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FIAT Auto – Alfa Romeo Performance Competence Center Handling Ride & Brake

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ABSTRACT

The paper presents the research activity managed to investigate the dynamics of a 4WD vehicle equipped considering drivelines with different layout. The procedure developed required to conceive an on purpose simulator to compare performance through virtual experimentation. Drivelines mechanical main characteristics and performance increasing due to control strategy were evaluated. Preliminary road test were performed with a single driveline layout, to evaluate simulation reliability and limits. The paper presents the 4WD vehicle simulator, the main equations applied to model open, torque sensing and limited slip differentials, some preliminary road test results showing torque sensing driveline performance.

INTRODUCTION

A Four Wheel Drive (4WD) vehicle can be designed according to general criteria like, e.g., increased longitudinal acceleration and safety due to a better directional behavior, that usually characterize the 4WD vehicle dynamics in compare with a Two Wheel Drive (TWD). Nowadays the driveline components available to carry out a driveline for 4WD are a number and, considering a same nominal typology, the characteristics can vary deeply by one Supplier to another one. Such differences permit to establish the desired performance choosing adequately the driveline components. The relationships about driveline characteristics, suspensions, tires, steer, operating conditions, driver behavior and so on, are so complicated for an 4WD vehicle that is unavoidable to design 4WD considering virtual experimentation. The paper presents how were modeled open, torque sensing and limited slip differentials, to consider the torque transmitted from the engine to the wheels depending on the driveline layout. Both torque sensing and limited slip differentials required a different, very accurate model of the transmitted torque through friction to satisfy the necessary precision required by virtual experimentation objectives.

Finally, the activity managed on the described subject was fundamental to evaluate the possible advantages following the overlapping of 4WD functionality with those of active systems devoted to control the vehicle dynamics like, e.g., Vehicle Dynamics Control (VDC). The results obtained are coherent with experimental tests and permitted to evaluate the main advantages of each driveline considered and the performance dependence on driveline component characteristics.

VEHICLE MODEL

The vehicle model was carried out merging different subsystems models either specially conceived or, eventually, coming from general purpose codes to cope with simulations aims. Generally driver, driveline layout, engine, steering system, braking system were developed on purpose. It was preferred to model separately such subsystems to apply provisions to them devoted to, e.g., change layout, simulate control strategies, modify the actuation systems and so on. Moreover, it was necessary to apply a special model of torque transmitted through friction to describe the transient behavior of open, torque sensing, limited slip differential. Simulations were carried out using Matlab/Simulink environment. The road test validation of the model was carried out considering stationary (ramp steer), transient (step steer) and frequency (sweep steer) maneuvers. The results obtained were generally satisfying. Figure 1 presents the main simulator window (a) and an example obtained comparing experimental and simulated sideslip angle during a sweep steer (b). The vehicle considered was a sedan prototype.

DRIVELINE MODELS

In the following are presented the driveline mathematical models considered to compare 2WD e 4WD vehicle performance due to different driveline layout:

- 2WD (front/rear, in the following *open*)

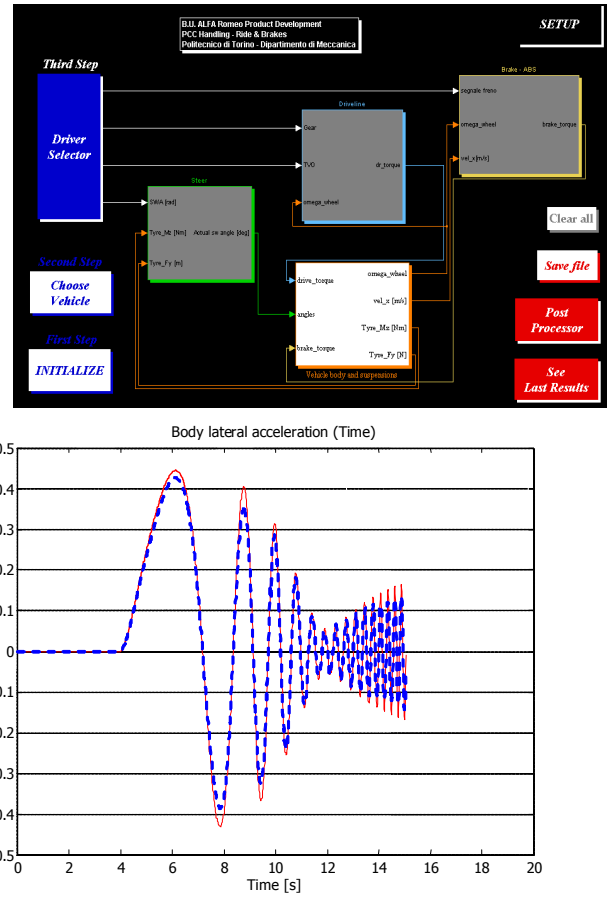

Figure 1. (a) Vehicle Model (b) Frequency (20° - 120km/h) Experimental data (dashed blue) and simulation result (red): high frequency response, obtained after the 12th second of simulation time, are not shown since the comparison is not appreciable due to figure dimension.

