

Development of an Advanced Torque Vectoring Control System for an Electric Vehicle with In-Wheel Motors using Soft Computing Techniques

Kiumars Jalali, Thomas Uchida, Steve Lambert and John McPhee
 Univ. of Waterloo

ABSTRACT

A two-passenger, all-wheel-drive urban electric vehicle (AUTO21EV) with four direct-drive in-wheel motors has been designed and developed at the University of Waterloo. A 14-degree-of-freedom model of this vehicle has been used to develop a genetic fuzzy yaw moment controller. The genetic fuzzy yaw moment controller determines the corrective yaw moment that is required to stabilize the vehicle, and applies a virtual yaw moment around the vertical axis of the vehicle. In this work, an advanced torque vectoring controller is developed, the objective of which is to generate the required corrective yaw moment through the torque intervention of the individual in-wheel motors, stabilizing the vehicle during both normal and emergency driving maneuvers. Novel algorithms are developed for the left-to-right torque vectoring control on each axle and for the front-to-rear torque vectoring distribution action. Several maneuvers are simulated to demonstrate the performance and effectiveness of the proposed advanced torque vectoring controller, and the results are compared to those obtained using the ideal genetic fuzzy yaw moment controller. The advanced torque vectoring controller is also implemented in a hardware- and operator-in-the-loop driving simulator to further evaluate its performance.

CITATION: Jalali, K., Uchida, T., Lambert, S. and McPhee, J., "Development of an Advanced Torque Vectoring Control System for an Electric Vehicle with In-Wheel Motors using Soft Computing Techniques," *SAE Int. J. Alt. Power.* 2(2):2013, doi:10.4271/2013-01-0698.

INTRODUCTION

The AUTO21EV vehicle model with four direct-drive in-wheel motors and an active steering system is an exciting platform on which to apply advanced motion control techniques, such as advanced slip control and torque vectoring systems, since the motor speeds and torques can be generated and controlled quickly, precisely, and independently at each wheel. An advanced fuzzy slip control system is developed and evaluated in [9] using predefined test maneuvers and a novel path-following driver model [10]. In addition, a genetic fuzzy yaw moment controller is developed in [11], the objective of which is to determine the corrective yaw moment required to minimize the vehicle yaw rate and sideslip errors. This genetically-tuned fuzzy yaw moment controller acts as a high-level supervisory module that assigns tasks to the lower-level controllers and actuators. In this paper, an advanced torque vectoring controller is developed for the AUTO21EV that distributes the task of generating the calculated corrective yaw moment to the in-wheel motors. The developed advanced torque vectoring controller consists of left-to-right and front-to-rear torque

vectoring components, which work together to distribute the calculated corrective yaw moment in an integrated approach.

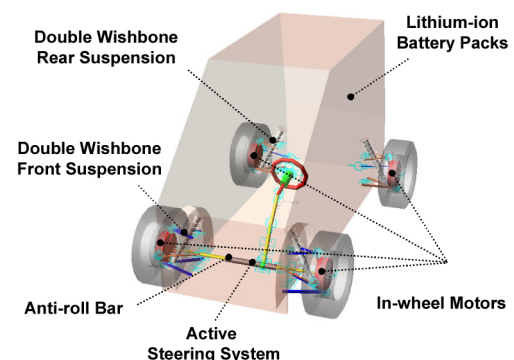


Figure 1. AUTO21EV concept vehicle

Figure 1 illustrates the AUTO21EV, which is a two-passenger, all-wheel-drive urban electric vehicle developed and modeled using the ADAMS/View software. Table 1 lists some of the relevant parameters used for the AUTO21EV model. The use of small but powerful direct-drive in-wheel motors allows for the implementation of the most advanced

all-wheel-drive system in which the optimal traction force can be generated and controlled on each wheel.

