

Development of an Integrated Control Strategy Consisting of an Advanced Torque Vectoring Controller and a Genetic Fuzzy Active Steering Controller

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ABSTRACT

The optimum driving dynamics can be achieved only when the tire forces on all four wheels and in all three coordinate directions are monitored and controlled precisely. This advanced level of control is possible only when a vehicle is equipped with several active chassis control systems that are networked together in an integrated fashion. To investigate such capabilities, an electric vehicle model has been developed with four direct-drive in-wheel motors and an active steering system. Using this vehicle model, an advanced slip control system, an advanced torque vectoring controller, and a genetic fuzzy active steering controller have been developed previously. This paper investigates whether the integration of these stability control systems enhances the performance of the vehicle in terms of handling, stability, path-following, and longitudinal dynamics. An integrated approach is introduced that distributes the required control effort between the in-wheel motors and the active steering system. Several test maneuvers are simulated to demonstrate the performance and effectiveness of the integrated control approach, and the results are compared to those obtained using each controller individually. Finally, the integrated controller is implemented in a hardware- and operator-in-the-loop driving simulator to further evaluate its effectiveness.

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INTRODUCTION

Over the last two decades, advances in electronics have revolutionized many aspects of automobiles, especially in the areas of engine management and vehicle dynamics safety systems such as the anti-lock braking system (ABS), traction control system (TCS), and electronic stability control (ESC) system. In these cases, the signals generated by the brake or accelerator pedal are modulated by an electronic control unit in order to control the tire slip of individual wheels in severe braking (ABS) or acceleration (TCS) situations, or to control the vehicle yaw rate through individual wheel braking (ESC). It is important to note that the U.S. National Highway Traffic Safety Administration (NHTSA) has passed federal legislation making the installation of ESC mandatory on all

passenger cars, multipurpose passenger vehicles, trucks, and buses [1]. The move to improve the safety, comfort, and performance of vehicles has led to an increase in the use of electronic control systems and the introduction of drive-by-wire systems. Today, the value added to the modern vehicle by electronic systems is approximately 20 percent. In luxury vehicles, for example, more than 90 control systems are used to control a variety of actuators. It is expected that this rate will consistently increase, reaching over 40 percent by 2015 [2].

Integrating various electronic control systems offers the potential to optimize driving behavior independently of the driving maneuver through the individual control and allocation of traction, steering, and braking forces. These unique features create new opportunities for controlling the

driving dynamics of a vehicle in ways that were not possible in the past. For example, integrating the active braking and active steering systems can avoid the vehicle side-pushing behavior when braking on a μ -split road [3]. In addition, by integrating all chassis control systems, sensor signals can be broadcast across the entire network of controllers, thereby avoiding sensor redundancy and reducing costs.

Recently, electric vehicles (EVs) have attracted a great deal of interest as an elegant solution to environmental and energy problems. EVs have no tailpipe emissions because they have no fuel, combustion, or exhaust systems, and they have achieved driving performance and efficiency metrics that are comparable to those of conventional internal combustion engine vehicles. Critics proclaim that EVs are simply “elsewhere emission vehicles” because they transfer emissions from the tailpipe to the smokestack. However, in the ideal scenario, EVs would be charged using renewable energy sources such as hydro, solar, or wind.

Advanced Stability Control System through Networked Chassis

Until the 1980s, chassis technology (which directly determines the dynamic performance of a vehicle) advanced exclusively within the mechanical engineering framework. Following the mid-1980s invention and practical application of the four-wheel-steering (4WS) system, the field of vehicle dynamics performance became a main stream of research and development for control technology. Since then, research and development of vehicle dynamics performance has been carried out as a collaboration between mechanical engineering and control engineering.