- 4WD central torque sensing (in the following *torsen*)
- AWD central limited slip differential (in the following *hang-on*)

At first it will be described the mathematical model adopted for each driveline, successively it will be presented how the driveline layout can be modeled considering each described differential with other common elements like axles, transaxle, engine, tires.

DIFFERENTIAL OPEN AND LOCKING

The efficiency η is defined as

$$\frac{T_{in}}{T_{out}} = \frac{1}{\eta} \qquad (1)$$

where T_{in} and T_{out} are respectively the differential input and output torques. The η is considered the efficiency between the two output shafts, using the open differential like a simple conventional bevel gear. Such efficiency is fundamental to define the differential locking effect. The differential split ratio ρ is the ratio between the two output shafts, without locking effect:

$$\rho = \frac{T_4}{T_5} \qquad (2)$$

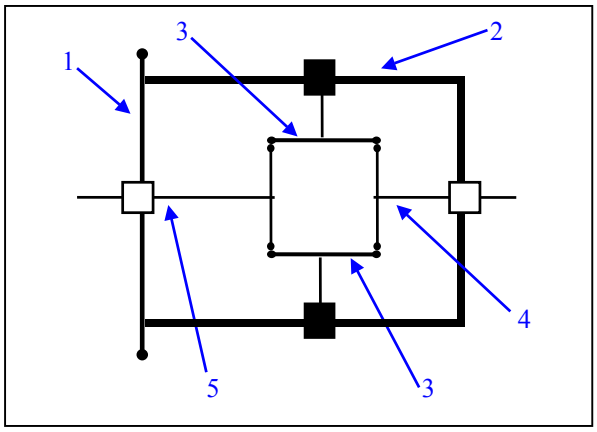

Figure 2. Open differential schematic: (1) Ring Gear, (2) Housing, (3) Pinions, (4) Right Gear and Axle, (5) Left Gear and Axle. Black squares mean that pinions are locked to housing, while white squares mean that axles unlocked to housing and ring gear.

The combined effect due to the interaction between split ratio ρ and efficiency η can be described locking the differential ring. In such condition if the output shaft 1 is used to supply power and the other one 2 as output, considering the efficiency, is:

$$T_5 = (T_4 / \rho) \cdot \eta \qquad (3)$$

The supply power is dissipated due to friction. The locking effect so generated introduces a torque bias that also in so-called open differential can be usually be observed around the 10% of power supply.

<u>Open and locking mathematical model</u>

Defined

T_p ring gear (planetary) torque

I_p ring gear inertia

$\dot{\Omega}$ ring gear acceleration

it is:

$$T_p - I_p \dot{\Omega} = T_2 + T_1 \qquad (4)$$

$$T_p - I_p \dot{\Omega} = T_2 + T_2 \cdot \frac{1}{\eta} \cdot \rho \qquad (5)$$

$$T_p - I_p \dot{\Omega} = T_2 \cdot \left(1 + \frac{1}{\eta} \cdot \rho\right) \qquad (6)$$

The equation describing the torques applied to output shafts depending on the torque applied to ring gear are:

$$T_2 = \frac{T_P - I_P \dot{\Omega}}{1 + \frac{1}{\eta} \cdot \rho} \quad (7)$$

$$T_1 = \left(T_P - I_P \dot{\Omega} \right) \cdot \frac{\frac{1}{\eta} \cdot \rho}{1 + \frac{1}{\eta} \cdot \rho} \quad (8)$$

The efficiency η variation depending on differential dynamics provokes the torque ratio between the axles. Such dynamics is described in the following torque sensing mathematical modeling.

TORQUE SENSING DIFFERENTIAL

The following characteristic parameters are defined:

- Torque Bias Ratio (TBR)

$$TBR = \frac{T_{hi}}{T_{lo}} = \frac{1}{\eta} \quad (9)$$

where η is the efficiency between the two output shafts. T_{hi} and T_{lo} are respectively the higher and lower torques transmitted when the torque sensing works partially locked.

- Locking Effect (LE)

$$\%LE = \frac{\Delta T}{T} = \frac{T_{hi} - T_{lo}}{T_{hi} + T_{lo}} \quad (10)$$

considering that

$$\%LE = \frac{TBR - 1}{TBR + 1} \cdot 100 \quad (11)$$

it is

$$TBR = \frac{LE + 1}{1 - LE} \quad (12)$$

- Differential split ratio ρ

The differential split ratio is defined, for example considering a central torque sensing differential, as the ratio between the front and the rear torques without locking effect

$$\rho = \frac{T_{front}}{T_{rear}} \quad (13)$$

- Torque Distribution Ratio (TDR)

$$TDR = \frac{T_{hi}}{T_{lo}} \quad (14)$$

The relationship between TBR and ρ is generally determined by the road conditions of front and rear wheels. In first approximation, the torque distribution ratio is considered equal to the product between TBR and ρ and is assumed constant. TDR parameter describes the relationship between the torques applied to the output shafts when the torque sensing differential presents a ρ different by 1.

The following equation describes the torques differential balance around the output shafts axes:

$$T_p - I_p \dot{\Omega} = T_{hi} + T_{lo} \quad (15)$$

Considering the torque sensing parameters formerly described is:

$$T_p - I_p \dot{\Omega} = T_{lo} \cdot TDR + T_{lo} \quad (16)$$

$$T_{hi} = T_{lo} \cdot TBR \cdot \rho \quad (17)$$

$$T_p - I_p \dot{\Omega} = T_{lo} \cdot TBR \cdot \rho + T_{lo} \quad (18)$$

$$T_p - I_p \dot{\Omega} = T_{lo} \cdot (TBR \cdot \rho + 1) \quad (19)$$

$$T_{lo} = \frac{T_p - I_p \cdot \dot{\Omega}}{1 + TBR \cdot \rho} \quad (20)$$

$$T_{hi} = T_{lo} \cdot TBR \cdot \rho \quad (21)$$

Describing the TBR depending on the LE is:

$$T_{lo} = \frac{T_p - I_p \cdot \dot{\Omega}}{1 + \frac{LE + 1}{1 - LE} \cdot \rho} \quad (22)$$

$$T_{hi} = T_{lo} \cdot \frac{LE + 1}{1 - LE} \cdot \rho \quad (23)$$

Consequently, the equations describing the output shafts torques depending on torque sensing parameters are:

$$T_{lo} = (T_p - I_p \cdot \dot{\Omega}) \cdot \frac{1 - LE}{LE \cdot (\rho - 1) + \rho + 1} \quad (24)$$

$$T_{hi} = (T_p - I_p \cdot \dot{\Omega}) \cdot \frac{(LE + 1) \cdot \rho}{LE \cdot (\rho - 1) + \rho + 1} \quad (25)$$

It is useful to precise that the locking effect LE has a dynamics tiny dependent on the output shafts velocity difference, due to its dependence on the efficiency η through the torque bias ratio TBR. A possible LE dynamics mathematical model is:

$$LE = sign(T_p) \cdot LE_{nom} \cdot \tanh[k(\omega_{rear} - \omega_{front})] \quad (26)$$

where

- LE_{nom} is the nominal locking effect.

- *k* is a differential tuning parameter introduced to describe how rapidly the LE joins the design nominal LE value depending on the relative velocity between the two output shafts.

- *sign(x)* is a function whose value is 1 if (x) is grater than zero, 0 if it equals zero and -1 if it is less than zero.

- *tanh(x)* is the hyperbolic tangent function.

Considering a central differential torque sensing it is necessary to define the final torques transmitted to the front and the rear axles. In general, during an acceleration transient the torques passes from the faster output shaft to the slower one. The contrary happens during a deceleration transient. For sake of simplicity, in the following it will consider ρ=1. During an acceleration transient (ring torque T_p>0), if the rear axle velocity is higher than the front axle one, the LE is positive. Consequently, in such hypothesis the higher torque is applied to the front axle. The equation to describe T_{hi} and T_{lo} are:

$$T_{rear} = (T_p - I_p \cdot \dot{\Omega}) \cdot \frac{1 - LE}{LE \cdot (\rho - 1) + \rho + 1} \quad (27)$$

$$T_{front} = (T_p - I_p \cdot \dot{\Omega}) \cdot \frac{(LE + 1) \cdot \rho}{LE \cdot (\rho - 1) + \rho + 1} \quad (28)$$

Open and torque sensing differential

The locking effect is the parameter to distinguish between an open and a torque sensing differential. The locking effect dynamics can be shown through Figure 3.

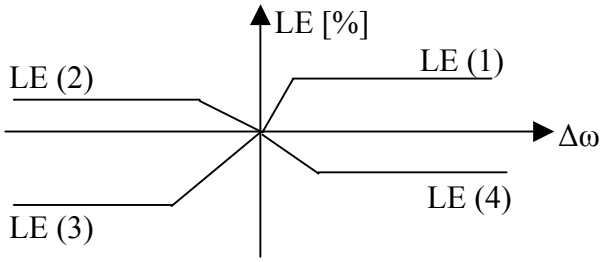

Figure 3. Locking effect dependence on output shafts relative velocity. Torque sensing differential grants four different operating conditions, depending on traction/release condition combined with angular velocity gradient sign.