Table 1. AUTO21EV model parameters

Parameter	Value	Unit
Vehicle mass (M)	1075	kg
Vehicle yaw moment of inertia (I_{zz})	1171	kg m ²
Distance from CG to front axle (a)	0.82	m
Distance from CG to rear axle (b)	0.98	m
Front track width (t_f)	1.275	m
Rear track width (t_r)	1.35	m
Front tire cornering stiffness ($C_{\alpha F}$)	45570	N/rad
Rear tire cornering stiffness ($C_{\alpha R}$)	45570	N/rad
Tire size	175/55 R15	–
Tire unloaded radius (r)	0.29	m

TORQUE VECTORING CONTROL SYSTEMS

In conventional four-wheel-drive (4WD) vehicles, either all the wheels of the vehicle are permanently driven, which is referred to as an all-wheel-drive (AWD) vehicle, or one of the two axles is always linked to the engine and the second can be engaged (either manually or automatically) when needed. In both cases, in order to transmit the available engine torque to the wheels, inter-axle differentials or clutches must be installed on the propshaft, between the front and rear axles, and on both drive axles in order to split the torque between the left and right wheels. A comprehensive overview of the drivelines and differentials available for 4WD and AWD vehicles can be found in [1, 2]. Although these powertrain configurations can enhance the traction and driving dynamics when faced with various road conditions, they suffer from higher costs, weight, and power consumption, as well as inefficiency.

In recent years, the market for AWD vehicles has evolved and expanded, and customer expectations of the driving dynamics and stability of such vehicles have increased dramatically. In addition, consumers are increasingly willing to request AWD on their new cars at extra cost [3]. In order to meet these requirements and remain competitive in the market, active differentials for driveline systems are being introduced, which are able to distribute the engine torque to the front and rear axles, as well as to the left and right wheels on each axle, depending on the driving maneuver and road conditions. The possibility of actively influencing the distribution of the available engine torque based on the driving situation and traction conditions has led to an upheaval of the AWD market [4]. However, pressures to reduce CO₂ emissions and increase fuel efficiency standards force the weight, cost, and power requirements of these actuators to be minimized.

Torque vectoring is the term introduced by the Ricardo Company [2] to describe a means of varying the distribution

of engine torque between two outputs of a differential unit by controlling the torque over a relatively small speed difference between the outputs. Torque is directed in proportion to the relative shaft speeds, and can be biased seamlessly from one output to the other. Active torque vectoring systems were later introduced into the powertrain system, presenting torque-on-demand capabilities, where a center differential distributes the available engine torque to the front and rear axle differentials. Each axle differential can be equipped with an active torque vectoring system as well, which can be used to distribute the torque between the left and right wheels. An active center differential apportions the torque depending on the driving dynamics and the traction potentials on each axle, whereas an active axle differential distributes the torque based on the vehicle dynamics and the traction potential on each wheel of a single axle.

The benefits of active torque vectoring technology are twofold: the enhancement of vehicle traction and the enhancement of vehicle stability. On the one hand, torque vectoring can be used to modulate the individual wheel speeds to keep each tire operating in its optimal longitudinal slip range for the best traction. On the other hand, torque vectoring can be used to enhance the vehicle handling response by generating a corrective yaw moment to influence the vehicle yaw behavior [5]. For instance, Figure 2 illustrates an active powertrain with an active center differential, an active rear differential, and an open front differential [6]. In normal driving conditions, where the differential actuators are not active, the center differential splits the torque generated by the engine in a fixed ratio of 40:60 percent to the front and rear axles, respectively, using a planetary gear. This fixed torque distribution ratio can be established based on the axle-load ratios, the design philosophy of the vehicle, or the desired handling characteristics of the vehicle [8].

