

Brake Based Torque Vectoring for Sport Vehicle Performance Improvement

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ABSTRACT

The most common automotive drivelines transmit the engine torque to the driven axle through a differential. Semi-active versions of this device ([4], [5], [6]) have been recently conceived to improve vehicle handling at limit and under particular conditions; these differentials are based on the structural scheme of the passive one but they try to manipulate the vehicle dynamics by controlling the distribution of the driving torque on the wheels of the same axle thus generating a yaw moment. Unfortunately a semi-active differential is not able to perform a complete yaw control since the torque can only be transferred from the faster wheel to the slower one; on the other hand, active differentials ([11], [12], [13]) allow to generate the most appropriate yaw moment controlling both the amount of transferred torque and its direction.

The application presented in this the paper aims at enhancing the dynamic behaviour (in terms of handling performance) of a rear-driven sport vehicle by creating the required yaw moment through brakes actuation and throttle control; the examined car is equipped with free differential, thus the proposed system does not require the introduction of additional devices. The feasibility study of the proposed concept is based on the comparison with a semi-active and active differential model.

INTRODUCTION

The conventional free differential is a mechanism that lets the driven wheels to assume different speeds while cornering with a uniform distribution of the driving torque on the two wheels of the same axle; this device shows its main limit when the adherence conditions of the two wheels are different: in this case a free differential is not able to transfer torque to the wheel with a higher adherence with the consequent result of a really poor traction of the vehicle.

The need for a better match between available power to actual road conditions has been partially covered by the use self-locking differentials. In presence of a difference between the internal and the external wheel speed, a locking action of these passive mechanisms transfers torque from the fastest wheel to the slowest one; this effect improves the traction since most part of the torque is directed towards the wheel with higher adherence but also leads to an asymmetric distribution of the longitudinal contact forces giving rise to a yaw moment which affects the turning behavior of the vehicle (Figure 1).

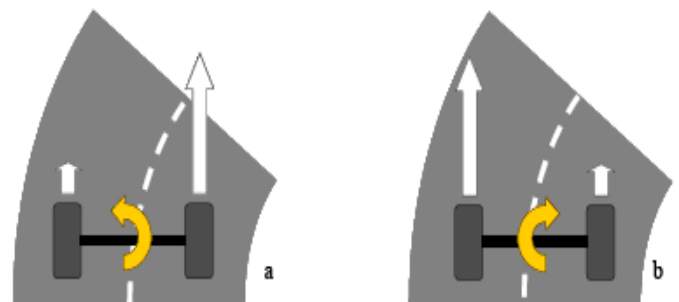


Figure 1: Torque distribution with a passive differential at low lateral acceleration (a), where the angular speed of the outer tyre is higher, and high lateral acceleration (b), where the angular speed of the outer tyre is lower because of the lateral load transfer.

Passive self-locking differentials are commonly adopted in sport vehicles in order to improve traction and stability. Semi-active and active differentials have been developed in order to provide further improvement on vehicle overall performance and driving feeling.

A semi-active differential system (Figure 2) is generally equipped with an internal electronic control logic tuned to apply the most appropriate locking torque for every dynamic condition of the vehicle. The weak point of this system is related with the lack of control of the direction of the transferred torque which is always shifted from the fastest wheel to the slowest one.

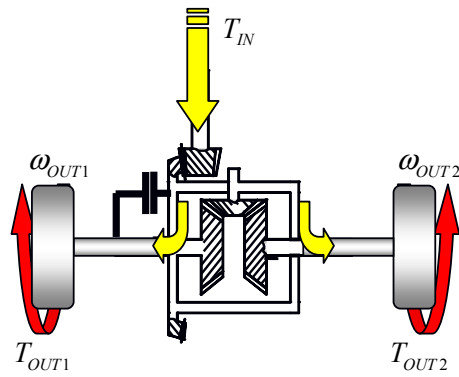


Figure 2: Semi-active differential. T_{IN} : engine torque; T_{OUT1} and T_{OUT2} : resultant torque on each wheel; ω_{OUT1} and ω_{OUT2} : wheel speed rotation.

On the other hand, a powertrain equipped with an active differential system achieves an higher degree of flexibility: an active differential (Figure 3) is designed to control both the locking torque (equivalent to the semi active one) and its direction; in this way it is possible to create a yaw moment regardless of the kinematic condition of the driven wheels, by transferring torque also from the slower one to the faster one. This flexibility produces a better compromise between traction and vehicle dynamics performance.

