partially or fully safe and operational. In fact, control allocation method has been widely applied for fault-tolerant control in aircraft control [3] and surface vessel control [4]. Rule-based control allocation methods were widely applied, in which the actuator failures were often modeled as control gain change or control output

saturation [2]. Redistributed pseudo-inverse (RPI) method, on the other hand, is a method often used in dealing with constrained control allocation problems [5]. As an iterative process, RPI works in a way similar to the pseudo inverse with the addition of position and rate limit. Quadratic program (QP) is another method used to solve the problems [6] with numerical iteration via, for example, active set method or interior point method. Some most recent research in this area can also be found from the literature [7-8].

However, the existing methods are often either rule-based that needs to be designed case by case, or iteration-based that does not always guarantee that a feasible or optimal solution exists or can be found with a reasonable number of iterations and/or within limited computation time. This is especially critical and necessary for real-time implementation. This paper proposed a novel allocation-based fault tolerant control method based on dynamic constrained optimization for electric vehicles with x-by-wire actuators. The main contribution of the paper is in presenting a novel fault-tolerant control method based on control allocation via dynamic constrained optimization. The cost function for the optimal target is set as a linear combination of the multiple control objectives and the specific problems due to tire-road contact and its nonlinear characteristics, actuators' faulty grades and other saturation factors are treated as constraints. The analytical solution is then explicitly expressed aimed to provide optimal solution to the problems with real-time computational efficiency.

Simulation is conducted using a high-fidelity vehicle model with the results showing the validity and effectiveness of the proposed method.

The remainder of the paper is organized as follows: the control system is described and the control problem is formulated first. Secondly, the total control command is obtained based on interpretation of driver's intention. Then the control distribution is discussed based on the dynamic constrained optimization method. An analytical yet explicit solution is then derived that forms a fault-tolerant control for electric vehicle with XBW systems. Finally, simulation is performed with results presented and discussed, and followed by conclusive remarks.

SYSTEM DESCRIPTION AND PROBLEM FORMULATION

Fig. 1 presents the structure of the proposed control system. It consists of a control command, which converts the driver's input to the desired body forces as the total control command, and a control allocation part that distribute the total control command to the control command of each actuator. In addition, a feedback control part is added to stabilize the system and to compensate model uncertainties and external disturbances.

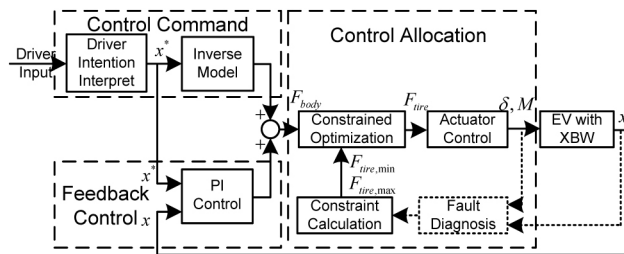


Figure 1. Schematic of the overall control structure

In the control structure above, the driver's intention is interpreted by the driver's input on acceleration and brake pedals, and steering wheel angle via driver interpreter. That leads to a set of vehicle state variables representing the desired vehicle motion in longitudinal speed, lateral speed and yaw rate. An inverse model of vehicle dynamics is used to the desired vehicle motion into the desired vehicle body forces on the vehicle center of gravity that drive the vehicle to follow the desired motion. A feedback control (e.g. PI control) is used to compensate model parameter uncertainties, un-modeled dynamics and external disturbances.

In the control allocation part, the desired body forces are distributed to each corner first via constrained optimization. It is assumed that the fault information can be obtained by the fault diagnosis module. The upper and lower bound of the tire forces which can be generated by the actuators after fault occur are calculated by the fault information. So the constraints of actuators are time-varying and are affected by the fault grade. Multiple objectives are considered in the defined cost function.

The tire force is finally converted to the direct control commands for motor control, which are the steering wheel angle on the road and drive/brake moment of the wheel.

CONTROL COMMAND

Driver Intention Interpreter

The driver intention is interpreted via a vehicle reference model. Both the intention of driver and the performance and driving environment of the vehicle are taken into account to avoid an unreasonable value of the desired vehicle motion when it is derived. Since it is for desired motion, a linear vehicle reference model with linear tire characteristics is employed.

Longitudinal

Desired acceleration is usually proportional to the pedal input in recent research [9], which is difficult to obtain a balance in both low and high speeds. In this paper, the pedal input and the accelerate/brake ability determine the desired acceleration together. The ability to accelerate/brake is represented by the maximum or minimum of the acceleration which the vehicle can reach at a given moment. The desired acceleration can be expressed by:

$$a_{xd} = \begin{cases} a_{acc,max} \cdot \bar{P}_{acc} (\bar{P}_{acc} \geq 0, \bar{P}_{brake} = 0) \\ a_{brake,max} \cdot \bar{F}_{brake} (\bar{P}_{acc} = 0, \bar{F}_{brake} \geq 0) \end{cases} \quad (1)$$

where $a_{acc,max} / a_{brake,max}$ is the maximum or minimum acceleration the vehicle can reach at current moment. It is largely determined by the vehicle weight, the tire-road adhesion, the actuators characteristic and the vehicle state. $\bar{P}_{acc}$, $\bar{F}_{brake}$ are normalized accelerate pedal position and normalized brake pedal force. In addition, the tire-road adhesion is taken into account as well. Then:

$$a'_{xd} = \min\{|a_{xd}|, \mu g\} \cdot \text{sgn}(a_{xd}) \quad (2)$$

where μ is the road friction coefficient.

