Evaluation of Combined Energy-Efficient and Stability Strategies Utilising Direct Yaw Moment Control

Peikun Sun^{1(✉)}, Annika Stensson Trigell^{1,2}, Lars Drugge^{1,2},
and Jenny Jerrelind^{1,2}

¹ KTH vehicle dynamics, KTH Royal Institute of Technology,
100 44 Stockholm, Sweden
peikun@kth.se

² Centre for ECO2 Vehicle Design, KTH Royal Institute of Technology,
100 44 Stockholm, Sweden
<https://www.kth.se/profile/peikun>

Abstract. For sustainability reasons it is important to reduce energy consumption during driving. One contribution to energysavings is by using proper wheel torque distributions during manoeuvring. An active energy-efficient direct yaw moment control (DYC) for electric vehicles has previously been proposed by the authors, taking the motor efficiency map into consideration. The results show a potential for reduced energy losses during driving, but it might result in stabilityproblems during safety-critical manoeuvres. In this work, consequences on stability due to this proposed energy efficient DYC is explored. Also an approach combining DYC both energy-efficiency and stability is proposed. The simulation results show that for the studied case the combination of DYC for energy-efficiency and stability can have an potential to both keep the vehicle safe and save considerable percentage of energy during both non safety-critical and safety-critical driving manoeuvres.

AQ1

Keywords: Energy-efficient · Direct yaw moment control · Stability

Nomenclature

m	Vehicle mass	I_z	Yaw moment of inertia
ψ	Yaw angle	h	Height of CoG
δ_f	Front steering angle	t_w	Vehicle track width
l_f	Distance from COG to front axle	F_{xi}	Longitudinal force
l_r	Distance from COG to rear axle	F_{yi}	Lateral force
F_{ar}	Aerodynamic resistance	v_x	Longitudinal velocity
v_y	Lateral velocity	T_i	Driving torque
f_{rr}	Rolling resistance coefficient	I_w	Rotational inertia of the wheel
ω_i	Wheel angular velocity	R_0	Effective radius of the tyre
κ_i	Longitudinal slipratio	a_y	Lateral acceleration
β	Body slipangle	C_{ar}	Aerodynamic resistance coefficient
C_f	Front axle cornering stiffness	C_r	Rear axle cornering stiffness

(CoG: centre of gravity)
($i=1$: front left, $i=2$: front right, $i=3$: rear left, $i=4$: rear right.)

1 Introduction

Direct yaw moment control (DYC) of in-wheel motor vehicles can reduce energy consumption by means of proper wheel torque distribution, however, it usually is a passive way to save energy [1–3]. Although stability control is prioritized, during non safety-critical driving manoeuvres, energy-efficient driving should play the dominant role. An active energy-efficient DYC method taking into account the motor efficiency map was proposed in previous work [4]. In this paper, stability judgements consisting of yaw rate and body slip angle are designed and the switching principles between energy-efficient DYC and stability DYC are devised. When the yaw rate and body slip angle are out of the set boundaries, stability control takes over energy-efficient DYC control until the working area is within the boundaries.

2 Methods

2.1 Vehicle Model

A two track vehicle model is used and front wheel steering is adopted. The longitudinal, lateral and yaw motions are shown in Eqs. (1)–(3)

$$m(\dot{v}_x - v_y \dot{\psi}) = F_{x1} \cos \delta_f + F_{x2} \cos \delta_f + F_{x3} + F_{x4} - F_{y1} \sin \delta_f - F_{y2} \sin \delta_f - F_{ar} \quad (1)$$

$$m(\dot{v}_y + v_x \dot{\psi}) = F_{x1} \sin \delta_f + F_{x2} \sin \delta_f + F_{y1} \cos \delta_f + F_{y2} \cos \delta_f + F_{y3} + F_{y4} \quad (2)$$

$$I_z \ddot{\psi} = (F_{x1} \sin \delta_f + F_{x2} \sin \delta_f + F_{y1} \cos \delta_f + F_{y2} \cos \delta_f) l_f - (F_{y3} + F_{y4}) l_r + (F_{x2} \cos \delta_f - F_{y2} \sin \delta_f - F_{x1} \cos \delta_f + F_{y1} \sin \delta_f + F_{x4} - F_{x3}) t_w / 2 \quad (3)$$