Combining the acceleration transient sign with the output shafts relative velocity sign, a torque sensing differential can be carried out with four different locking effect values. From the mathematical point of view, to join the design locking effect values can be then required also four different values of tuning parameter k. An open differential can have only one locking effect value, computed depending on the efficiency η according to the following equation:

$$LE = \frac{\Delta T}{T} = \frac{T_{hi} - T_{lo}}{T_{hi} + T_{lo}} = \frac{T_{lo}}{T_{lo}} \cdot \frac{\frac{T_{hi}}{T_{lo}} - 1}{\frac{T_{hi}}{T_{lo}} + 1} = \frac{\eta - 1}{\eta + 1} \quad (29)$$

Usually an open differential locking effect is around 10 %. A torque sensing can work at four different locking effect with different split ratio.

FRICTION-CLUTCH-BASED LIMITED SLIP DIFFERENTIAL

A friction-clutch-based limited slip differential transmits torque due to slipping existing between surfaces appropriately loaded. The friction force distribution along slipping surfaces (discs) devoted to generate torque is extremely non-homogeneous and depends on phenomena not yet completely described. The mathematical model considers a resulting torque due to a total force applied in the friction disc medium radius. When the clutch is in a slipping transient, the torque is computed as:

$$C_f = n \cdot f \cdot N \cdot \frac{(R_1 + R_2)}{2} \cdot \mathrm{sgn}(\omega_m - \omega_r) \quad (30)$$

where

n number of slipping surfaces.

f clutch friction dynamic coefficient.

N normal load applied on clutch disc.

R_1, R_2 outer and inner clutch disc radius.

ω_m, ω_r angular velocities of faster and slower discs.

A clutch engagement respects the fundamental that torque is transmitted through friction without discontinuity, then no abrupt changes can occur during a transient to friction coefficient from slipping to disc surfaces adherence. It was conceived a torque transmitted mathematical model devoted to describe the transient from slipping to adherence avoiding discontinuities. Such result was obtained computing every simulation sample time the torque it should be transmitted if the disc surfaces should be engaged.

Friction-clutch-based limited slip differential devoted to AWD are equipped with actuation provisions devoted to apply on clutch discs a desired force N electronically controlled.

SIMULATIONS

Different driveline layout has been tested with objective maneuvers. All simulation presented in the following as example are carried on considering high adherence conditions. Figure 4 shows the comparison between different vehicle assembly during a ramp steer maneuver (15°/s steering rate, 100km/h). Vehicle understeering

characteristic and sideslip angle versus lateral acceleration do not present appreciable differences. The vehicle proceed at constant speed during this test, so the magnitude of longitudinal forces generated by tires is not great and does not trigger lateral force saturation. In this particular situation vehicle lateral dynamics is not influenced by different distribution of longitudinal force due to 4WD drivelines.

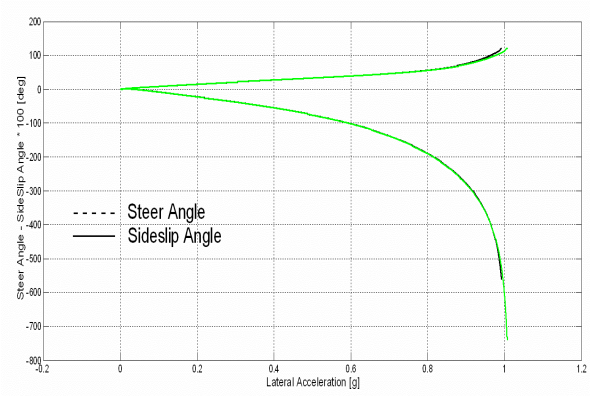

Figure 4. Ramp Steer at constant speed. Steer angle and Sideslip angle versus Lateral Acceleration

Driveline layout influence on vehicle dynamics can be better observed during vehicle longitudinal acceleration. Figure 5 shows results of a ramp steer, during which throttle position has been set constant, in order to gradually increase vehicle speed. While understeer gradient and sideslip gradient are not affected by driveline layout.

