CONTRIBUTIONS TO THE SIMULATION OF VEHICLE LONGITUDINAL DYNAMICS

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Abstract: *The paper presents some Matlab-Simulink models imagined and used by the authors to simulate the operation of drivetrain subassemblies. The behavior of a tractor-semitrailer was studied in different traveling conditions: standing start, gear shifting, engine braking, travel on uneven road, tire friction variations. In addition, sub-models conceived for actuation cylinder, solenoid or electronic control devices permitted to simulate the working of vehicle's servo or automated systems.*

Keywords: *vehicle dynamics, drivetrain, actuation system, command logic, computer model.*

1. Dynamic Model of Drivetrain

The first step in drivetrain simulation is to envision a **dynamic model**, which means to mentally replace the complex mechanical system with an assembly containing inertial elements (flywheels or translational masses) connected by idealized (without mass) shafts, springs or couplings.

The dynamic model must be complex enough to serve the aim of study and as simple as possible to be easily transposed in a **mathematical model** and then solved with no major difficulties.

The **object of study** presented here as example is a road train composed by a two-axle rear-drive truck tractor and a two-axle semitrailer. The total mass of tractor-semitrailer combination considered for the simulation was 38200 kg, from which 6500 kg and 10000 kg were distributed on the tractor axles.

The tractor is powered by a direct-injection turbo-diesel engine developing 235 kW at 2600 rpm and 1000 Nm at 1600 rpm. The drivetrain includes a two-disk dry clutch, a synchromesh gearbox with 8+2 speeds, a cardan driveshaft and a drive axle with locking differential and two-stage final drive (middle mounted and hub mounted). The gearbox includes a front splitter unit, its total ratios spread is 8.52 and the 7^{th} and 8^{th} gears are direct drive and overdrive.

Figure 1 presents the dynamic model used for the study of the road train longitudinal dynamics. It consists of seven flywheels, connecting shafts (having only stiffness as properties) and tree couplings (representing the clutch, the synchromesh and the wheel-road interface).

The signification of symbols used stands for: J_e – moment of inertia for parts rotating with the engine angular speed (mobile components of the engine and driving parts of the clutch); J_{cd} – moment of inertia for the friction disk of the clutch; J_{is} – moment of inertia for driven parts of the clutch, for gearbox input and counter

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shafts and for all the gearwheels upstream the synchromesh; J_{os} – moment of inertia for gearbox output shaft and half of the cardan driveshaft; J_{dif} – moment of inertia for half of the cardan driveshaft and for ring and pinion axle drive; J_{dw} – moment of inertia for the vehicle's driving wheels; J_{mfw} – moment of inertia corresponding to vehicle's mass and non-driving wheels; M_e – effective moment of the engine; M_{res} – moment corresponding to driving resistances.

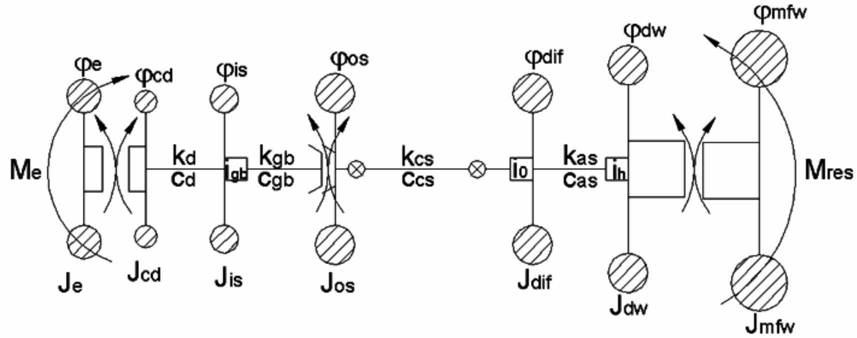

Fig. 1. *Drivetrain dynamic model of a tractor-semitrailer combination*

Based on their long-term experience, the authors introduced two innovative elements in the dynamic model:

– idealised gear mechanisms (symbolised by rectangles with inscription of their transmission ratios: i_{gb} for the gearbox, i_0 for ring and pinion axle drive, i_h for the hub mounted planetary gearset);
– torsional moments explicitly marked between pair of flywheels: M_{fc} – frictional moment of the clutch; M_{sync} – frictional moment of the synchromesh; M_{ad} – moment of adherence (wheel-road friction).