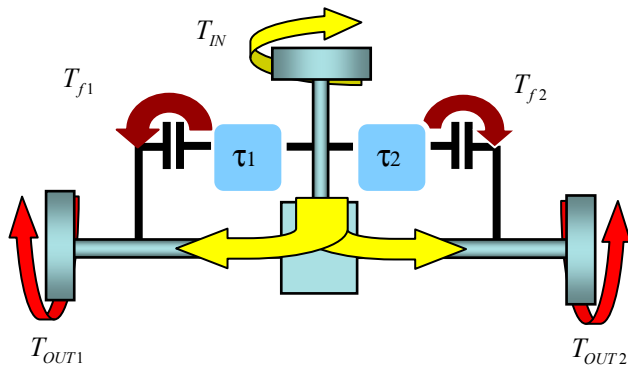


Figure 3: Active differential. T_{IN} : engine torque; T_{OUT1} and T_{OUT2} : resultant torques on the two wheels; T_{f1} and T_{f2} : torque's transferred by the friction; τ_g and τ_d : gearbox and differential gear ratio.

Unfortunately, the implementation of such a complicated device is associated with an increase of costs in terms of tuning, space and funding. These drawbacks have a significant impact on the automotive companies which are usually interested in more economical solutions involving minimum changes in the architecture of the vehicle. For this reason devices like Active Brake Control (ABC [11]) might be preferred even if, as far as the handling point of view is concerned, active differentials may have an inherent advantage since they operate by directly reapportioning the torque supplied to the wheels. On the other hand the intervention of the ABC leads to a reduction of the total driving torque and therefore this system cannot operate without slowing down the vehicle speed.

HANDLING TARGET

The handling performance of a high speed vehicle equipped with a semi-active differential is illustrated by the understeer curve reported in Figure 4.

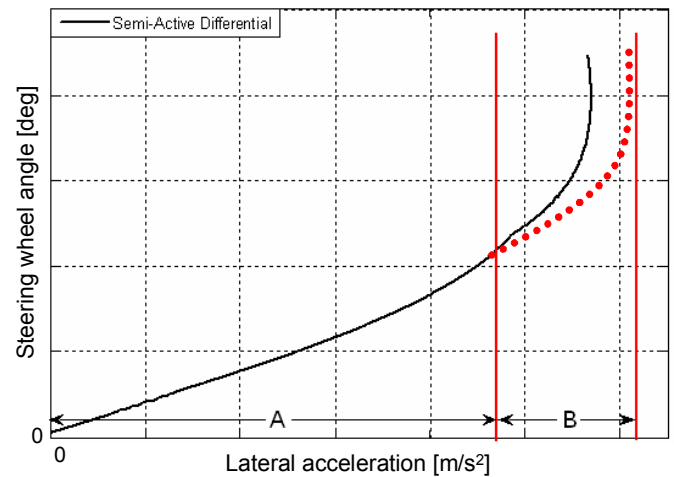


Figure 4: Understeer curve of the reference model equipped with the semi-active device. Red dotted line is the handling target.

The handling diagram of Figure 4 can be divided into two regions of interest: the linear region (A) where the response of the tires is still in the linear range and the non-linear region (B) where the tires gradually reach the saturation of the frictional force.

The generation of a yaw moment by means of active and semi-active differential or brake actuation can affect the shape of the understeer diagram both in region A and B; even if region B represents the zone of interest for a sport vehicle, where the target is the highest lateral acceleration achievable associated with vehicle stability. As first step it should be important to understand how the resultant yaw moment can affect and improve the lateral dynamics regardless of the way it is generated.

Considering a single track vehicle model, the lateral acceleration A_y of the centre of gravity (c.o.g.) of a vehicle can be simply expressed as:

$$A_y = \dot{V}_y + \psi \dot{V}_x \quad (1)$$

where V_y and V_x respectively represent the lateral and longitudinal component of the c.o.g. speed, while ψ is the yaw rate.

During a steady-state maneuver at constant velocity the first term of eq. 1 can be considered negligible; this implies that, a higher lateral acceleration can be developed if the yaw rate is increased, i.e. imposing higher yaw moment.