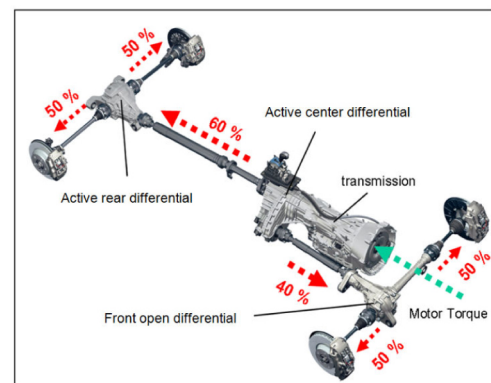


Figure 2. An active powertrain system with active center and rear differentials [6]

In addition, the front and rear differentials split their received torques in half using open differentials. Using torque vectoring technology for traction enhancement, the torque vectoring ratios on the center and rear differentials can vary depending on the traction potentials available on each axle or

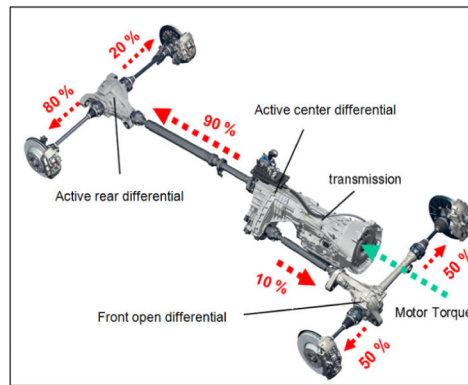


Figure 3. Torque vectoring in an active powertrain to enhance the vehicle traction [6]

tire, such that the highest traction force on each wheel is attained. Figure 3 illustrates the case where the front tires are on ice with almost no traction, and more traction is available on the rear-right tire than the rear-left tire. As can be seen, the active center differential is capable of sending 90 percent of the available engine torque to the rear axle, where more traction is available. Meanwhile, the active rear differential splits the torque received from the center differential asymmetrically by a ratio of 20:80 percent for the left and right wheels of the rear axle, respectively.

Torque vectoring can also be used as a stability control system, where the available drive torque is distributed among the wheels in order to generate a corrective yaw moment about the vertical axis of the vehicle. The corrective yaw moment can be generated in a left-to-right torque vectoring mode, where the available drive torque is distributed asymmetrically to the left and right wheels of an axle, or in a front-to-rear mode, where the available engine torque is distributed asymmetrically to the front and rear axles. For instance, an oversteering situation, in which the adhesion potential has been reached at the rear axle, can be corrected using left-to-right torque vectoring on the front axle, where more torque is transmitted to the front-inside tire (Figure 4-a). Conversely, an understeering situation, in which the adhesion potential has been exceeded on the front axle, can be corrected using left-to-right torque vectoring on the rear axle, where a higher percentage of the available axle torque is transmitted to the rear-outside tire in order to generate the required corrective yaw moment (Figure 4-b).

The driving dynamics of a vehicle can also be influenced by varying the front-to-rear torque distribution. The front-to-rear torque vectoring mode takes advantage of the interconnection between the longitudinal and lateral tire forces, where changing one force will automatically influence the other. For instance, an understeered vehicle can be controlled by transmitting more of the engine torque to the rear axle than the front axle. This strategy not only increases the lateral force potential at the front axle and, therefore, enhances the ability of the driver to steer the vehicle, but it also reduces the lateral force potential at the rear axle by

increasing the longitudinal traction force. The lateral force difference between the two axles generates the required corrective yaw moment about the vertical axis of the vehicle, helping the vehicle to travel in the direction in which it is pointed (Figure 5-a). Conversely, an oversteered vehicle can be controlled by transmitting more of the available engine torque to the front axle, thereby increasing the essential lateral force potential on the rear axle. At the same time, this strategy will reduce the lateral force potential on the front axle by elevating its longitudinal force, thus helping the vehicle to develop more understeering behavior (Figure 5-b).

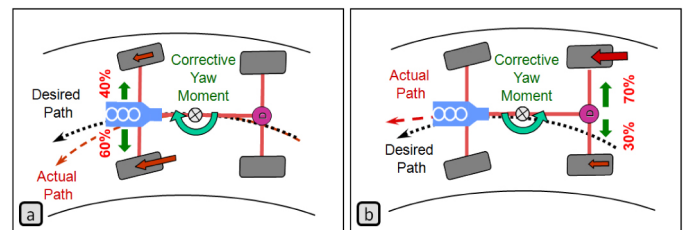


Figure 4. (a) Stability control of an oversteered vehicle through left-to-right torque vectoring on the front axle, and (b) stability control of an understeered vehicle through left-to-right torque vectoring on the rear axle

