Vehicle Dynamics and Stability Analysis with Matlab and Adams Car

N.Arborio, P.Munaretto, M.Velardocchia
Dipartimento di Meccanica Politecnico di Torino
C.so Duca degli Abruzzi, 24 – 10129 Torino Italy
Tel: +39 011 5646931; fax: +39 011 5646999: email: velardocchia@polito.it

N. Riva, E. Suraci
FIAT Auto - Direzione Tecnica Autotelaio
C.so Settembrini, 40 – 10135 Torino - Italy

ABSTRACT

The automotive active dynamics and stability control is nowadays available from many manufacturers which offer different hardware and software solutions. To compare the performance obtainable considering different active systems, a mathematical model of a new car was implemented through Adams Car. The model was completed with a powertrain specifically conceived in Matlab environment to overcome problems due to an Adams Car modelling not suitable to describe every operating condition (e.g. standing start, gearshift).
A strategy to overcome the problems introduced by the dependence of nominal yaw rate value on the instantaneously available tire-terrain adhesion coefficient was considered.
The most important results obtained adopting the described strategy are presented through animations.

INTRODUCTION

Many Authors described how active systems could increase handling and comfort car's performance. Active steering systems, vehicle dynamics control, continuous damper control are an immediate example of active systems nowadays available. From car manufacturer's point of view, such large availability requires to increase the know-how both about the active systems and their integration. Great importance acquires, moreover, the capability to compare the performance obtainable considering different active systems. To develop the necessary evaluation, it is particularly useful to employ software that permits to manage a sensitivity analysis. With this aim, a mathematical model of a new FIAT automobile was implemented through the Adams Car code.
Adams Car 10.0 permits to simulate a lot of different manoeuvres, but presents some limits when has to model the torque transmitted by clutch, synchroniser, brakes, considering operating conditions that require a continuous change of the slipping velocity sign. During transients that determine those changes, oscillations appear due to a discontinuous model of friction transmitted torque. To overcome such limit, extending the Adams Car ability to model the vehicle dynamics also including active systems, the model was completed with a powertrain specifically conceived in Matlab environment. The driveline conceived permits to simulate, for example, operating condition like standing start and gearshift, and to consider correctly the wheel's dynamics during manoeuvres typical of vehicle dynamics control (VDC).

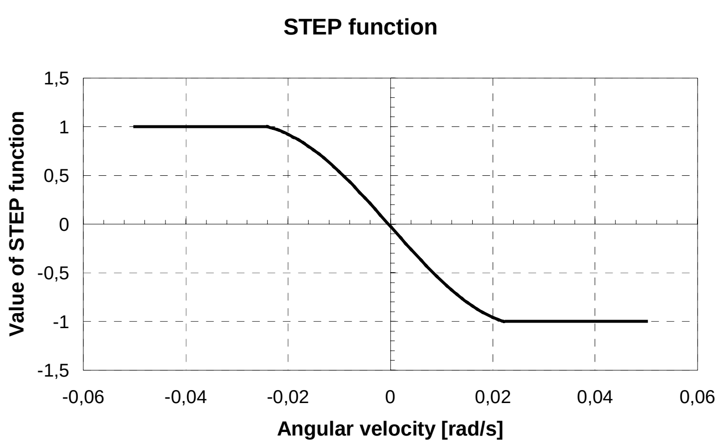
Different strategies were tested to actively control the vehicle dynamics. Considering available the only signals of steering wheel angle, yaw rate, lateral acceleration and wheels angular velocities (signals present on board vehicle), several magnitudes were estimated (e.g. tire forces, tire-terrain friction available). A strategy to overcome the problems introduced by the dependence of yaw rate nominal value from the characteristics velocity and on the instantaneously available tire-terrain adhesion coefficient was considered.
The methodology adopted to carry out the integration between the vehicle model developed in Adams Car, the powertrain and vehicle dynamics control developed in Matlab are presented.