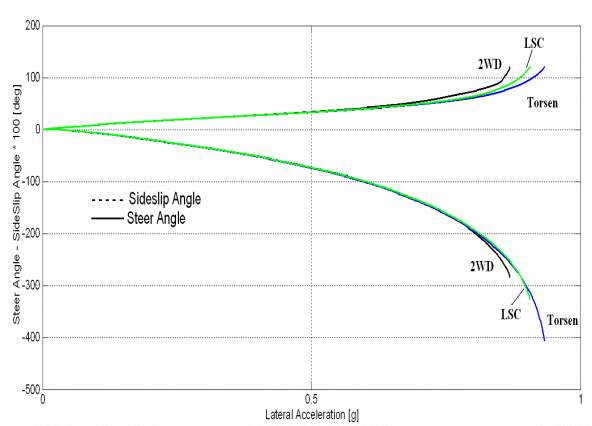

Figure 5. Ramp Steer at increasing speed. Steer angle and Sideslip angle versus Lateral Acceleration

Vehicle linear behavior field is extended to greater lateral acceleration. Central differential choice still does not produce changes in vehicle lateral dynamic.

The third maneuver consists of a step steer (100° maximum steering wheel angle, 400°/s steering rate, 100km/h) with power on. Results of previously described layout are compared with the behavior of a vehicle equipped with a central clutch (C.C.) actuated hydraulically depending on the relative velocity between the shafts connected through such components. Figure 6 summarizes the main vehicle dynamics variables.

Figure 7 shows tyres longitudinal forces. The first second is not significant because of the intervention of speed control to reach the proper manoeuvre speed. The turn is toward right and different longitudinal forces can be observed on the left tyres after load transfer.

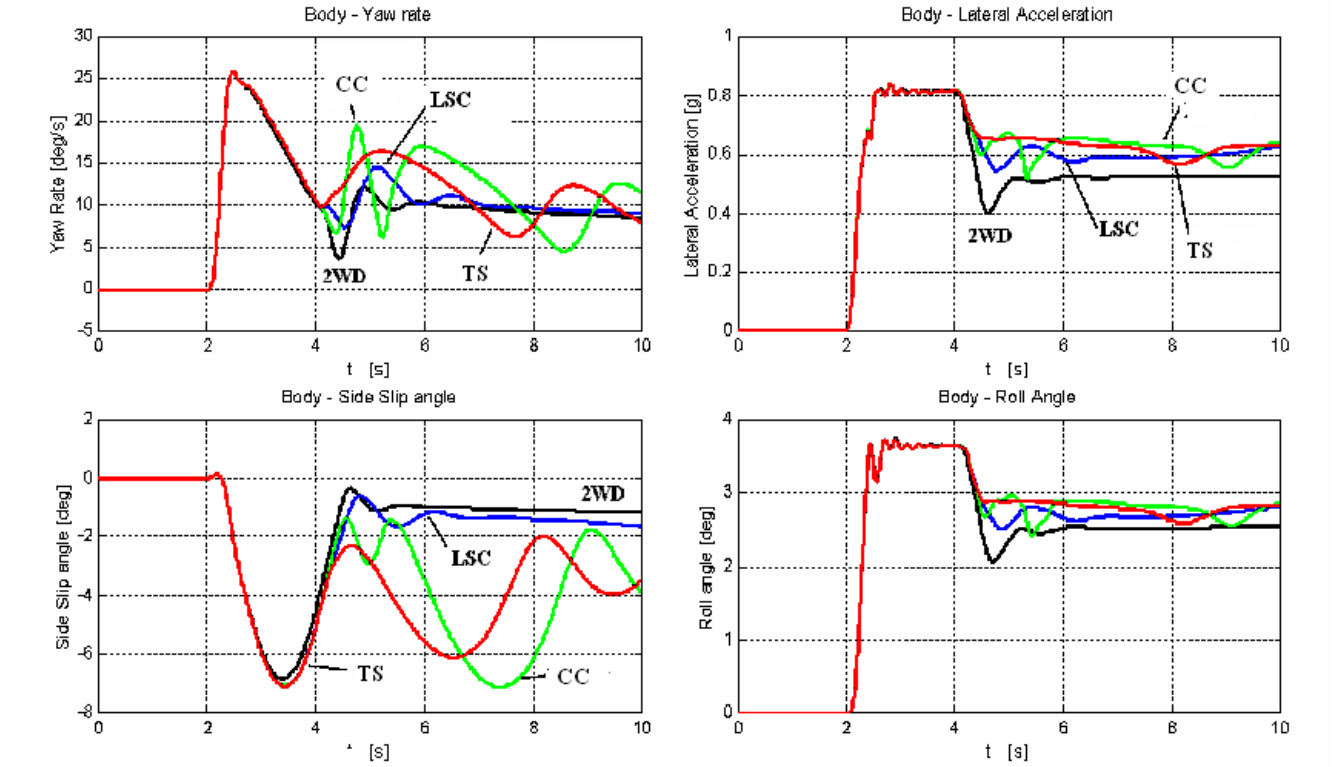

Figure 6. Step Steer with acceleration. Yaw rate, lateral acceleration, sideslip angle and roll angle versus time