Current chassis control systems are distinguished by the way the individual subsystems work. While some subsystems affect only one dynamic domain (i.e., longitudinal, lateral, or vertical dynamics), many subsystems can influence two of the three domains, as illustrated in Figure 1. For instance, a torque vectoring system can influence both the longitudinal and lateral dynamics of a vehicle. Furthermore, the influences of individual subsystems are tightly coupled through the tire-road characteristics, especially during extreme maneuvers. Thus, the optimum driving dynamics can be achieved only when the tire forces on all four wheels and in all three coordinate directions can be influenced and controlled precisely. In order to achieve this level of control, the vehicle must not only be equipped with various active chassis subsystems, but these subsystems must be networked together, cooperatively controlling the tire forces to meet the driving dynamics, safety, and comfort requirements [4].

Taking advantage of the strengths of each subsystem, the ideal stability control system can be obtained by activating the most effective subsystem or subsystems based on the driving maneuver and road conditions. For example, an ESC system uses braking intervention of individual wheels in order to influence the longitudinal forces on the tires and, ultimately, the yaw behavior of the vehicle. However, in

many situations, a vehicle can be stabilized without applying the brakes. For example, if the driver wishes to maintain a constant vehicle speed while negotiating a curve, torque vectoring techniques can be used to control the driving dynamics without slowing the vehicle.

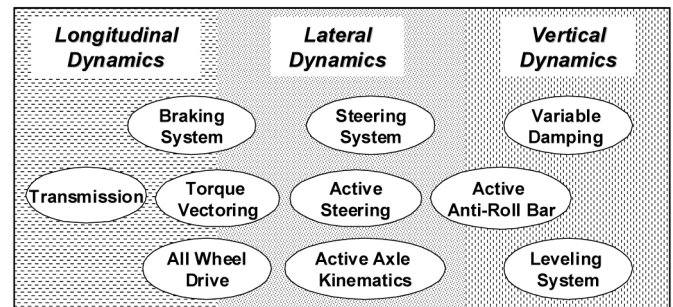


Figure 1. Domains affected by driveline and chassis subsystems [4]

Figure 2 illustrates the interdependencies among longitudinal, lateral, and vertical dynamics. As indicated, a friction ellipse couples the longitudinal and lateral tire forces, a “longitudinal friction coefficient versus slip ratio” relationship couples the longitudinal and vertical tire forces, and a “lateral friction coefficient versus slip angle” relationship couples the vertical and lateral forces of a tire. Thus, most vehicle control principles can be related to the linear and nonlinear characteristics of the tire-road contacts. The factors influencing the nature of this contact can be summarized as direct effects of the steering angle, slip angle, and camber angle of the tires on lateral forces, and direct effects of traction and braking intervention on longitudinal forces. The wheel vertical load, however, influences both longitudinal and lateral forces directly by defining the maximum possible adhesion potential.

The fundamental question of which configurations are both effective and feasible given a specific set of driving conditions can be answered only if the strengths and limitations of each active chassis subsystem have been identified. A popular method of addressing such issues is to analyze the effects of each subsystem on the resulting tire-road friction ellipse or circle. Since the driver is limited by the friction constraints of the tires, the vehicle controls are expected to provide the driver with predictable authority over longitudinal and lateral accelerations, within the physical constraints of the vehicle “friction circle”, and subject to perceived customer acceptability of the frequency- and amplitude-dependence of the vehicle responses [5]. Therefore, given that friction limits change with speed, road surface conditions, and so forth, these vehicle control systems are required to provide adequate feedback of such changes. The illustrations shown in Figure 3, presented by Toyota [6], indicate the domain of operation of some typical vehicle control systems. Although the diagrams should not be taken too literally, they clearly underline the fact that integrated control can enlarge the dynamic response domain of the

vehicle by taking advantage of the control system with the most effectiveness for a particular driving maneuver and set of road conditions.

