ARTICLE IN PRESS

Mechatronics xxx (2015) xxx–xxx

Contents lists available at ScienceDirect

Mechatronics

journal homepage: www.elsevier.com/locate/mechatronics

Direct yaw moment control actuated through electric drivetrains and friction brakes: Theoretical design and experimental assessment

Leonardo De Novellis [a], Aldo Sorniotti [a,*], Patrick Gruber [a], Javier Orus [b], Jose-Manuel Rodriguez Fortun [b], Johan Theunissen [c], Jasper De Smet [c]

[a] University of Surrey, United Kingdom
[b] Instituto Tecnológico de Aragón, Spain
[c] Flanders' Drive, Belgium

ARTICLE INFO

Article history:
Received 8 June 2014
Accepted 22 December 2014
Available online xxxx

Keywords:
Direct yaw moment control
Fully electric vehicle
Yaw rate
Sideslip
Friction brake

ABSTRACT

A significant challenge in electric vehicles with multiple motors is how to control the individual drivetrains in order to achieve measurable benefits in terms of vehicle cornering response, compared to conventional stability control systems actuating the friction brakes. This paper presents a direct yaw moment controller based on the combination of feedforward and feedback contributions for continuous yaw rate control. When the estimated sideslip exceeds a pre-defined threshold, a sideslip-based yaw moment contribution is activated. All yaw moment contributions are entirely tunable through model-based approaches, for reduced vehicle testing time. The purpose of the controller is to continuously modify the vehicle understeer characteristic in quasi-static conditions and increase yaw and sideslip damping during transients. Skid-pad, step-steer and sweep steer tests are carried out with a front-wheel-drive fully electric vehicle demonstrator with two independent drivetrains. The experimental test results of the electric motor-based actuation of the direct yaw moment controller are compared with those deriving from the friction brake-based actuation of the same algorithm, which is a major contribution of this paper. The novel results show that continuous direct yaw moment control allows significant "on-demand" changes of the vehicle response in cornering conditions and to enhance active vehicle safety during extreme driving maneuvers.

1. Introduction

The majority of the fully electric vehicles currently on the market have a basic drivetrain configuration, consisting of a single on-board electric motor drive, which is connected to the driven wheels through a single-speed transmission, an open differential and half-shafts [1–5]. However, many industrial and academic researchers are developing drivetrain layouts with multiple motors [6,7], which promise considerable performance enhancements in terms of vehicle behavior and active safety. Hence, the assessment and optimization of the performance achievable through different drivetrain configurations for fully electric vehicles is one of the main areas in automotive research.

For instance, two electric motors installed on the same axle allow a direct yaw moment control (also defined as torque-vectoring), i.e., the generation of a yaw moment through an asymmetric wheel torque distribution [8,9]. The yaw moment can be achieved without varying the overall wheel torque in traction or braking conditions, unless the electric motor drives are operating close to their torque limits. A similar decoupling between yaw moment and wheel torque demand can be achieved through the adoption of a central electric motor drive and a torque-vectoring differential [10], or through the drivetrain concept presented in [11], consisting of a main motor for vehicle traction and a second motor providing the required torque-vectoring effect.

Direct yaw moment control is also the fundamental idea behind existing vehicle stability control systems for internal combustion engine-driven vehicles [12–14]. These systems keep the vehicle within its stability limits, through engine torque reduction and actuation of individual friction brakes. However, in this case the yaw moment generation is achieved at the price of an increased overall braking torque, which reduces vehicle speed. Therefore, the friction brake-based intervention of stability control systems is mainly carried out as an emergency measure, only when the offset between the reference and actual values of vehicle yaw rate or sideslip angle goes beyond an assigned threshold [12,15].

* Corresponding author. Tel.: +44 (0)1483 689688.
E-mail address: a.sorniotti@surrey.ac.uk (A. Sorniotti).

http://dx.doi.org/10.1016/j.mechatronics.2014.12.003
0957-4158/© 2015 The Authors. Published by Elsevier Ltd.

Nomenclature

The subscripts 'F' and 'R' respectively refer to the front axle and the rear axle. The symbols '$\dot{}$' and '$\ddot{}$' respectively indicate the first and second time derivatives of a variable. The symbol '$\wedge$' indicates an estimated variable. The symbol '$_0$' indicates an initial condition or a steady-state value. The subscripts 'MIN' and 'MAX' respectively indicate the minimum and maximum values of a variable.