The equation for the forward speed of each wheel can be expressed as

$$\begin{cases} v_1 = v_x - \dot{\psi} t_w / 2; & v_2 = v_x + \dot{\psi} t_w / 2 \\ v_3 = v_x - \dot{\psi} t_w / 2; & v_4 = v_x + \dot{\psi} t_w / 2 \end{cases} \quad (4)$$

The angular velocity of the i^{th} can be expressed as

$$\omega_i = (1 + \kappa_i) v_i / R_0 \quad (5)$$

The equation of tyre motion in the longitudinal direction can be written as

$$T_i - f_{rr} F_{zi} R_0 - F_{xi} R_0 = I_w \dot{\omega}_i \quad (6)$$

The aerodynamic resistance F_{ar} can be expressed as

$$F_{ar} = 0.5 C_{ar} \rho A v_x^2 \quad (7)$$

The vehicle is designed to be a four-in-wheel motor electric vehicle and the chosen vehicle parameters are shown in Table 1. In the present study, Magic formula tyre models [5] for the longitudinal force and lateral force are used. Based on the characteristics of in-wheel motor data in Chen [1], a simplified in-wheel motor efficiency map, shown in Fig. 1, is designed where the efficiency is a function of the motor torque.

Table 1. Vehicle parameters

Parameter	Value	Parameter	Value	Parameter	Value	Parameter	Value
m	1500 kg	l_r	1.5 m	l_f	1.2 m	I_z	1700 kgm ²
t_w	1.65 m	I_w	1 kgm ²	h	0.48 m	R_0	0.3 m
C_{ar}	0.3	A	2 m ²	f_{rr}	0.01	C_f	-87 kN/rad
C_r	-69 kN/rad

Fig. 1. The motor efficiency map

2.2 Potential of DYC on Energy-Saving During Steady-State Cornering

By analysing the vehicle characteristics in steady-state cornering, the fundamental vehicle motion characteristics can be understood [6]. The potential of DYC on energy-saving can be analysed through quasi-steady-state cornering when $\dot{v}_y=0$, $\dot{\psi}=0$ and $\dot{\omega}_i=0$. During quasi-steady-state cornering, with a given v_x , a_y and M_z , the total torque can be deduced. The offline optimisation can be developed where the procedure is shown in Fig. 2. Constant velocity cornering with $\dot{v}_x = 0 \text{ m/s}^2$ and $v_x = 80 \text{ km/h}$ is analysed. The range of a_y is from 0 m/s^2 to 4 m/s^2 and for each a_y , M_z is varied from -1000 Nm to 1000 Nm .

The optimisation results for $v_x = 80 \text{ km/h}$ are shown in Figs. 3. In Fig. 3(a), two lateral accelerations results, $a_y = 1 \text{ m/s}^2$ and $a_y = 2 \text{ m/s}^2$, are presented. There are two local minimal power points for each a_y . In the a_y range from 0 m/s^2

Fig. 2. The minimal power optimisation procedure under M_z

Fig. 3. Optimisation results for $v_x = 80 \text{ km/h}$: (a) Power versus M_z for $a_y = 1 \text{ m/s}^2$ and $a_y = 2 \text{ m/s}^2$; (b) Total torque versus a_y and M_z ; (c) δ_f versus a_y and M_z ; (d) Power consumption versus a_y and M_z ; (e) Percentage of energy saving compared to four-wheel equal torque driving; (f) Percentage of energy saving compared to two-wheel equal torque driving.

to 4 m/s^2 , the black lines in Fig. 3(d) represent local minimal power points. It can be seen in Fig. 3(e) and (f) that considerable percentage of energy saving can be achieved at local minimal power points compared to both two-wheel equal torque driving and four-wheel equal torque driving. The grey planes in Fig. 3(e) and (f) represent 0% energy saving. From Fig. 3(b), for each a_y , M_z does not change T_{all} so much. From Fig. 3, it is shown that DYC can be used to reduce the power consumption.