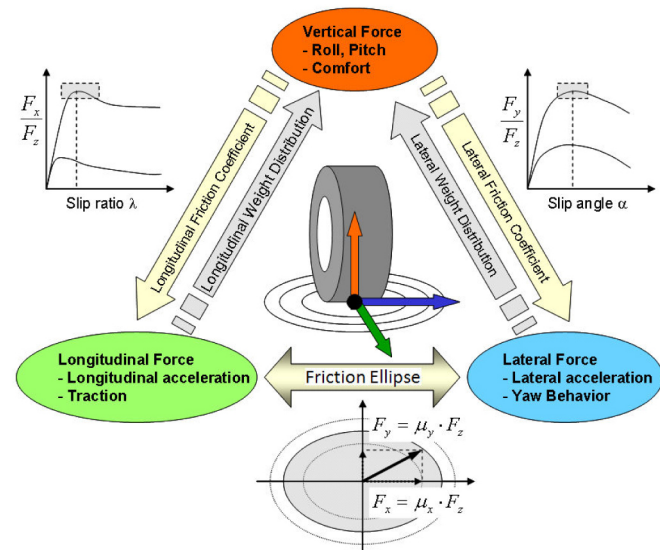


Figure 2. Interdependencies among longitudinal, lateral, and vertical dynamics

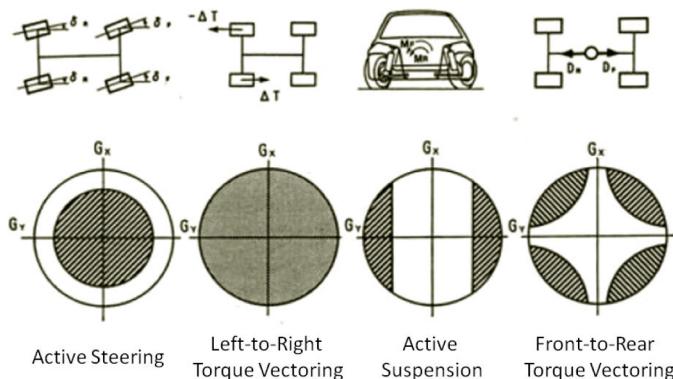


Figure 3. Effective range of various control systems based on the resulting tire-road friction circle [6]

The majority of the stability control systems currently on the market stabilize the vehicle in critical driving situations by intervening with only one type of active chassis subsystem at a time, which limits the performance of the vehicle. Recently, there has been a move towards networking the individual subsystems in order to take advantage of synergies and increase the performance of the vehicle. Until very recently, however, mainly due to marketing strategies, chassis subsystems have been treated as stand-alone systems in a so-called “coexistence” architecture, which requires no overhead but still suffers from suboptimal performance. This architecture can lead to a situation in which, for instance, a vehicle with four active chassis subsystems is equipped with as many as four independent sets of sensors, state estimators, reference models, and state controllers [3, 4]. Since the

simultaneous actuation of these subsystems may affect the same degrees-of-freedom of the vehicle and have counterproductive results, the overlapping of actuator effort must be addressed in a more coordinated way. One solution to this problem is a “hierarchical coexistence” of the subsystems with a unidirectional flow of information, where one system acts independently and the others adapt as necessary.

An “integrated” approach is a more sophisticated means of addressing the coordination of several actuators [5, 7]. In this approach, each chassis subsystem has one basic function. In contrast to the coexistence approach, there is only one set of sensors, one state estimator, one reference model, and one state controller. Based on the desired and actual behavior of the vehicle, the system can calculate the required generalized forces and moments to maintain the course desired by the driver. These generalized forces and moments are then applied by the actuators based on their effectiveness, ensuring the best overall safety, ride quality, and driving pleasure.

DEVELOPMENT OF AN INTEGRATED CHASSIS CONTROL SYSTEM

A two-passenger, all-wheel-drive urban electric vehicle (AUTO21EV) with four direct-drive in-wheel motors and an active steering system has been designed and developed at the University of Waterloo. The AUTO21EV is modeled in both ADAMS/View and MapleSim, and is an exciting platform on which to apply advanced motion control techniques. Figure 4 illustrates the AUTO21EV, which has a similar configuration to the commercially-available Smart fortwo.