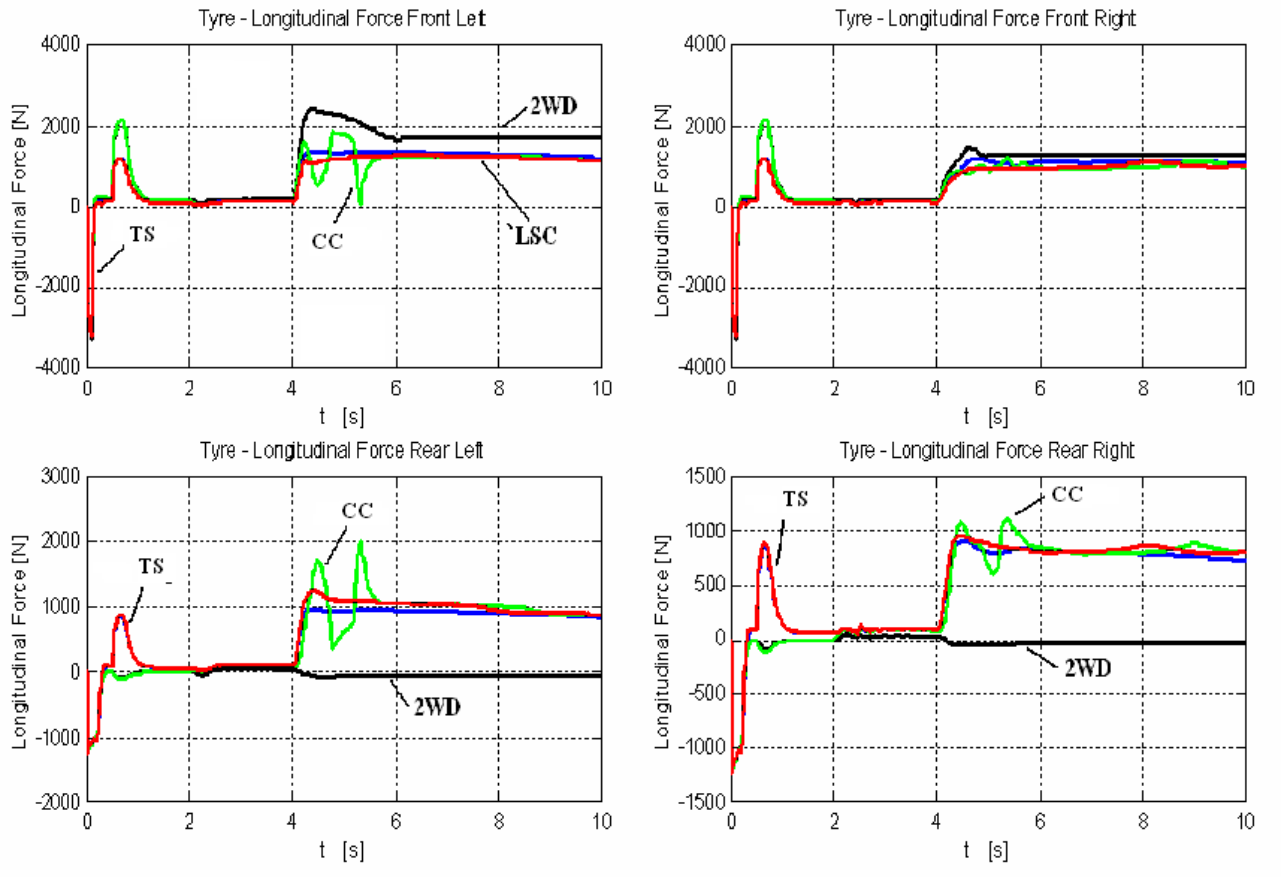

Figure 7. Step Steer with acceleration. Tires longitudinal forces

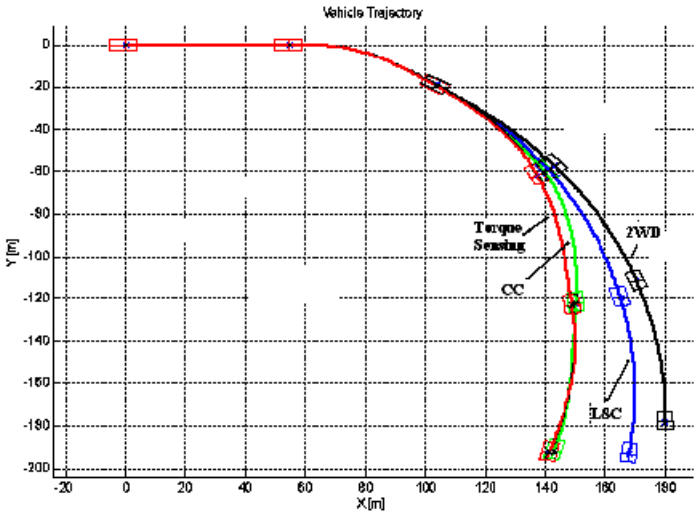

Figure 8. Step Steer with acceleration. Trajectory Comparison

Figure 8 describes the trajectory of the four vehicles. Torque sensing layout grants the lesser trajectory radius, reducing understeering behaviour of the car. Central clutch hydraulically actuated intervention is delayed in time