Therefore, the movement equations write easier; there is no need to recalculating equivalent inertia, stiffness or damping when gear ratios change; new inertial elements can be added with no difficulty, if a more complex or a new model is needed; the real values of the flywheels' angular velocities compute directly; values for the torques acting on shafts or couplings (that are always of interest) are already available. Last but not least, these additions to the traditional dynamic models permit the simulation of gear shifting.

Containing all the powertrain elements controlled by the driver (engine, clutch and gearbox), this model permits to obtain valuable information about the longitudinal vehicle behavior and drivability, both qualitative and quantitative.

2. Mathematic Model of Drivetrain

The second step in drivetrain simulation is to obtain, and then to solve, the **mathematical model**. This represents a system of differential and algebraic equations that describe the dynamic behavior of the studied system.

To obtain the mathematical model, it writes the movement equations for all the inertial elements. By solving these equations, there is firstly obtained the angular accelerations for each flywheel and then, by integration, the velocities and spaces.

As an example, according to figure 1, the movement equations of driving wheels are

$$\dot{\omega}_{dw} = \frac{M_{as}\, i_h - M_{ad}}{J_{dw}}, \qquad \omega = \omega_0 + \int_0^t \dot{\omega}\, dt$$

$$\varphi = \varphi_0 + \int_0^t \omega\, dt \tag{1}$$

where

$M_{as} = (\varphi_{dif} - \varphi_{dw} i_h)k_{as} + (\omega_{dif} - \omega_{dw} i_h)c_{as}$
is the torsional moment acting on the axle shafts and have an elastic and a damping components.

The traditional way to formulate the mathematical model was to **write** by hand all the equations. This stage represented a laborious and tricky task, because human errors could always appear. The difficulty increases with the complexity of the dynamic model. Having the system of equations, these could be solved by "wiring" it in an analogical computer, also a time consuming task.

Another option to **solve** the system was given by the accelerated evolution of digital computers and computing languages. Skilled researchers took a lot of time writing, testing and tuning programs using high level languages as C, Pascal or Fortran. Unfortunately, the efforts are directed not only to solve equation, but also to realize the user interface (data introduction and results presentation) or to document the program.

A modern option for simulation is nowadays offered by specialized simulation software, as Matlab-Simulink, designed to reduce the effort of the researcher. The model is a graphical representation of symbols (blocks – realized by the user or predefined and stocked in libraries) connected by informational links. The block diagram of the model became its simulation program and, therefore, this graphic language can be easier understood by engineers. In addition, component elements can be grouped to form functional blocks that can be organized by levels of complexity in hierarchical tree structures. Other advantages are that Simulink transcripts automatically the model in instructions of classical programming language and computer code (writes with no error the system of differential and algebraic equations) and also provide tested and powerful algorithms to numerically solve the system of equations.

3. Simulink Model

The Simulink model, conceived by the authors in concordance with figure 1, permits to simulate:

- the work of engine, clutch and synchromesh;
- the wheel-road interaction;
- the functioning of clutch and gearbox actuating systems, including variants with pneumatic assistance;
- the driving regimes;
- the generation of driver's command actions.

The main module, shown in figure 2, represents the block schematic of the algorithm that solves the mathematic model and contains:

- seven blocks for the main mechanical subsystem, which determine the kinematics (acceleration, speed and space) of engine, clutch, gearbox, differential, driving wheels and vehicle's body (translational mass); their speeds ω and spaces φ are the output signals of the blocks and are grouped (joined) in line **w_fi** to simplify the schematic and his understanding;
- block **Commands**, which generates the driver commands applied to the drivetrain; these are functions of time and kinematic quantities (from line w_fi) and are grouped in line **ctrl**, suggesting the multiplex network of modern vehicles; this block can also be used if part or all the commands come for an automated system that assist the driver;
- (left-side) blocks, which compute (based on kinematic and command

quantities) the torsional moments and forces that are input values for the blocks of the mechanical subsystems;

– block **Gear ratios**, which establishes the current values of the gear ratios, accordingly with the gear selected;