a front semi-wheelbase
a_x longitudinal vehicle acceleration
$a_{x,8s}$ the average value of measured vehicle acceleration during the 8 s following the steering wheel input
a_y lateral vehicle acceleration
a_y^* maximum lateral acceleration in the linear region of the understeer characteristic
A, B, C, D matrices of the continuous state-space formulation of the system
A_{piston} area of the brake caliper piston
A_r, B_r, C_r, D_r, E_r, F_r, G_r terms of the yaw rate transfer functions
b rear semi-wheelbase
B_{CA}, C_{CA}, D_{CA}, H_{CA}, u, γ matrices and weight defining the wheel torque distribution criteria (control allocation)
C axle cornering stiffness
c_{AVC} Active Vibration Controller gain
c_{hs} half-shaft torsion damping coefficient
c_p stiffness parameter of the brush tire model
C_{valve} valve coefficient (including orifice dimension and discharge coefficient)
c_1, c_2 weighting factors used within the PSO algorithm
d track width
$\overline{D}$ disturbance term of the state-space formulation
DR damping ratio
e_r yaw rate error
e_t anti-windup variable, equal to the difference between the unsaturated yaw moment, $M_{z,tot}^{+}$, and the saturated yaw moment, $M_{z,tot}$
e_β sideslip angle error
f number of step steers considered within the PSO algorithm
f_r tire rolling resistance coefficient
F_x longitudinal tire force
F_y lateral tire force
F_y^* lateral tire force contribution in the matrix $\overline{D}$ of the single-track vehicle model
F_z vertical tire force
G_{dyn}^{FF} transfer function from δ to $M_{z,dyn}^{FF}$
G_f increment of torque demand oscillation frequency per unit time during the tests of Figs. 4 and 5
$G_{r,dyn}$ transfer function between $r_{ref,SS}$ and r when the effect of the dynamic feedforward contribution is not included
$\overline{G}_{r,dyn}$ reference transfer function from $r_{ref,SS}$ to r_{ref}
G_{M_z} transfer function from M_z to r
G_δ transfer function from δ to r
h index corresponding to a specific particle of the swarm
h_{CG} height of the center of gravity
i_{t1} gear ratio of the first transmission stage
i_{t2} gear ratio of the second transmission stage
j index referring to the step steers considered within the PSO algorithm
J cost function to be minimized within the control allocation algorithm
J_{hs} mass moment of inertia of the half-shafts
J_m mass moment of inertia of the rotating parts of the electric motor
J_r cost function to be minimized by the PSO algorithm
J_{t1} mass moment of inertia of the transmission primary shaft
J_{t2} mass moment of inertia of the transmission secondary shaft
J_{t3} mass moment of inertia of the transmission output shaft
J_w mass moment of inertia of the wheel
J_z vehicle yaw moment of inertia
k discretization index
k_{hs} half-shaft torsion stiffness
K_P proportional gain of the yaw rate controller
K_b fitting factor of the caliper volume displacement model
K_U^{lin} understeer gradient in the linear part of the understeer characteristic
K_β proportional gain of the sideslip controller
L wheelbase
l_p half length of tire contact patch
m vehicle mass
M_z generic yaw moment
M_z^{FB} feedback part of the yaw moment contribution based on yaw rate
$M_{z,dyn}^{FF}$ dynamic part of the feedforward yaw moment contribution
$M_{z,r}$ reference yaw moment from the yaw rate controller
$M_{z,stat}^{FF}$ static part of the feedforward yaw moment contribution
$M_{z,tot}^{+}$ reference yaw moment before saturation
$M_{z,tot}$ reference yaw moment after saturation
$M_{z,\beta}$ sideslip part of the yaw moment
n number of iterations of the PSO algorithm
OS_r yaw rate overshoot
p_{acc} accumulator pressure in the electro-hydraulic braking system unit
p_b caliper pressure
$\mathbf{p}_h$ velocity of the particle h
$P_{m,MAX}$ maximum drivetrain power
$\widetilde{\mathbf{q}}$ global best position of the swarm
$\mathbf{q}_h$ position of the particle h
$\bar{\mathbf{q}}_h$ best position of particle h
Q_b flow rate through the equivalent valve of the electro-hydraulic braking system
r yaw rate
$r_{MAX,j}$ peak value of vehicle yaw rate during the specific step steer test
$r_{ref,SS}$ steady-state reference yaw rate
r_{ref} reference yaw rate
r_{MAX} maximum value of yaw rate measured during a step steer test
r_1, r_2 randomly generated numbers with uniform distribution between 0 and 1
R skid pad radius
R_l laden radius of the tire
s Laplace operator
t time
t_{MAX} time at which r_{MAX} is achieved
t_{rise} rise time
t_{settling} settling time
T_b friction brake torque
T_D derivative parameter of the PID controller
T_{hs} half-shaft torque
T_I integral parameter of the PID controller
T_m electric motor torque
$T_{m,amp}$ electric motor torque amplitude during the tests of Figs. 4 and 5
$T_{m,av}$ average torque of the electric motor during the tests of Figs. 4 and 5
$T_{m,dem}$ electric motor torque demand

Please cite this article in press as: De Novellis L et al. Direct yaw moment control actuated through electric drivetrains and friction brakes: Theoretical design and experimental assessment. Mechatronics (2015), http://dx.doi.org/10.1016/j.mechatronics.2014.12.003