if compared with torque sensing but, after the beginning of backward traction transfer, the vehicle increases its yaw rate and reach the inner trajectory.

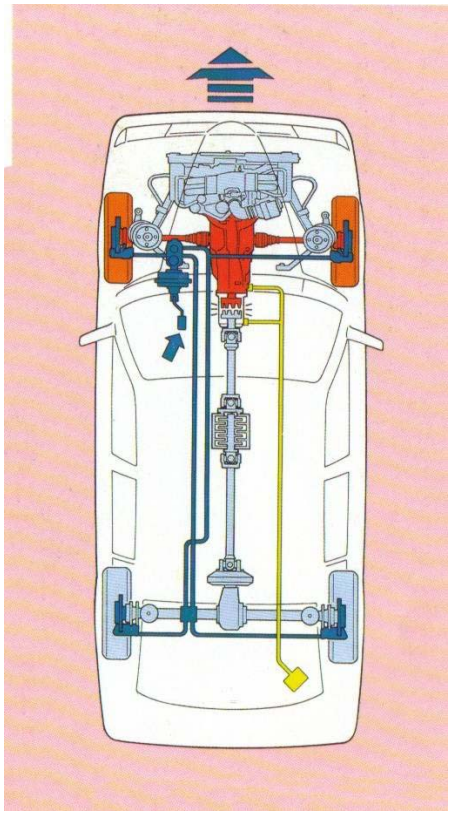

Figure 9. Schematic view of the vehicle equipped with central LSC differential.

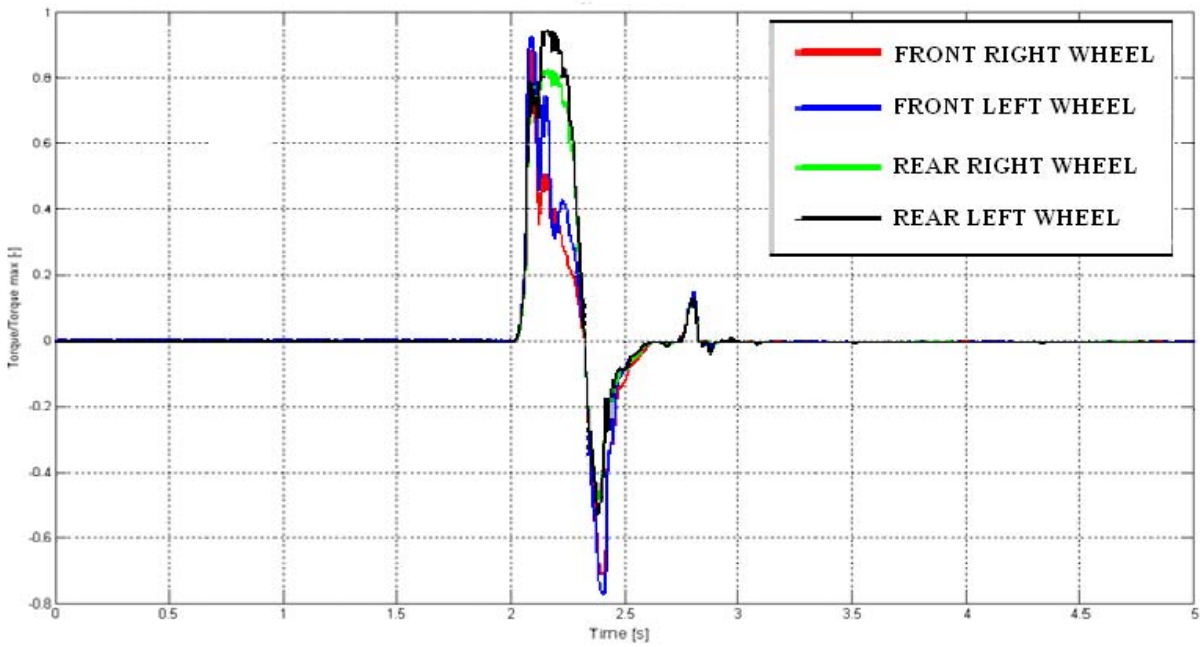

Figure 10. Experimental road test: Snap Star Maneuver, Wheel Torque (normalized value according to maximum torque reached).