DRIVELINE

The driveline available by Adams Car 10.0 is suitable to consider manoeuvres which does not require the change of the slipping velocity sign to compute the torque transmitted by components like clutch synchroniser and brakes. Consequently, to employ the Adams Car driveline model could introduce problems during transients typical of active systems like ABS, VDC, automatic gear, etc.
The unsatisfactory driveline Adams Car modelling is due to a not good description of torque transmitted by mechanical components through friction. For example, the torque transmitted by the clutch is described through the following function:

IF(VARVAL(.MDI_Demo_Vehicle.TR_Powertrain.cis_transmission_demand_adams_id):1,0,1)
**(1-VARVAL(.MDI_Demo_Vehicle.TR_Powertrain.cis_clutch_demand_adams_id))*
**(step((DIF(.MDI_Demo_Vehicle.TR_Powertrain.engine_omega_xl)-*
VARVAL(.MDI_Demo_Vehicle.TR_Powertrain.transmission_input_omega)),
(-1.MDI_Demo_Vehicle.TR_Powertrain.pvs_clutch_torque_threshold/.MDI_Demo_Vehicle.TR_Powertrain.pvs_clutch_damping),*
(-1.MDI_Demo_Vehicle.TR_Powertrain.pvs_clutch_torque_threshold),*
(.MDI_Demo_Vehicle.TR_Powertrain.pvs_clutch_torque_threshold/
.MDI_Demo_Vehicle.TR_Powertrain.pvs_clutch_damping), (.MDI_Demo_Vehicle.TR_Powertrain.pvs_clutch_torque_threshold))
*+(.MDI_Demo_Vehicle.TR_Powertrain.pvs_clutch_stiffness)*DIF(.MDI_Demo_Vehicle.TR_Powertrain.clutch_slip))*if(mode-*
5:1,0,1)

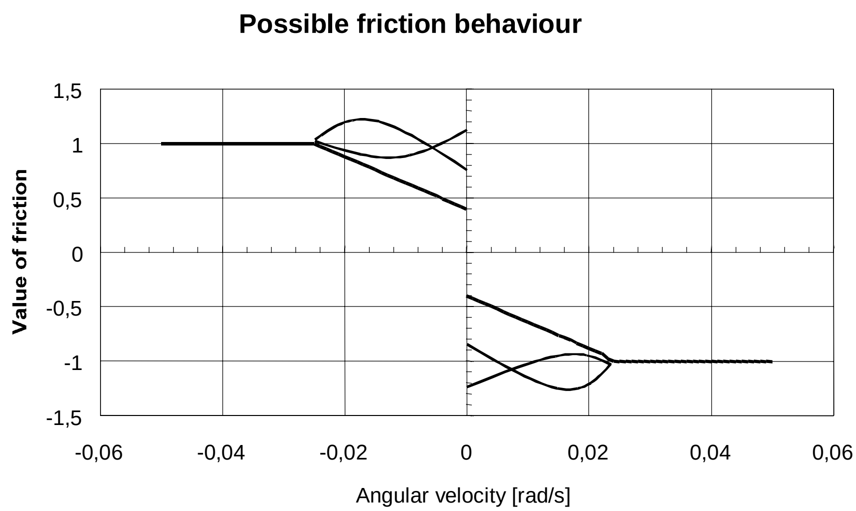
Besides the physical factors, the expression presents a logical function, named STEP, which appears as follows:



This function is defined time to time:

$$\text{STEP(t)}=\begin{cases} C_0 & x \le x_0 \\ ax^3 + bx^2 + cx + d & x_0 < x < x_1 \\ C_1 & x \ge x_1 \end{cases}$$

Adams Car uses STEP function to compute the clutch transmitted torque according to a Coulomb's friction mathematical description. The same approach is used to model torque transmitted through brakes. The function described presents some disputable points. First, it represents an imposed course for friction, which is physical and always different due to the dynamics and laden conditions of the system. Secondly, corresponding to zero value of clutch slipping angular velocity (intended the difference between clutch disk and flywheel) it passes from zero which means a zero value of



transmitted torque. Adams uses this device to prevent the changing sign of the clutch torque when reaching adherence.

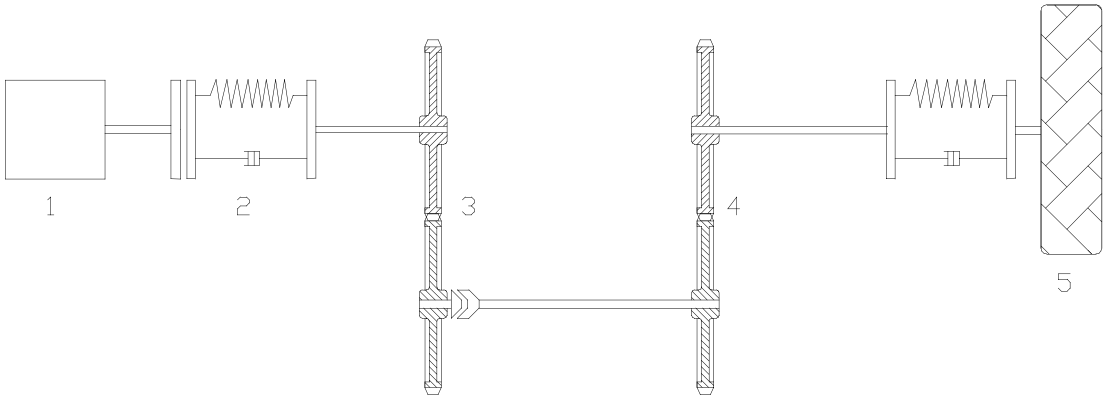
It is to be intended that the mathematical description of friction would be more precise with the following aspect:

Figure 1. Typical behaviour of friction coefficient during clutch engagement

The figure represents some possible behaviour for friction. It is not known the actual behaviour of friction when the system passes from slipping to adherence and vice-versa. Anyway, when the system has reached a synchronous condition (e.g. clutch slipping velocity equal to zero), the clutch transmitted torque must not have a zero value. Its value is determined by the value that friction assumes on y axis, when angular velocity or relative difference of angular velocity falls to zero. This considerations are not possible using a description as in Adams Car (STEP function). The approach used in Adams Car seems to be the discontinuous one, where the clutch transmitted torque is evaluated only in the slipping condition.

Furthermore, the driveline used in the Adams vehicle model does not permit to carry out gearshift or beginning simulations with a zero value of initial velocity of the vehicle. Gear has to be imposed at the beginning of the simulation and it is not possible to carry out any either upshift or downshift.

For these reasons it has been conceived a new driveline in Matlab, which describes the dynamic behaviour of the following layout:



1) Engine and Flywheel
2) Clutch
3) Gear
4) Differential
5) Semi-axis and wheel

Figure 2. Driveline scheme

The driveline is mathematically modelled considering the equilibrium equations which describe the dynamic behaviour of every component (e.g. Engine, Clutch, Gear Differential and so on).
It has been said that the transmission in Adams seems to be a discontinuous one, where, after the adherence between flywheel and clutch disc, the system looses the perception of the clutch transmitted torque due to a not good description of friction phenomenon.
The driveline realised in Matlab presents a continuous clutch model based on a new approach to friction phenomenon modelling. To reach this aim it has been necessary to evaluate the clutch transmitted torque at engagement concluded, so starting from the hypothesis that the engagement is completed when the angular acceleration of the flywheel and the clutch disc would be the same:

$$\ddot{\vartheta}_m = \ddot{\vartheta}_f \tag{1}$$

results that the clutch transmitted torque assumes the following expression:

$$C^*_{clt} = \frac{I_{clt}}{I_{eng} + I_{clt}} \cdot C_{eng} - \frac{I_{clt}}{I_{eng} + I_{clt}} \cdot res + \frac{I_{eng}}{I_{eng} + I_{clt}} \cdot \left[k_{clt} \cdot \left(\vartheta_{clt} - \vartheta_{ps}\right) + \beta_{clt} \cdot \left(\dot{\vartheta}_{clt} - \dot{\vartheta}_{ps}\right)\right] \tag{2}$$

where:

C_{clt} clutch transmitted torque
I_{clt} clutch inertia
I_{eng} engine inertia
res friction engine torque

The expression in square parenthesis represents the torque transmitted by the clutch damper.

