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ABSTRACT

This paper focuses on modeling of torque-biasing devices of a four-wheel-drive system used for improving vehicle stability and handling performance. The proposed driveline system is based on nominal front-wheel-drive operation with on-demand transfer of torque to the rear. The torque biasing components of the system are an electronically controlled center coupler and a rear electronically controlled limited slip differential. Kinematic modeling of the torque biasing devices is introduced including stage transitions during the locking stage and the unlocking/slipping stage. Analytical proofs of how torque biasing could be used to influence vehicle yaw dynamics are also included in the paper. A yaw control methodology utilizing the biasing devices is proposed. Finally, co-simulation results with Matlab®/Simulink® and CarSim® show the effectiveness of the torque biasing system in achieving yaw stability control.

INTRODUCTION

Vehicle stability control systems are being increasingly used in the automotive industry and are becoming standard equipment in many vehicles. Most vehicle stability control systems in the market are brake-based. Brake-based stability control systems use Anti-Lock Braking System (ABS) hardware to apply individual wheel braking forces in order to correct yaw dynamics. While the use of such brake-based system is acceptable in many situations, it can deteriorate longitudinal performance of the vehicle, especially in situations where the driver desires to accelerate the vehicle [1].

The last two decades have witnessed significant growth in the application of four-wheel-drive (4WD) systems to passenger vehicles [2]. To overcome the limitations of brake-based stability control systems, the use of active torque distribution based stability control in 4WD systems is proposed. Such a system provides vehicle stability control without deteriorating the longitudinal performance of the vehicle.

Recent developments in vehicle yaw control using torque management methodologies include the use of front-back torque control couplers by Nissan V-TCS [3], Haldex LSC [4], BMW xDrive [5], Bosch CCC [6]; the use of limited slip differentials together with on-demand couplings by GKN TMD [7], Dana Dynamic Trak [8], Ricardo [9]; and the use of right-left torque control systems by Honda SH-AWD [10], and Mitsubishi AYC [11][12][13] to enhance cornering performance and stability.

The selected system discussed in this paper is for a 4WD vehicle with the driveline system based on nominal front-wheel-drive operation with on-demand to the rear. The torque-biasing components of the system include an electronically controlled center coupler and a rear electronically controlled limited slip differential. Both torque-biasing devices are widely used and available in the automotive market.

Simulation has become common practice in automotive development because it speeds up the development cycle and decreases costs [14]. Unfortunately, dynamic models of torque-biasing devices are not yet available. Some researchers [15][3] have proposed models of torque distribution but they are useful only for steady-state analysis. Commercial vehicle dynamic software such as CarSim®, ADAMS/Car® or Dymola® only provide steady state look-up tables. Torque-biasing devices are somewhat special because their internal connection structure changes due to the engaging of clutches and brakes. Hence, for vehicle yaw control that involves dynamic response, dynamic modeling of the torque-biasing devices is necessary. In this paper, modeling of the torque-biasing devices is presented. Kinematic modeling of the torque biasing devices is introduced including the dynamics during stage transition of the locking stage and the unlocking/slipping stage.

A yaw control concept using the aforementioned electronically controlled center coupler and rear limited slip differential to enhance the vehicle lateral dynamics while preserving longitudinal motion is also presented in this paper.

To evaluate the effectiveness of the proposed driveline modeling, the torque-biasing models were generated in Matlab®/Simulink® environment. A full vehicle model developed in CarSim was modified so that co-simulation can be performed.

The outline of this paper is as follows. First the proposed system configuration is presented. Second the modeling of the torque-biasing devices is presented. Third, the yaw control methodology is proposed. Finally, co-simulation results with CarSim show the effectiveness of the torque biasing system.

SYSTEM CONFIGURATION

The driveline configuration for vehicle yaw control is shown in Figure 1. It is an on-demand 4WD vehicle. The front axle is primarily and directly driven by the engine. The rear axle is indirectly driven via a power transfer unit (PTU) and a center coupler. The center coupler can be categorized to be one of the following three types: speed sensing passive coupler; torque sensing passive coupler and electronically controlled coupler [7]. An electronically-controlled limited slip differential (ELSD) is used to bias the rear prop-shaft torque to left and right wheels. The clutch torque level can be excited either by hydraulic or electromagnetic systems. Regardless of the mechanical construction of either torque-biasing device, the clutch response time needs to be sufficient to guarantee the effectiveness of the stability control system.

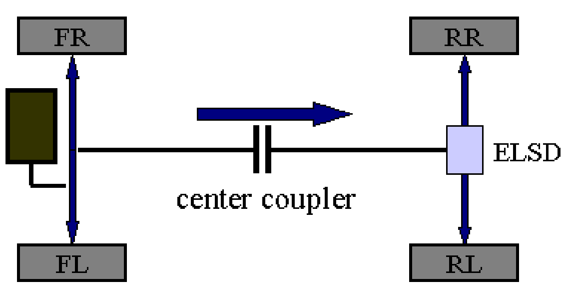


Figure 1. Driveline configuration

An example of such a system is Eaton's active torque-biasing control system. The system consists of a Hydraulic Torque Coupling (HTC-I™) and an electronically controlled limited slip differential (EGerodisc-II™).

The HTC-I shown in Figure 2 is an electronically controlled all wheel drive coupler designed as an integrated component of the vehicles rear drive module. The HTC-I transfers power smoothly and rapidly from the vehicles drive shaft to the hypoid pinion gear of the rear drive module in response to orchestrated signals from an electronic control unit (ECU). Power transfer is provided by an actively controlled wet multi-plate clutch disposed between the coupling shaft and the hypoid pinion gear. Clutch engagement limits the slip between the vehicles driveshaft and the hypoid pinion gear and in doing so torque is transferred from the driveshaft to the hypoid pinion gear, the magnitude of which will be less then or equal to the clutch torque. The HTC-I provides fast coupling torque application and removal as is required for both driveline torque control based vehicle dynamic operation (as is the focus of this paper), and also for compatibility with many of the current brake based vehicle dynamic intervention systems. Design work was performed on both the electro-hydraulic actuator and clutch to provide engagement and disengagement times of less than 50 milliseconds.