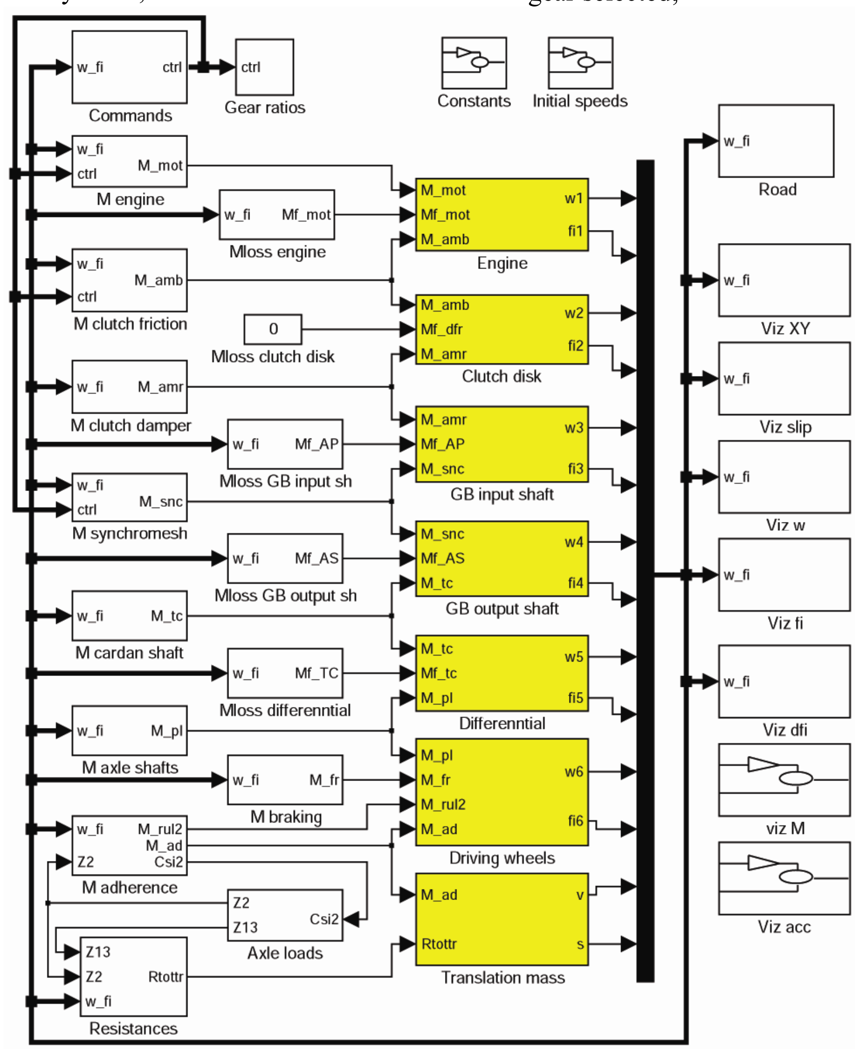

Fig. 2 *Main module of the Matlab-Simulink model used to simulate the functioning of a tractor-semitrailer drivetrain*