Starting from the expression above it is possible to evaluate the friction coefficient corresponding to an engaged condition using the well known expression for the clutch transmitted torque C_f^* in the slipping condition.

$$C_f^* = f^* \cdot n \cdot N \cdot R_m \tag{3}$$

where

*f** is torque transmitted coefficient
n is the number of clutch disc surfaces
N is the load applied on clutch discs
R_m is the medium clutch disc radius

The torque transmitted coefficient *f** can be used in an opportune equation describing the variations in function of the angular velocity difference existing between the flywheel and the clutch disc [16].
The driveline briefly described has been integrated with the Adams vehicle. Adams receives traction torque from the Matlab driveline before the differential, and passes to Matlab the wheel-terrain reactions

VEHICLE DYNAMICS CONTROL STRATEGY

The vehicle dynamics control is carried out applying an adequate pressure to each wheel. The reference magnitudes considered are the sideslip angle β and the yaw rate $\dot{\Psi}$. The control inputs are the steering wheel δ, vehicle velocity (lateral v_y and longitudinal v), yaw rate, wheels velocity. The output are the reference pressure for each wheel. The equations used to compute the reference signals are:

$$\dot{\Psi}_{No} = \frac{\delta \cdot V}{L + Kus \cdot V^2} = \frac{\delta \cdot V}{L\left(1 + \frac{V^2}{V_{ch}^2}\right)} \tag{4}$$

$$\beta_{No} = \delta \cdot \left(\frac{b}{L + Kus \cdot V^2} - \frac{V^2}{gC_\alpha \left(L + Kus \cdot V^2\right)} \right) \tag{5}$$

Where L is the carriage step, v_{ch} expresses the characteristic velocity, b is the rear step, C_α is the lateral stiffness, g is the acceleration of gravity. Nominal yaw rate and nominal sideslip are limited according to:

$$\dot{\psi}_{NoMax} = \frac{\mu_{HF}\, g}{V} \tag{6}$$

$$\beta_{NoMax} = \mu \cdot \left(\frac{bg}{V^2} - \frac{1}{C_\alpha} \right) \tag{7}$$

where μ_{HF} is the maximum tire-wheel friction coefficient. A variable $\Delta\beta = \beta_{no} - \beta$ is considered. If Δβ is inside a range close to zero, the vehicle control is not activated.

The friction coefficient is estimated according to

$$\mu_{estim} = (1 - k_{mu} * \Delta\beta) \tag{8}$$

The equation (8) considers that a side-slip angle variation can be an index of a different tire-terrain friction coefficient. In equation (6), the μ_{HF} is computed according to the equation (8).

The k_{mu}. was determined through a number of simulations carried out considering terrains with a friction coefficient known.

RESULTS

A number of different manoeuvres were simulated applying the Adams Car – Matlab co-simulation to the car modelled together with the vehicle dynamics control strategy formerly described. The manoeuvres considered generally are not carried out according to standard ISO or SAE, due to the high simulation computing time required. Consequently, easier manoeuvres were considered, like step steer or a lane change. Moreover, to evaluate the vehicle dynamics performance obtainable through an active system, it is important to manage manoeuvres also in non standard conditions.

Each manoeuvre was repeated considering three tire-terrain friction coefficient: low (0.15), medium (0.45), high (1).