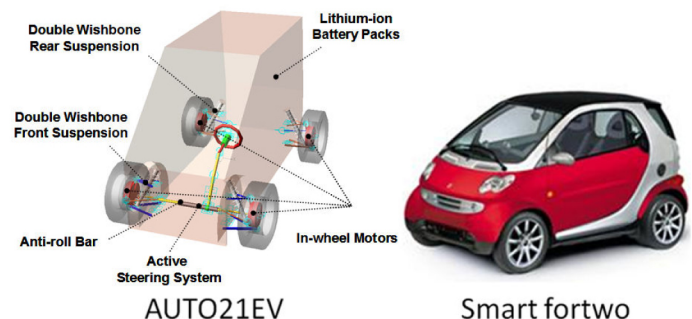


Figure 4. AUTO21EV concept vehicle (left) and Smart fortwo [8] (right)

Table 1 lists some of the relevant parameters used for the AUTO21EV model. Based on this vehicle model, an advanced fuzzy slip control system [9] was developed and tested using some predefined test maneuvers and a novel path-following and speed-controlling driver model [10]. A genetic fuzzy yaw moment controller was also developed [11], the objective of which was to determine the corrective yaw moment required to minimize the vehicle yaw rate and sideslip errors. The genetically-tuned fuzzy yaw moment

controller acts as a high-level supervisory module that assigns tasks to a lower-level torque vectoring controller [12], which then 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. Moreover, a genetic fuzzy active steering controller was developed using the AUTO21EV vehicle model [13], and the performance and effectiveness of this controller were studied using a driving simulator. In this work, we investigate whether the integration of these stability control systems enhances the performance of the vehicle in terms of handling, stability, path-following, and longitudinal dynamics. An integrated approach is introduced that distributes the required control effort between the in-wheel motors and the active steering system.

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

INTEGRATION OF THE ADVANCED TORQUE VECTORING AND GENETIC FUZZY ACTIVE STEERING CONTROLLERS

As mentioned above, the optimum driving dynamics can be achieved only when the tire forces on all four wheels and in all three coordinate directions are monitored and controlled precisely. This advanced level of control is possible only when the vehicle is equipped with several active chassis control systems that are networked together in an integrated fashion. Taking advantage of the strengths of each active chassis subsystem, the ideal traction and stability performance of the vehicle can be obtained by activating the subsystem or subsystems that will be most effective given the deviation between the desired and actual behavior of the vehicle. The performance and effectiveness of the advanced torque vectoring controller (ATVC) and the genetic fuzzy active steering controller (GFASC) were studied and evaluated individually in previous papers [12, 13]. In this work, we investigate whether the integration of these stability

control systems enhances the performance of the vehicle in terms of handling, stability, path-following, and longitudinal dynamics. An integrated approach is introduced that distributes the required control effort between the in-wheel motors and the active steering system.

The GFASC developed in [13] has confirmed that superimposing the steering input provided by the driver with a correction generated by the active steering system is considered to be a continuous process, and is not perceived by the driver as being disruptive. It is also advantageous to employ steering intervention rather than braking or driving individual wheels when controlling the vehicle on slippery surfaces, since steering intervention requires less frictional force between the tire and the road to generate a corrective yaw moment. However, the GFASC is not of substantial help when the vehicle is driven near its handling limits due to its limited range of effectiveness (caused by actuator restrictions). In [12], on the other hand, it has been confirmed that the ATVC is very effective at improving vehicle stability and handling, even when driving the vehicle near its handling limits. It has also been observed, however, that using the in-wheel motors to generate a corrective yaw moment can cause oscillations in the vehicle states, which might be perceived by the driver as being annoying. Thus, the GFASC and ATVC appear to be complementary.

In this work, the integration of the ATVC and GFASC is realized by using the activation function illustrated in Figure 5. This activation function is a standard Gaussian curve generated using the following exponential function:

$$\chi_{ATVC}(\delta_{corr}) = \begin{cases} \exp\left[-\frac{(\delta_{corr} - \delta_{corr,max})^2}{2\sigma^2}\right] \cdot 100\% & , |\delta_{corr}| \leq 3^\circ \\ 100\% & , |\delta_{corr}| > 3^\circ \end{cases} \quad (1)$$

where $\chi_{ATVC}(\delta_{corr})$ is the ATVC activation function, which is defined as a function of the corrective steering angle (δ_{corr}); $\delta_{corr,max}$ is the actuator range limit of the active steering controller, which is set to 3° [13]; and σ is the standard deviation, which is set to 0.7 in order to form the bell curve shown in Figure 5. Note that the shape of this activation function is designed such that the contribution of the ATVC is introduced gradually rather than abruptly. In other words, depending on the driving conditions and the difference between the desired and actual behavior of the vehicle, the GFASC first attempts to stabilize the vehicle without receiving any support from the ATVC (since the ATVC can have a negative effect on driving enjoyment). As the required corrective steering angle increases, the activation function gradually activates the ATVC to support the GFASC in its effort to stabilize the vehicle. If the required corrective steering angle is larger than 3° , thus exceeding the actuator range limit of the active steering controller, the activation function fully activates the ATVC and both controllers are fully deployed to help stabilize the vehicle. In critical or

emergency driving situations, it is acceptable to sacrifice some amount of passenger comfort to maintain vehicle stability and, ultimately, passenger safety.

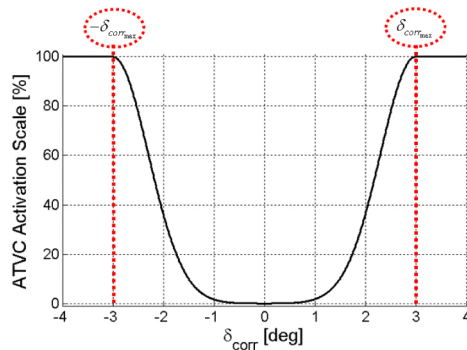


Figure 5. Activation function used for the integration of the ATVC and GFASC

TEST MANEUVERS FOR EVALUATING THE EFFECTIVENESS OF THE INTEGRATED CONTROL STRATEGY

Many test maneuvers have been developed for evaluating the quality of the handling and performance characteristics of a vehicle. Many of these test maneuvers are based on ideal driving conditions, and some of them are motivated by the examination methods typically used for control systems, such as step-steer and swept-sine-steer maneuvers. An extensive overview of different test maneuvers and their detailed descriptions can be found in the publications of Roenitz, Braess, and Zomotor [14, 15]. A comprehensive evaluation of the dynamic characteristics of a vehicle and the effectiveness of different chassis control systems can be achieved only when the results obtained from a variety of test maneuvers are combined and evaluated as a whole. Several test maneuvers have been used in the past to provide important information about different aspects of the dynamic behavior of the vehicle and the effectiveness of each individual chassis control system, namely the advanced slip control system [9], the advanced torque vectoring controller [12], and the genetic fuzzy active steering controller [13]. In this work, we use the same test maneuvers to evaluate the effectiveness of our integrated chassis control strategy. These test maneuvers are chosen such that all aspects of vehicle dynamics are addressed and act to quantify the advantages of each control method. A comprehensive description of the chosen test maneuvers and the specific requirements for a desirable vehicle response during each maneuver can be found in [16]. In order to evaluate the performance and effectiveness of the integrated control strategy, the AUTO21EV is driven through four test maneuvers.

ISO Double-Lane-Change Maneuver

The performance of the integrated control system consisting of the ATVC and the GFASC is first evaluated by driving the AUTO21EV through the ISO 3888 double-lane-change maneuver [17] with an initial speed of 75 km/h and using the path-following driver model developed previously [10]. Figure 6 illustrates the vehicle trajectory and demonstrates that the driver is able to negotiate the maneuver without striking the cones when the integrated control strategy is used. Note that the path-following driver model is not able to negotiate this maneuver at higher speeds without striking the cones when only the GFASC is active [13].

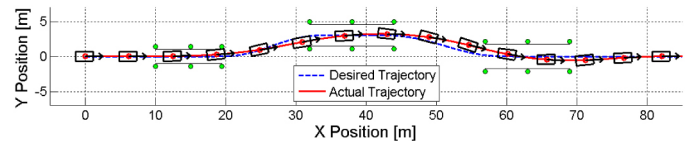


Figure 6. Desired and actual vehicle trajectories when driving through the double-lane-change maneuver with an initial speed of 75 km/h using the path-following driver model and the integrated control of the ATVC and GFASC

Figure 7 illustrates the vehicle yaw rate and sideslip angle during this maneuver. In contrast to the individual performance of the ATVC and GFASC, the integrated control approach using both controllers is able to match the actual vehicle yaw rate with the desired yaw rate (which is calculated using a reference bicycle model). Note that the oscillations in the actual vehicle yaw rate that were observed when the ATVC was used on its own [12] are not present when the integrated control approach is used. Figure 8-a illustrates the vehicle lateral acceleration as a function of time. As can be seen, the lateral acceleration when using the integrated control approach is, in some regions, similar to that observed when using the GFASC; in other regions, it is similar to that observed when using the ATVC. At the handling limits, oscillations can be seen in the lateral acceleration of the vehicle, which are caused by the activation of the in-wheel motors, but they are mostly damped out. Figure 8-b confirms that, except during the second lane change, the driver requires about the same amount of steering wheel input as is the case when only the ATVC is used.

Figure 8-c illustrates the vehicle speed during the double-lane-change maneuver, and confirms that the vehicle loses the least amount of speed when the integrated control approach is used. This lack of deceleration can be attributed to the fact that only relatively small steering angles are needed to negotiate the maneuver (Figure 8-b) and the fact that the couples generated at the front and rear axles do not slow the vehicle. As a result, the speed reduction during this maneuver when using the integrated control approach is even smaller than that observed when the genetic fuzzy yaw moment controller (GFYMC) is active [11]. Figure 8-d illustrates the handling performance of the vehicle and

indicates that the hysteresis of this plot is less than that of the analogous plots for the GFASC and the ATVC. In other words, the responsiveness and agility of the vehicle are considerably improved compared to the cases where only individual controllers (namely, the GFASC and the ATVC) are active. However, the responsiveness and agility of the vehicle are not as good as they are when the ideal (but not directly realizable) GFYMC is used to apply the required corrective yaw moment directly to the vehicle center of gravity.

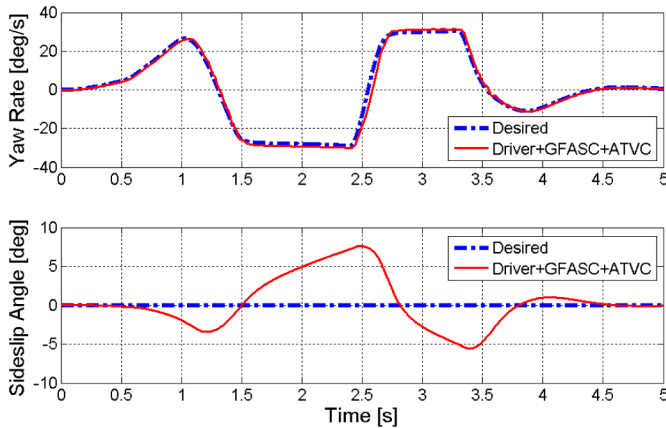


Figure 7. Desired and actual vehicle yaw rates (top) and sideslip angles (bottom) when driving through the double-lane-change maneuver with an initial speed of 75 km/h using the driver model and the integrated control of the ATVC and GFASC

Figure 9 illustrates the torque of each in-wheel motor during the double-lane-change maneuver. Comparing this figure to Figure 12 in [12], it is clear that the use of the ATVC has been reduced to a minimum by the activation function. In other words, the ATVC is activated only when the GFASC is unable to stabilize the vehicle on its own, which generally occurs only when the vehicle is driven near its handling limits. Figure 12 indicates that the GFASC is able to stabilize the vehicle most of the time, and the ATVC is only activated at three time periods. For instance, between 2.4 and 2.65 seconds of the simulation, during which time the vehicle experiences the maximum lateral acceleration of 8.5 m/s^2 (Figure 8-a), the left-to-right torque vectoring controller has ordered the left wheels to brake and the right wheels to accelerate, thereby generating a positive corrective yaw moment to compensate for an oversteering situation, where the actual yaw rate is larger than the desired one (Figure 7).

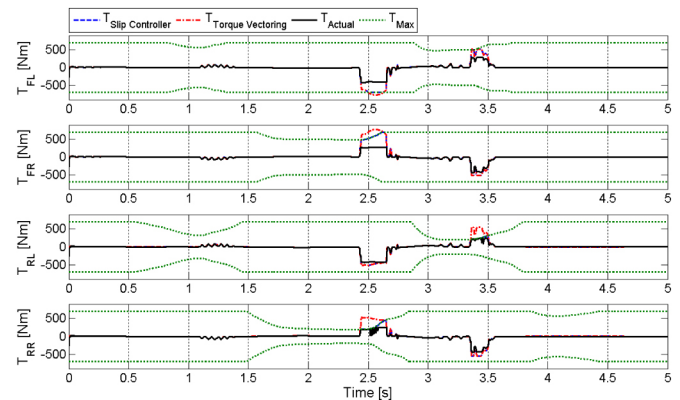


Figure 9. Requested and actual motor torques at each wheel when driving through the double-lane-change maneuver using the driver model with the integrated control of the ATVC and GFASC

From the torque vectoring ratios shown in Figure 10, it can be confirmed that the front-to-rear torque vectoring controller supports both the GFASC and the left-to-right torque vectoring controller in a coordinated effort to stabilize the vehicle. For instance, between 2.4 and 2.65 seconds, the front-to-rear torque vectoring controller has requested that the front motors generate up to 61% of the total required corrective yaw moment. Note that generating more torque with the front motors and less with the rear ones reduces the lateral force potential at the front axle and increases that at the rear axle. The asymmetric lateral force potentials on the front and rear axles help to generate the required corrective yaw moment. The activation of the front-to-rear torque vectoring controller can also be confirmed in Figure 9. For instance, between 2.4 and 2.65 seconds, the front-to-rear torque vectoring controller has requested the front-left motor to generate up to 770 Nm of braking torque and the front-right motor to generate up to 770 Nm of driving torque while, at the same time, the controller has requested up to 510 Nm of braking and driving torques from the rear-left and rear-

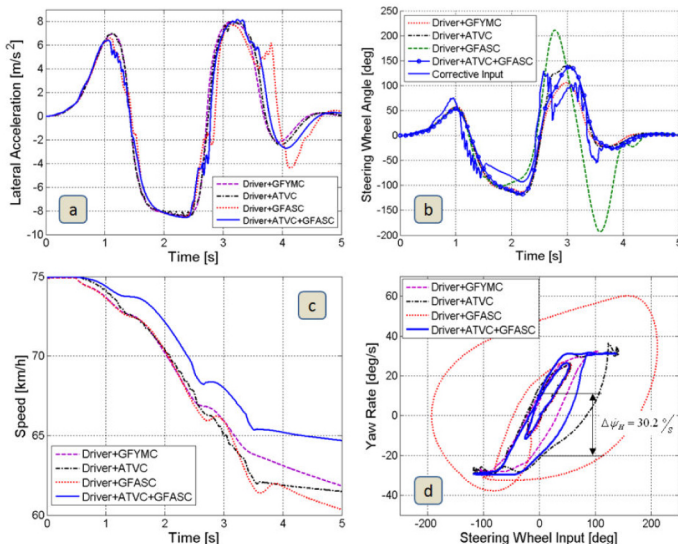


Figure 8. (a) Lateral acceleration, (b) steering wheel angle, and (c) vehicle speed as functions of time; and (d) vehicle yaw rate as a function of the steering wheel input when driving through the double-lane-change maneuver with the GFYMC, the ATVC, the GFASC, and the integrated control of the ATVC and GFASC