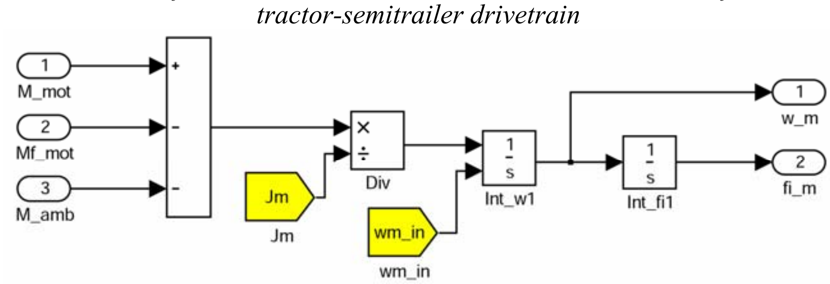

Fig. 3 *Submodule for engine dynamics*

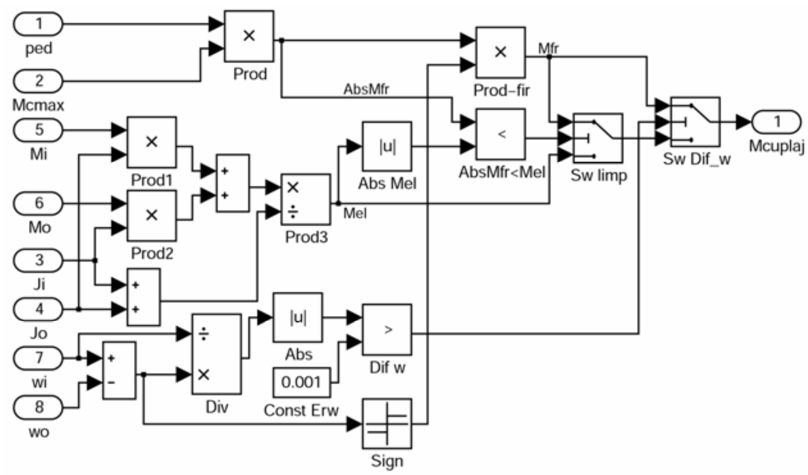

Fig. 4 Submodule to simulate a frictional coupling

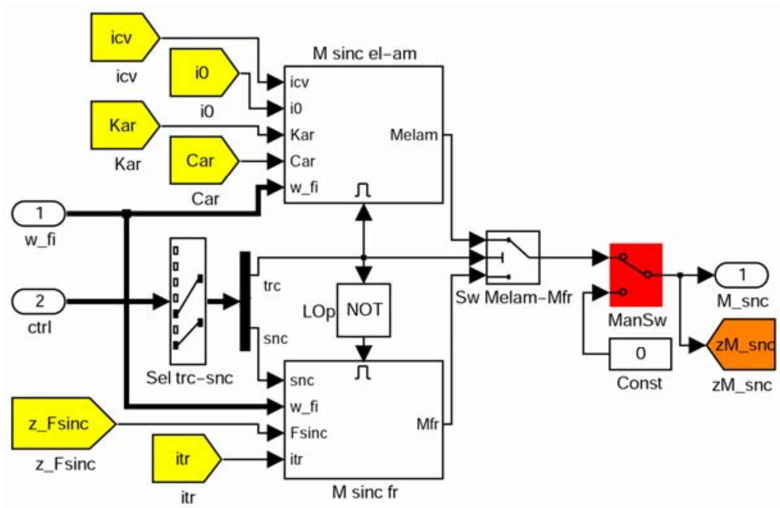

Fig. 5 *Submodule for the determination of synchromesh acting torque Msnc*

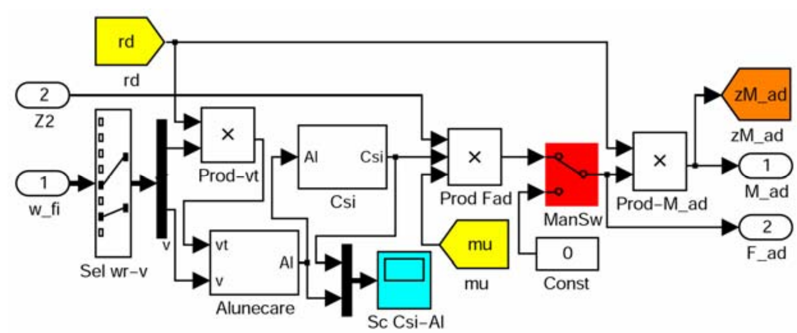

Fig. 6 *Submodule to calculate the wheel-road moment of friction M_{ad}*

– block **Constants**, which permits the introduction of vehicle's main characteristics, constant during simulation;
– block **Initial speeds**, which permits to assign integration initial values for the kinematic quantities;
– block **Road**, which generates the road characteristics (rolling resistance coefficient, grade, tyre friction coefficient) as functions of speed and traveled space;
– blocks **Viz** (on right-side), which realise graphical representation of calculated interest quantities, as time histories or characteristics (one vs. other), with their true or engine-reduced values;
– blocks **Mloss…**, which can simulate specific losses in the powertrain components by taking into account normal efficiencies or supplementary influences like cold or abnormal functioning;
– blocks **M braking**, which calculate the braking moment applied to the driving wheels;
– block **Axle loads**, which compute the instantaneous vertical load on each axle of vehicle, considering the load transfer between axles due to the current grade or acceleration;
– block **Resistances**, which compute the rolling, grade and aerodynamic resistances and their resultant.

