AIAS – ASSOCIAZIONE ITALIANA PER L'ANALISI DELLE SOLLECITAZIONI
44° CONVEGNO NAZIONALE, 2-5 SETTEMBRE 2015, – UNIVERSITÀ DI MESSINA

AIAS 2015 - 568

A STUDY ON THE INFLUENCE OF A SEMI-ACTIVE DIFFERENTIAL ON THE VEHICLE BEHAVIOUR WITH A SOFTWARE-IN-THE-LOOP APPROACH

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Sommario

Nel campo dell'industria automobilistica il sempre maggior interesse per l'aumento della sicurezza dei passeggeri e, al contempo, delle performance ha portato ad investire ingenti risorse nello sviluppo di sistemi di controllo attivi. In questa memoria viene presentato un innovativo differenziale automobilistico semi-attivo, gestito per mezzo di un sistema di controllo elettro-idraulico, capace di modificare in maniera continua la distribuzione di coppia fra le ruote motrici.
Grazie a questa peculiarità, oltre ad un incremento della trazione, si ha la possibilità di influenzare il comportamento della vettura migliorandone l'handling. Il funzionamento è però fortemente dipendente dall'algoritmo di controllo adottato, perciò il principale obiettivo di questa ricerca è la verifica delle effettive capacità del sistema al variare della logica testata. I risultati presentati sono stati ottenuti utilizzando un approccio numerico avanzato conosciuto con il nome di *Software-in-the-Loop*.

Abstract

In the automotive field the growing interest for the enhancement of the passengers' safety and, at the same time, of the performances has led to invest resources in the development of active control systems. In this paper is shown an original semi-active car differential, actuated through an electro-hydraulic control system, which is able to continuously manage the torque distribution among the driving wheels. Thanks to this feature, in addition to an enhancement of the tractive capabilities, there is the possibility to influence the vehicle attitude improving the handling. As the functioning is strongly influenced by the control algorithm embedded, the main target of this research has been the assessment of its actual capabilities by testing different control logics. The results illustrated were obtained using an advanced numerical approach known as *Software-in-the-Loop*.

Parole chiave: semi-active differential, software-in-the-loop, vehicle dynamics.

1. INTRODUCTION

The automotive differential provides a fundamental function enabling an equal subdivision of the driveline torque between the two driving wheels and, simultaneously, allowing the latter to keep their own angular speed during turning manoeuvres. This last feature is essential to avoid the tyres scrubbing over the road surface.
However, as a secondary effect, it impedes an optimal split of the tractive force because the maximum value is always limited by the minimum value of friction available on the two wheels of the same axis. Consequently, the loss of traction is the cause of a vehicle control reduction that becomes very dangerous in case of high travelling speed.

In the last decades, the stability control topic has been faced and examined far and wide and several solutions have been proposed; the most considerable ones, tested and extensively used by now, act on the brake callipers of the four wheels independently and on the power unit.
Although these solutions have good functional features, the vehicle is slowed down and the efficiency level is reduced because of the energy waste; therefore, with the idea to equip race cars with the device object of this work, these two points are crucial.
Furthermore, as Fox M. and Grogg J. claim [1], their functioning tend to make a worse driving perception feeling a reduced vehicle control.
This paper will provide an overview about an electro-mechanic car differential, which should constitute an optimal solution to this issue. In particular, it will present the assessment of the actual possibilities of this device. The system was born with the primary goal to change the torque distribution between the two driving wheels in order to keep a high traction threshold; but at the same time, the idea was to make use of the force imbalance to generate a yaw moment that makes possible the modification of the lateral behaviour of the car.
In the first part of this document, the main system features are described in details. Then, the uncommon approach used for the assessment phase (known as *SiL - Software in the Loop*) is illustrated, whereas in the end the experimental campaign and the results are presented.

2. SAD – SEMI ACTIVE DIFFERENTIAL

The mechatronic device proposed is an automotive differential, electrically controlled by an ECU – Electronic Control Unit, which belongs to the semi-active family.
Its design originates from a classic self-locking, and therefore passive, differential supplied with friction discs. This, typically called LSD - Limited Slip Differential, has a structure similar to the structure of an open differential but with the addition of one or more disc packs (generally oil immersed) that constitute the internal clutch. All these discs are alternatively coupled with the differential cage and one of the half shaft through a splined coupling: for this reason they are constrained for the revolution but are free for the axial movements. The task of this clutch is the creation of a friction torque capable to modify how the torque is split between the driving wheels [2].
In the LSD the clutch engagement is *automatic*, thanks to the axial thrust exerted by proper cups that have a different shape depending on the structural layout. Thus, the system realizes the locking even in those conditions when usually it is not necessary. To illustrate this concept we may refer to a classic example regarding a car that is moving on a constant radius curve.
Until the lateral acceleration level is high, the vertical load on the internal wheel might not be enough to keep a good adherence: in these cases, the clutch locking is required in order to maintain a tractive effort. On the contrary, if the lateral acceleration is low, the vertical load on the wheels is well distributed on both sides: the car is in the adherence region and the engagement is not required. Rather, in some cases, the actuation is counter-productive and damaging due to the oversteering and understeering phenomena produced, which worsen the vehicle behaviour.
Another basic point refers to the locking level: in the LSD, once the inner clutch is engaged, the level is essentially fixed and is dependent on the ramp angle of the thrust cups, set in the design stage. Thus, the change of the locking value is possible only with an invasive intervention to replace a mechanical part of the device.
The introduction of such an electronically controlled device allows to overcome these two negative points: a progressive locking over a wide range of functioning (theoretically, the system should reach the complete locking conditions) is assured and the clutch actuation takes place only when it is positive for the enhancement of the driving conditions. For this purpose, to reach the best results in all the situations the use of a multi-level control logic is preferable.
The request to design this *in-house* system arises from the will to realize an appropriate device to be used on race cars, with distinguished features of lightness and small dimensions impossible to reach with similar *off-the-shelf* differentials.

2.1. Structure

The structure could be divided in three main parts: the right cartridge, the left cartridge and the housing. The latter, which is the central element, receives the engine torque by the outer crown fixed to this, and transfers it to the two solar gears; this is possible thanks to four planet bevel gears, which are connected to the central housing through the differential cross.
The two bevel wheels, each fixed to a half shaft and belonging to a different cartridge, generate an axial load due to their geometry; thus, in addition to radial bearings, each cartridge is equipped with an axial bearing to sustain this thrust.
The structure of the right cartridge is more complicate as it contains the actuation system. This is composed by a main piston, placed in a suitable chamber, which is pushed by the oil in pressure: in detail, an external actuator increases the pressure using another piston activated by a ball screw mechanism and a servomotor. The main piston is constituted by two parts: one in contact with the friction discs and the other with the oil chamber. An angular contact bearing, that allows the connection between these two parts, is necessary: the one that faces the oil chamber moves only axially while the other is also free of rotating around the revolution axis.
The axial thrust on the clutch pack generates a friction action useful to create a torque that contrasts the relative rotation between the two half shafts: more torque is always delivered towards the slowest wheel, with the purpose to reduce the speed difference.
According to [3], the torque distribution is described by:

$$\begin{cases} T_L = \dfrac{T_{Diff}}{2} - T_f \dfrac{|\omega_L - \omega_R|}{(\omega_L - \omega_R)} \\ T_R = \dfrac{T_{Diff}}{2} + T_f \dfrac{|\omega_L - \omega_R|}{(\omega_L - \omega_R)} \end{cases}$$

where:

T_L, T_R = the torque on the left and right half shafts

T_{Diff} = the input torque to the differential

T_f = the internal friction torque

ω_L, ω_R = the angular velocity of the left and right half shafts.

Actually, the behaviour is often described through *b*, which is the parameter that identifies the *locking ratio* of the differential, defined as:

$$b = \frac{|T_L - T_R|}{T_{Diff}}$$

Again, the ratio can be modified acting on the actuator pressure, once the geometry, the number of discs and the tribological characteristics of the friction material are designated.

A key point concerns the vehicle trim. If, in the first approximation, we may neglect the aerodynamic forces (which are limited at low speed), it is well known that the car exerts forces on the environment only through its tyres. Furthermore, an unbalanced distribution of force on the same axis inevitably generates of a yaw torque that modifies the vehicle trim: the idea is to operate properly on the torque distribution to control the yaw angle of the vehicle, as well as the traction level.
That said, it is easy to understand the considerable influence of the logic embedded in the control unit. For this reason, after a first and significant design stage of the mechanism, we focused the attention on the definition and the development of a suitable control logic, suitably realized for the first application on the Formula SAE car.

3. ASSESSMENT

The assessment of the actual system capabilities to change the vehicle attitude, which is the main topic of this work, has been carried out with a uncommon numerical approach that makes easy the analysis and the modification of the control logic.
In detail: the test vehicle was modelled in a multibody environment while the control logic was developed in a specific numerical computing environment. The technique used, generally known as *SiL – Software in the Loop*, expects a component (the ECU in this case) to be completely emulated by a proper software that is connected to the remaining part of the assembly (the car that is virtually emulated too).
In the last decades this method, evolved in the areas of software and control systems design, has rapidly spread in the mechatronic design area and, on account of this, it may be considered as a part of the development process called V-Cycle [4], [5]. The name originates from the shape of the structure, where the two branches of the V identify the design and the product assessment stages respectively.
The design process requires the designer to move from the whole assembly towards the single components definition, whereas the second stage expects to begin from the assessment of the single part (using a simple mono-dimensional model for instance) up to the tests of the entire system placed in its work environment. Thus, the integration level of the hardware parts with those virtually emulated is gradually increased. The method allows to proceed in a systematic way for the product design that is positive for the *time-to-market* reduction. Furthermore, not only time but even costs are lower because neither advanced nor expensive models are necessary.

3.1. Multibody modelling

For the tests, the reference vehicle is the first car that is going to be equipped with this special device. As said, it is the single seat, open wheels, rear wheel drive car for Formula SAE challenge, designed and developed in the school of engineering of the University of Florence.
During the work development, the vehicle at issue was not available yet, due to the last activities to be done. Therefore, for the multibody modelling we uniquely referred to the design documentation, trying to keep high standards of accuracy, above all for the mass distribution and the geometric characteristics of the suspension assemblies, which have great effects on the vehicle dynamics.

- Mass $[kg]$ = 200
- Moment of inertia $[kg \cdot mm^2]$ = Ixx: $2.5 \cdot 10^7$ Iyy: $1.1 \cdot 10^8$ Izz: $3.9 \cdot 10^8$
- Wheelbase $[mm]$ = 1600
- Front track $[mm]$ = 1200
- Rear track $[mm]$ = 1100

Referring to the tyres, specific considerations must be underlined: the tyres are fundamental to understand deeply the vehicle behaviour, because, as said above, are responsible for the contact forces generating process on the road surface.
Given that, the tyre modelling is not a secondary but a fundamental stage. In this work we decided to use Pacejka 2002 Magic Formula Tyre Model designed by prof. H.B.Pacejka [6]: this is a semi-empirical relationship specific for each tyre and characterized by a large number of coefficients. The method to extract them is quite complex and it is a prerogative of specialized companies; this is why we made use of coefficients published by one of this company, about the tyres that equip the car, which are *Hoosier 20.5/6.0 X13 R25B Racing Slick*. The Figure 1 shows the characteristic curves obtained with the following conditions: an inflation pressure of 12 psi, a longitudinal speed of 20 km/h, a vertical load of 650 N and no camber angle.

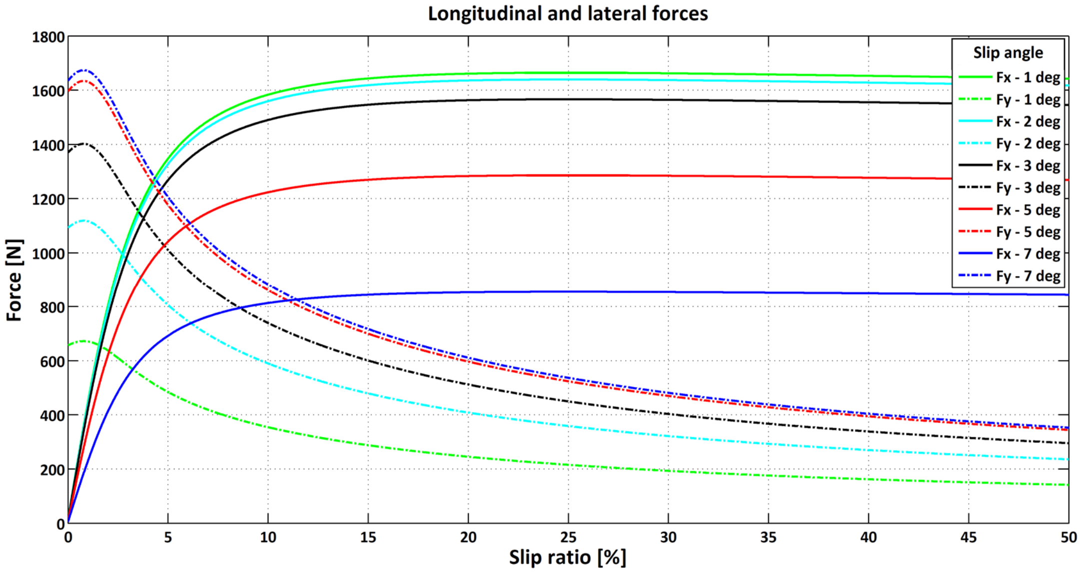

Figure 1 –Effects of the slip ratio and the slip angle on the longitudinal and lateral forces.

Referring to the differential model, we took the decision to adopt a simplified criterion: the main goal is indeed the evaluation of the influence, on the vehicle dynamics, of the torque distribution between the driving wheels and not the study of the device behaviour.
Therefore, the system is a purely mathematic model that follows the equations below:

$$2\cdot\Omega_{Diff} - \omega_L - \omega_R = 0$$

$$\begin{cases} T_L = +\dfrac{T_{Diff}}{2} + T_{Input\ Signal} \\ T_R = -\dfrac{T_{Diff}}{2} + T_{Input\ Signal} \end{cases}$$

where:

Ω_{Diff} = the angular velocity of the differential cage

$T_{Input\ Signal}$ = the artificial torque used to represent the torque distribution.

Once the modelling stage was concluded, given that the car had not been ready yet, the validation phase was not realized. However, we focused on a specific point, which is the mass distribution. In detail, taking into account the vertical wheel loads evaluated with the CAD model, we made use of a specific tool to reach the mass properties required by changing the mass of a multibody component (the chassis). When the vehicle is assembled, we are planning to realize experimental tests matching the actual results with those obtained with the numerical model emulating the same maneuvers. By doing so, it is possible to achieve a high level of fidelity in the model.

3.2. Control system

The ECU was programmed in another specific work environment: this is a high-level numerical environment that allows to develop models with block flow diagrams and that supports a model-based design method.
Actually, the model created is complex and includes several elements: both the high and the low levels of the control unit, and an artificial subsystem that converts the pressure signal in the friction torque of the differential.

This means that in input the on-board vehicle sensor signals are read, whereas in output the $T_{Input\ Signal}$ value is directly provided.
The list of the sensors, which belong to the control system created in the multibody model, is listed below:

- TPS (Throttle Position Sensor) = the opening percentage of the butterfly valve
- RPM (Run Per Minute) = the engine speed
- $\Delta\omega$ = the difference of the angular velocities between the rear wheels
- Gear = the transmission ratio engaged
- r = the yaw rate.

Please note that all the quantities evaluated during the numerical simulations are related to measurable values, thanks to the on-board sensors pack of the real vehicle.
The picture below illustrates the flow diagram of the system and the main components that define the model.

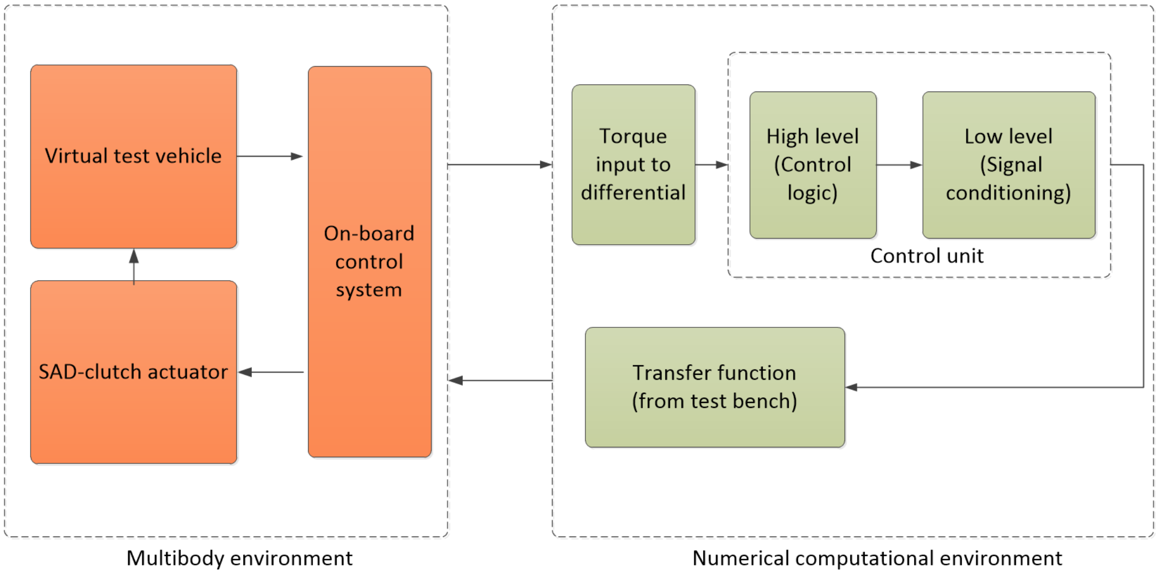

Figure 2 – The flow diagram of the *Software-in-the-Loop* model.

The first subsystem (*Torque input to differential*) is needed to estimate the input torque to the differential cage. The input signals are: the gear engaged, the opening percentage of the throttle valve and the engine speed, together with the information about the transmission ratios available, the transmission efficiency and the engine torque map. This evaluation is possible through a *look-up table* that identifies the engine functioning thanks to two input values: the engine speed and the opening percentage of the throttle valve. Clearly, this might be considered accurate enough only in steady-state conditions, whilst there is a certain error in the transient phase; however, we followed this procedure because this is how the real control unit works.

The second subsystem (*Control unit*), split into two parts only for convenience, is the real control unit. The first part identifies the control algorithm, which might be modified to emulate different functionings of the device. The other one, instead, gathers a set of common elements for all the algorithms that may be embedded. Among the main ones, there is a block for the dynamic saturation of the signal, which avoids the pressure value required to be higher than the maximum feasible, and a similar block that limits the increase/decrease speed of the signal. Another important element is the block that creates a *dead-zone* around zero to avoid the actuation beneath a certain threshold.
Furthermore, there is a module to convert the pressure signal in a friction torque value. Clearly, this is not realistic because the authentic output of the ECU is electric and corresponds to a certain target of

pressure that the actuator must exert. Again, we decided to act following this procedure because we had not exactly defined the tribological model that characterizes the contact behaviour of the discs.

The third and last subsystem here specified (*Transfer function*) has been added recently to take the real dynamic response of the actuation system into account, considering all the inertial, the elastic and the damping features. Indeed, the future developments expect the use of a *HiL - Hardware in the Loop* procedure necessary to test the real behaviour of the system components [7]. With this purpose in mind, the first specific test rig *(ATB – Actuator Test Bench)* has been designed and realized to evaluate the behaviour of the actuation system. In brief, only the left cartridge is accommodated in this test rig with its clutch, its external hydraulic circuit, the actuator and the control unit board.
Thanks to the tests executed at the test rig, we understood its attitude and we had the opportunity to characterize the dynamic response. Then, considering the system as time-invariant (we neglected all the mechanical effects of temperature on the mechanical part and on the working fluid) and linear for the typical working conditions, we managed to deduce a transfer function that characterizes it: this was embedded in the third subsystem.

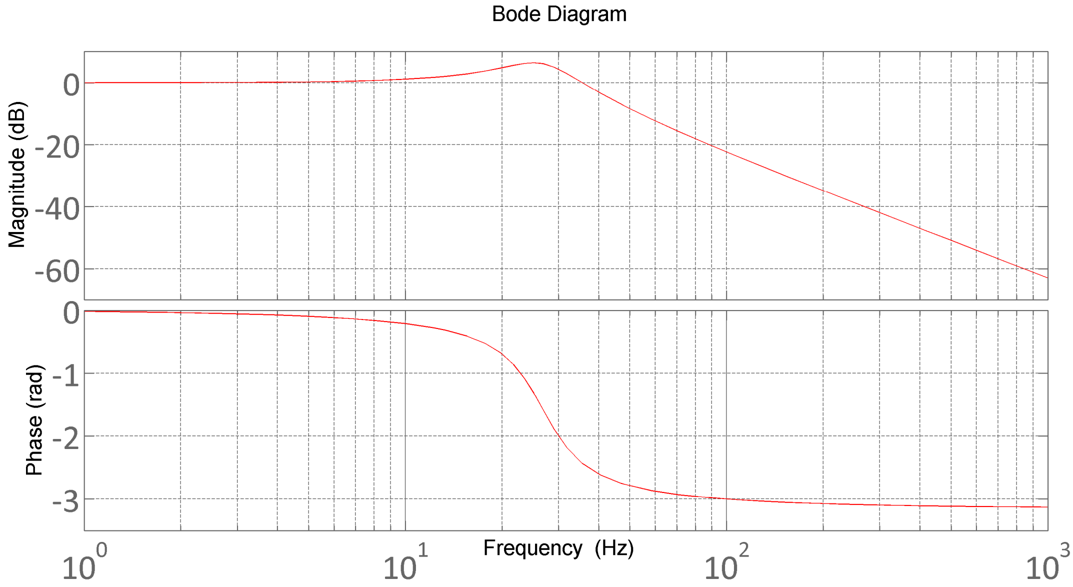

Figure 3 – Bode diagram of the test-rig transfer function.

In addition, each signal, related to sensors, was sampled using the same frequency of the data acquisition process of the sensors. Moreover, we used another block that allows to emulate the correct frequency at which the *serial* CAN-bus transmits the signals to and from the control unit.
This was necessary to evaluate the delay time of the actuation and the necessity of the implementation of an algorithm for the vehicle behaviour prediction.

3.3. Control logics

Regarding the model, after the creation of the basic structure, we focused the attention on the control logic to be implemented.
Initially we made the decision to let the device work as a classic self-locking differential; in detail, we picked two versions with different operating principles: a LSD belonging to the torque-sensitive category and a viscous differential belonging to the speed-sensitive category.
The first one, already described above, has an internal multi-discs clutch that is activated when the driving torque flows into the device: therefore, the friction torque generated contrasts with whichever difference of angular speed between the two wheels.
The second one, instead, is composed by a series of properly shaped discs immersed in a viscous liquid; thus, the locking torque is proportional to the speed difference between the half shafts, up to a certain level of saturation.

The different functioning might be highlighted in the following graph that shows the locking torque trend versus the difference of the angular speed of the half shafts. It is remarkable to note that, except for a slight initial peak due to *stiction*, the torque value is almost constant in the LSD.

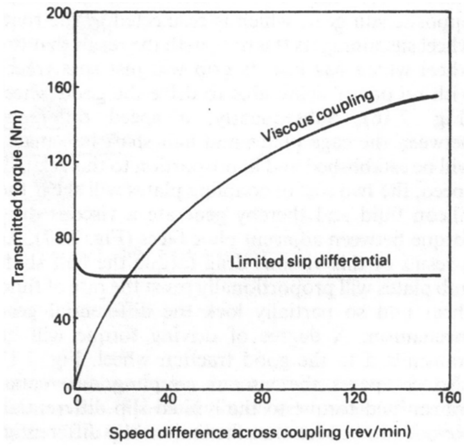

Figure 4 – Torque transmitted to wheel with respect to speed difference between half shafts [8].

Subsequently, we compared their functioning with that of a specific control logic developed for the device object of this work and described in [9].
In brief, this consists of a vertical architecture with two different intervention levels: while the bottom one has only the task to enhance the traction, while the top one employs the *torque-vectoring* technique to improve the stability through the yaw angle control. More in detail, in this paper only the results obtained with the first level are presented, which originates from the behaviour of a classic LSD, to which some improvements were made. In fact, as indicated in the previous paragraph, the LSD locking takes place whenever a difference of speed between the two driving wheels is present and if a determined quantity of power flows into the differential. Besides, it is not possible to operate on the locking level once the system has been designed and assembled; thus, in spite of an enhancement of the car traction, we may verify a reduction of the directional control.
On account of this, the proposed algorithm compares the sign of the angular speeds difference signal $(\Delta\omega)$, measured by the phonic wheels, with the sign of the yaw rate value $(\dot{\varphi})$.
Considering:

$$\Delta\omega = \omega_L - \omega_R$$

and the vertical axis of the vehicle's frame of reference pointed towards the ground, such that $\dot{\varphi}$>0 if the car is turning right, the clutch will be engaged only if the two signs are different.

	$\Delta\omega > 0$	$\Delta\omega < 0$
$\dot{\varphi} > 0$	*Disengaged*	*Engaged*
$\dot{\varphi} < 0$	*Engaged*	*Disengaged*

4. TESTING

As is so often the case, in this work the behaviour of the vehicle equipped with this device was evaluated performing both steady-state and transient tests. Indeed, generally the first metrics, that must be taken into consideration during the vehicle dynamics assessment, are generally deduced from steady-state tests. Then, after a correct and deep understanding of this behaviour, it is possible to move towards the analysis of the transient attitude with great care.
In both cases we referred to recommendations suggested by the norms. Here below, together with the test description, we will present the results of each manoeuvre with the most adequate metrics.

4.1. Steady-state test: ramp steer

According to [10], the characterization of the steady-state behaviour takes place through open-loop manoeuvres, where the vehicle control is independent from the driver's skills. Indeed, the trend of the control variables, such as steering wheel angle or throttle position, are defined in advance and are maintained independently from the vehicle response.
The norm prescribes three different tests: *constant radius, constant speed* and *constant steering wheel angle*. Theoretically, given the steady-state nature of the tests, the expected results are the same for all the three; in other words, it is sufficient to hold any one of the three main quantities (steering wheel angle, turning radius and vehicle speed), vary the second and measure the third.
Unfortunately, we might notice that the results are different and for this reason, in order to make some comparisons, the choice and the application of one test only is a good practice.
For this work we decided to run *Ramp steer* tests, where the car is kept at a certain value of speed and the driver acts on the steering wheel angle following a ramp with a proper slope (this is also known as *spiral* for the geometry of the vehicle trajectory).
The norm, composed to describe the passenger cars, supplies information about the values to be considered during the manoeuvre; despite of that, in our case the values were modified because the vehicle is a race car with different features in comparison with passenger ones.
The tests were run considering a constant longitudinal speed of 50 km/h with a linear actuation of the steering wheel: the slope of 1 deg/s is considered adequate to keep the car close to the steady-state conditions, avoiding the generation of dynamic effects.

4.1.1. Ramp steer test results

In this section the tests results are shown and discussed making use of proper and synthetic metrics. It is essential to highlight that, as it almost happens for the study of the vehicle dynamics, the quantities are not interesting for their specific values but, above all, for the relative comparisons. Given that, we will comment the results following this principle, bringing in comparison the three control logics introduced with the traditional open differential.
We may begin presenting the steering wheel angle versus the lateral acceleration trend, which typically is the common quantity used to describe the steady-state.

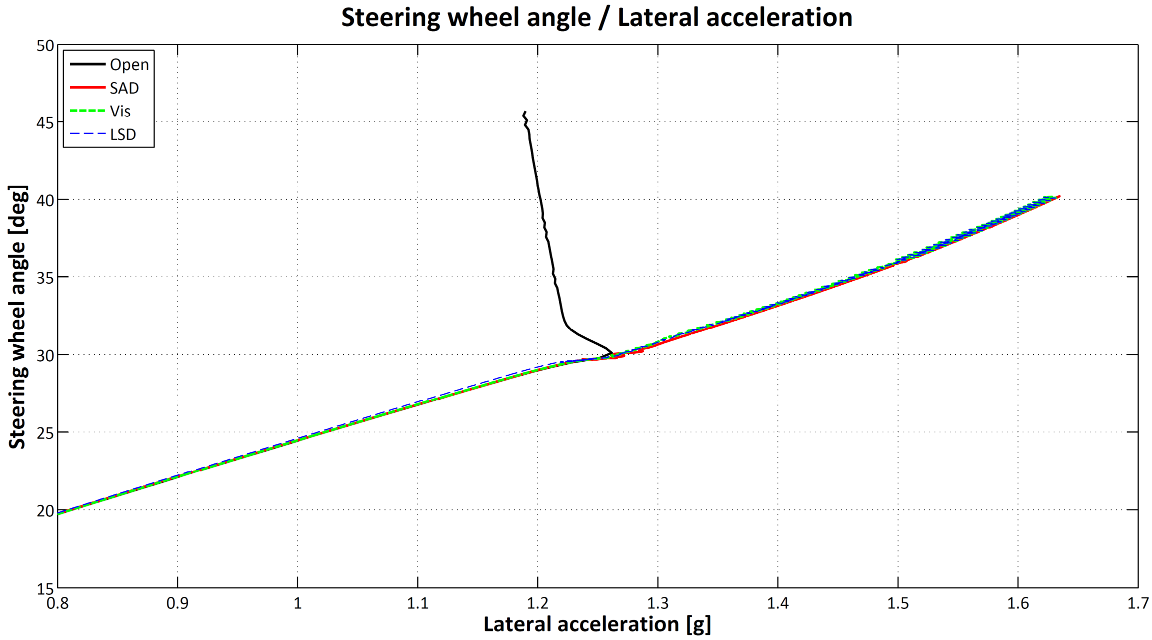

Figure 5 – Steering wheel angle versus lateral acceleration.

Unfortunately, in this case the information earned from the graph are few: until the test vehicle is moving in the adherence region and both the driving wheels have grip, all the curves are substantially overlapped.
Overcoming that condition the curve related to the open differential has a plateau caused by a dramatic reduction of the tractive capabilities. In other words, once the value of lateral acceleration is reached, only using an advance device it is possible to maintain the vehicle speed during the reduction of the path radius.
The SAD maintains the same increasing trend of the other two self-locking differentials as wanted. In fact, the critical lateral acceleration demarcates the zone where the self-locking differentials have an optimal functioning. In this acceleration interval we can not appreciate any difference between the devices because the boundary conditions (substantially the difference of the angular velocities) are demanded, thus the devices work at the edge of their possibilities with a similar behaviour.
Instead, different results between the devices would be expected in the first part of the test, where the acceleration is low, but this is not appreciable in this graph and the reason is likely linked to the power distribution. Each system, for its nature, operates in a different manner and distributes the tractive force differently, but this is not evident because the power required for the manoeuvre is very low. In fact, the small drag coefficient, the limited frontal area and the low speed, as well as the low values of rolling resistance of the tyres, entail a very low total resisting force. In order to consider this phenomenon, we may refer to a more proper graphic, which shows the power distribution to the wheels during the manoeuvre.

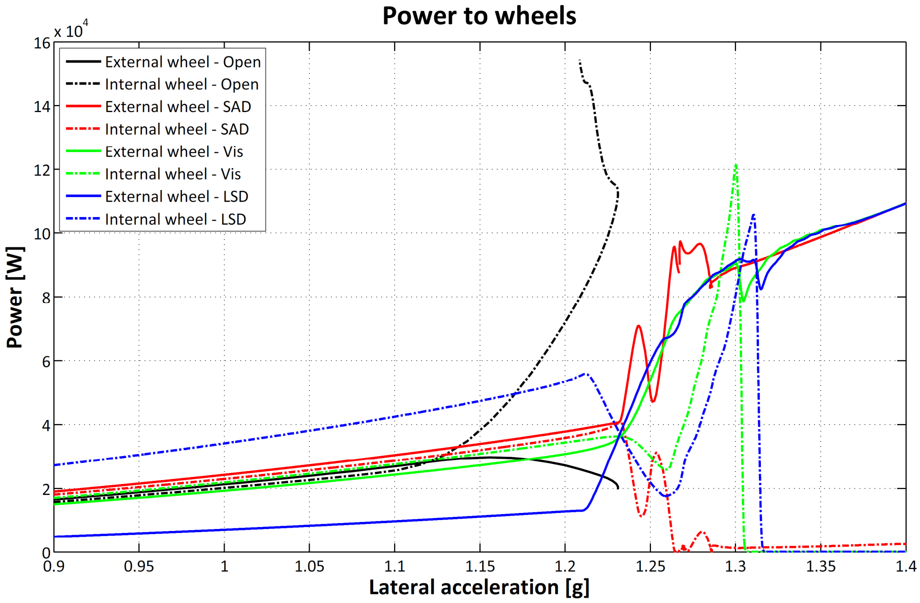

Figure 6 – Power to wheels versus lateral acceleration.

Observing it, we can validate the previous consideration. In the first part the three systems act differently: the SAD clutch is not actuated until the sign of the difference of the angular velocities changes, thus the torque is equally distributed as in open differentials. This means that the greater power is trasmitteted to the external wheel that has a greater speed. The other two devices, on the contrary, operate to minimize this speed difference, distributing a greater percentage of power on the internal wheel. From the comparison, the most abrupt behaviour of the LSD is here well noticeable.
In the second part, beyond the critical acceleration, the power distribution of the open differential is completely different from that of the others: the internal wheel starts spinning so the whole power flows through it and the traction is dramatically reduced. This is the key point: the SAD, as well as the other self-locking differentials, can oppose the wheel spinning, maintaining traction. The important aspect not yet investigated is about how this device influences the handling, above all when the lateral acceleration is low.
In these tests in fact, the driving forces were low and, furthermore, the tyres worked far from their limits, thus the little change of the longitudinal forces did not heavily influence the lateral force generation:

these are the reasons that explain a limited variation of the yaw moment and, in turn, of the vehicle dynamics. In the future developments we are going to analyse it, with the purpose to demonstrate all the SAD advantages, so far only theorized.

4.2. Transient test: sine sweep

The second test performed, characterized by another reference norm [11], is necessary to partially assess the lateral transient behaviour of the vehicle.
As in the previous case, the norm illustrates the possible manoeuvres, but these are not equivalent. Thus, each test allows to identify slightly different characteristics, considering the analysis of results in terms of time and frequency.
Typically, the most used ones are the *step-steer* and *sine-sweep* tests, but we focused the attention on the second one only. The explanation of this choice is related to the fact that, as we highlighted during the preliminary stages of the work, this test gives us more interesting indications about the transient phase.
The norm prescribes that this is performed over a smooth, clean and perfectly dry surface (with the highest level of grip available for tyres) with an initial velocity of 50 km/h. After, at least three seconds of straight line, the driver acts on the steering wheel following a sinusoidal signal and increasing the frequency from a minimum of 0.2 Hz to a maximum of about 2 Hz. The amplitude is set as the angle necessary to reach a lateral acceleration equal to 0.3g in steady-state conditions, to let the tyres work in the linear field.
These indications, related to passenger vehicles, were taken as reference but were modified for this project. Indeed, the test car is markedly different from a common passenger car for the geometric features, the mass, and the stiffness. Therefore, we decided to increase the frequency peak up to 6 Hz, with the amplitude equal to 10 deg .

4.2.1. Sine sweep test results

The study of the transient behaviour is a complex subject and several quantities have to be taken into account to have an overall understanding. Here below only two quantities are shown and discussed: for a better comprehension, in both cases, the metrics are presented, in magnitude and phase, in the frequency domain.

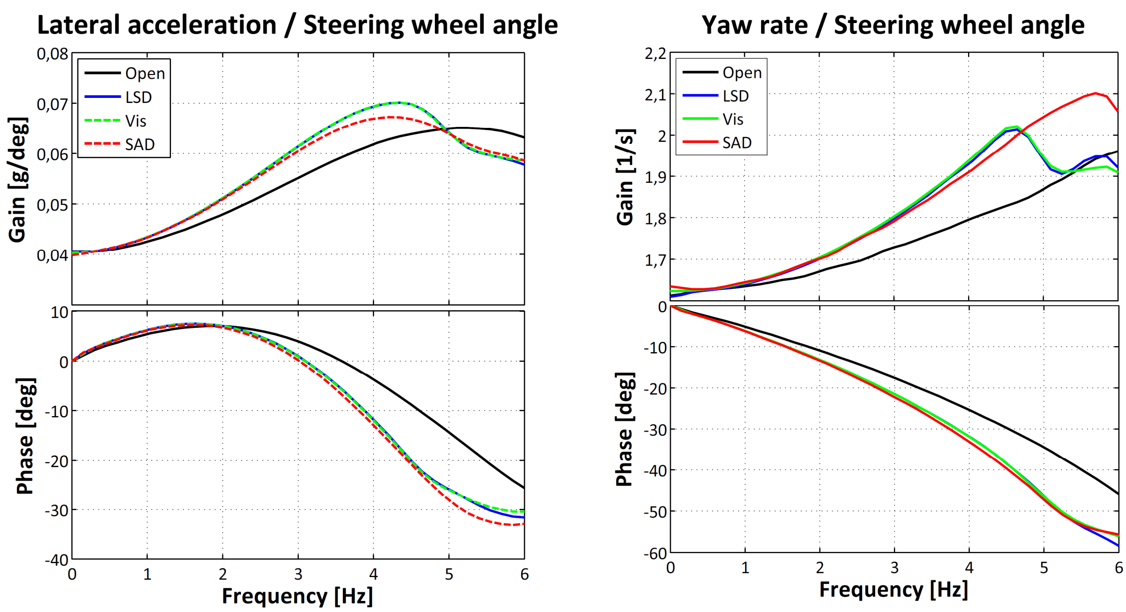

Figure 7 – a) Lateral acceleration versus steering wheel angle
b) Yaw rate versus steering wheel angle.

The first one is the ratio between the lateral acceleration and the steering wheel angle. Considering the magnitude, it is easy to see that the SAD and the self-locking differentials have a similar progress; in particular the peak (which corresponds to the resonance) is reached at a lower value of frequency compared to the open one. This would probably mean that the vehicle is more responsive at lower frequency, with advantages in terms of driving precision.
Typically, the peak is followed by a decreasing trend, as for a simple mechanical system with one degree of freedom, due to the high damping of the vehicle.
The phase supplies information about the delay of the dynamic response of the vehicle undergone the driver's input. In this case, its natural reduction, with correspondence to the resonance frequency, is limited facilitating the stability conditions.
The second plot shows the ratio between the yaw rate and the steering wheel angle, which is a more straight measure of the vehicle directionality. Focusing the attention on the magnitude, we notice that the SAD has a different trend compared to the self-locking ones, with the peak achieved at a higher value of frequency. In other words, this can be considered as the sign that the car has a better dynamic response over a wider range of frequency.
Summarizing, even with this test we assessed the positive qualities of the SAD, which allows to enhance the dynamic characteristics of the car keeping a high level of driveability.

5. CONCLUSIONS AND FINAL REMARKS

This work dealt with the assessment of the SAD capabilities to affect the vehicle behaviour, making use of a specific method called *Software-in-the-Loop*. For this purpose, we tested the car in two different manoeuvres that gave the possibility to characterize both the steady-state and the transient driving behaviour.
The data acquired were compared to those obtained with a classic open differential and with those obtained by using the control laws that emulate the functioning of two common self-locking differentials.
The investigation of the results firstly proves that the proposed device is able to positively affect and enhance the vehicle dynamics and, furthermore, that it supports an increment of the tractive capabilities as expected. Thus, the original target of the research to assess the system capabilities has been fulfilled.
This paper describes the first part of the work only, during which most of the energies and the resources were devoted to elaborate and to validate the working method. Future developments are expected to carry out a wide simulation campaign to outline the behaviour of a vehicle equipped with this device focusing specifically on the handling characteristics.

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