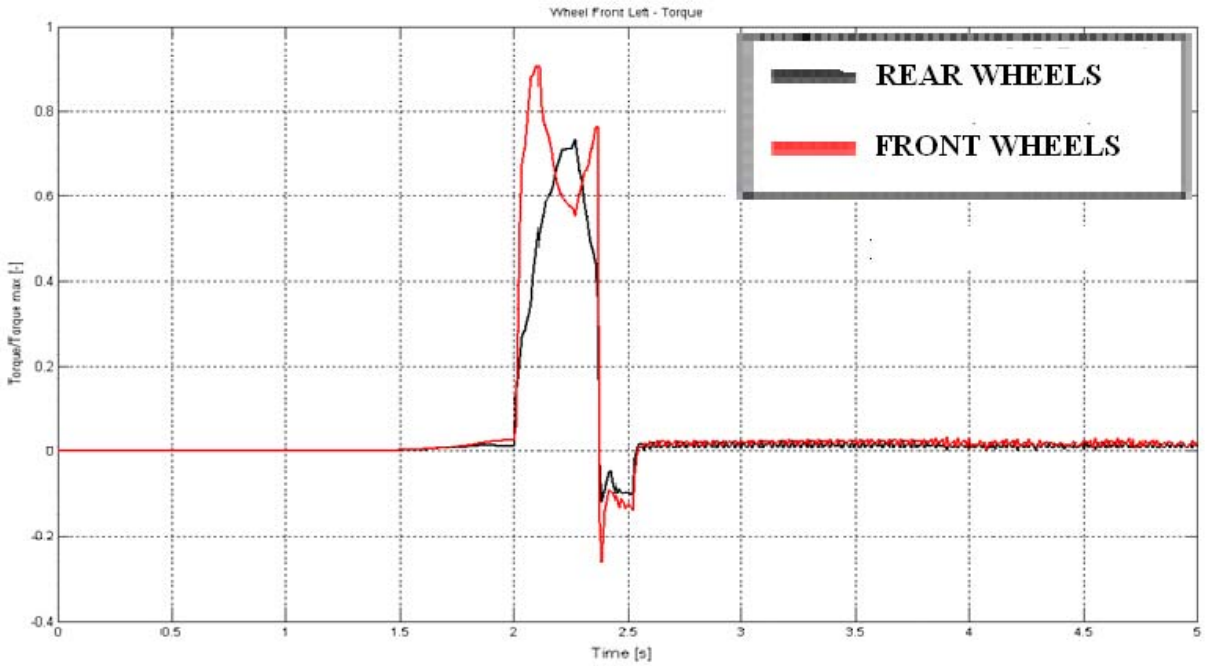

Figure 11. Simulation Data: Snap Star Maneuver, Wheel Torque (normalized value according to maximum torque reached). Left and rear wheel on the same axle show the same torque value.

ROAD TEST EXAMPLE

An example of comparison between road test results and simulation is presented. Test car was a sedan prototype equipped with central torque sensing differential. Test manoeuver consists in a Snap Start (start speed = 0 km/h, throttle wide open and quick clutch release). Snap Start manoeuver is finalized to evaluate 4WD vehicle traction capability, front/rear torque distribution and wheel torque oscillations. Figure 9 and 10 shows experimental and simuated wheel torque at the four wheels.

Firstly, asimmetry of right and left experimental wheel torques can be observed, while functional model evaluates torque difference only between front and rear axles. Secondly, torque oscillations are not appreciable in simulation results, as long as the torque backlash after

throttle release. Mean differences are due to subjectiveness of Snap Start manoeuver, expecially to the lack of driver action on throttle and clutch time history. Mathematical models are fit to simulate objective manoeuvers like those shown previously. Slight different torque increase and decrease rates are probably due to not appropriate engine torque characteristic.

Rear axle torque delay is appreciable in both test and simulation: central torque sensing effect is then qualitatively reproduced. This preliminary comparison demonstrates that the model needs further development in order to be validated through road test, being subjective or objective. Performance gradient due to different driveline layouts is still trustful because it is mainly a qualitative result, depending on well established equations shown above.

CONCLUSIONS

Main parameters and characteristics of open, torque sensing and limited slip differentials have been focused. A mathematical model has been developed in order to describe different behavior and merged with a vehicle dynamics model. Different driveline layouts performances have been compared through virtual experimentation and main results presented. Road tests demonstrated that subjective maneuvers are hardly reconstructed using a mathematical model, even if the behavior of the vehicle is well reproduced. Further model development is actually work in progress and will be presented in future publications.

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