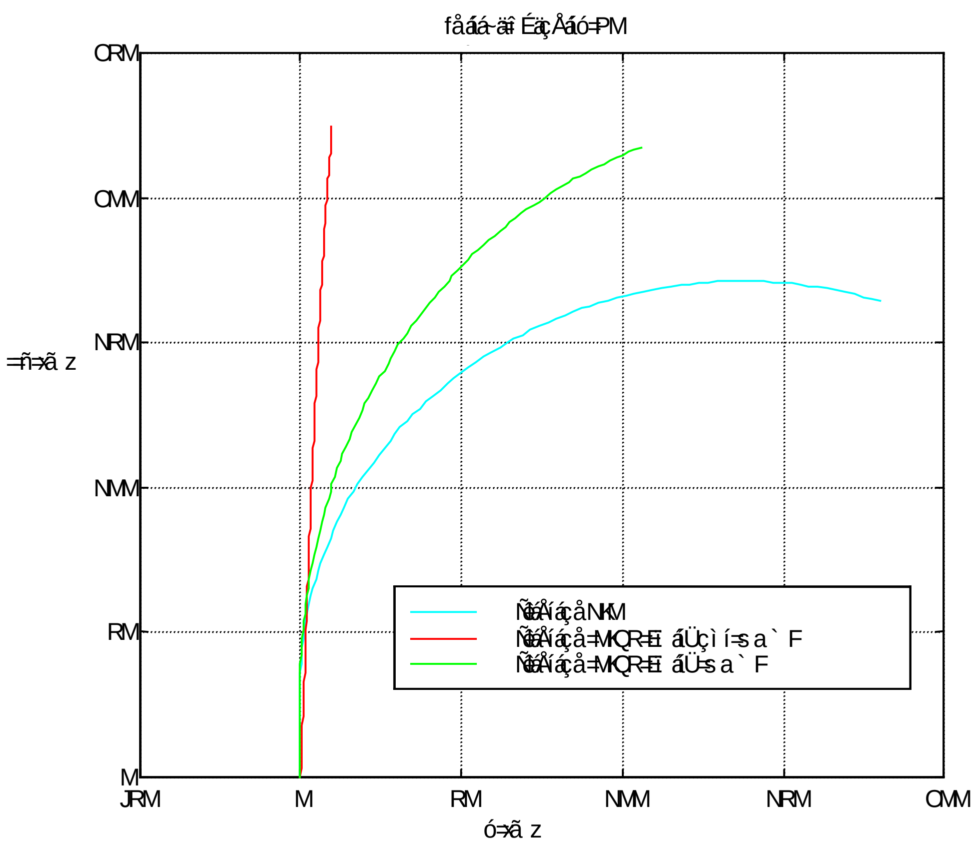


Figure 3. Step steer with high friction and medium friction (without and with vehicle dynamics control)

Figure 3 presents an example of the vehicle trajectory computed considering a step steer with high friction and with medium friction. In the latter case, are compared performance obtained without and with vehicle dynamics control.
Figure 4 presents the comparison between the trajectory established considering a lane change carried out with vehicle dynamics control active or not.