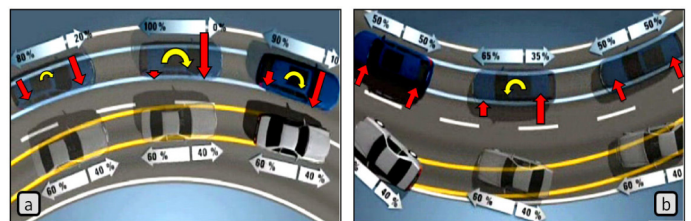


Figure 5. Front-to-rear torque vectoring (a) in an understeered vehicle, and (b) in an oversteered vehicle [8]

Although the theory behind the torque vectoring technique is similar to that used for an electronic stability control (ESC) system, torque vectoring is more effective, especially at generating a corrective yaw moment at higher

vehicle speeds and during emergency maneuvers near the handling limits of the vehicle [5]. In general, torque vectoring can affect the vehicle driving dynamics and traction while causing almost no change in the total driving force of the vehicle. Active braking, on the other hand, is hampered by a net braking effect and drive torque reduction, which not only increases the inefficiency of the vehicle by eliminating power that has already been produced, but it has also been reported by drivers as being disruptive due to the unexpected speed reduction. Moreover, in contrast to active braking, which is allowed only a limited operation time to ensure a safe reserve of fade-free braking performance, torque vectoring can be employed much more actively to enhance driving dynamics and vehicle traction even in normal driving conditions to improve the everyday driving experience. However, it is important to note that a torque vectoring system can be effective only when a driving torque exists in the first place. In other words, torque vectoring is not able to intervene when the driver releases the throttle or brakes the vehicle. In such cases, an ESC system must maintain the stability of the vehicle. Therefore, torque vectoring and active braking should be considered to be complementary technologies whose full potential can be realized only if a holistic integrated approach is used to operate both systems under a common supervisory controller.

In summary, to control the vehicle traction and driving dynamics during both braking and acceleration maneuvers, the vehicle should be equipped with an antilock braking system (ABS), a traction control system (TCS), ESC, and torque vectoring control systems, and all of these systems should be networked together in an integrated fashion. However, equipping a vehicle with all the aforementioned control systems and actuators is a very expensive and complex task. It is for this reason that such a degree of vehicle control is currently available in only luxury-class vehicles, where the customer is prepared to pay for these technologies.

CONTROL METHOD FOR LEFT-TO-RIGHT TORQUE VECTORING DISTRIBUTION

The objective of the advanced torque vectoring system is to distribute the calculated corrective yaw moment from the high-level yaw moment controller to the individual in-wheel motors in order to stabilize the vehicle driving dynamics. Assuming that M_z is the total required corrective yaw moment that is calculated by the genetic fuzzy yaw moment controller, the tire forces on each axle must be adjusted such that each axle generates a portion of the total corrective yaw moment:

$$M_z = M_{z,front} + M_{z,rear} = \theta_f \cdot M_z + \theta_r \cdot M_z \quad (1)$$

where $M_{z,front}$ and $M_{z,rear}$ are the portions of the required corrective yaw moment that must be generated at the front

and rear axles, respectively. In other words, θ_f and θ_r are the percentages of the total required corrective yaw moment M_z that must be generated at the front and rear axles, such that $\theta_f + \theta_r = 100\%$. The relationship between θ_f and θ_r defines the front-to-rear torque vectoring distribution, and will be discussed below.

