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by Ivan Christian Hernando

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Dynamic Analysis of Four-Wheel Electric Vehicle: Effect of Torque Vectoring

Hernando, Ivan Christian[1, a)] and Wasiwitono, Unggul[2, b)]

Author Affiliations

[1]*Mechanical Engineering Department of Petra Christian University, Indonesia*

[2]*Mechanical Engineering Department of Institut Teknologi Sepuluh Nopember, Indonesia*

Author Emails

[a)] Corresponding author: ivan.hernando@petra.ac.id

[b)]unggul@me.its.ac.id

Abstract. A vehicle has many parameters that can be measured while moving. To increase stability, these parameters need to be controlled by adjusting the torque value for each wheel. This control is often referred to as Torque Vectoring (TV). Understanding how the vehicle responds when input is given, in this case, the torque on each wheel is important in case of design a control system. Vehicle stability is indicated by the yaw rate and side slip angle value. In this study the effect of longitudinal velocity, steering angle, and correction torque on vehicle dynamics. Increasing torque correction value will increase the yaw rate and the side slip angle value linearly. Increasing steering angle will increase the yaw rate and the side slip angle linearly. As the longitudinal velocity increases, the yaw rate and the side slip angle value will increase exponentially. The side slip angle has a higher stability limit than for the yaw rate. This is means that the yaw rate stability needs more attention.

INTRODUCTION

One way to support the reduction of global warming is to reduce the use of internal combustion vehicles and replace them with electric vehicles. Many countries have already started planning to ban internal combustion vehicles. Norway will ban the use of internal combustion vehicles in 2025. There has been an increase in electric vehicle purchases in recent decades. In 2020, the electric vehicle sales have increased by up to 41% globally[1].

This increase certainly needs to be balanced with various infrastructure and innovations that support the safety and comfort of driving using electric vehicles. Electric vehicles provide convenience, especially in terms of control and flexibility in terms of performance, compared to internal combustion engine vehicles. In addition, electric cars also have a faster actuator response [2]. These advantages will make controlling the electric vehicle more accessible.

There are a lot of control models to optimize the control of electric vehicles. Some of the previous techniques that have been used include the anti-lock braking system (ABS), traction control system (TCS), and electronic stability control (ESC). ABS and TCS work inversely to control vehicle stability. ABS will work when the vehicle is decelerating, while TCS will work when the vehicle is accelerating. The ESC controls the stability by braking each car wheel [3].

Recently, the demand for increased vehicle stability getting higher, which of course, will not be easily achieved by the previously stated systems. For that, there needs to be a different approach. One of the techniques is to control the torque applied to each car wheel. This control is often referred to as Torque Vectoring (TV), which was first proposed by Ricardo Company [4]. The control system of electric cars is easier because there is no need to add mechanical components (just with electronic control systems), so the development of the TV system is relatively rapid for the stability control of electric cars.

The TV system is modeled and controlled with various control systems in its development. Some of them are the fuzzy logic control method [3], linear parameter varying (LPV) control method [5], model predictive control (MPC) model control method [6], proportional control method integral derivative (PID) [7], and the sliding mode controller (SMC) control method [8]. Besides the various control methods, the car dynamics models vary, ranging from 2 to 14 degrees of freedom (DOF). The tire model used also varies depending on the approach taken, namely, the Pacejka tire model [9], the Dugoff tire model[10], and the Salani tire model [11].

TV development is still being carried out because of several obstacles, including a complicated algorithm that burdens the microcontroller. In addition, until now, there is still no general agreement regarding the potential utilization of this TV system [12]. For this reason, there is an opportunity in making a TV system that can improve the stability and control of electric cars. A vehicle has many parameters that can be measured while moving, including linear speed, linear acceleration, angular velocity, angular acceleration, side slip angle, wheel slip, and others. These parameters will affect the stability of a vehicle. In order to increase stability, these parameters need to be controlled by adjusting the torque value for each wheel (TV).

Jalali, et al [3] using the slip ratio and the error to the slip ratio as the value to increase the vehicle stability by the TV control. The wheels are always expected to provide the greatest lateral and longitudinal forces that the rubber wheels can do by controlling the slip ratio. Kaiser, et al. [13] choose the longitudinal force value that is needed and the yaw moment value to determine the torque value. These values are calculated based on the value of the yaw rate value of the vehicle and the side slip angle value of the vehicle when the vehicle is in a linear state at that speed. Kaiser, et al. [14] control the vehicle's longitudinal speed value and the yaw rate value of the vehicle. The control system then continued with the control of the motor torque limit and the tire slip limit. Siampis et al. [15] carry out vehicle control so the vehicle can be in a steady state turning condition. These controls include the yaw rate value of the vehicle, the side slip angle value of the vehicle, the longitudinal speed value of the vehicle, and the rear wheel angular speed value of the vehicle. Alipour, et al. [16] control the longitudinal speed and the yaw rate value so that the vehicle is stable.

Conclusion from the literature above, control based on yaw rate is the most control used to ensure vehicle stability. In addition to the yaw rate control, one more control is needed to ensure the wheel is always in a linear state. Control using the side slip angle value is a control that is able to ensure the wheel is always in its linear condition [17]. It is necessary to understand how the vehicle responds when designing a control system for a vehicle especially when input is given, in this case, the torque on each wheel. To the best of the author's knowledge, there is still no literature that observed the response of a vehicle for a given torque correction and different steering angle. Therefore, this study will observe how the car responds when given torque correction. It would be observed from the yaw rate value of the vehicle and side slip angle value of the vehicle. This is because the yaw rate value of the vehicle will maximize the planar stability of the vehicle. In other way, the side slip angle value of the vehicle will ensure the wheels are in a linear condition. The novelty of this research is seeing how the car responds at different speeds and steering angle as measured by the yaw rate value of the vehicle and side slip angle value of the vehicle based on the torque correction given.

VEHICLE MODEL

The vehicle model used is the 14-DOF model. The first 3-DOF is obtained from the lateral, longitudinal, and vertical movement of the vehicle body. The other 3-DOF obtained from the rotational motion of pitch, roll, and yaw. Each wheel has 2-DOF, namely rotational movement about the wheel axis and vertical movement due to suspension. Numbering on the tires is done clockwise, starting from the left front wheel with the number 1 and ending with the left rear wheel with the number 4. Mathematically, the 6-DOF equation of motion can be seen from equations 1 to 8 [18].

$$F_x = \sum_{i=1}^{4} F_{xi} + F_{xg} \quad , i = 1,2,3,4 \tag{1}$$

$$F_y = \sum_{i=1}^{4} F_{yi} + F_{yg} \quad , i = 1,2,3,4 \tag{2}$$

$$F_z = \sum_{i=1}^{4} F_{zi} + F_{zg} \quad ,i = 1,2,3,4 \tag{3}$$

$$M_x = \sum_{i=1}^{4} h_i F_{yi} + (-1)^{i-1} b \cdot F_{zi} \quad ,i = 1,2,3,4 \tag{4}$$

$$M_{yi} = \begin{cases} -h_i F_{xi} - a_f \cdot F_{zi} & ,i = 1,2 \\ -h_i F_{xi} + a_r \cdot F_{zi} & ,i = 3,4 \end{cases} \tag{5}$$

$$M_y = \sum_{i=1}^{4} M_{yi} \quad ,i = 1,2,3,4 \tag{6}$$

$$M_{zi} = \begin{cases} (-1)^i b \cdot F_{xi} + a_f F_{yi} & ,i = 1,2 \\ (-1)^i b \cdot F_{xi} + a_r F_{yi} & ,i = 3,4 \end{cases} \tag{7}$$

$$M_z = \sum_{i=1}^{4} M_{zi} \quad ,i = 1,2,3,4 \tag{8}$$

where b is half of the thread, a_f is the distance between vehicle body center of gravity to the vehicle front axle, a_r is the distance between vehicle body center of gravity to the vehicle rear axle, and h_i is the distance between tire ground to the vehicle body in each tire. F_{xi} is a force from the tire in the longitudinal direction of the vehicle frame which is obtained from the tire model. F_{yi} is a force from the tire in the lateral direction of the vehicle frame which is obtained from the tire model. F_{zi} is a force from the tire in the vertical direction of the vehicle frame. F_{zi} is a force that is caused by the suspension. F_{xg} is a force from the weight of the vehicle in the longitudinal direction of the vehicle frame. F_{yg} is a force from the weight of the vehicle in the lateral direction of the vehicle frame. F_{zg} is a force from the weight of the vehicle in the vertical direction of the vehicle frame.

The equation for rotational movement of vehicle wheel can be seen in equation 9 [18].

$$I_{yw}\,\dot{\omega}_{ywi} = T_{ywi} - (F_{xwi} + 0.015 k_w h_{wi}) R_l \quad ,i = 1,2,3,4 \tag{9}$$

where T_{ywi} is the torque in the wheel shaft. F_{xwi} is a force from tire in longitudinal direction of the wheel frame which is obtained from the tire model. k_w is the tire spring constant, h_{w1} is the displacement of the vehicle wheel on the vertical direction, and R_l is the vehicle wheel radius. Equation for the vertical movement of the wheel can be seen in equation 10 [18].

$$m_w \ddot{h}_{wi} = -k_w h_{wi} + k_{si}(h_{ci} - h_{wi}) + d_{si}(\dot{h}_{ci} - \dot{h}_{wi}) + F_{zgw} \quad ,i = 1,2,3,4 \tag{10}$$

where m_w is the wheel mass, k_{si} is the spring constant, and d_{si} is the damping constant. h_{ci} is the displacement of the car corner, i.e., h_{c1} is refer to the displacement of the front left car corner. F_{zgw} is a force from the weight of the vehicle wheel. This force is in the vertical direction of the wheel frame.

Salaani's tire model is used in this study to estimate the lateral force of the tire and the longitudinal force of the tire [16]. The selection of this tire model is based on the ease of equations for this tire model and the accuracy of the tire model. The tire model P225/60R/16 is chosen for this mathematical model. The value of the steering angle in each front tire for this study is gotten from the Ackerman steering model. The steering ratio is 18.3719 [16]. Equations 9 until 11 show the Ackerman steering model formula [19].

$$\delta = \frac{\delta_{sw}}{\gamma} \tag{9}$$

$$\delta_L = \tan^{-1}\left(\frac{(a_f + a_r)\tan(\delta)}{(a_f + a_r) - 0.5b\tan(\delta)}\right) \quad (10)$$

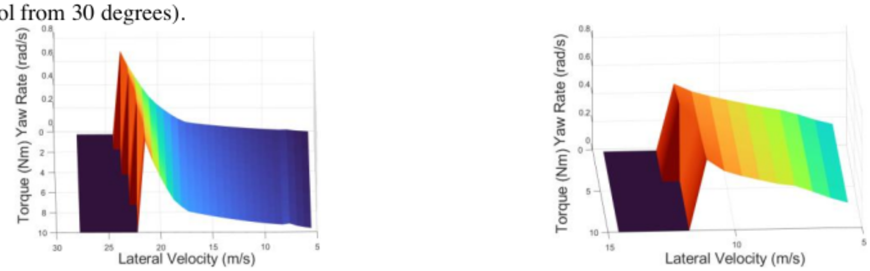
$$\delta_R = \tan^{-1}\left(\frac{(a_f + a_r)\tan(\delta)}{(a_f + a_r) + 0.5b\tan(\delta)}\right) \quad (11)$$

where the steering wheel angle is symbolled by δ, the steering ratio is symbolled by γ, the front left steering angle is symbolled by δ_L, and the front right steering angle is symbolled by δ_R.