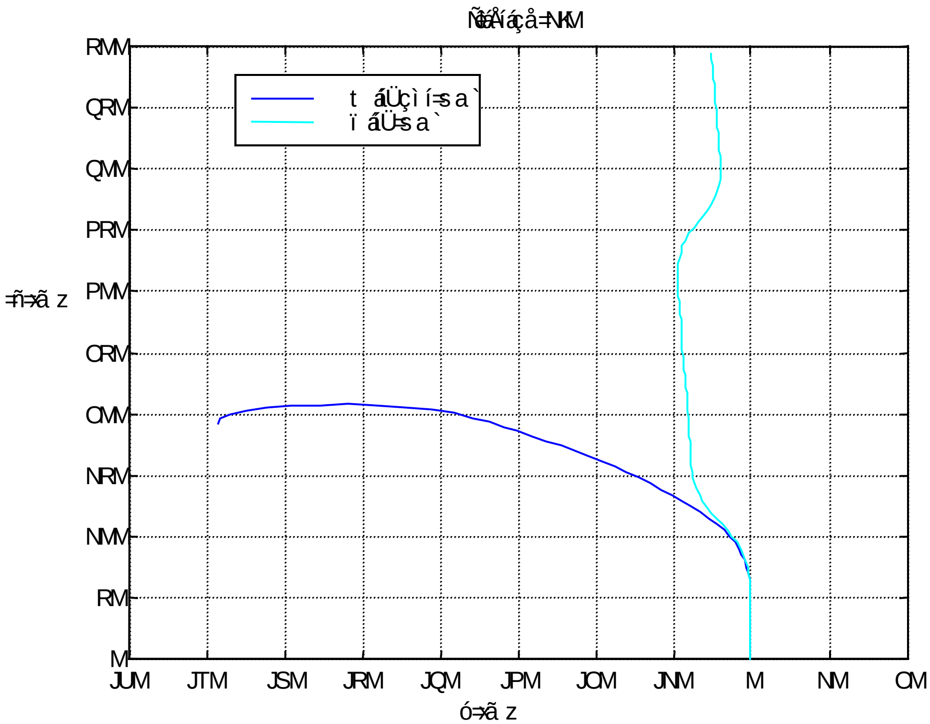


Figure 4. Lane change without and with vehicle dynamics control

Simulation results demonstrated the system ability to well control the vehicle dynamics, permitting to maintain its lateral stability in every operating condition simulated. It was unfortunately not possible to simulate operating conditions based on a different tire-terrain friction coefficient among the wheels, like e.g. μ-split, due to the tire model available in Adams Car.
According to the vehicle dynamics control strategy adopted, it was verified the necessity to control contemporarily both yaw rate and side-slip angle.

CONCLUSION

A methodology was developed to simulate the vehicle dynamics through Adams Car and Matlab co-simulation. The general scheme, usually adopted, which considers the Adams Car car model and the control strategy carried out through Matlab, was modified. To overcome problems due to an incomplete Adams Car driveline modelling, an original model was substituted to that available in Adams Car. That provision was effective to increase the Adams Car potentiality.
The methodology developed was applied to evaluate a control strategy conceived to carry out the vehicle dynamics control. Works are in progress to develop new vehicle dynamics control strategies.

References

1. A. van Zanten, R. Erhardt, G. Pfaff, “VDC, The Vehicle Dynamics Control System of Bosch”, SAE Technical Paper Series SP-950759
2. ABS-TCS-VDC Proceedings SAE, 1996, SAE PT-57
3. A. van Zanten, W. Ruf., A. Lutz, “Measurement and Simulation of Transient Tire Forces”, SAE Technical Paper Series SP-890640
4. M. Watanabe, N. Noguchi, “A New Algorithm for ABS to Compensate for Road-Disturbance”, SP-900205
5. G. Buschmann, H.T. Ebner, W. Kuhn, “Electronic Brake Force Distribution Control – A Sophisticated Addition to ABS”, SP-920646
6. E. Bakker, H. Pacejka, L. Lidner, “ A New Tire Model with an Application in Vehicle Dynamics Studies”, SP-890087
7. S. Matsumoto, H. Yamaguchi, H. Inoue, Y. Yasuno, “Improvement of Vehicle Dynamics Through Braking Force Distribution Control”, SP-920645
8. *Automotive Brake Systems*, Edited by Bosch, 1st Edition, ISBN 1-56091-708-3, Distributed by SAE Society of Automotive Engineers, USA.
9. *Electronic Braking, Traction and Stability Controls*, Edited by Ronald K. Jurgen, SAE, PT-76, 1999
10. *Actuators*, Edited by Ronald K. Jurgen, SAE, PT-74, 1998
11. *Automotive Handbook*, Edited by Bosch, 4th Edition, ISBN 1-56091-918-3, Distributed by SAE Society of Automotive Engineers, USA.
12. *Vehicle Dynamics and Simulation 2000*, Proceedings of SAE2000 World Congress, Detroit, Michigan, USA, March 6-9, 2000, SP-1526.
13. *Transmissions and Driveline System Symposium 1999*, Proceedings of SAE1999, World Congress, Detroit, Michigan, USA, March 6-9, 2000, SP-1440.
14. *Electronic Steering and Suspension Systems*, SAE, PT-77, 1999
15. *Vehicle Dynamics and Stability*, Proceedings of SAE Conference, Troy, Michigan, 15-17 May 2000
16. M. Velardocchia, S. Mauro, N. Arborio“ Non-linear Continuous Model for Clutch Engagement”, proposed for next SAE2001 Conference - Technical Paper 01-PC190