THE DEVELOPMENT OF A BRAKE BASED TORQUE VECTORING SYSTEM FOR A SPORT VEHICLE PERFORMANCE IMPROVEMENT

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Keywords: Vehicle dynamics, Passive differential, Semi-active differentials, Active differentials, Controlled brake system, Performance.

Abstract: In every driving condition powertrain and vehicle dynamics deeply influence each other. The main role of powertrain influence is played by the differential, which transmit the driving torque mainly with respect to wheel kinematics. Semi-active controlled versions of this device have been recently conceived to improve vehicle handling basing their function on the wheels kinematical conditions. On the other hand, active differentials 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 behavior 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. Performance measures relate to both open-loop and closed-loop driving demands, and include limit handling maneuvers.

1 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. There are many examples of controlled differential systems in the literature (Pedrinelli, Cheli 2007) (Resta, Teuschl, Pedrinelli, Zorzutti 2005), (Zorzutti, Pedrinelli, Cheli, 2007). The vast majority employ a limited slip differential (LSD) similar to the passive gerodisc type where a friction clutch is employed effectively to provide a connection between the two drive shafts. The distinguishing feature of this type of LSD is that it will always transfer torque to the slower wheel. Such control systems thus have no control over the direction of torque transfer and are only able to modulate the applied magnitude.

The advent of the “overdriven” differential (Hancock, Williams, Gordon, Best, 2005), (Granzow, Gruhle, Spiess, Denzier, Baasch, Peter, 2007), (Leffler, 2007), however (Figure 1), makes it possible to control both the magnitude and direction of torque transfer. This allows the direction of the resulting yaw moment to be controlled and has led to the development of active yaw control systems (Tomari, Mori, Shibahata, 2005) which utilize controlled torque transfer. a powertrain equipped with an active differential system achieves an higher degree of flexibility: an active differential 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 kinematical 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.

2 HANDLING TARGETS

The handling performance in steady state conditions of a high speed vehicle equipped with a semi-active differential is illustrated by the understeer curve reported in Figure 1.

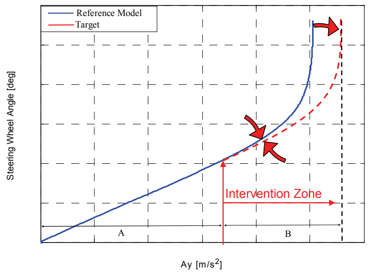

Figure 1: Reference understeer and target diagram.

Table 1: Selected test manoeuvres and performance targets.

Dynamic Conditions	Manoeuvre	Performance Index	Target
Steady State	Ramp Steer (ISO 4138:1996)	Understeer gradient	Reduce
		A_{ymax}	Increase
		Sideslip gradient	Increase
Power on	Ad-hoc	A_{ymax}, A_{xmax}	Increase
Mixed	Virtual race track	Time lap	Reduce
		Power dissipation	Reduce

The handling diagram of Figure 1 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 frictional force saturation.

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 main target is the highest lateral acceleration (red dotted line in Figure 1) achievable associated with vehicle stability. In Table 1 all the dynamic conditions among with their correspondent dynamic targets are summarized.

3 BRAKE TORQUE VECTORING SYSTEM (BTV)

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.

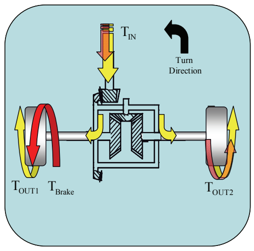

Figure 2: BTV general scheme. T_{IN}: input torque; T_{OUT1} and T_{OUT2}: resultant torque to each output shafts; T_{Brake}: braking torque.

Figure 2 represents a scheme of the BTV system intervention: 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:

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, imitating the function of an active differential. 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.

4 CAR MODEL AND SIMULATION ENVIRONMENT

A 15 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 implemented in Matlab/Simulink. The tires behavior has been described using MF-Tyre model version 2002 (Pacejka, 2003), taking into account combined slip effects. The model has been validated comparing its outputs to experimental data relevant to the passive vehicle equipped with a rear free differential.

In Figure 3 the understeer curve (steer angle vs. lateral acceleration) is shown for an ISO steering pad maneuver (ISO 4138:1996). The scaling is not reported in all figures in this paper because of confidentiality agreements.