2.3 Energy-Efficient and StabilityDYC Design

The idea of active energy-efficient DYC is to find the most energy-efficient yaw moment for a given reference lateral acceleration a_{yr} . The steering angle follows

Fig. 4. The controller structure for energy-efficient DYC [4]

the same direction of a_{yr} to avoid counter-steering. The offline torque distribution rule is optimised based on the motor efficiency map in Fig. 1. The structure of energy-efficient DYC is shown in Fig. 4.

The principle of stability DYCs is that the desired yaw rate should be tracked. The reference yaw rate $\dot{\psi}_r$ [6] as a function δ_f can be expressed as Eq. 8.

$$\dot{\psi}_r = \frac{v_x}{L + \frac{mv_x^2}{L} \left(\frac{l_f}{C_r} - \frac{l_r}{C_f} \right)} \delta_f \quad (8)$$

The stability yaw moment M_{zs} is then

$$M_{zs} = I_z \ddot{\psi}_r - ((F_{y1} + F_{y2})l_f - (F_{y3} + F_{y4})l_r) \quad (9)$$

The vehicle stability working area can be a combination of the yaw rate $\dot{\psi}$ and the body slip angle β . In Rajamani [7], the simplified suggested ranges for yaw rate and body slip angle are expressed as

$$\begin{cases} |\dot{\psi}| \leq 0.85 \frac{\mu g}{v_x} \\ |\beta| \leq \text{atan}(0.02 \mu g) \end{cases} \quad (10)$$

The $\dot{\psi}$ - β phase-plane method [8,9] and Lyapunov method [10] can also divide the vehicle working area into a stable and a non-stable region. The comparison of simplified method, phase-plane method and Lyapunov method are shown in Fig. 5.

It can be seen that the simplified working area in Eq. (10) is within the phase-plane method and the Lyapunov method results in unreasonable slip angle and yawrate. Therefore, the simplified stability area described in Eq. (10) was chosen for the present study.

Fig. 5. Comparison of stability regions for $v_x = 80$ km/h and $\mu=0.8$.

The strategy for both energy-efficient and stability DYC is that during nonsafety-critical manoeuvres, energy-efficient DYC plays the dominant role and during safety-critical manoeuvres, stability DYC control takes charge. The switching principle between energy-efficient DYC and stability DYC is: when both of $\dot{\psi}$ and β are within 0.6 times maximum values, energy-efficient DYC works; when any of $\dot{\psi}$ and β is out of 0.6 times maximum values, stability DYC will be used until both of $\dot{\psi}$ and β are within 0.58 times maximum values.

3 Results and Conclusions

The ISO 3888-1 double lane change [11], shown in Fig. 6, is used to verify the effectiveness of the controller during transient cornering. For comparison, based on ISO 3888-1, an extended double lane change is designed. The path parameters are shown in Table 2.

Fig. 6. The double lane change path

Table 2. Path parameters

Parameter	Extended	ISO3888-1
L_1 (m)	15	15
L_2 (m)	60	30
L_3 (m)	25	25
L_4 (m)	50	25
L_5 (m)	30	30
W_1 (m)	2.065	2.065
W_2 (m)	2.3125	2.3125
W_3 (m)	2.395	2.395
B (m)	3.58	3.58

Table 3. Average power for all simulated manoeuvres

Average power (kW)	Extended	ISO3888-1
Four-wheel equal torque	10.26	13.22 (unsafe)
Two-wheel equal torque	8.89	11.69 (unsafe)
Energy-efficient DYC	8.52	11.65 (unsafe)
Stability DYC	8.74	12.57
Combined DYC	8.52	11.84

Constant velocity $v_x = 80$ km/h during these two paths are simulated and the results are shown in Figs. 7 and 8. The line colors are explained as follows: black: two-wheel equal torque driving; green: energy-efficient DYC; red: stability DYC; blue: combined DYC. The combined DYC is a combination of the proposed energy-efficient and stability controls. The average power consumptions of four-wheel equal torque driving, two-wheel equal torque driving, energy-efficient DYC, stability DYC and combined DYC are summarised in Table 3.