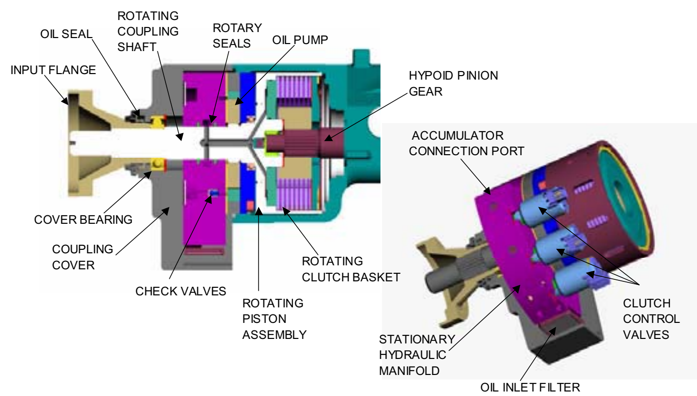


Figure 2. Eaton's center coupler (HTC-I)

The EGerodisc-II utilized in this paper is built upon a solid "Open" differential foundation. Limited slip functionality is provided by an actively controlled wet multi-plate clutch disposed between one of the bevel "side" gears and the differential housing. Clutch engagement limits the slip between the "side" gear and differential housing and in doing so limits the slip between both output axle shafts thru the bevel gear set. This slip limiting function results in the ability to produce a torque bias between output axle shafts, the magnitude of which will be less then or equal to the clutch torque. The actuator for the clutch is a design adaptation from Eaton's EGerodisc™ differential. When ever there is rotational motion between the differential housing and the axle housing a gerotor pump displaces oil from the axle sump to a discharge passage that is in direct communication with both the clutch actuation piston and pressure regulation valve(s). When the valves are de-energized, oil flows freely thru the valves resulting in little hydraulic pressure within the clutch actuation piston and assures fail-safe behavior of the actuator. When the valves are energized, oil flow is restricted thru the valves creating hydraulic pressure within the actuation piston locking the clutch to a level proportional to that of the hydraulic pressure. Both the torque bias engagement and disengagement times are rated at 50milliseconds.

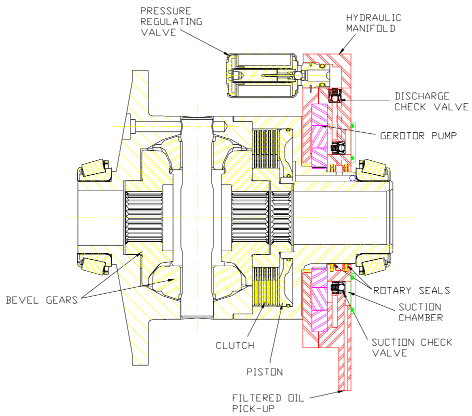


Figure 3. Eaton's ELSD (EGerodisc-II)

TORQUE-BIASING DEVICE MODELING

In this section, the models of the center coupler and the rear ELSD are developed for vehicle control system evaluation. Both models are built based on the dynamic properties of the clutch. The modeling focuses on locking and unlocking (or slipping) conditions. The conditions for the transitions between unlock (or slipped) stage and locked stage are also presented.

CENTER COUPLER

A center coupler is a device that can be used to transfer the torque from front/back to back/front depending on the clutch torque value. The control input is usually the electrical signal from vehicle ECU, and the control signal could be amplified electro-hydraulically, electro-mechanically, or electro-magnetically. Since our research focuses on investigating how to improve the vehicle stability and performance by dynamically distributing the torque to each wheel, we need to know both the steady state torque and the transient process of torque variation. A schematic diagram of the torque-biasing center coupler is shown in Figure 4.

Let's define T_{CT} to be the torque transferred thru the center coupler. Note that this torque is not necessarily the same as the applied clutch torque level controlled by the vehicle ECU, depending on locked, unlock, or slipped stages. Assuming zero inertia, the clutch is simply modeled as a spring-damper torsional element:

$$T_{CT} = c \cdot \Delta\omega_c + \int k \cdot \Delta\omega_c dt \qquad (1)$$

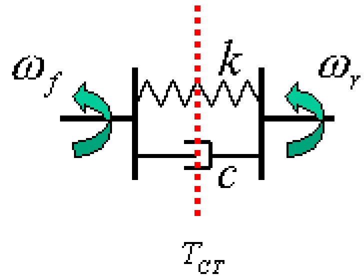


Figure 4. Center coupler model

where *c* is the clutch damping coefficient, *k* is the clutch spring coefficient, and $\Delta\omega_c := \omega_f - \omega_r$ is the speed difference between the front axle and the rear axle. It is worth mentioning that even though the physical model of the center coupler could be very complicated, this simple model can be used to successfully explain two fundamental properties of the center coupler:

1. If the speed difference is nonzero ($\Delta\omega_c \neq 0$), some amount of torque is transferred. This can be modeled as a damper.
2. Even if the speed difference is zero ($\Delta\omega_c = 0$), or the center coupler is locked, torque can still be transferred. This can be modeled as a spring.

Then we have

$$T_f = T_{in} - T_{CT} \qquad (2)$$

$$T_r = T_{CT} \qquad (3)$$

where T_f is the net torque to the front axle inertia, T_r is the net torque to the rear axle inertia, T_{in} is the engine torque injected into the front inertia.

Taking the derivative of the above equation gives

$$\dot{T}_f = \dot{T}_{in} - c \cdot \Delta\dot{\omega}_c - k \cdot \Delta\omega_c \qquad (4)$$

$$\dot{T}_r = c \cdot \Delta\dot{\omega}_c + k \cdot \Delta\omega_c \qquad (5)$$

where $\Delta\dot{\omega}_c = \dot{\omega}_f - \dot{\omega}_r$.

If we consider the equations of motion:

$$I_f\dot{\omega}_f = T_f - r_{eff}F_f \qquad (6)$$

$$I_r\dot{\omega}_r = T_r - r_{eff}F_r \qquad (7)$$

where F_f and F_r are the tractive forces from the front tires and the rear tires respectively and r_{eff} is the effective radius, then $\Delta\dot{\omega}_c$ is given by

$$\Delta\dot{\omega}_c = \frac{T_f - r_{eff}F_f}{I_f} - \frac{T_r - r_{eff}F_r}{I_r} \qquad (8)$$

REAR LIMITED SLIP DIFFERENTIAL

An Electronic Limited Slip Differential (ELSD) has the same components as an open differential except for a clutch that provides an additional path for torque transfer. In Figure 5, T_r is the torque transferred to rear prop shaft by center coupler, T_{diff} is the torque transferred thru the differential gears, and T_{CT_r} is the torque transferred thru the clutch. Also, note that this torque is not necessarily the same as the applied clutch torque level controlled by the vehicle ECU, depending on locked, unlock, or slipped stages.

Assume that

1. The efficiency of the torque transmission is 100%;
2. The differential gear ratio from the prop shaft to the differential is 1; and
3. The differential has no mass inertia.

Then we have

$$T_r = T_{CT\_r} + T_{diff} \qquad (9)$$

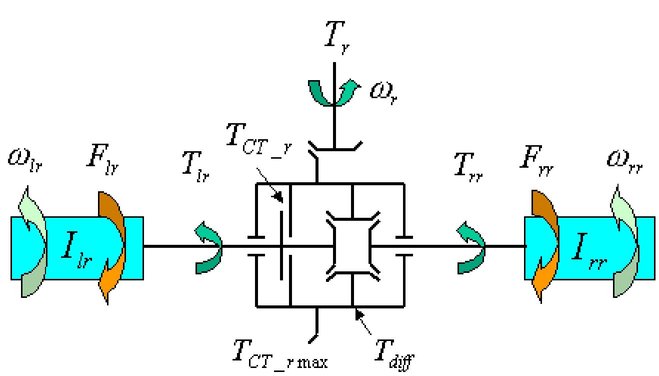


Figure 5. Schematic of side-to-side ESLD (rear)

Since T_{diff} is equally distributed to the left and right axle, then the net torque to the rear-left inertia and the rear-right inertia are given by

$$T_{lr} = T_{CT\_r} + \frac{T_{diff}}{2} \qquad (10)$$

$$T_{rr} = \frac{T_{diff}}{2} \qquad (11)$$

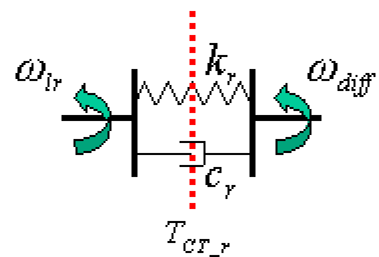


Figure 6. Model of clutch in ESLD

Similar to the center coupler, the clutch in ELSD can be modeled as a spring-damper torsion torque:

$$T_{CT\_r} = c_r \cdot \Delta\omega_r + \int k_r \cdot \Delta\omega_r dt \qquad (12)$$

where c_r is the clutch damping coefficient, k_r is the clutch spring coefficient, $\Delta\omega_r := \omega_{diff} - \omega_{lr}$. From Equation (12), T_{diff} is calculated as

$$T_{diff} = T_r - T_{CT\_r} = T_r - (c_r \cdot \Delta\omega_r + \int k_r \cdot \Delta\omega_r dt) \quad (13)$$

Then

$$T_{lr} = T_{CT\_r} + \frac{T_{diff}}{2} = \frac{T_r + (c_r \cdot \Delta\omega_r + \int k_r \cdot \Delta\omega_r dt)}{2} \qquad (14)$$

$$T_{rr} = \frac{T_{diff}}{2} = \frac{T_r - (c_r \cdot \Delta\omega_r + \int k_r \cdot \Delta\omega_r dt)}{2} \qquad (15)$$

Taking the derivative of the above equations gives

$$\dot{T}_{lr} = \frac{1}{2}\left(\dot{T}_r + c_r \cdot \Delta\dot{\omega}_r + k_r \cdot \Delta\omega_r\right) \qquad (16)$$

$$\dot{T}_{rr} = \frac{1}{2}\left(\dot{T}_r - c_r \cdot \Delta\dot{\omega}_r - k_r \cdot \Delta\omega_r\right) \qquad (17)$$

where $\Delta\dot{\omega}_r = \dot{\omega}_{diff} - \dot{\omega}_{lr}$.

Next we will derive $\Delta\dot{\omega}_r$. First consider the following dynamic equations of the rear left and rear right shafts:

$$I_{lr}\dot{\omega}_{lr} = T_{lr} - r_{eff}F_{lr} \quad (18)$$

$$I_{rr}\dot{\omega}_{rr} = T_{rr} - r_{eff}F_{rr} \quad (19)$$

In addition, according to the physical principle of differential, we have

$$\omega_{diff}(=\omega_r) = \frac{\omega_{lr}+\omega_{rr}}{2}$$,or

$$\dot{\omega}_{diff} = \frac{\dot{\omega}_{lr}+\dot{\omega}_{rr}}{2} \quad (20)$$

Substitute Equations (18) and (19) into (20), we have

$$\Delta\dot{\omega}_r = -\frac{T_{lr}-r_{eff}F_{lr}}{2I_{lr}} + \frac{T_{rr}-r_{eff}F_{rr}}{2I_{rr}} \quad (21)$$

by noting that $2\Delta\dot{\omega}_r = 2(\dot{\omega}_{diff}-\dot{\omega}_{lr}) = \dot{\omega}_{lr}-\dot{\omega}_{rr}$.

STAGE TRANSITION MODELING

The previous modeling represents the situations where the applied clutch torque is bigger than the torque difference between the clutch plates, hence, it describes the locking dynamics. However, when the applied clutch torque is smaller than the torque difference between the clutch plates, the clutch will unlock or slip.

In this section, we will address the transition models between the unlock/slipped stage and the locked stage. As aforementioned, given a control input *u,* a clutch can only provide a certain amount of transfer torque up to the corresponding upper bound. For center coupler, the upper bound is $\max_{u_{CT}}|T_{CT}|$ where u_{CT} is the control input to the center coupler; while ELSD's upper bounded torque is $\max_{u_{CT\_r}}|T_{CT\_r}|$ where u_{CT_r} is the control input to ELSD. Obviously transition between the two stages is directly related to the upper bounds.

<u>Condition for center coupler</u>: given a control input to the center coupler u_{CT},

1. Transition from unlock/slipped stage to locked stage occurs if the absolute value of transferred clutch torque is less than its upper bound. Or

   $$|T_r| < \max_{u_{CT}}|T_{CT}|$$

2. Transition from lock stage to unlock/slipped stage occurs when

   $$|T_r| = \max_{u_{CT}}|T_{CT}|$$

<u>Conditions for ELSD</u>: given a control input to the rear limited slip differential u_{CT_r},

1. Transition from unlock/slipped stage to locked stage occurs if the absolute value of transferred clutch torque is less than its upper bound. Or

   $$|T_{lr}-T_{rr}| < \max_{u_{CT\_r}}|T_{CT\_r}|$$

2. Transition from locked stage to unlock/slipped stage occurs when

   $$|T_{lr}-T_{rr}| = \max_{u_{CT\_r}}|T_{CT\_r}|$$

   From Equations (10) and (11) we have

   $$T_{CT\_r} = T_{lr} - \frac{1}{2}T_{diff} = T_{lr} - T_{rr}$$

Utilizing the above stage transition conditions, the dynamics of the torque-biasing devices can be implemented in simulation software such as Matlab/Simulink. The discrete-time modeling of both devices can be summarized as:

<u>Center Coupler</u>:

Changing from locked stage to unlock/slipped stage,

$$T_f(n) = T_{in}(n) - T_{CT\_\max}(n) \quad (22)$$

$$T_r(n) = T_{CT\_\max}(n) \quad (23)$$

Changing from unlock/slipped stage to locked stage,

$$\begin{aligned} T_f(n+1) &= T_f(n) + T_{in}(n+1) - T_{in}(n) \\ &\quad - k\cdot\delta t\cdot\Delta\omega_c(n) - c\cdot\delta t\cdot\Delta\dot{\omega}_c(n) \end{aligned} \quad (24)$$

$$T_r(n+1) = T_r(n) + k\cdot\delta t\cdot\Delta\omega_c(n) + c\cdot\delta t\cdot\Delta\dot{\omega}_c(n) \quad (25)$$

where $\Delta\dot{\omega}_c(n) = \frac{T_f(n)-F_f(n)}{I_f} - \frac{T_r(n)-F_r(n)}{I_r}$ and δt is the sampling period.

<u>Rear ELSD</u>:

Changing from locked stage to unlock/slipped stage,

$$T_{lr}(n) = \frac{T_r(n) + T_{CT\_r\max}(n)\cdot \mathrm{sgn}(\Delta\omega_r(n))}{2} \quad (26)$$

$$T_{rr}(n)=\frac{T_r(n)-T_{CT\_r\max}(n)\cdot \mathrm{sgn}(\Delta\omega_r(n))}{2} \tag{27}$$

Changing from unlock/slipped stage to locked stage,

$$\begin{aligned} T_{lr}(n+1)=T_{lr}(n)+\frac{T_r(n+1)}{2}-\frac{T_r(n)}{2} \\ +\frac{k_r\cdot\delta t}{2}\cdot\Delta\omega_r(n)+\frac{c_r\cdot\delta t}{2}\cdot\Delta\dot{\omega}(n) \end{aligned} \tag{28}$$

$$\begin{aligned} T_{rr}(n+1)=T_{rr}(n)+\frac{T_r(n+1)}{2}-\frac{T_r(n)}{2} \\ -\frac{k_r\cdot\delta t}{2}\cdot\Delta\omega_r(n)-\frac{c_r\cdot\delta}{2}t\cdot\Delta\dot{\omega}(n) \end{aligned} \tag{29}$$

TORQUE-BIASING EFFECTS ON VEHICLE YAW DYNAMICS

In this section, the effects of the above torque-biasing devices on vehicle yaw dynamics are studied.

SIDE-TO-SIDE TORQUE TRANSFER

An electronically-controlled limited slip differential (ELSD) is used to bias the rear prop-shaft torque to left and right wheels. If the differential clutch torque is applied while the vehicle is turning, the device only transfers torque from the outside wheel to the inside wheel, thus generating a yaw moment in the opposite direction of the turn. This would increase the understeer tendency of the vehicle. The phenomena can be explained by considering equations (16) and (17). The speed of the outside wheel is normally larger than the speed of the inner wheel while turning. When the differential clutch torque is applied, it will try to bring the speeds of both outer wheel and inner wheel to the same value. The outer wheel speed and acceleration will be reduced, along with the driving torque, and vice versa, the driving torque at the inner wheel will be increased.

FRONT-TO-BACK TORQUE TRANSFER

Unlike side-to-side torque transfer, which typically generates understeering tendency, front-to-back torque transfer can produce a much more complicated response. It is known from the tire model literature that tire steering forces/lateral forces will reduce when accelerating or braking during cornering due to the effect of the friction circle [11]. Next, we will analyze the vehicle yaw response to front-to-back torque transfer by utilizing the tire friction circle concept.

First consider the yaw dynamics of a vehicle

$$I_r\dot{r}=L_f F_{xf}\sin\delta+L_f F_{yf}\cos\delta-L_r F_{yr} \tag{30}$$

where I_r is the yaw inertia, L_f and L_r are the distances between CG and the front axle and the rear axle, respectively, δ is the steered angle, F_{xf} , F_{yf} , F_{yr} are the tires forces, in whose subscripts *x* and *y* represent longitudinal direction and lateral direction respectively, and the subscripts *f* and *r* represent front and rear wheel.

A simple front-to-back torque transfer case is considered: in a time interval $[t_0, t_f]$, engine torque, T_{in}, is being transferred from front wheels to rear wheels. Then, at $t=t_0$, equation (30) yields

$$\frac{I_r}{L_f}\dot{r}(t_0)=F_{xf}(t_0)\sin\delta+F_{yf}(t_0)\cos\delta-\frac{L_r}{L_f}F_{yr}(t_0) \tag{31}$$

Now we would like to compare the yaw rate at $t=t_f$ by using the friction circle concept. Figure 7 shows a function of cornering force with respect to slip angle (the left half side) and longitudinal force (the right half side), respectively.

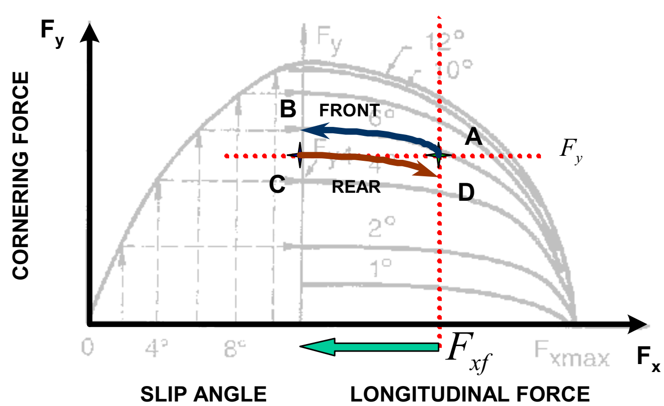


Figure 7. Front-to-back torque transfer effect on friction circle

For any slip angle, note that cornering force monotonously decreases as longitudinal force increases, and vice versa. In Figure 7, "A" and "C" denote the initial conditions for the front and rear wheels, respectively. By assuming that the torque transfer is much faster than the dynamics of slip angle change, we can use a constant slip angle in the following analysis. In transition procedure $[t_0, t_f]$, front wheel force will track the curve passing thru "A" up to point "B" while keeping slip angle identical. Similarly, rear wheel will track the curve passing thru "C" down to point "D" while keeping slip angle identical as well. According to the monotonic property of the curves, we have

$$F_{yf}(t_0) < F_{yf}(t_f), F_{yr}(t_0) > F_{yr}(t_f)$$

In other words,

$$F_{yf}(t_f) = F_{yf}(t_0) + \rho_1 \quad (32)$$

$$F_{yr}(t_f) = F_{yr}(t_0) - \rho_2 \quad (33)$$

where ρ_1 is the cornering force gained at the front wheels ($\rho_1 > 0$), and ρ_2 is the cornering force lost at the rear wheels ($\rho_2 > 0$) during the transition.

Based on the above analysis, we will have the following condition to judge the yaw response for front-to-back torque transfer.

Lemma 1*:* Front-to-back torque transfer will generate oversteering if and only if

$$-F_{xf}(t_0)\sin\delta + \rho_1\cos\delta + \frac{L_r}{L_f}\rho_2 > 0 \quad (34)$$

where ρ_1 and ρ_2 are given by equation (32).

Proof:

At $t = t_f$, equations (30), (31), and (32) gives

$$\begin{aligned}\frac{I_r}{L_f}\dot{r}(t_f) &= F_{xf}(t_f)\sin\delta + F_{yf}(t_f)\cos\delta - \frac{L_r}{L_f}F_{yr}(t_f) \\ &= F_{xf}(t_f)\sin\delta + F_{yf}(t_0)\cos\delta - \frac{L_r}{L_f}F_{yr}(t_0) \\ &\quad + \rho_1\cos\delta + \frac{L_r}{L_f}\rho_2 \\ &= \frac{I_r}{L_f}\dot{r}(t_0) + \rho_1\cos\delta + \frac{L_r}{L_f}\rho_2 - \rho_3\sin\delta\end{aligned}$$

where $\rho_3 = -F_{xf}(t_f) + F_{xf}(t_0)$ is the longitudinal force lost during transition ($\rho_3 > 0$).

Then the oversteering, $\dot{r}(t_f) - \dot{r}(t_0) > 0$, is equivalent to

$$-\rho_3\sin\delta + \rho_1\cos\delta + \frac{L_r}{L_f}\rho_2 > 0 \qquad \blacklozenge$$

Basically Lemma 1 implies that as long as the torque contributed by lateral force variation is larger than the effect of longitudinal force variation and the magnitude of the steering angle, oversteering can occur.

Remark 1: Lemma 1 is valid for the general case, since monotonicity of the curves in the tire friction circle model still holds. However, ρ_1, ρ_2 and ρ_3 need to be calculated accordingly.

Remark 2: For a small steering angle $\delta \approx 0$, inequality in equation (34) certainly holds. Front-to-back torque transfer always generates oversteering in this case.

Remark 3: If the initial front longitudinal force, F_{xf}, is small, it would be more difficult to induce oversteering. This can be seen in the friction circle in Fig. 7, close to point "B" or "C" where the profiles for a certain slip angle are really "flat". Lateral force variation has very little effect on yaw dynamics in this case.

YAW CONTROL METHODOLOGY

As discussed in the previous section, the control of drive torque distribution can be used to change vehicle tractive forces at the wheels and consequently influence the dynamic yaw response of the vehicles.

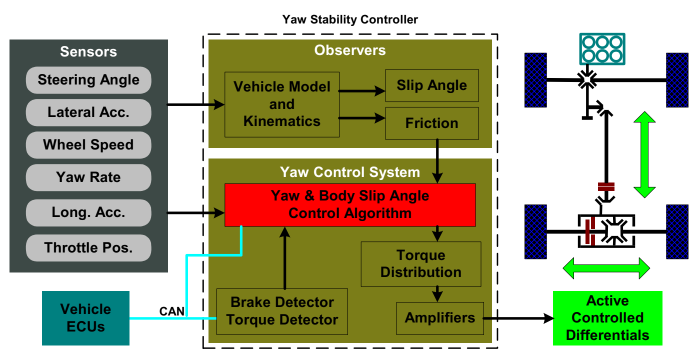


Figure 9. Vehicle stability control

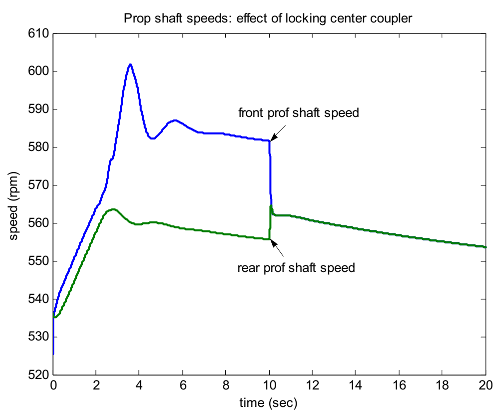
A yaw control strategy has been developed to improve vehicle yaw stability and handling performance using torque biasing. The vehicle stability control architecture is shown in Figure 9., The control strategy with the given driveline layout is based on the following two principles: (1) transferring the tractive force from front wheels to rear wheels with the active center coupler induces less understeering, and (2) locking the rear differential induces more understeering behavior. In combination with vehicle dynamic signals, the clutch torques can be adjusted to tune the desired vehicle yaw dynamics behavior to be suitable for specific driving conditions. The clutch torque level can be excited either by hydraulic or electromagnetic system. However, it needs to have fast enough response time to guarantee the effectiveness of the yaw dynamics control.

The clutch excitation values for the ELSD and the center coupler can be increased linearly depending on the yaw rate difference from the desired yaw rate. In addition, in order to guarantee transferring torque from front wheels to rear wheels by locking the center coupler, the front axle speed needs to be larger than the rear axle speed. A more detailed discussion of the control system will be reported in a future paper.

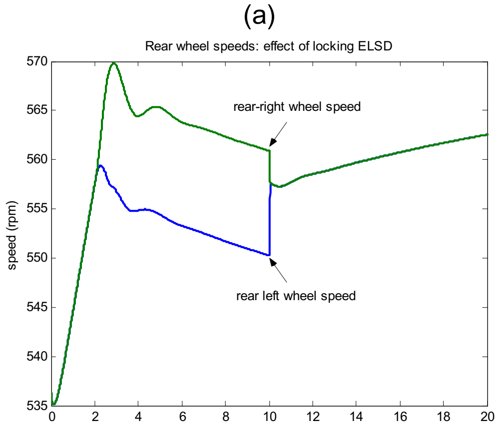
SIMULATION RESULTS

The torque-biasing device dynamic models were generated in Matlab/Simulink environment. A full vehicle model developed by CarSim was used and modified to fit with the new torque biasing devices so that co-simulation can be performed.

Figure 9 shows the validation of the developed models. When a high clutch torque was applied to the center coupler at 10 seconds for a turning maneuver, the speeds of the front prop shaft and the rear prop shaft became the same within an engagement time. Similar result was observed when the rear ELSD was locked, the left wheel speed and the right wheel speed became the same.



(a)



(b)

Figure 9. Speed curves with locked (a) center coupler (b) rear ELSD

A turning maneuver was simulated under various conditions to validate the effects of locking torque-biasing devices on vehicle yaw dynamics. An SUV model was created in CarSim. All conditions had the same initial speed of 50 mph and step steering wheel angle of 70 degrees started at 2 seconds.

The torque-biasing devices were simulated for the following 4 scenarios.

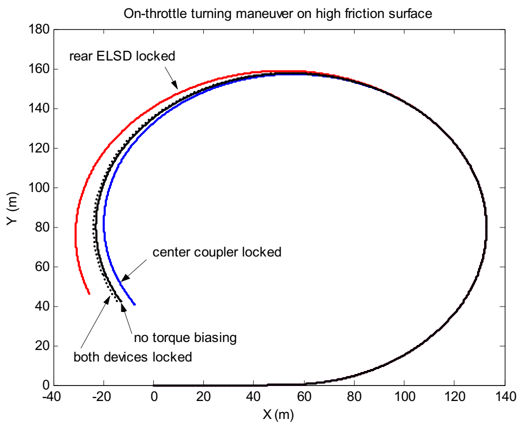
- Both the center coupler and the rear ELSD were open.
- The center coupler was locked at 10 seconds.
- The rear ELSD was locked at 10 seconds.
- Both the center coupler and the rear ELSD were locked at 10 seconds.

The turning maneuvers were then performed on different tire-road friction surfaces, $\mu = 0.2$ and $\mu = 0.85$, with on and off throttle.

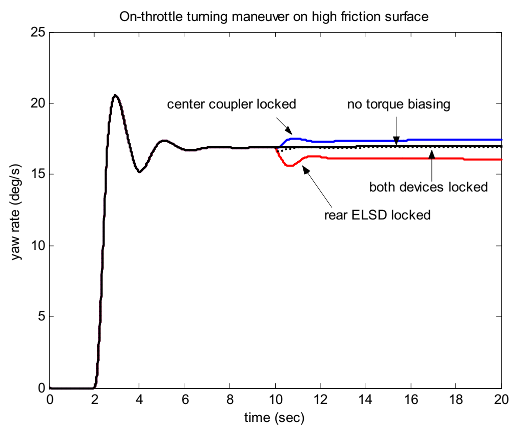
Figures 10-13 show the vehicle paths and yaw responses of the aforementioned settings. Figure 10 shows an on-throttle turning performance on the high friction surface. As expected, the vehicle tends to show less understeer with the tractive force transferred from the front wheels to the rear wheels affected by the locked center coupler. It tends to exhibit more understeer with the locked rear ELSD. When both biasing devices were locked, compromised performance was observed. Figure 11 shows the performance comparison for an on-throttle turning maneuver on the low friction surface. It indicates similar results to the previous case but the rear ELSD has less affect on the low friction surface and locking the center coupler creates more oversteering affect. Figures 12 and 13 shows similar responses with no throttle on both high and low friction surface. The resulting plots indicates that locking the center coupler has very minimal affect on the vehicle dynamics while the rear ELSD has a big influence to create more understeering.

The effectiveness of locking biasing devices is summarized as follows,

1. Locking center coupler influences vehicle dynamics more with accelerating maneuvers and induces less understeering. However it has minimal affects with off-throttle maneuvers.
2. Locking rear ELSD induces more understeering on high friction surfaces and has less affect on low friction surface.
3. Compromised performance using both center coupler and rear ELSD can be achieved only with accelerating maneuvers. In other words, it is difficult to induce less understeering with off-throttle maneuver.

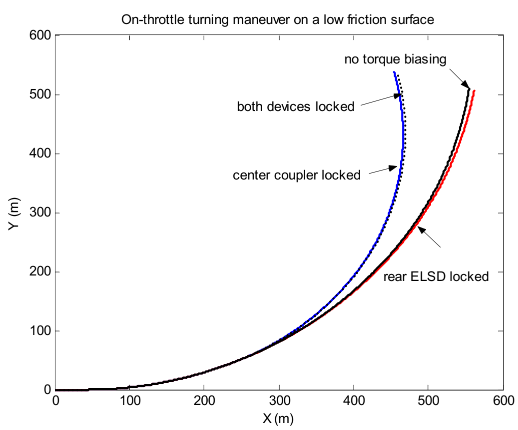


(a)

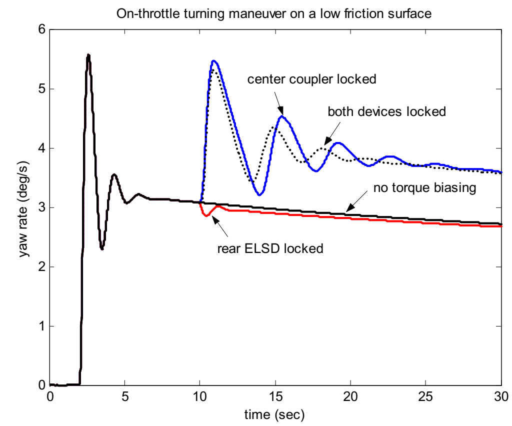


(b)

Figure 10. On-throttle turning maneuver on a high-mu surface. (a) vehicle path (b) yaw rate

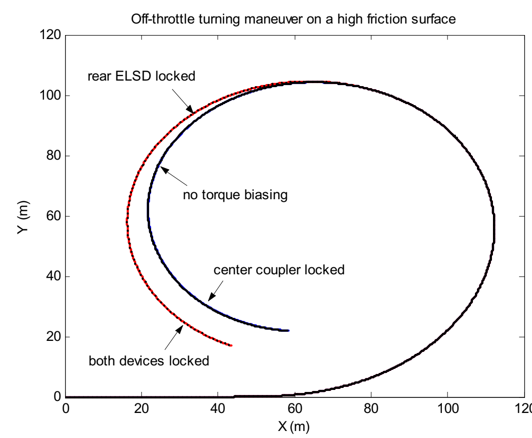


(a)

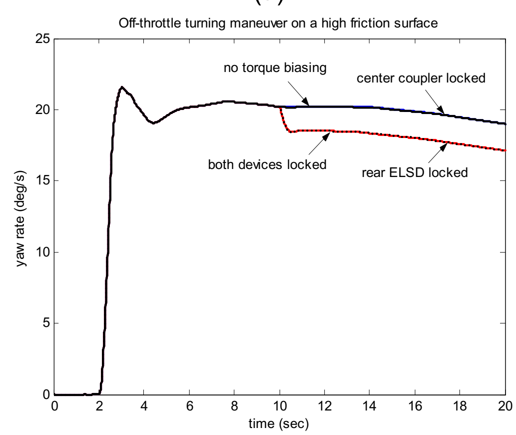


(b)

Figure 11. On-throttle turning maneuver on a low-mu surface. (a) vehicle path (b) yaw rate

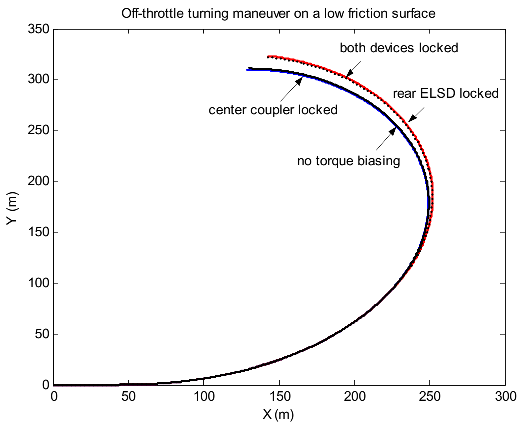


(a)

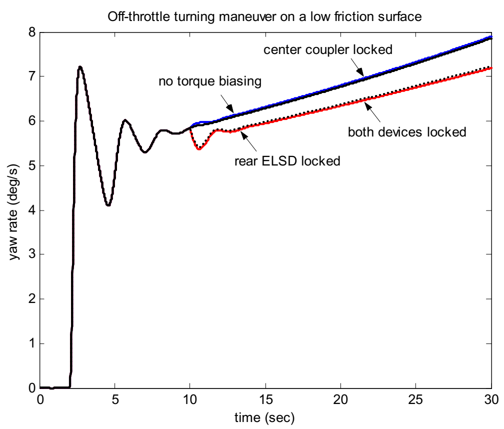


(b)

Figure 12. Off-throttle turning maneuver on high-mu surface. (a) vehicle path (b) yaw rate



(a)



(b)

Figure 13. Off-throttle turning maneuver on low-mu surface. (a) vehicle path (b) yaw rate

CONCLUSION

The concept of providing yaw stability control on a front-wheel-drive based 4WD vehicle using torque-biasing devices has been discussed. The torque-biasing devices utilized are an electronically controlled center coupler and a rear electronically controlled limited slip differential. Mathematical dynamic models of torque-biasing devices, center couplers and rear limited slip differentials, have been developed including stage transitions of the locking stage and the unlocking/slipping stage. Analytical proofs of the torque biasing effect on the yaw vehicle dynamics are presented. Finally, co-simulation results with Matlab/Simulink and CarSim show the effectiveness of the developed model and of the proposed strategy for yaw control.

REFERENCES

1. H. Elmqvist, S.E. Mattsson, H. Olsson, J. Andreasson, M. Otter, C. Schweiger and D. Bruck, “Realtime Simulation of Detailed Vehicle and Powertrain Dynamics,” *SAE Paper* No. 2004-01-0768.
2. R.P. Osborn and T. Shim, “Independent Control of All-Wheel-Drive Torque Distribution,” *SAE Paper* No. 2004-01-2052.
3. S. Torii, E. Yaguchi, K. Ozaki, T. Jindoh, M. Owada and G. Naitoh, “Electronically Controlled Torque Split System for 4WD Vehicles,” *SAE Passenger Car Meeting & Exposition*, Dearborn, Michigan, September 1986.
4. E.F. Drenth, “Verification of the Haldex LSC System Performance,” *15th ADAMS European Users' Conference*, Rome, 2000.
5. G. Fischer, W. Pfau, H.S. Braun and C. Billig, “xDrive: The New Four-Wheel Drive Concept in the BMW X3 and BMW X5,” ATZ Worldwide, Vol. 106, February 2004.
6. E.K. Liebemann, K. Meder, J. Schuh and G. Nenninger, “Safety and Performance Enhancement: The Bosch Electronic Stability Control (ESP),” *SAE Paper* No. 2004-21-0060.
7. H. Huchtkoetter and T. Gassmann, “Vehicle Dynamics and Torque Management Devices,” *SAE Paper* No. 2004-01-1058.
8. J. Park and W.J. Kroppe, “Dana Torque Vectoring Differential Dynamic Trak,” *SAE Paper* No. 2004-01-2053.
9. J.C. Wheals et al., “Torque Vectoring Driveline: SUV-based Demonstrator and Practical Actuation Technologies,” SAE Paper No. 2005-01-0553.
10. Y. Atsumi, “Development of SH-AWD (Super Handling-All Wheel Drive) System,” Presentation, Vehicle Dynamics EXPO 2005, Stuttgart, Germany, June 2005.
11. Y. Ushiroda, K. Sawase, N. Takahashi, K. Suzuki and K. Manabe, “Development of Super AYC,” *Technical Review*, No. 15. 2003.
12. K. Sawase and Y. Sano, “Application of Active Yaw Control to Vehicle Dynamics by Utilizing Driving/Braking Force,” *JSAE Review* 20, Paper No. 9930801, pp. 289-295, 1999.
13. Y. Ikushima and K. Sawase, “A Study on the Effects of the Active Yaw Moment Control,” *SAE International Congress and Exposition*, Detroit, Michigan, February 1995.
14. C. Liu, V. Monkaba, H. Tan, C. McKenzie, H. Lee, and S. Suo, “Driveline Torque-Bias-Management Modeling for Vehicle Stability Control,” *SAE Paper* No. 2002-01-1584.
15. H. Dugoff, P.S. Fancher and L. Segal, “Tyre Performance Charecteristics Affecting Vehicle Response to Steering and Braking Control Inputs,” *Final Report, Contract CST-460, Office of Vehicle Systems Research*, US National Bureau of Standards, 1969.
16. G. Genta, *Motor Vehicle Dynamics*: Modeling and Simulation, Series on Advances in Mathematics for Applied Science - Vol. 43, World Scientific.
17. R. Rajamani, *Vehicle Dynamics and Control*, Springer Verlag, New York, 2005.
18. J.Y. Wong, *Theory of Ground Vehicles*, Third Edition, John Wiley & Sons, 2001.