Available online at www.sciencedirect.com

ScienceDirect

IFAC-PapersOnLine 49-11 (2016) 412–417

A Torque Vectoring Optimal Control Strategy for Combined Vehicle Dynamics Performance Enhancement and Electric Motor Ageing Minimisation ⋆

Angelos Kampanakis, Efstathios Siampis, Efstathios Velenis and Stefano Longo

Cranfield University, Cranfield, Bedfordshire, MK43 0AL UK (e-mail: kampanakis.a@gmail.com, e.siampis@cranfield.ac.uk, e.velenis@cranfield.ac.uk, s.longo@cranfield.ac.uk)

Abstract: In this paper we propose a control architecture that combines velocity, sideslip angle and yaw rate regulation with motor temperature regulation on a electric vehicle with four independent electric motors. The linear controller incorporates both the vehicle dynamics and the electric motor dynamics by combining a four-wheel vehicle model with a motor degradation model. It is found that the resulting controller not only enhances the vehicle stability of the vehicle, but also extends the lifetime of motors by regulating their temperatures.

Keywords: automotive control; electric vehicles; quadratic regulators; motor control

1. INTRODUCTION

The rising ecological sensitivity along with the ever tighter government regulations has driven the automotive industry and academia to look for more fuel efficient transport solutions that produce less emissions. One of the current aims of this research is the development of Electric Vehicles (EV) as a viable, cost effective solution. The use of electric motors has distinctive advantages over a conventional driverline such as fast response and high energy efficiency, and in an electric vehicle topology such as the one considered here with four independent electric motors it can dramatically change the behaviour of the vehicle in a positive way by controlling both the direction and magnitude of the torques on the wheels, a method commonly known as torque vectoring. However, using electric motors in an automotive environment also poses specific problems which are mainly connected to safety and cost issues. In particular, motor degradation is of great importance since it can lead to complete motor failure: while this can occur due to mechanical, thermal, chemical and electrical reasons as pointed by Thorsen and Dalva (1995), studies reveal that more than 90% of electric drives fail due to overheating in the windings insulation or excessive mechanical stresses on the bearings, see Rahman et al. (2010) and Haifang et al. (2011). Especially the insulation temperature of a motor is considered of vital importance since, even for motors with a insulation class H, a temperature rise of 10°C above 160°C can lead to halving the residual lifetime of the motor. Predicting these issues at the development stage is considered crucial for developing highly optimized electric machines, but a very effective way to reduce the likehood of failure is to prevent overheating of the motor during operation. Based on this observation the main goal of this paper is to develop a control strategy that regulates both the vehicle dynamics according to the driver's commands and the motor temperatures to prevent overheating by using the torque vectoring capabilities of a Four Wheel Drive (FWD) electric vehicle.

The study of electric motor degradation for use on electric vehicles has so far progressed independently from the development of active vehicle dynamics systems. Motor degradation surveys deal mainly with the residual lifetime of an electric motor based on temperature and focus on conducting thermal analysis on the motor before it is assembled from its different components, as found in Kimotho and Hwang (2011), Driesen et al. (2001) and Fodorean and Miraoui (2008). Use of Computational Fluid Dynamics (CFD) suites in such studies provides good estimations not only for the steady state temperature, but also for short term overloads. On the other hand extensive research has been carried out on vehicle stability control on Hybrid Electric Vehicles (HEV) and EVs with a variety of control methodologies, ranging from Model Predictive Control (MPC) strategies and Linear Quadratic Regulators (LQR) as found in Siampis et al. (2013), to simpler ones such as Proportional-Integral-derivative (PID) controllers, see Pinto et al. (2010). All of these studies aim to achieve not only better drivability, but also safer operation of the vehicle near the limits of lateral acceleration, but neglect to take into account the effect of high operational motor temperatures on the life of the motor.

In this paper we propose an optimal control strategy that combines a vehicle dynamics model with an electric drive thermal-degradation model to both achieve better vehicle performance and extend the motor life by optimally dis-

⋆ The authors would like to acknowledge the financial support from EPSRC via the 'FUTURE Vehicle' project (grant number EP/I038586/1) and the Impact Acceleration Account (grant number EP/K503927/1).

10.1016/j.ifacol.2016.08.061

tributing the torques on the four wheels of a FWD electric vehicle. After introducing the vehicle, tyre and motor models to be used, the optimal control strategy is explained. Finally, simulation results show the effectiveness of the proposed solution under both normal driving conditions and a critical turn.

2. VEHICLE AND MOTOR MODELS

In this section we introduce the vehicle and tyre models, along with the motor model which will be used to provide the life expectancy of the four electric motors.