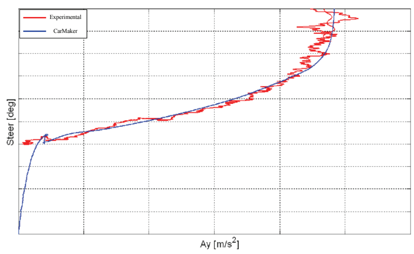

Figure 3: Steering pad constant radius. Understeering curve numerical vs. experimental.

Such a relative validation has allowed achieve a better confidence in the presented numerical results.

Due to the significance of a proper clutch stiction and slip phenomena modelling, the powertrain model has been developed using a mathematical approach appropriate for this kind of analysis (Cheli, Pedrinelli, Zorzutti, 2007).

5 CONTROL LOGIC ARCHITECTURE

In paragraph 3 the target of this project has been pointed out as the maintenance of the stability at limit and, above all, the vehicle performance improvement in regard to the same car equipped with a semi-active differential or an active differential.

The control logic is not based on modern control theory (LQR, etc.), but the simpler way of a feed-forward to guaranty a quick response and PID controllers to better adjust the overall algorithm output is chosen.

First of all the algorithm foresees that the car state has to be detected (Kakalis, 2009): the system, then, applies dedicated sub-algorithms, one for steady-state/step steer/power on and one for power off (Kakalis, 2009) which results the desired brake torque.

5.1 Steady State

As said in the previous sections, the resultant yaw moment should not lead to an oversteering condition. Therefore, the control system must work only when it can guarantee a sensible gain in vehicle performance. Because of that the feed-forward part is constituted of a 3D map whose values correspond to the maximum oversteering moment tolerable by the car in various adherence levels. The applied yaw moment should follow certain rules. At low speed and lateral acceleration the gain in terms of understeering gradient is narrow so that the driver shouldn't perceive a major handling improvement. On the other hand, at high lateral acceleration the gain in maximum lateral acceleration should be hugely influenced by the logic intervention.

Based on the 3D map, BTV is capable of generating a high asymmetry distribution (braking inner wheel) of the rear longitudinal forces due to the simultaneous action on brakes and throttle. However, such an extreme torque vectoring can generate an uncomfortable feeling (tank steering), so that a standard lateral torque distribution (LTD) was imposed on both the models (Figure 5) where, as limit case, the internal wheel is kept in free rolling condition.

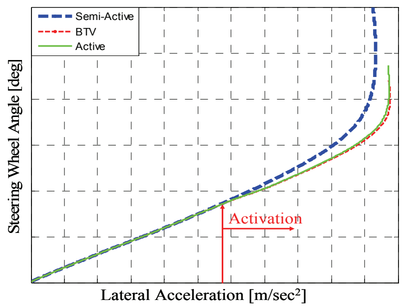

Figure 4: Understeer curve.

Figure 4 presents an example of an understeer curve for a fixed velocity of 100km/h, comparing the response of the same car equipped with semi-active differential, active differential and BTV. It can be easily noted that beyond 6 m/s^2 (activation threshold) the understeer gradient is reduced for both BTV and active differential; both the systems have produced a better exploitation of the frictional forces thus allowing to reach higher lateral acceleration with respect to the semi-active differential (+5%). The examination of Figure 6 suggests that this improvement is obtained with an increase of the sideslip angle.

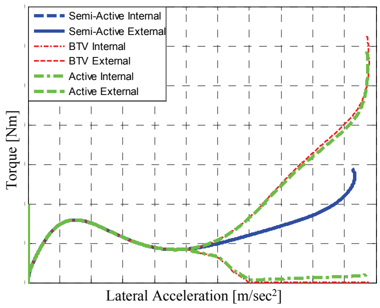

Figure 5: Lateral torque distribution.

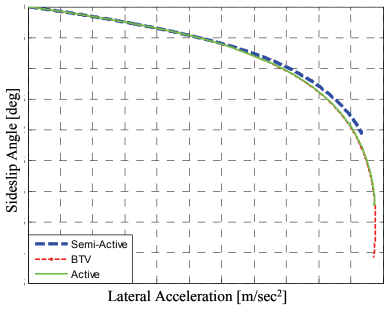

Figure 6: Sideslip angle.