Generating a couple (equal and opposite traction forces) on each axle is the best strategy for creating the required corrective yaw moment for two reasons: first, all tires participate in generating the required corrective yaw moment; second, the vehicle speed will not be influenced by the activities of the torque vectoring system as they would in an ESC system, provided the forces on the sides of each axle can be generated without being restricted by the traction potential of the tires or the performance of the in-wheel motors. In addition, since all four tires are involved in generating the corrective yaw moment, and since each tire can be accelerated or braked independently, this strategy is more efficient than conventional ESC and torque vectoring systems, where only some tires are involved in generating the corrective yaw moment. Figure 6 illustrates the proposed torque vectoring strategy, where a couple is generated on each axle. The generated yaw moment on each axle is calculated as follows:

$$\left. \begin{aligned} M_{z,front} &= F_{x,FR} \cdot \frac{t_f}{2} + F_{x,FL} \cdot \frac{t_f}{2} \\ F_{x,FR} &= F_{x,FL} = F_{x,F} \end{aligned} \right\} \Rightarrow$$

$$M_{z,front} = F_{x,F} \cdot t_f \Rightarrow F_{x,F} = \frac{M_{z,front}}{t_f}$$

$$(2)$$

$$\left. \begin{aligned} M_{z,rear} &= F_{x,RR} \cdot \frac{t_r}{2} + F_{x,RL} \cdot \frac{t_r}{2} \\ F_{x,RR} &= F_{x,RL} = F_{x,R} \end{aligned} \right\} \Rightarrow$$

$$M_{z,rear} = F_{x,R} \cdot t_r \Rightarrow F_{x,R} = \frac{M_{z,rear}}{t_r}$$

$$(3)$$

where $F_{x,FR}$, $F_{x,FL}$, $F_{x,RR}$, and $F_{x,RL}$ are the longitudinal tire forces of the front-right, front-left, rear-right, and rear-left wheels, respectively; t_f and t_r are the front and rear wheel track widths.

In order to calculate the required motor torques, a torque balance is formed for each wheel. Figure 7 illustrates the model of a single wheel, where $I_{yy,w}$ denotes the moment of inertia of the wheel about its spin axis, T_m denotes the motor torque, r_{dyn} is the tire dynamic radius, ω_w is the angular velocity of the tire, F_z is the tire vertical force, and F_x is the tire longitudinal force. Note that the tire rolling resistance and the aerodynamic drag of the vehicle are neglected in this

model for simplicity. By specifying the equation of motion of the wheel, the traction force at the tire-road contact patch can be estimated as follows:

$$T_m - F_x \cdot r_{dyn} = I_{yy,w} \cdot \dot{\omega}_w \Leftrightarrow F_x = \frac{1}{r_{dyn}} \cdot (T_m - I_{yy,w} \cdot \dot{\omega}_w) \quad (4)$$

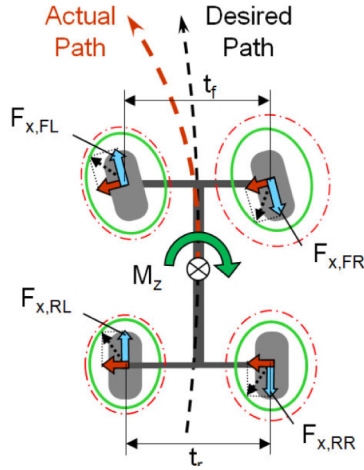


Figure 6. Advanced torque vectoring control strategy using couple generation on each axle (the dash-dotted ellipse surrounding each tire indicates the adhesion potential of that tire; the solid ellipse indicates the actual friction ellipse)

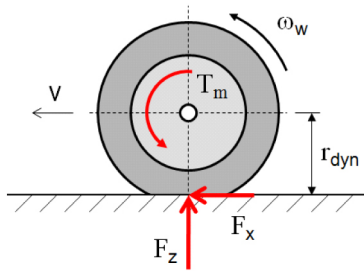


Figure 7. Torque balance at the tire-road contact patch

Substituting the traction force F_x from equation (4) into equations (2) and (3), the required motor torque at each wheel can be calculated as follows:

$$T_{m,i} = \frac{r_{dyn,i}}{t_f} \cdot M_{z,front} + I_{yy,w} \cdot \dot{\omega}_{w,i} \quad (5)$$

where $i \in \{FR, FL\}$

$$T_{m,i} = \frac{r_{dyn,i}}{t_r} \cdot M_{z,rear} + I_{yy,w} \cdot \dot{\omega}_{w,i} \quad (6)$$

where $i \in \{RR, RL\}$

Note that when the required corrective yaw moment is positive, the wheels on the right side of the vehicle must be driven and the wheels on the left side must be braked; when the required corrective yaw moment is negative, the wheels on the right side of the vehicle must be braked and those on the left side must be driven.

CALCULATION OF TIRE ADHESION POTENTIAL

To prevent the tires from spinning out or locking up during the couple generation, the maximum possible traction force of each tire is estimated at each time step of the simulation, and is used as a limit in the controller. In addition, the adhesion potential of each tire is calculated and used to define the extent to which the tire forces have been saturated. Estimating the adhesion potential requires information about the horizontal and vertical forces acting on the tire, as well as the friction coefficient between the tire and the road, the estimation of which is discussed below.

The longitudinal and lateral tire forces are estimated using the well-known “Magic Tire Formula” [4, 5]. This model is a semi-empirical set of curve fits that takes into account the coupling between the longitudinal and lateral tire forces through combined-slip characteristics, a limited tire adhesion potential, the variation in cornering stiffness with tire load, and the influence of the tire-road friction coefficient. The combined-slip horizontal tire forces are estimated using the following equations [12]:

$$F_{x,i} = F_{xo,i} \cdot \cos\left(C \cdot \arctan\left(B \cdot \alpha_i - E \cdot (B \cdot \alpha_i - \arctan(B \cdot \alpha_i))\right)\right) \quad (7)$$

$$F_{y,i} = F_{yo,i} \cdot \cos\left(C \cdot \arctan\left(B \cdot \lambda_i - E \cdot (B \cdot \lambda_i - \arctan(B \cdot \lambda_i))\right)\right) \quad (8)$$

where $i \in \{FL, FR, RL, RR\}$. Note that these forces are calculated with respect to the wheel coordinate system. $F_{xo,i}$ and $F_{yo,i}$ represent the pure-slip tire forces in the longitudinal and lateral directions, respectively, and are calculated as follows [12]:

$$F_{xo,i} = D \cdot \sin\left(C \cdot \arctan\left(B \cdot \lambda_i - E \cdot (B \cdot \lambda_i - \arctan(B \cdot \lambda_i))\right)\right) \quad (9)$$

$$F_{yo,i} = D \cdot \sin\left(C \cdot \arctan\left(B \cdot \alpha_i - E \cdot (B \cdot \alpha_i - \arctan(B \cdot \alpha_i))\right)\right) \quad (10)$$

In this tire model, the interdependence between the longitudinal and lateral tire forces is considered, where peak factors D , shape factors C , stiffness factors B , and curvature factors E are different for equations (7),(8),(9),(10), and for the longitudinal and lateral directions [12]. All the parameters required by this model are taken from the Pacejka 2002 tire data obtained for a 175/55 R15 tire. In addition, the slip ratio (λ) of each tire is calculated using equations (1) and (2) in [9].

and the sideslip angle (α) of each tire is approximated using a bicycle model, as indicated below [14]:

$$\alpha_F = \delta + \beta - \frac{a \cdot \dot{\psi}}{v_x} \quad (11)$$

$$\alpha_R = \beta + \frac{b \cdot \dot{\psi}}{v_x} \quad (12)$$

where δ is the steering angle, β is the sideslip angle of the vehicle, a and b are the distances of the front and rear axles from the vehicle center of gravity, v_x is the vehicle forward speed, and $\dot{\psi}$ is the vehicle yaw rate.