The desired longitudinal speed can be derived by the time integral of desired acceleration:

$$u_d = u_0 + \int_0^t a'_{xd}(t) dt \quad (3)$$

Lateral

A bicycle model with steady-state characteristics is used to represent the desired vehicle lateral responses. The desired sideslip angle at vehicle center of gravity (C.G.) is zero, while the desired yaw rate is expressed by:

$$r_d = \frac{u/L}{1 + Ku^2} \delta_{sw} \quad (4)$$

where δ_{sw} is the steer wheel angle; u is the actual vehicle longitudinal velocity; L is the wheelbase; K is the understeer characteristic. Similar to the longitudinal acceleration, the correction of the desired yaw rate due to the road friction coefficient is given as:

$$r'_d = \min \left\{ |r_d|, \left| \frac{\zeta \mu g}{u} \right| \right\} \text{sgn}(\delta_{sw}) \quad (5)$$

where ζ is a factor to compensate the error caused by the absence of longitudinal acceleration and the differential of lateral velocity [9].

Inverse Vehicle Model

An inverse vehicle model with three degrees of freedom (DOF) is used to calculate the desired body forces. This method can improve the response of the whole system and contribute most part of the desired body forces when the model is matching the vehicle. The express of forward model is:

$$\begin{cases} \dot{u} = vr + \frac{1}{M} (\sum F_x - F_{air}) \\ \dot{v} = -ur + \frac{1}{M} \sum F_y \\ \dot{r} = \frac{1}{I_z} \sum M_z \end{cases} \quad (6)$$

$$F_{air} = C_{air} \cdot A_{air} \cdot \varphi_{air} \cdot u^2 / 2 \quad (7)$$

where u , v , r are the longitudinal velocity, lateral velocity and yaw rate respectively; $\sum F_x$, $\sum F_y$, $\sum M_z$ are the body force; M is the total mass; I_z is the yaw inertia; F_{air} is the air drag force which can be calculated by the drag coefficient factor C_{air} , the frontal cross-section area A_{air} , the air density φ_{air} and the longitudinal velocity. So the desired body forces calculated by the inverse vehicle model are obtained:

$$\begin{cases} F_{xivm} = M \cdot (\dot{u}_d - v_d r_d) + C_{air} \cdot A_{air} \cdot \varphi_{air} \cdot u_d^2 / 2 \\ F_{yivm} = M \cdot (\dot{v}_d + u_d r_d) \\ M_{zivm} = I_z \dot{r}_d \end{cases} \quad (8)$$

where u_d , v_d , r_d are the desired longitudinal velocity, desired lateral velocity and desired yaw rate which are obtained by the driver intention interpreter.

FEEDBACK CONTROL

The feedback control is used to compensate errors between the desired and the actual values caused by the model unknown or uncertainties, and external disturbance. A simple PI control is applied with the control law to be:

$$\begin{cases} F_{xpb} = P_1 \cdot (u_d - u) + I_1 \int_0^t (u_d - u) dt \\ F_{yfb} = P_2 \cdot (v_d - v) + I_2 \int_0^t (v_d - v) dt \\ M_{zfb} = P_3 \cdot (r_d - r) + I_3 \int_0^t (r_d - r) dt \end{cases} \quad (9)$$

where P_i , I_i ($i=1,2,3$) is the proportional and integral gains.

CONTROL ALLOCATION

The control allocation algorithm is designed to distribute the desired body forces into the individual tire force of each corner. The relationship between the desired body forces and the tire forces can be derived using Fig. 2. According to the geometric relationship, it can be expressed by:

$$\sum F = M_u f \quad (10)$$

where $\sum F \in R^3$ is the desired body forces; $M_u \in R^{3 \times 8}$ is the control effectiveness matrix; $f \in U \subset R^8$ is the tire forces and U represents the feasible region of tire force due to the physical constraints and fault. These can be expressed by:

$$\begin{aligned} \sum F &= \begin{bmatrix} \sum F_x & \sum F_y & \sum M_z \end{bmatrix}^T \\ &= \begin{bmatrix} F_{xivm} + F_{xpb} & F_{yivm} + F_{yfb} & M_{zivm} + M_{zfb} \end{bmatrix}^T \end{aligned} \quad (11)$$

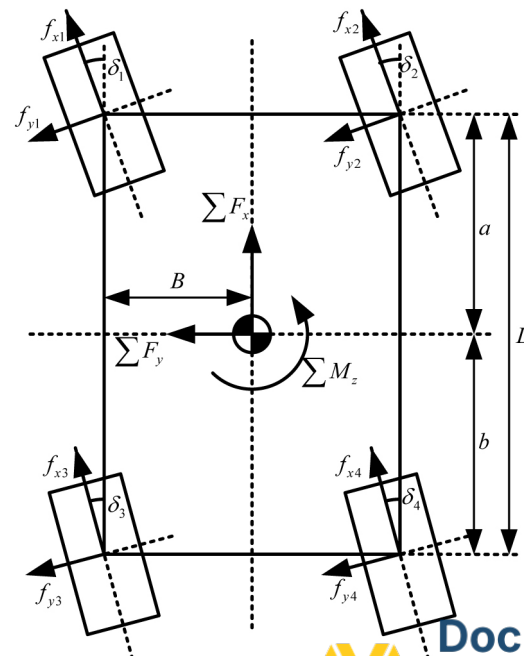


Figure 2. Desired body forces and tire forces