5.2 U-turn (Power on)

The maintenance of the longitudinal acceleration thresholds under medium and high lateral acceleration imposes the need to combine two fundamental arguments: stability in the limit area and optimal traction.

As examined before the control logic of both BTV and active differential in steady-state cornering, generates a strongly asymmetric torque distribution (0–100%). However, if a simultaneous longitudinal acceleration is required by the driver, the external tire can't guarantee alone all the traction force and the lateral one without saturating and generating oversteer. Because of that, the transferred torque to the external wheel must be limited by changing the distribution ratio, i.e. the internal wheel should be progressively accelerated. This action will reduce the inwards moment whose amplitude is directly governed by the longitudinal dynamic state of the inner wheel. The acceleration of the inner wheel causes the longitudinal slip boost and thus the longitudinal force increase. It's important to not exceed the longitudinal slip peak (normally around 12% and 14%) to avoid the wheel spinning and a huge engine rpm increase. In case of BTV the progressive reduction of the torque distribution is necessary and it is particularly complicated also because an excessive braking action would dissipate a lot of engine power that could be used to accelerate the vehicle.

The optimization of the longitudinal slip is based on a PID controller. The error signal is given by the difference between the actual longitudinal slip and the optimal one (12%-14%).

As far as BTV is concerned, the controller directly commands the braking torque applied on the inner wheel while in the active differential regulates the outer clutch.

In order to compare the performance of the three models under power on conditions, an ad-hoc maneuver was designed (Figure 7), consisting of two parts: in the first one the vehicle enters a curve and progressively reaches steady-state conditions (Steady State phase) achieving maximum performance (maximum velocity and lateral acceleration). In this part of the maneuver both BTV and the active differential impose a 0-100% LTD ratio. It has to be underlined that, in order to extract meaningful conclusions, the driver model forces the three vehicles to follow the same trajectory.

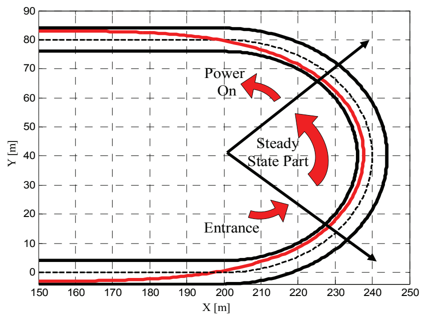

Figure 7: U-turn (Radius = 40m).

The second part (Power On phase) begins when the driver accelerates (full throttle) and exits the curve following the defined trajectory. During the steady state phase BTV and active differential clearly show their superiority in respect to the semi-active model by describing the fixed trajectory with a higher velocity (+2%).

As far as the transient phase (power on) is concerned, BTV accelerates several meters before the semi-active model and the active one. Any attempt, for both the vehicle equipped with semi-active and active differential, to accelerate before would cause an oversteering response and exit from the track.

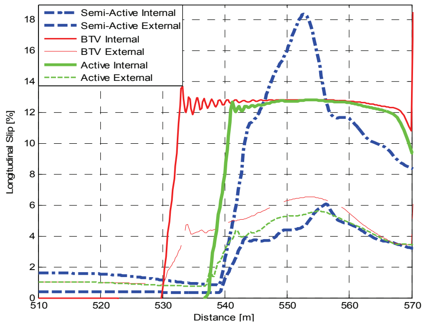

Figure 8: U-turn. Longitudinal slip on internal and external wheel vs. distance.

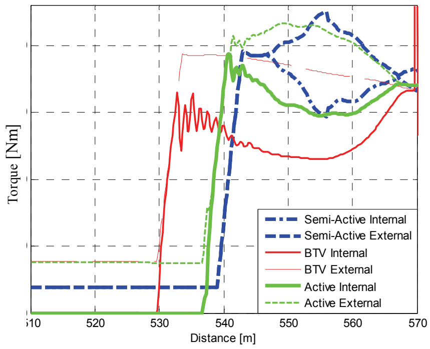

Figure 9: U-turn. Net torque on the real left and right semi-axle vs. distance.

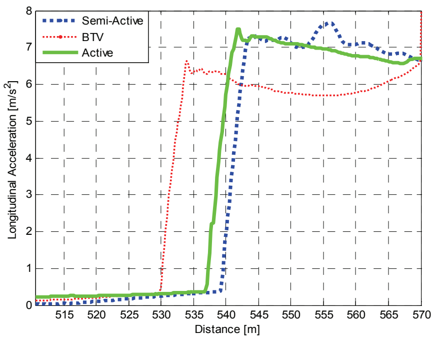

Figure 10: U-turn. Longitudinal acceleration during the exit phase vs. distance.

The need to follow the reference longitudinal slip (Figure 8) would produce an excessive drive torque transfer to the outer wheel (Figure 9) causing its saturation. The BTV yaw moment generation mechanism is instead more flexible since the torque on the inner wheel can be controlled without the need of transferring the same torque to the external one. Such property makes it possible to initiate the power on phase much earlier. Although both the semi-active and the active differential lead the vehicle to accelerate several meters after BTV, both the systems allow a better exploitation of the remaining longitudinal adhesion and achieve a higher longitudinal acceleration (Figure 10).

Judging by the distance history of the longitudinal velocity (Figure 11), BTV slightly improves the performance of the active differential and it presents a considerable advantage over the semi-active.

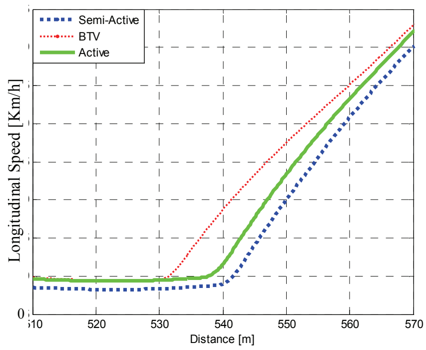

Figure 11: U-turn. Longitudinal speed vs. distance.

5.3 Virtual Race Track

As a last test, the performance offered by the three control systems was tested comparing their performance on an entire race track.

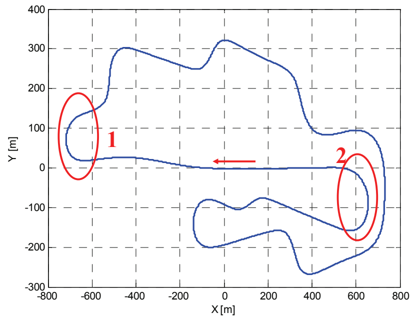

Figure 12: Selected race track.

The choice to validate the performance for all the three systems on the virtual track of Figure 12, showed the need to increase the robustness of their logic in order to extract more meaningful results. Such a test implies the fact that all models should have a common state-recognizing switch governed by the same principles and then the same power-off strategy in order to eliminate great trajectory

variations. The simulations were carried out in CarMaker™ environment; the virtual driver of IPG™ was chosen to perform the simulations with a driving style very close to the one of a real driver.
Active differential and BTV have been actuated by the same control logic previously presented for steady-state curve and power on transient; this implies that in steady-state the internal wheel does not transmit any traction force to the ground. Once power-on conditions is recognized, the optimization of the internal's wheel longitudinal slip will take place.

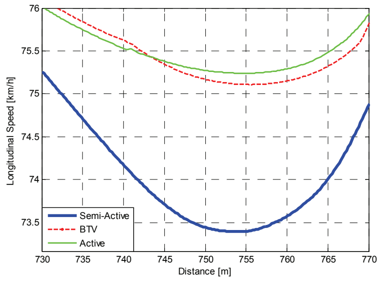

Figure 13: Longitudinal speed vs. distance in turn 1.

Considering a total lap time of approximately 140 s, BTV and active differential allow a reduction of 1.9% and 2.0% respectively when compared to the performance produced by the vehicle equipped with the semi-active differential. The time difference between the three systems can be explained by analyzing the dynamic performance in different circuit sections. By observing at the first turn speed profile (Figure 13) BTV and active differential achieve a longitudinal velocity 2.5% higher with respect to vehicle equipped with the semi-active differential.

6 TEMPERATURE ANALYSIS

6.1 Brake Temperature Estimation

Increased power dissipation produced by repeated brake actuations, might pose concerns around their temperature and efficiency; it is therefore required to estimate the expected temperature increase in the brake system to complete the feasibility analysis of the proposed concept. It has to be underlined that the authors feel to provide only a short description of the developed thermal model because its complexity and the assumptions taken into consideration would require a more detailed analysis which can be found in (Sabbioni, Cheli, 2008) and (Limpert, 1999).

The thermal model takes into consideration the heat transfer due to conduction between:

- the rotor and the braking pad;
- the braking pad and the caliper;
- rotor and disc's hub;
- disc's hub and wheel carrier;
- and to forced convection between:
- caliper, rotor and braking pad and the air;
- disc's hub and wheel carrier with the air;

The validation of the numerical model was carried out by using ten consecutive laps test results recorded on a race track using as a test vehicle the reference model equipped with the semi-active differential. The temperature was measured through a temperature sensor positioned in the braking pad.

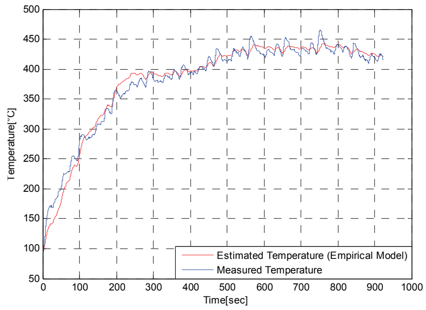

Figure 14: Comparison between measured and estimated braking pad temperatures.

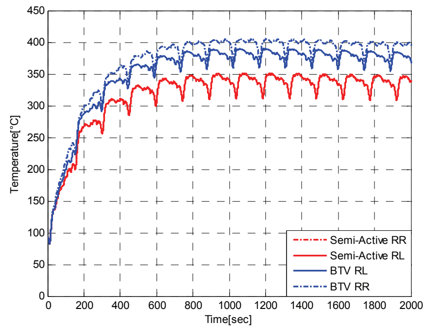

Figure 15: Brake pad temperature estimation.

Figure 14 shows the comparison between the temperature measured on the braking pad and the one obtained by the numerical brake model.
The brake model was fed with the data obtained through the simulation on the test track; this procedure allowed to estimate the discs temperature time history and thus evaluate the increased thermal load associated with the BTV logic. Figure 15 collects the results obtained from the control systems: the brake temperature gradually increases with time and reaches a mean operating temperature after about 5-6 laps. In terms of temperature, the

dissipated power differences presented in Figure 15 correspond to a disc's temperature rise of approximately 50 ^{o}C for the rear left brake and 75 ^{o}C for the rear right one. This temperature difference between the two models may be considered limited and tolerable since telemetry data on the real car indicated operating temperatures above 400 ^{o}C (Figure 15).

7 CONCLUSIONS

This paper presented the feasibility study of a system designed for the improvement of the handling characteristics of a sport vehicle based on the yaw moment control. The proposed system, named BTV, generates an asymmetric distribution of the longitudinal forces on the driving axle through an independent actuation of the brakes and a control of the throttle valve. As far as handling performance is concerned, BTV showed its superiority with respect to the semi-active differential and allows to get the same improvement provided by an active differential under several operating conditions. Besides this, BTV presents an important advantage related to its implementation on a real vehicle which would not require any additional electronic or mechanical component. On the other side, active differential still appears superior as far as the mechanism of generation of the yaw moment is concerned: BTV produces a torque difference by dissipating the energy supplied to one of the wheels in the form of heat, while the active differential simply attempts to reapportion the torque that is supplied to the wheels. The mechanism by which this is achieved - the friction clutch - still leads to some energy loss, but this is generally much lower than the energy dissipated in the brakes. The low energy consumption of the active differential gives it the potential to apply yaw moment control throughout the operating range of the vehicle.

The increased thermal solicitation of the brake system was also examined through a thermal model of brakes; according to the model results the expected increase of the temperature of the discs after a series of laps on a race track will not compromise the brake efficiency.

Obviously remains still in discussion the problem of the adherence level identification. This difficult task can be handled through the definition and the implementation of a self-governing recognizing algorithm which, based on the observation of the on board measured sizes, can replace the manual control regulation made by the driver which now is the implemented solution on the reference vehicle. A major step towards the adherence recognition can be considered the new generation of Cyber Tires, (Pasterkamp, Pacejka, 1997), (Mancosu and others, 2008).

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