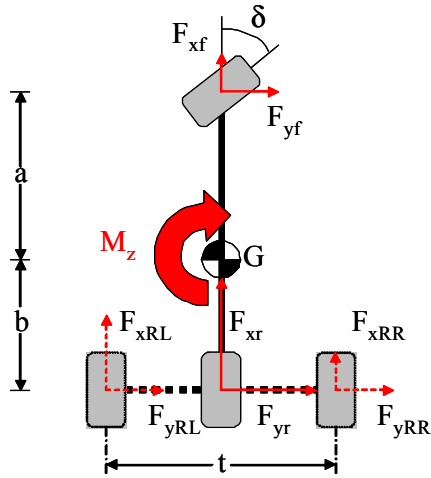


Figure 5: Single track model. M_z : resultant yaw moment caused by the asymmetric distribution of the longitudinal forces.

Considering the symbols of Figure 5, it is possible to write down the equation of motion along the z-axis of a simplified single track model:

$$J_z \ddot{\psi} = F_{yf}a - F_{yr}b + M_z \quad (2)$$

where M_z represents the yaw moment produced with the asymmetric distribution of the longitudinal forces on the rear axle:

$$M_z = \frac{F_{xRL} - F_{xRR}}{t/2} \quad (3)$$

In (2) J_z is the yaw moment of inertia, a the distance of the front axle from the center of gravity, b the distance from the rear axle and t the track width. Under the hypothesis of a steady-state condition the following expression can be obtained:

$$F_{yf} = \frac{F_{yr}a + M_z}{b} \quad (4)$$

The equilibrium around z axis will be achieved only if the rear tires are able to generate higher lateral forces and balance the increased yaw moment; this will cause a raise of the slip angles on both the rear tires.

Considering the improvement of the vehicle performance as target, this work explores the feasibility of a mechanism of yaw moment generation, based on the control of throttle and independent actuation of the brakes on the driven wheels of a rear driven vehicle, equipped with a free differential. The ultimate advantage of the system (hereafter named BTV) is that it doesn't require any additional electronic or mechanical component but it is only exploiting the existent ones. The improvement of vehicle performance associated with the use of BTV has been investigated through numerical simulation thus allowing also a comparison with the

performance obtained by the equivalent dynamic model of a semi-active differential (Electronic Differential) and an active one.

BRAKE TORQUE VECTORING (BTV)

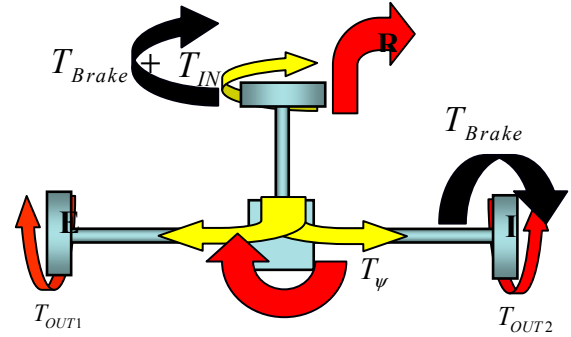


Figure 6: Free Differential. T_{IN} : input torque, transmitted by the gear box to the differential; T_{OUT1} and T_{OUT2} : resultant torque to each output shafts; T_{Brake} : braking torque; T : resultant yaw moment; R : direction of the turn (right); L stands for inner and E for outer.

The concept of BTV is based upon the generation of a yaw moment through independent brake actuation on the driven wheels. With respect to systems like Vehicle Dynamics Control (VDC), the main focus of BTV is the global enhancement of the vehicle performance; for this reason BTV acts also on the throttle valve to avoid the speed reduction associated with brake actuation. Even if the system is designed to increase lateral acceleration and promptness during transients, stability at limit is obviously included among the targets.

Figure 6 represents a scheme of the intervention of BTV system: assuming a steady state condition during a right turn of a rear driven car equipped with a free differential, a brake torque on the internal axle has been applied. As a consequence the external wheel must receive a torque equal to the braking torque applied on the internal one to keep the vehicle speed. The additional torque applied to the external wheel has been applied by accelerating the engine thus compensating energy dissipation produced by the brake actuation:

$$T'_{IN} = T_{IN} + \frac{T_{Brake}}{r} \quad (6)$$

where r in (6) is the differential gear ratio.

The asymmetric torque distribution on the rear axle can clearly affect the traction force balance and create a yaw moment mechanism. Compared to a passive or semi-active differential, this mechanism can be created independently from loading and adherence on the ground. This implies that, during a turn, the system has the ability not only to transfer all the driving torque to the external wheel and maintain the internal one in free rolling condition but also to further amplify the yaw moment by creating a negative traction force on the

internal wheel and increasing the driving torque on the external one.