YAW RATE VEHICLE RESPONSE

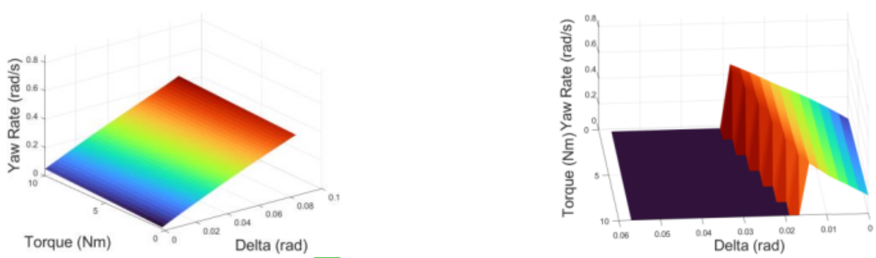
The yaw rate controls are done by providing opposite correction torque on both wheels, in this case are the left wheel and the right wheel. First, the simulation is conducted with a constant input of the steering angle condition. The longitudinal vehicle speed and correction torque are varied. The longitudinal speed varies from 20km/h to 100km/h. The torque varies from 0.25Nm to 10Nm. Two cases are considered: low steering angle of 5 degrees and high steering angle of 90 degrees. Figure 1 shows the vehicle yaw rate response for different speeds and correction torque. It showed that with an increase in velocity, the yaw rate would increase exponentially. Figure 1 also shows that with an increase in correction torque, the yaw rate would increase linearly. The yaw rate keeps increasing until at some point the vehicle loses control (marked by the black color). The trend is the same between small and big angle, but at bigger steering angle, the vehicle has a bigger yaw rate and which means it will quickly lose control (at 75 degrees and correction torque 0.25Nm, the vehicle loses control from 42.5km/h).

Next, we simulate the case vehicle have a constant longitudinal speed and at the same time, the vehicle steering angle and correction torque values are varied. The steering angle is varies from 0 degrees to 90 degrees. Two cases are considered: low speed of 35 km/h and high speed of 65 km/h. The yaw rate response for these cases is shown in Figure 2. Figure 2 showed that with an increase in steering angle and correction torque, the yaw rate would increase linearly. The yaw rate keeps increasing until at some point the vehicle loses control (marked by the black color). The trend is the same between low and high longitudinal velocity, but at higher longitudinal velocity, the vehicle has a bigger yaw rate and which means it will quickly lose control (at 65 km/h and correction torque 0.25Nm, the vehicle loses control from 30 degrees).



(a) (b)

FIGURE 1. Yaw rate response at a constant steering angle (a) 5 degrees and (b) 75 degrees



(a) (b)

FIGURE 2. Yaw rate response at a constant longitudinal speed (a) 35 km/h and (b) 65 km/h

SIDE SLIP ANGLE VEHICLE RESPONSE

The side slip angle controls are done by providing same torque correction on all wheels. First, the simulation is conducted with a constant steering angle condition, and the longitudinal speed and correction torque are varied. The longitudinal speed varies from 20km/h to 100km/h. The torque varies from 0.25Nm to 10Nm. Two cases are considered: low steering angle of 5 degrees and high steering angle of 90 degrees. From Figure 3, the vehicle side slip angle response for different speeds and correction torque. Figure 3 showed that with an increase in vehicle longitudinal velocity, the vehicle side slip angle value would increase exponentially. Figure 3 also shows that increasing the correction torque will make the vehicle side slip angle value increase linearly until the vehicle near loses control then the vehicle side slip angle value would increase rapidly. The side slip angle keeps increasing until at some point the vehicle loses control (marked by the black color). The trend is the same between small and big angle, but at bigger steering angle, the vehicle has a bigger side slip angle and which means it will quickly lose control (at 75 degrees and correction torque 0.25Nm, the vehicle loses control from 45km/h).

Next, we simulate the case vehicle runs at a constant longitudinal speed, at the same time, the vehicle steering angle and correction torque values are varied. The vehicle steering angle value varies from 0 degrees to 90 degrees. Two cases are considered: low speed of 35 km/h and high speed of 65 km/h. The side slip angle response for these cases is shown in Figure 4. Figure 4 showed that with an increase in vehicle steering angle value and correction torque, the vehicle side slip angle value would increase linearly until the vehicle near loses control then the vehicle side slip angle value would increase rapidly. The side slip angle keeps increasing until at some point the vehicle loses control (marked by the black color). The trend is the same between low and high longitudinal velocity, but at higher longitudinal velocity, the vehicle has a bigger side slip angle and which means it will quickly lose control (at 65 km/h and correction torque 0.25Nm, the vehicle loses control from 32.5 degrees).

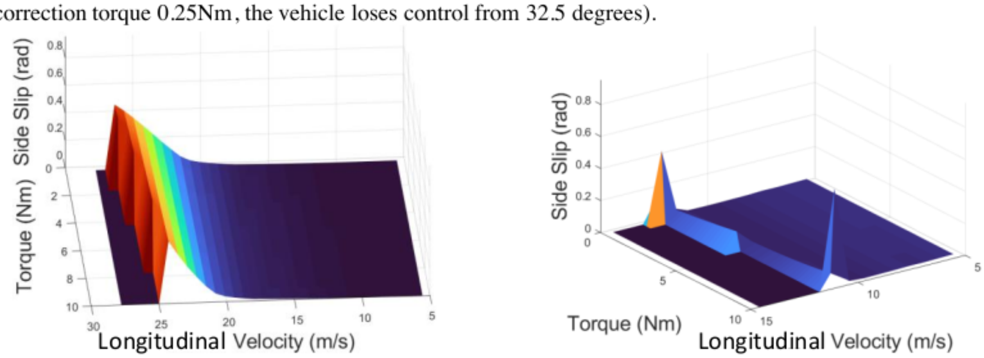


FIGURE 3. Side slip angle response at a constant steering angle (a) 5 degrees and (b) 75 degrees

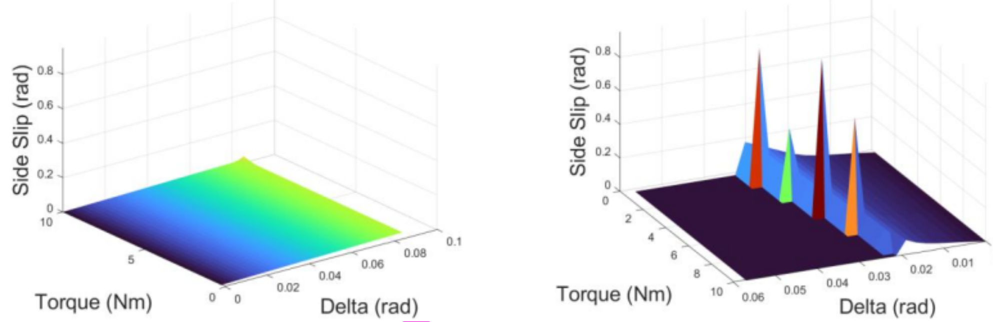


FIGURE 4. Side slip angle response at a constant longitudinal speed (a) 35 km/h and (b) 65 km/h

CONCLUSION

The vehicle yaw rate and the vehicle side slip angle response are generally similar. Both of the vehicle yaw rate value and the vehicle side slip angle value will increase linearly with increasing torque correction value. Those values are also will increase linearly with increasing steering angle. Both of the vehicle yaw rate value and the vehicle side slip angle value will increase exponentially as the vehicle longitudinal velocity value is increases. The difference between Both of the vehicle yaw rate value and the vehicle side slip angle value responses is seen when the vehicle approaches limits of stability with increasing torque correction value. The rise in yaw rate remains linear, while the increase in side slip angle becomes rapid. Based on the simulation, the vehicle stability limit for the vehicle side slip angle is higher than for the vehicle yaw rate so the vehicle yaw rate stability needs more attention to make sure the vehicle always stable. It is hoped that by knowing the response to vehicle stability parameters, the controller design becomes more accurate and can more optimally improve vehicle stability.

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