Approximating the longitudinal and lateral tire forces using equations (7) and (8), the tire adhesion potential utilization η can be estimated using the following elliptic relation:

$$\eta_i = \sqrt{\left(\frac{F_{xi}}{F_{xi,max}}\right)^2 + \left(\frac{F_{yi}}{F_{yi,max}}\right)^2}, \quad 0 \leq \eta_i \leq 1 \text{ and } i \in \{FL, FR, RL, RR\} \quad (13)$$

where $F_{xi,max}$ and $F_{yi,max}$ represent the maximum possible forces in the longitudinal and lateral directions, respectively. With the nominal tire load F_{z0} and the tire-road friction coefficients μ_i , the maximum longitudinal and lateral tire forces can be approximated as follows [13]:

$$F_{xi,max} = \mu_i \cdot F_{zi} \cdot k_{x,i} \cdot \left(1 + l_x \cdot \frac{F_{zi} - F_{z0}}{F_{z0}}\right) \quad (14)$$

$$F_{yi,max} = \mu_i \cdot F_{zi} \cdot k_{y,i} \cdot \left(1 + l_y \cdot \frac{F_{zi} - F_{z0}}{F_{z0}}\right) \quad (15)$$

where $k_{x,i}$ and $k_{y,i}$ depend on the actual tire camber angle, and parameters l_x and l_y , which define the degressive behavior of the tire horizontal forces [12], are set to 1. For the sake of simplicity, the camber angles in the ADAMS/View model of the AUTO21EV are assumed to be zero and constant throughout the simulation.

The actual vertical force applied to each tire (F_{zi}) is approximated by neglecting the coupling between the vehicle roll and pitch motions, and by disregarding the suspension dynamics. To this end, two half-car models are defined, one for the longitudinal direction and one for the lateral direction. Figure 8-a illustrates a half-car model in the longitudinal direction, where a_x indicates the longitudinal acceleration of

the vehicle. The inertial force due to the longitudinal acceleration at the vehicle center of gravity (CG) causes a weight shift to the rear axle, which simultaneously reduces the front axle load and increases the rear axle load. Constructing the torque balance at the rear axle contact point yields the following expression for the front axle load ($F_{z,F}$) [15]:

$$\begin{aligned} \sum M_y = 0 &\Rightarrow \\ (a+b) \cdot F_{z,F} - m_{CG} \cdot g \cdot b + m_{CG} \cdot a_x \cdot h_{CG} &= 0 \Rightarrow \\ F_{z,F} &= \frac{b}{a+b} m_{CG} \cdot g - \frac{h_{CG}}{a+b} m_{CG} \cdot a_x \end{aligned} \quad (16)$$

where a and b are the distances of the front and rear axles from the vehicle center of gravity, m_{CG} is the total mass of the vehicle, h_{CG} is the height of the vehicle center of gravity, and g is the gravitational acceleration constant. a_x is the longitudinal acceleration of the vehicle, and can be measured using an accelerometer. Subtracting the front axle load ($F_{z,F}$) from the total vehicle weight, the rear axle load ($F_{z,R}$) is calculated as follows:

$$\begin{aligned} \sum F_z = 0 &\Rightarrow F_{z,F} + F_{z,R} - m_{CG} \cdot g = 0 \Rightarrow \\ F_{z,R} &= \frac{a}{a+b} m_{CG} \cdot g + \frac{h_{CG}}{a+b} m_{CG} \cdot a_x \end{aligned} \quad (17)$$

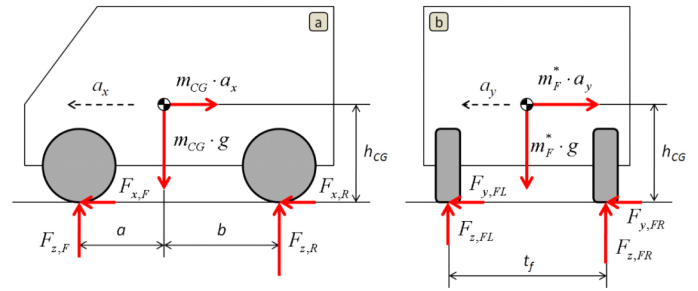


Figure 8. (a) Longitudinal weight shift during acceleration, and (b) lateral weight shift during cornering

During cornering, the lateral acceleration causes a weight shift to one side of each axle, whose distribution between the front and rear axles depends on the axle loads. The two axles are considered to be decoupled from each other, and a half-car model is used to calculate the lateral weight shift on each axle. For instance, Figure 8-b illustrates the half-car model for the front axle, where the virtual mass of the front axle (m_F^*) is calculated as follows: