

Development of an e-LSD Control Strategy Considering the Evolution of the Friction Torque with the Wear Depth

Amedeo Tesi, Francesco Vinattieri, and Renzo Capitani
Universita degli Studi di Firenze

Claudio Annicchiarico
Meccanica 42

ABSTRACT

The Electro actuated Limited Slip Differential (e-LSD) can help increasing the dynamic features of the vehicle, but to implement a well designed control logic it is necessary a deep knowledge of the actual friction torque built up by the differential clutch. This work presents the development of such a control law that takes into account the wear depth progression. To carry out this task, an alternative method has been used to study the clutch discs engagement depending on the wear rate.

The method takes advantages from a mixed approach with a numerical and an experimental part. Using a general purpose block-on-ring test bench, the tribologic analyses were performed following the ASTM G77 standard; thus, the friction coefficient has been investigated in the contact between discs with molybdenum treatment and steel alloy discs, as well as its variation depending on the wear rate. The results were input in a numerical algorithm aimed at evaluating the friction torque of the clutch as a function of the pressure and the wear depth. The results, besides providing useful hints for the clutch design, were used to numerically assess the differential effects on a vehicle equipped with it. To accomplish this goal, a technique generally known as *SiL* - Software in the Loop was applied to multibody analyses. The paper is aimed at comparing the vehicle behavior using three different devices (open, self-locking and e-LSD) and performing standard (ISO) manoeuvres. The outcomes prove the advantages of the e-LSD in terms of handling, lateral dynamics and traction in comparison with the other solutions.

CITATION: Tesi, A., Vinattieri, F., Capitani, R., and Annicchiarico, C., "Development of an e-LSD Control Strategy Considering the Evolution of the Friction Torque with the Wear Depth," *SAE Int. J. Engines* 9(3):2016, doi:10.4271/2016-01-1136.

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 maneuvers. This last feature is essential to avoid the tires 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 adherence available on the two wheels. Consequently, the loss of traction is the cause of a vehicle control reduction that becomes very dangerous in case of high travelling speed. This is the reason why in the last decades the stability control topic has been faced and examined carefully with the aim to define a proper solution. The most incisive one proposed, which takes the generic name of ESC (Electronic Stability Control) system, is extremely effective. For instance, this is demonstrated by several studies, based on European statistics, which show a significant reduction of the road fatalities occurred in during the years [1].

ESC systems act on the brake calipers of the four wheels independently and on the power unit, thus, to get their good functional features in terms of safety, they slow down the vehicle and reduce the efficiency level. This are the main reasons that have impeded the diffusion on race cars [2]. Furthermore, as Fox M. and Grogg J. claim [3], their functioning worse the driver perception of the car, who feels a reduced vehicle control.

All the previous problems encouraged the design of the device proposed in this paper, which belongs to a class of active transmission systems specifically born for high performance and race cars. In order to achieve these objectives a numerical and experimental approach was chosen. The structural design of the e-LSD was the first step of this process, followed by an experimental campaign on the molybdenum-based treatment aimed at characterize its tribologic properties. Defined a reliable correlation between pressure and friction torque, the results were used to define a numerical model of the differential. Its implementation in a multibody environment permitted to perform standard maneuvers with the test vehicle, to assess the impact on the vehicle dynamics

e-LSD - Electro actuated Limited Slip Differential

The mechatronic device presented is an automotive differential, electrically controlled by an ECU - Electronic Control Unit, which belongs to the semi-active differential's family. Its design originates from a classic self-locking, and therefore passive, mechanism known as LSD - Limited Slip Differential.

Structure

The structure can 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 one fixed to a half shaft and belonging to a different cartridge, generate an axial load due to their geometry; hence, 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 and the internal clutch. The actuation 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. Thus, in order to allow a relative movement, an angular contact bearing is necessary. The internal clutch is formed by a discs' pack (oil immersed). All these discs are alternatively coupled with the differential cage and the half shaft through a splined coupling: this means that they are constrained for the revolution but free for the axial movements.

The goal of the clutch is the generation of a friction torque that contrasts the relative rotation of the two half shafts: more torque is always delivered towards the slowest wheel, with the purpose to reduce the speed difference and to avoid a wheel spin. In addition to this, an unbalanced distribution of force on the same axis inevitably generates a yaw torque on the vehicle that modifies its attitude: the idea is to operate properly on the torque distribution to control the yaw angle of the vehicle, as well as the traction level [4].

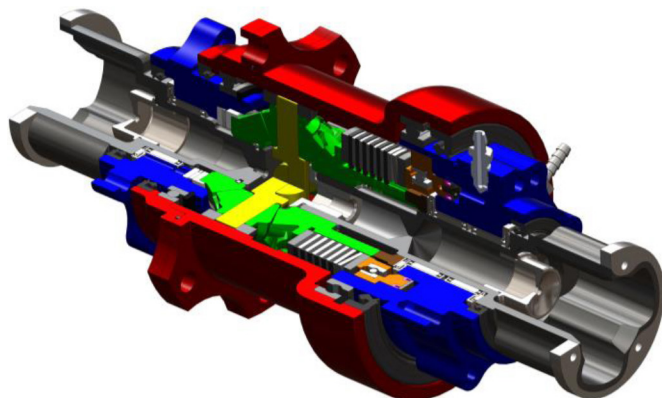


Figure 1. Cutaway of the e-LSD structure.

Control Logic

The great advantage of this system lies in the possibility to modify the functioning, and in turn the vehicle behavior, only acting on the control logic. In this work, the results obtained with an *ad-hoc* logic have been compared with those obtained by means of a LSD and an open differential.

In the LSD the clutch engagement is automatic, thanks to the axial thrust exerted by the internal cups. Unfortunately, if a determined quantity of power flows into the differential, the device locks whenever a difference of speed between the two driving wheels is present, even in those conditions in which it is not necessary. Moreover, once the inner clutch is engaged, the locking level is essentially fixed depending on the design specifications. Both these effects may cause a reduction of the directional control in spite of an enhancement of the car traction.

The introduction of a custom control logic allows to overcome these two negative points: a progressive locking over a wide range of functioning is assured (the system can be configured to reach even the complete locking conditions) and the clutch actuation takes place only when it is positive for the enhancement of the driving conditions [5]. This originates from the behavior of a classic LSD but, as improvements, avoids the unwanted activations. The proposed algorithm introduces the *WS* parameter to identify the wheel that is spinning.

$$WS = \text{sign}(\omega_L - \omega_R) \cdot \text{sign}(r)$$

where $\text{sign}(x)$ is the function providing the ratio between the signal x and its modulus $|x|$, (ω_L, ω_R) are the angular speeds of the left and right axle respectively and r is the yaw rate of the vehicle.

Assuming the vertical axis of vehicle's frame of reference pointed towards the ground, such that r is positive if the car is turning right, *WS* is equal to 1 when one wheel starts spinning, otherwise it assumes -1. The final relation is expressed as below:

$$T_L = \max[0, WS] \cdot PID(\omega_L - \omega_R) \cdot T_{\max}(p) \quad (1)$$

where T_L is the locking torque provided by the internal clutch, *PID* is a Proportional Integral Derivative controller and $T_{\max}(p)$ is the maximum locking torque available depending on the input oil's pressure.

Clutch Characterization

The relation (1) entails the knowledge of the internal clutch behavior depending on the different working conditions, but the large number of parameters obliges the use of a proper approach. Since a dedicated test-rig for the clutch was not available, a specific approach was elaborated, which considered both an experimental and a numerical part. In detail, the experimental stage allows to examine the friction coefficient and the wear trend, whereas the numerical stage allows to process the results to obtain the friction torque trend.

Experimental Tests

The experimental tests were carried out using a block-on-ring test bench following the ASTM G77 standard [6], where a block was pushed against a rotating ring. Obviously, both the elements replicated the material characteristics of the differential discs, with the aim to guarantee a structural equivalence of the contact. To obtain the same surface hardness (of about 450 HK), each component was quenched following [7]. Molybdenum treatment application was made directly on the specimens by means of APS - Air Plasma Spray technique. All the test campaign was carried out in a lubricated domain using a synthetic oil.

Test Description

The tests were made using a Metrocom "PIN RING" 640 block-on-ring test bench. The main input parameters (ring rotational speed, normal load and oil temperature) were set trying to replicate the working conditions usually present in an e-LSD during its use.

Table 1. Block-on-ring input parameters

Ring rotational speed [m/s]	0.39
Normal load [N]	130
Oil temperature [$^{\circ}C$]	95 $\div$ 100

During each test, the friction load and the wear were acquired using two linear position transducers: the friction load is acquired measuring the deformation of a flexure hinge rigidly connected to the block and the wear is obtained measuring the vertical displacement of the block. In addition, the oil temperature was acquired using a thermostat and two thermocouples: one to measure the oil temperature in the tank and the other the temperature upon the heating plate. A continuous and simultaneous monitoring permitted to keep both the temperatures at a constant value of about 95 - 100 $^{\circ}C$ during the tests. The campaign was carried out following ASTM G77 standard for conforming contact and the test duration was fixed in one hour of work. It is important to highlight that before the test campaign other two steps were required to prepare the specimens: the cleaning process using acetone and the weighing using a scale with a resolution of $1.10^{-7}kg$. The cleaning process is repeated at the end of each test, using an ultrasonic apparatus, in order to remove all the oily residue attached to the specimens' surfaces and, in turn, to facilitate the following weighing, with the main purpose to evaluate the weight loss.

Results

The wear rate evaluation method used for the blocks complies with the assumption written in the ASTM G77 standard, which makes use of specific equations to evaluate the worn volume of each block. It is possible to explain this choice because, for little components as the blocks, a weight loss evaluation could be inaccurate due to the possibility of a material transfer from the steel to the molybdenum surface. Furthermore, as it is possible to see in [figure 2](#), a plastic deformation occurred on the contact surface of the blocks, so a weight loss analysis could mislead the wear rate evaluation.



Figure 2. Plastic deformation of a block.

On the contrary, for the rings wear rate evaluation, it was possible to measure the worn volume as the difference between the rings weight before and after each test. This approach is suitable for those materials with a high hardness (like molybdenum), where an evaluation of the worn track dimensions could not be so accurate due to the very low amount of worn material reached after a complete test. The equation used for volume loss evaluation is the following one:

$$V_r = \frac{m_{lr}}{\rho_r}$$

where V_r is the volume loss by the ring [m^3], m_{lr} is the mass loss by the ring [kg] and ρ_r is the density of the molybdenum [kg/m^3]. Again, following ASTM G77 standard, the equation used to define the friction coefficient is:

$$f = \frac{F}{W}$$

where f is the friction coefficient, F is the friction load [N] and W is the normal load acting on the contact surfaces [N].

Data obtained from tests' campaign were acquired using a Picolog data logger. According to the pictures below, an oscillation in the data acquisition was observed; for this reason, a Butterworth Lowpass filter has been applied to obtain an average trend of the wear and the friction coefficient ([figs. 3-4](#)). Both the friction coefficient and the wear rate values reported are normalized on their maximum value.

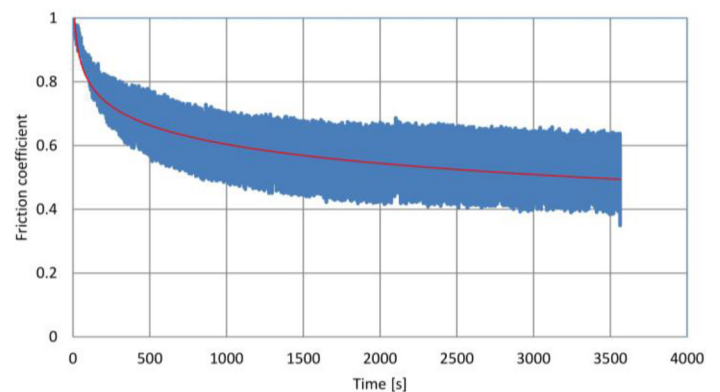


Figure 3. Friction coefficient trend.