From Figs. 7(b), 8(b) and (e), when the lane change length reduces, the working area increases while using two-wheel equal torque driving and energy-efficient DYC. During the extended double lane change in Fig. 7, the working area is within the stability boundaries and the combined control is energy-efficient only. During ISO 3888-1 double lane change, only stability DYC and combined DYC can keep the working area within the stability boundaries. From Figs. 7(c), 8(c) and Table 3, it can be seen that the energy-efficient DYC during extended double lane change consumes the minimal power. Energy-efficient DYC can save 2.5% energy compared to stability DYC, 4.1% compared to two-wheel equal torque driving and 16.9% compared to four-wheel equal torque driving. During the ISO 3888-1 double lane change, combined DYC can keep vehicle safe and can save 5.8% compared to stability DYC.

Fig. 7. Simulation results of extended double lane change

Fig. 8. Simulation results of ISO 3888-1 double lane change

Acknowledgment. The authors greatly appreciate the support from China Scholarship Council (CSC), the Centre for ECO2 Vehicle Design at KTH funded by the Swedish Innovation Agency Vinnova (Grant Number 2016-05195) and the strategic research area TRENOP.

References

1. Chen, Y., Wang, J.M.: Adaptive energy-efficient control allocation for planar motion control of over-actuated electric ground vehicles. *IEEE Trans. Control Syst. Technol.* **22**(4), 1362–1373 (2014). <https://doi.org/10.1109/TCST.2013.2287560>
2. Han, Z.L., Xu, N., Chen, H., Huang, Y.J., Zhao, B.: Energy-efficient control of electric vehicles based on linear quadratic regulator and phase plane analysis. *Appl. Energy* **213**, 639–657 (2018). <https://doi.org/10.1016/j.apenergy.2017.09.006>
3. Zhai, L., Hou, R.F., Sun, T.M., Kavuma, S.: Continuous steering stability control based on an energy-saving torque distribution algorithm for a four in-wheel-motor independent-drive electric vehicle. *Energies* **11**(2), 350 (2018). <https://doi.org/10.3390/en11020350>
4. Sun, P., Stensson Trigell, A., Drugge, L., Jerrelind, J.: Energy-efficient direct yaw moment control for in-wheel motor electric vehicles utilising motor efficiency maps (2019). (Submitted for publishing)
5. Pacejka, H.: *Tire and Vehicle Dynamics*, 3rd edn. Elsevier, Amsterdam (2012)
6. Abe, M.: *Vehicle Handling Dynamics: Theory and Application*. Butterworth Heinemann, Amsterdam (2015)
7. Rajamani, R.: *Vehicle Dynamics and Control*. Springer, Boston (2012). <https://doi.org/10.1007/978-1-4614-1433-9>
8. Bobier, C.G., Gerdes, J.C.: Staying within the nullcline boundary for vehicle envelope control using a sliding surface. *Veh. Syst. Dyn.* **51**(2), 199–217 (2013)

9. Ono, E., Hosoe, S., Tuan, H.D., Doi, S.: Bifurcation in vehicle dynamics and robust front wheel steering control. *IEEE Trans. Control Syst. Technol.* **6**(3), 412–420 (1998)
10. Yin, G., Jin, X., Qing, Z., Bian, C.: Lateral stability region conservativeness estimation and torque distribution for FWIA electric vehicle steering. *Sci. China Technol. Sci.* **58**(4), 669–676 (2014)
11. Passenger cars—Test track for a severe lane-change manoeuvre—Part 1: Double lane-change, International Standards Organization (ISO) (2018). ISO: 888-1:2018(E)