CAR MODEL AND SIMULATION ENVIRONMENT

A 14 degrees of freedom model (IPG CarMaker®) of the examined sport car has been used to test and compare the performance of various control systems; the vehicle model has been integrated with the models of actuators and of the control logic. The model has been validated comparing its outputs to experimental data relevant to the passive vehicle equipped with a rear free differential.

In Figure 7 and Figure 8 the understeer curve (steer angle vs. lateral acceleration) and the side slip curve (sideslip angle vs. lateral acceleration) are respectively shown for an ISO steering pad maneuver. The scales are not reported in all figures in this paper because of confidentiality agreements.

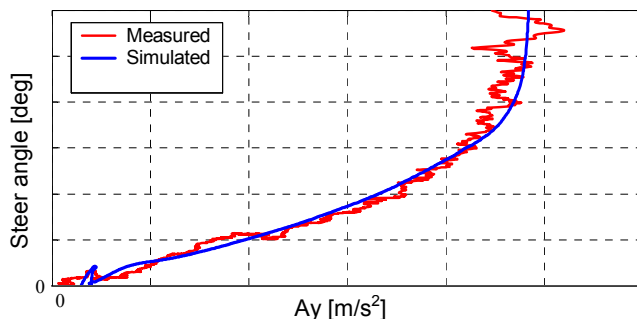


Figure 7: Steering pad constant radius. Understeering curve numerical vs. experimental.

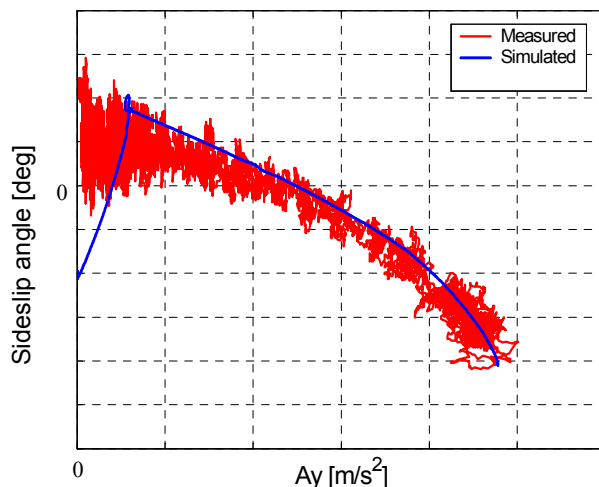


Figure 8: Steering pad constant radius. Side slip curve numerical vs. experimental.

As well known, tire behavior deeply affects the handling dynamics and influences the differential and clutches response. For this reasons the combined behaviour of the tires has been studied with great care, thus allowing get a good reproduction of the vehicle's response also in the non linear range. Such a relative validation has allowed achieve a better confidence in the numerical results presented in the following.

Because of the significance of a proper modelling of the clutch stiction and slip phenomena, the powertrain models has been developed using a mathematical approach appropriate for this kind of analysis ([6], [7], [8], [9]).

PERFORMANCE EVALUATION - HANDLING

In this section of the paper an attempt will be made in order to examine the BTV performance in a more quantitative perspective. The BTV effect on the vehicle dynamics is compared with those obtained by equipping the car with a semi-active differential and an active one. The comparison was made through the following simulations:

- standard ISO maneuvers (Steering Pad Constant Velocity & Steering Pad Constant Radius);
- exit in full throttle from an U-turn;
- virtual race track.

It must be pointed out that BTV logic is not yet been optimized and therefore a preliminary logic is used, which is adequate to demonstrate the advantages achievable with respect to the semi-active differential and to compare it with the performance of an active differential both in terms of vehicle handling and energy losses.

1. STEERING PAD CONSTANT VELOCITY (STEADY STATE CONDITIONS, ISO 4138)

The purpose of this test is to verify that both the active differential and BTV can achieve a similar handling performance and to determine the handling improvements with respect to the semi-active model.

As mentioned before, the resultant yaw moment should not lead to an oversteering condition; therefore, the imposed driving torque distribution must guarantee a sensible gain in vehicle performance without compromising its stability. The main part of the logic is thus based on a 3D map whose values correspond to the maximum oversteering moment tolerable by the car as function of both speed and lateral acceleration.

The applied yaw moment should follow certain rules: if speed and lateral acceleration are low, the gain in terms of understeering gradient is narrow so that the driver will not perceive a handling improvement. On the other hand, at high lateral acceleration the understeering gradient should be hugely influenced by the logic intervention.

The 3D map is a function of both longitudinal speed (V_x), and lateral acceleration (A_y). The procedure followed to tune the feed-forward map was: