A NEW CONTROL STRATEGY FOR A SEMI-ACTIVE DIFFERENTIAL (PART I)

Federico Cheli[1], Michele Giaramita[2], Marco Pedrinelli[1], Germano Sandoni[3], Gian Claudio Travaglio[4]

[1]*Politecnico di Milano*
Via La Masa 34, 20156 Milano, Italy

[2]*Ferrari S.p.A. – Gestione Ind.*
Via Abetone Inferiore 4, 41053 Maranello, Modena, Italy

[3]*MAGNA STEYR Fahrzeugtechnik AG & Co KG,*
Liebenauer Hauptstrasse 317, A-8041 Graz, Austria

[4]*Consultant of Ferrari S.p.A.*

Abstract: while Vehicle Dynamics Control systems usually operate on the engine torque and on brake pressures, new automotive applications try to use semi-active or active differentials in order to optimize the torque distribution on the wheels for traction maximization, driving comfort, stability and safety of the vehicle. The system presented in the paper comes out from the cooperation of Ferrari, MAGNA STEYR and Politecnico di Milano in the development of a semi-active differential. In the paper a description of either the physical layout of the system, its capabilities, and of the control software is given. Experimental results on an existent vehicle are shown.

Keywords: Differential, Vehicle, Control, Software, Logic, Automotive.

1. INTRODUCTION

In the recent past new active mechatronic systems have been introduced on the cars to help the driver, see (Van Zanten, *et al.*, 1995): ABS, ESP, Traction Control, AMT etc. that act on engine, clutch, gearbox, brakes and recently steering. Transfer-case and controlled differentials have been introduced in few applications, more over for the centre differential in four wheel driven vehicles, such as some Sport Utility Vehicle as BMW X3. The application presented in the paper has the aim of influence the dynamic behaviour of a rear driven sportive vehicle by controlling the locking torque of the differential. The conventional free differential is a mechanism that let the driven wheels to assume a different speed in turning condition, thus guaranteeing a uniform distribution of the driving torque. The lack of this mechanism is that, in not uniform conditions of adherence of the wheels (μ-split), it can't bring higher torque to the wheel with higher adherence with the effect of a really poor traction of the vehicle. This problem has been mechanically approached with passive devices that contrast the relative motion of the driven shafts, named self-locking differentials. This mechanisms influence the turning behaviour of the vehicle, due to the relative motion of the wheels imposed by kinematics in these conditions. In fact, in presence of a difference of wheel speeds, a locking action of the differential transfers torque from the fastest to the slowest one. This improves the traction because the wheel with higher adherence gets higher torque, although it generates asymmetric distribution of the longitudinal contact forces which translates into a Yaw moment acting on the vehicle, as shown in Fig. 1.

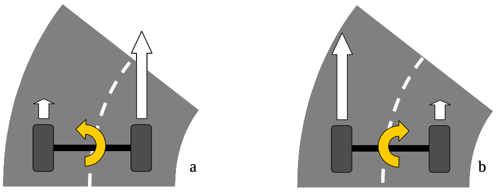

Fig. 1. Influence of locking torque on vehicle dynamics at low lateral acceleration (a) and high lateral acceleration (b).

At low lateral acceleration, or low driving torque, the inner wheel is slower: the torque transferred by the action of a self-locking differential would change the contact forces causing an understeering effect, negative for the performance and the driving feeling of the vehicle (Fig. 1a). At high lateral acceleration, and high driving torque, the load transfer, due to acceleration, can let the inner wheel spin. In this case (Fig. 1b) a locking action of the differential would transfer torque to the outer wheel, improving vehicle performance and traction, but generating an oversteering effect that can affect vehicle stability. The effects of this phenomena on vehicle dynamics are clarified in the understeering diagram shown in Fig. 2, where is reported the steer angle as a function of lateral acceleration in steady-state conditions, referring to a generic free differential vehicle and to a self-locking differential vehicle. The free differential lets the vehicle be stable, but not able to increase lateral acceleration, thus increasing the steer angle. The self-locking differential lets the vehicle reach higher lateral acceleration values but it leads to the spin of the car. The area between the two lines shows the intervention field of a semi-active controlled differential between the oversteering self-locking vehicle and the stable, but with less performance, free differential.

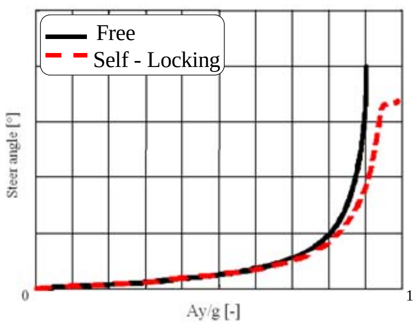

Fig. 2. Generic understeer diagram: influence of the Free and Self-locking differentials and controlled differential intervention field.

Due to the potential of this kind of controlled differential it has been decided to develop the controlled system presented in this paper. The subjects involved in the development of the system were Ferrari as the project leader, with its vehicle dynamics experience, MAGNA STEYR, with the development of the mechanical device (SAD) and of the inner control loops, and Politecnico di Milano with the Vehicle Dynamics Control logic.

The project is basically composed of a semi-active differential system, where the electronic control system can decide the locking torque of the differential, but not the direction of the torque transferred, as better explained in the following sections. The research proceeded through the development of a vehicle model, used in control logic definition and in numerical validation of algorithms. According to V Model software development process, see (Wünsche and Elser, 2004) and (Schäuffle and Zurawka, 2003), a phase of rapid prototyping and experimental tests has been conducted, till the reach of the target performances.

The paper presents a detailed description of the system and a general overview of the SW architecture. In the following sections a more detailed description of the most important modules (Actuation Layer, Vehicle Controller Layer) is given. Experimental results for the inner control loops and of interaction between all the system components will be presented. The second part of the paper (Resta, et al., 2005) will describe the development procedure of the Vehicle Dynamics Logic and its components. The experimental results of the whole system will be discussed from the vehicle dynamics point of view.

2. SEMIACTIVE DIFFERENTIAL DESCRIPTION

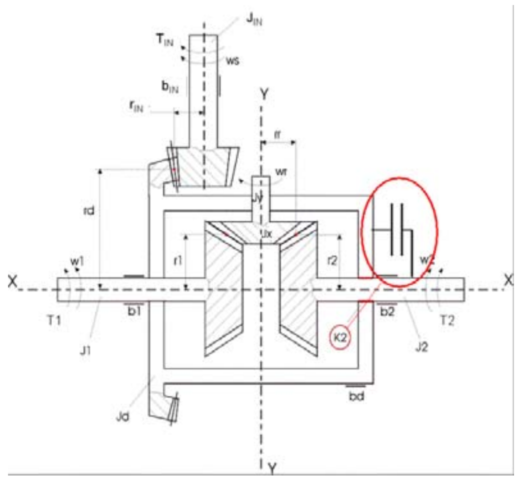

Fig. 3. Mechanical scheme of a differential.

In Fig.3 is shown a simplified scheme of a differential, used for understanding purposes. The Semi Active Differential (SAD) adopts friction plates, indicated with K2, inserted between the differential box and an output shaft. This friction is activated by an electro-hydraulic actuator in order to lock the differential in a wide range of operating conditions. The input driving torque, transmitted by the gearbox to the differential, is named T_{IN}, while the torques at the output shafts are defined as T_1 and T_2. The speeds of the output shafts are named ω_1 and ω_2, the rotation speed of the input shaft is named ω_{IN}, while the rotation speed of the differential box is defined as ω_D. The kinematical relationships between these quantities are the Willis ratio φ and the bridge ratio r:

$$r = \frac{\omega_{IN}}{\omega_D} \quad ; \quad \varphi = \frac{\omega_D - \omega_1}{\omega_D - \omega_2} = -1 \Rightarrow \omega_D = \frac{\omega_1 + \omega_2}{2} \qquad (1)$$

In this application the Willis ratio is fixed as φ = -1: the differential box rotation speed, ω_D, is always the average of the output shafts speeds. In steady state conditions, if there is no action on the friction plates, the output torques T_1 and T_2 are related to the input torque T_{IN}, by the following equations:

$$T_1 = r \cdot T_{IN} \cdot \frac{1}{1-\varphi} = \frac{r \cdot T_{IN}}{2} \qquad T_2 = r \cdot T_{IN} \cdot \frac{\varphi}{\varphi - 1} = \frac{r \cdot T_{IN}}{2} \qquad (2)$$

This condition of the system is the same as the free differential, which uniformly distributes the torque to the output shafts.

When an action on the friction plates is present a torque T_f is transferred from the fastest output shaft to the slowest one. Making the hypothesis of a coulomb friction between the plates, the torque generated by the friction plates, according to equation (1), can be defined as:

$$T_f = |T_f| \cdot sign(\omega_D - \omega_2) = |T_f| \cdot sign\left(\frac{\omega_1 + \omega_2}{2} - \omega_2\right)$$
$$= |T_f| \cdot sign(\omega_1 - \omega_2) \qquad (3)$$

The absolute value of the transferred torque is a function of the pressure applied at the clutch (P_d), the temperature ($T°$) and the friction coefficient between the plates (μ):

$$|T_f| = f(P_d, T°, \mu) \qquad (4)$$

Basing on this definition, adopting the convention of a positive value of T_f when $\omega_2 < \omega_1$, it is possible, in steady-state conditions, to evaluate the output torques T_1 and T_2:

$$T_1 = (r \cdot T_{IN} - T_f)\frac{1}{1-\varphi} = \frac{r \cdot T_{IN}}{2} - \frac{T_f}{2}$$
$$T_2 = T_f + (r \cdot T_{IN} - T_f)\frac{\varphi}{\varphi - 1} = \frac{r \cdot T_{IN}}{2} + \frac{T_f}{2} \qquad (5)$$

Observing the equations (3) and (5) it is easy to understand that this kind of system can only control the amount of torque transferred T_f, in other words its absolute value, while the direction of the transferring action is established by the system state (ω_1, ω_2). This is the reason why this system is called Semi-Active differential.

A proper hydraulic system (shown in Fig. 4) controls the pressure on the friction plates P_d in order to regulate the amount of torque transferred T_f from one wheel to the other. The supply pressure P_l is generated through an electrical motor and a pump, then the oil is stored in an accumulator, in order to stabilize the high pressure circuit. The pressure of the oil moves a piston that creates friction between the plates; such pressure P_d is controlled by a Pressure proportional valve driven with a PWM technique. Pressure sensors in the piston and in the accumulator are available in order to control the supply pressure P_l and the locking force $F_d = P_d \cdot A_d$, where A_d the area of the piston.

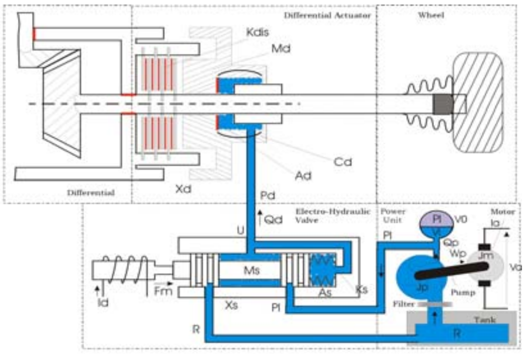

Fig. 4. Electro-hydraulic circuit that controls the Semi-Active differential.

3. SOFTWARE ARCHITECTURE AND DEVELOPMENT PROCESS

The structure of the SW architecture, reported in Fig.5, is basically composed of three parts:

- Basic Software
- Actuation Layer
- Vehicle Controller Layer

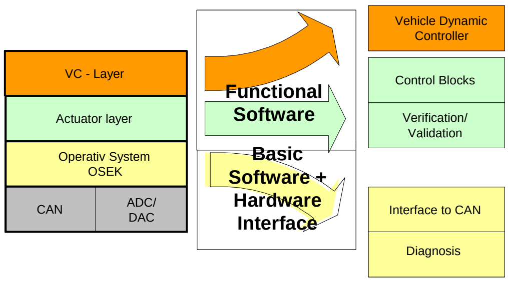

Fig. 5. Software Architecture.

The Basic Software is an interface to the HW (Sensors and CAN) that makes the operations of the Functional Software almost independent from the HW implementation. Actuation Layer and Vehicle Controller (VC) layer are the main parts of the functional Software and they will be described in details in the following sections.

The SW development process was accomplished in accordance with the well-known V-model see (Wünsche and Elser, 2004) and (Schäuffle and Zurawka, 2003), shown in Fig. 6. After the definitions of systems requirements a first design of the control algorithms is done in an off-line simulation (all the components are developed in Matlab/Simulink environments). The Vehicle Controller algorithms have been developed through co-simulation with a validated vehicle model, as stated in the second part of the paper (Resta, et al., 2005). A following rapid controller prototyping phase (with Micro Autobox of DSpace) allows to analyze and optimize the controller calibration in the vehicle using real time data. Afterwards the series code from the control algorithm is generated automatically (with the tool TargetLink) in order to work on standard production ECUs. Hardware In the Loop (HIL) tests and final validations in the cars are necessary in order to control all the failure conditions.

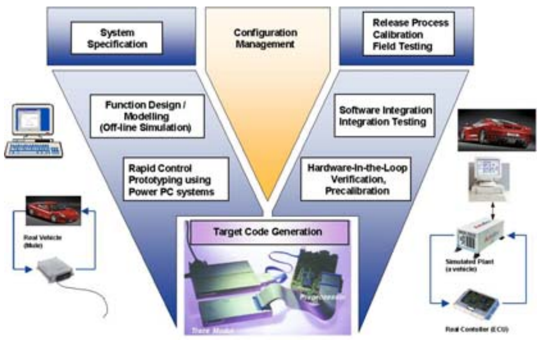

Fig. 6. V Model: SW development process.

3.1 *Actuation Layer.*

The main blocks that compose the Actuation Layer are shown in Fig 7. Basically the module 'Torque Controller' converts, in open loop, the torque reference $T_{f,ref}$, that come from the Vehicle Controller Layer (VC Layer), in a pressure reference Pd_{ref} for the 'Pressure Controller'.

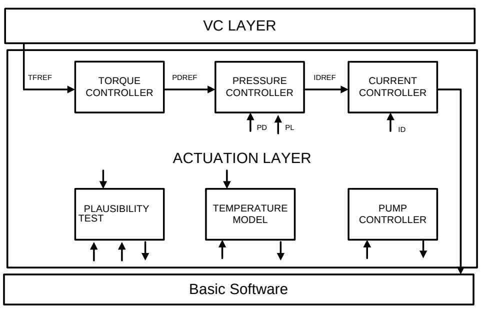

Fig. 7. Actuation Layer of the SAD SW.

The Pressure Controller is basically the heart: it is composed of a PID controller that uses the measured pressure in the actuator *Pd* and in the accumulator *Pl* in order to change the proportional, integral and derivative gains, whose expressions are:

$$Kp = f_1(Pd, Pl) \quad Ki = f_2(Pd, Pl) \quad Kd = f_3(Pd, Pl) \tag{6}$$

where $f_1()$, $f_2()$ and $f_3()$ are proper nonlinear functions that consider that the dynamic behaviour of the valve changes in dependency of this two pressures. In particular for higher pressure *Pl* the mechanical internal loop, of the pressure proportional valve, make the dynamics of the output pressure *Pd* faster.

The Current Controller is a second internal loop of control that sets up the proper input at the PWM driver of the valve.
The Pump Controller is a simple On Off relay, with hysteresis, that directly drives the pump in order to keep the correct supply pressure *Pl*.
Plausibility and Temperature Models are modules that check the correct behaviour of system, even if the Basic Software prevent and controls the main hardware error actions and conditions.

3.2 *Vehicle Controller Layer.*

Aim of the Vehicle Controller Layer is to evaluate the vehicle state and, basing on it, to calculate the reference target torque $T_{f,ref}$ for the Actuation Layer. The main blocks that compose the Vehicle Controller Layer are shown in Fig. 8. The 'Fail Safe Supervisor' receives the signals coming from the CAN bus of the car directly from the Basic Software. These are information coming either from the sensors installed on the vehicle (i.e. Lateral Accelerometer, Yaw Rate Sensor, Steer, Wheels Speeds, etc.) either from the other ECU on board, like ESP, Engine ECU, ABS or Gearbox. Dealing with these signals, the software makes some plausibility checks and reads also the results of the checks made by the Basic Software or even published on the CAN bus by the other ECUs on board: for example it is able to recognize the presence of a flat tyre. If some fault is found, the system evaluates its impact on the functionality of the logic and takes the subsequent decisions.

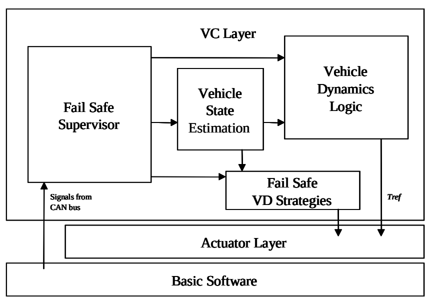

Fig. 8. Vehicle Controller Layer of the SW.

For example if some sensor fail is detected, the software verifies the opportunity to substitute the corrupted signal with an estimation based on other sensors. Referring to the goodness of the estimation itself, the system evaluates the consequences on the Vehicle Dynamics Logic behaviour and takes the proper decisions. In particular it can let the Vehicle Dynamics Logic work using the estimation just evaluated, or it can turn off some components of the Vehicle Dynamics Logic, giving the highest priority to the vehicle safety, or, if it is necessary, it can turn it off and adopt one specific Fail Safe Vehicle Dynamics Strategy, present into the 'Fail Safe VD Strategies' block. This block basically applies open loop strategies based upon specific subsets of the signals normally available on board. As an example: if the lateral acceleration and the Yaw rate measurements are not available nor reliable the system can adopt a specific Fail safe strategy based on the remaining signals. In this way the possibility of a shutdown of the system is clearly reduced.

The 'Vehicle State Estimation' block is responsible of all the estimations needed for the correct functionality of the Vehicle Dynamics control logic. Some of them are used only if necessary, i.e. a fail condition detected by the 'Fail Safe Supervisor', while others, like the linear speed of the car, are always evaluated. No complex and often unreliable estimations, such as sideslip angle, are used in the system. Basically the software has to evaluate the vehicle speed basing on the wheel speed sensors present on the car: since it deals with a rear driven vehicle the best estimation comes from the average of front wheel speed. The driving torque is evaluated by means of the information coming from the Engine ECU. All the information coming from sensors, as Lateral Accelerometer, is properly pre-processed before being used as inputs quantities in the Vehicle Dynamics Logic.

3.3 Vehicle Dynamics Logic.

The true core of the system is the 'Vehicle Dynamics Logic' block. As shown in Fig. 8, inside the Vehicle Controller Layer, this block evaluates the reference locking torque $T_{f,ref}$ the Actuation Layer has to realize. The aim of the controller is to achieve different targets such as improving vehicle stability, traction and performance thus improving the driving easiness. These targets are often contrasting, as clarified in the introduction. This fact led to a logic based on the driving condition of the vehicle.

Basically the 'Vehicle Dynamics Logic' block is supposed to reach the optimum compromise between traction and stability in different driving conditions, such as:

- Steady-State, Power On and Power Off turning conditions;
- Straight-Line and μ-split conditions.

The system recognizes the driving condition of the vehicle by means of inputs from the driver, such as Steering Wheel Angle δ and throttle position or the Engine Target Torque (T_{Engine}), published on the CAN bus by the Engine ECU (eventually corrected by other control systems, as ASR, or ESP), and of handling quantities as Lateral Acceleration A_y, Vehicle Speed V_X and Yaw Rate ψ_{Rate} communicated by the 'Vehicle State Estimation' block. When a different driving condition is detected the system handles the transition between the different algorithms (and targets) with the aim of giving the car a smooth behaviour, in other words predictable by the driver.

A detailed description of the logic will be presented in the second part of the paper (Resta, et al., 2005), while here the general outline is discussed.
The general structure of the logic adopted foresees a feedforward and a feedback component, whose parameters vary in function of the driving condition detected. The reference locking torque passed to the Actuation Layer is composed by the combination of the values calculated by these components.

$$T_{f,ref} = T_{feedforward} + T_{feedback} \tag{7}$$

Feedforward component ($T_{feedforward}$). The system evaluates a first attempt reference torque as a non-linear function of vehicle state, whose parameters depend on the driving conditions.

$$T_{feedforward} = f\left(V_X, A_Y\right) \tag{8}$$

The feedforward component, moreover, fixes the threshold of difference of wheel speeds, used by the feedback component as a reference. Depending on driving condition the system adopts different nonlinear functions of car state, driving torque and driver input.

$$\Delta V_{X,threshold} = f\left(V_X, A_Y, \dot{\psi}, T_{Engine}\right) \tag{9}$$

The functions consider the typical nonlinearity of the vehicle. In particular approaching the limits of adherence the car need a higher locking torque for Power Off turning conditions and a smoother action in Power On conditions. These functions have been defined through numerical simulations with a validated vehicle model. More detailed information concerning the adopted procedure to find out these functions and their characteristics can be found in the second part of the paper (Resta, et al., 2005).

Feedback component ($T_{feedback}$). It exploits the actual vehicle state and adapts the feedforward torque ($T_{feedforward}$) to it. In most driving condition it is basically a PID controller where the error ε is based on the quantities evaluated in feedforward ($T_{feedforward}$, $\Delta V_{X,threshold}$). The controller proportional and integral gains depend on the error ε itself and its integral.

$$Kp = Kp(\varepsilon) \qquad Ki = Ki\left(\int \varepsilon\right) \tag{10}$$

In this way a nonlinear, smooth behaviour of the controller is achieved in order to deal between the challenging targets of the system.

3.4 Experimental Results

In this section the measured experimental results, relatively to the inner control loops and of the general system behaviour, are shown.

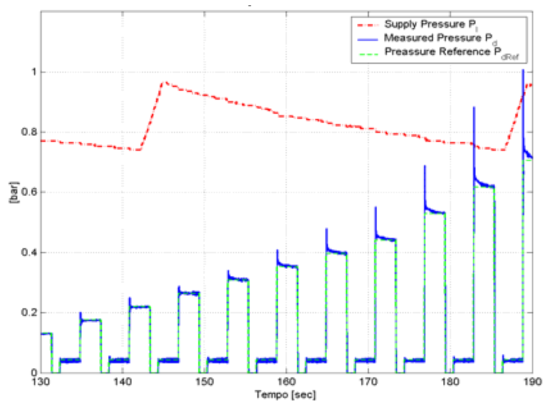

Fig. 9. Behaviour of the Pressure Control Algorithm at different reference torque steps.

In Fig. 9 is reported the dynamic behaviour of the actuator, in terms of supply pressure (*Pl*) and of measured system pressure (*Pd*), at different reference pressure (Pd_{ref}) steps. The nonlinear behaviour of the actuator, which is faster with higher steps, is evident. As described in Section 3.1 these nonlinearities are due to different pressure values P_l in the accumulator and to the dynamics of the valve. Despite all this issues, in the figure is shown the good behaviour of the proposed controller.

This system has been developed and implemented on a rear driven sport car, for the experimental phase. In order to show the influence of the car state on the performance of inner control loop, in the following figures the experimental data acquired directly on the vehicle are shown. In Fig. 10 time histories referring Steering Wheel δ, Vehicle Speed *Vx* and Lateral Acceleration A_Y are shown. Basing on these signals the system evaluates the reference threshold of difference of wheel speeds $\Delta V_{X,threshold}$ and the reference locking torque $T_{f,ref}$.

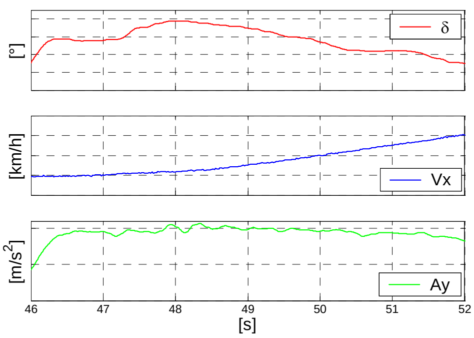

Fig. 10. Steering Wheel δ (a), Vehicle Speed V_X (b), Lateral Acceleration A_Y (c).

In Fig. 11 (a,b) the time histories of the reference and actual values of the difference of wheel speed and the reference $T_{f,ref}$ and the applied T_f value of locking torque are shown. The feedback controller behaves correctly beginning its action only when ΔVx is higher than threshold reference value. In this way it doesn't cause any understeering. The response time delay of the actuator is quite negligible compared to the dynamics of the vehicle.

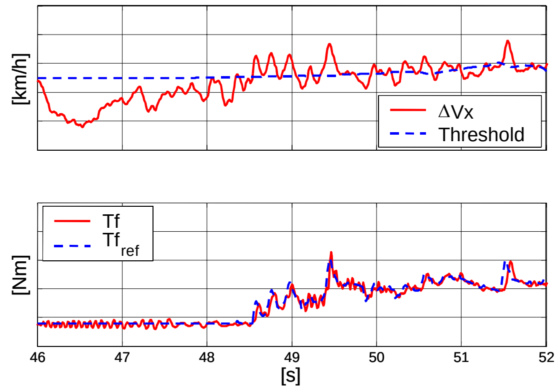

Fig. 11. Actual and reference threshold values of ΔVx (a) and of Locking Torque (b).

4. CONCLUSIONS

In this part of the paper it is discussed the structure of the proposed control system for a semi-active differential. The aim of this work is to optimize the wheel torque distribution, in order to improve both traction and vehicle dynamics. The main software layers, Actuation and Vehicle Controller, have been analysed, showing experimental results concerning the capability of the system in following the references fixed by the Vehicle Controller. In the second part of the paper (Resta, et al., 2005) a deeper analysis of Vehicle Controller structure and of the procedure adopted in its targets definition is conducted.

REFERENCES

Pedrinelli, M. and A. Zorzutti (2002), *Sviluppo ed implementazione di una logica di controllo per il differenziale di una vettura sportiva*, Degree Thesis A.A. 2001-2002 Politecnico di Milano (in Italian).

Resta, F., G.Teuschl, M.Zanchetta and A. Zorzutti (2005), A new control strategy of a semi-active differential (Part II), *Paper presented at the 16th IFAC WORLD CONGRESS*

Sandoni, G. (1999), *Modellistica dinamica e controllo di una frizione per F1*, Degree Thesis A.A. 1998-1999, Università degli studi di Modena e Reggio Emilia (in Italian).

Sandoni, G. (2002), *Modelling and control of a car driveline*, PhD Thesis A.A.2001-2002, Università degli studi di Modena e Reggio Emilia.

Schäuffele, J. and T. Zurawka (2003), *Automotive Software Engineering*, Verlag.

Van Zanten, A.T., R. Erhardt and G. Pfaff (1995), VDC, The Vehicle Dynamics Control System of Bosch, *SAE Technical Paper Series*, no.**950759**.

Wünsche, M. and J. Elser (2004), Model based development of Cruise Control for Mercedes Benz Trucks, *International Automotive Conference, Stuttgart Germany*.