2.1 Vehicle and Tyre Model

The vehicle and tyre models used in this paper are similar to the one found in Siampis et al. (2013), where the interested reader can refer for more details. The equations of motion for a vehicle travelling on a horizontal plane are (Fig. 1):

$$\begin{aligned} m\dot{V} &= (f_{FLx} + f_{FRx})\cos(\delta - \beta) \\ &\quad - (f_{FLy} + f_{FRy})\sin(\delta - \beta) \\ &\quad + (f_{RLx} + f_{RRx})\cos\beta \\ &\quad + (f_{RLy} + f_{RRy})\sin\beta, \qquad (1) \\ \dot{\beta} &= \frac{1}{mV}\left[(f_{FLx} + f_{FRx})\sin(\delta - \beta)\right. \\ &\quad + (f_{FLy} + f_{FRy})\cos(\delta - \beta) \\ &\quad - (f_{RLx} + f_{RRx})\sin\beta \\ &\quad \left.+ (f_{RLy} + f_{RRy})\cos\beta\right] - \dot{\psi}, \qquad (2) \\ I_z\ddot{\psi} &= \ell_F\left[(f_{FLy} + f_{FRy})\cos\delta\right. \\ &\quad \left.+ (f_{FLx} + f_{FRx})\sin\delta\right] - \ell_R(f_{RLy} + f_{RRy}) \\ &\quad + w_L(f_{FLy}\sin\delta - f_{FLx}\cos\delta - f_{RLx}) \\ &\quad + w_R(f_{FRx}\cos\delta - f_{FRy}\sin\delta + f_{RRx}) \qquad (3) \\ I_w\dot{\omega}_{ij} &= T_{ij} - f_{ijx}r, \qquad i = F, R,\ j = L, R. \qquad (4) \end{aligned}$$

where m and I_z are the mass of the car and the moment of inertia about the vertical axis respectively, V is the velocity at the center of mass, β and ψ are the sideslip and yaw angle of the vehicle. I_w is the moment of inertia of each wheel about its axis, r is the wheel radius and ω_{ij} is the angular speed of each wheel (i is marking the Front or Rear wheels and j the Left and Right). The steering angle is δ and T_{ij} is the applied torque to each wheel; f_{ijk} are the longitudinal and lateral forces which are stressed on the wheel during the driving conditions and ℓ_F,ℓ_R,w_L,$w_R$are the distances of each wheel from the center of mass.

The tyre forces f_{ijk} in the above equations are described in this paper using the Magic Formula from Pacejka and Bakker (1991). Assuming, for simplification reasons, that the camber and toe angles at each wheel are zero, the tyre forces can be found as functions of the longitudinal and lateral slips

$$s_{ijx} = \frac{V_{ijx} - \omega_{ij}r_{ij}}{\omega_{ij}r_{ij}}, \quad s_{ijy} = \frac{V_{ijy}}{\omega_{ij}r_{ij}}, \qquad (5)$$

where V_{ijk} $(i = F, R,\ j = L, R,\ k = x, y)$ is the

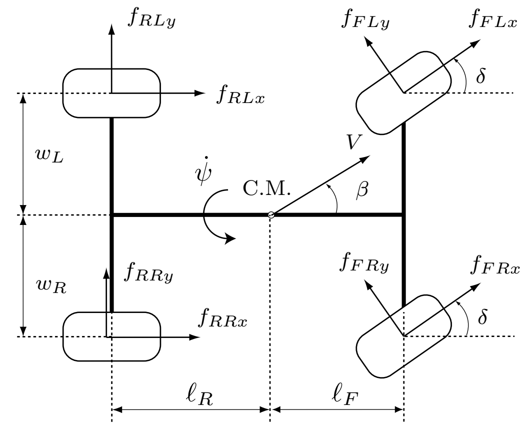

Fig. 1. *Four-wheel vehicle model.*

longitudinal velocity at the center of each of the four wheels. If we also assume a linear dependence of the tyre friction forces to the normal forces acting on each tyre, we get

$$\mu_{ij} = f_{ij}/f_{ijz}, \qquad \mu_{ijk} = f_{ijk}/f_{ijz}, \qquad (6)$$

where $f_{ij} = \sqrt{f_{ijx}^2 + f_{ijy}^2}$ is the total friction force acting on each tyre, μ_{ij} is the total tyre force coefficient, μ_{ijk} are the longitudinal and lateral tyre force coefficients, and f_{ijz} are the vertical forces on each of the four wheels. The total tyre force coefficient is calculated using the MF

$$\mu_{ij}(s_{ij}) = MF(s_{ij}) = D\sin(C(\text{atan}(Bs_{ij}), \qquad (7)$$

where $s_{ij} = \sqrt{s_{ijx}^2 + s_{ijy}^2}$. Then, assuming symmetric tyre characteristics in the longitudinal and lateral direction, we can find the longitudinal and lateral tyre force coefficients using the friction circle equations

$$\mu_{ijk} = -\frac{s_{ijk}}{s_{ij}}\mu_{ij}(s_{ij}). \qquad (8)$$

Finally, neglecting the pitch and roll rotation along with the vertical motion of the sprung mass of the vehicle, the vertical forces f_{ijz} on each wheel can be found using the static load distribution on the car and the longitudinal/lateral load transfer caused from longitudinal and lateral acceleration. Taking for example the front-left wheel the vertical tyre force is

$$f_{FLz} = f_{FLz}^0 - \Delta f_L^x - \Delta f_F^y, \qquad (9)$$

where the static load is given by

$$f_{FLz}^0 = \frac{m\,g\,\ell_R, w_R}{(\ell_F + \ell_R)(w_L + w_R)}, \qquad (10)$$

and the longitudinal load transfer is given as a function of the longitudinal acceleration by

$$\Delta f_L^x = \frac{m\,h\,w_R}{(\ell_F + \ell_R)(w_L + w_R)}a_x, \qquad (11)$$

where h the distance from the Center of Mass of the