Submodule **Engine**, presented in figure 3, computes kinematic quantities of engine's rotating parts. His input values are the engine's torque M_{mot}, the clutch's frictional torque M_{amb} and the supplementary frictional moment M_{fmot}; the output values are the engine angular speed w_1 and space fi_1. Almost similar submodules are used for the other inertial elements considered in the dynamic model.

To establish the input values for the seven modules that determine the inertial elements kinematics it was necessary to conceive a submodule for each torsional moment. Therefore, submodule **M engine** computes the engine torque M_{mot}, used as input value for the submodule in figure 3. This obtains by biparametric interpolation of test bench data, as function of accelerator pedal position *pac* and engine speed *wm*.

The clutch frictional torque and the synchromesh torque are modeled in separate blocks. In both submodules **M_clutch_friction** and **M_synchromesh** the main component is a submodule that simulates the way a frictional coupling act (fig. 4). During simulation, this permits the transmission of power in both directions. Consequently, it can simulate not only the engagement, but also the disengagement, with eventual slippage occurring at very high stress. Clutch frictional moment M_{amb} calculates as function of clutch pedal position *pamb* and kinematics of both input and output shafts.

The synchromesh acting moment M_{snc} computes by the submodule **M_synhcromesh** (fig. 5). Function of engaged gear (indicated by the signal *trc* in line **ctrl**), activates exclusively one of the two blocks **M_sinc_fr** or **M_sinc_el-am**, first being used during synchronizing time and the second when the gear is engaged.

Block **M_sinc_el-am** determine the torsional moment *Melam* that stress the synchromesh sleeve when the corresponding gear is engaged. The moment has two components (elastic and damping) that compute function of gearbox ratio i_{gb} and of gearbox stiffness and damping coefficients (k_{gb} and c_{gb}), represented in the dynamic model (fig. 1). Constants k_{gb} and c_{gb} correspond to gearbox components between clutch hub and synchromesh sliding sleeve, their values being reduced to the synchromesh.

Block **M_sinc_fr**, determine torsional moment M_{fr} stressing the synchromesh sleeve when the gearbox is not (yet) engaged. Two cases can appear, as indicated by the signal *snc* (of line **ctrl**), generated by the block **Commands**. In the first case, the gearbox is on neutral: *snc*=0 and M_{fr}=0. In second case, the gearbox lever actuates to shift in the desired gear: *snc*=1 and M_{fr} is the frictional moment generated by synchromesh.

The block **M_clutch_damper** corresponds to the clutch damper, generating the progressive characteristic torque vs. relative rotation angle and including the hysteresis produced by dry friction.

Figure 6 presents submodule that compute adherence moment M_{ad} on the driving wheels, this being used as input value in blocks determining the kinematics of the driving wheels and vehicle's body.

First, using as input data the theoretical speed $v_t = w_r\ r_d$ and real speed v, submodule **Alunecare** calculates the driving wheels slip *al*, according to next equation:

$$al = \begin{cases} 1 - \dfrac{v}{v_t} & if \quad v < v_t \\ 0 & if \quad v = v_t \\ -1 + \dfrac{v_t}{v} & if \quad v > v_t \end{cases} . \qquad (2)$$

The slip can take values between -1 (wheels locked during braking) and +1 (traction, spinning wheels and immobile vehicle). Easy to define, the slip is tricky to implement in computer algorithm, because at very low speeds (for driving wheels and vehicle body), numerical errors become very annoying and must be treated carefully. Based on the slip, submodule **Csi** calculate the used friction, defined by the relation

$$\xi = \frac{X}{\mu Z}, \qquad (3)$$

where X is tangential force (for traction or braking); Z – dynamic load on driving axle; X/Z – specific tangential force; μ – adherence (friction) coefficient. The graphic representation of used friction as function of slip ($\xi = f(s)$, output versus input in submodule **Csi**) shows in figure 7.

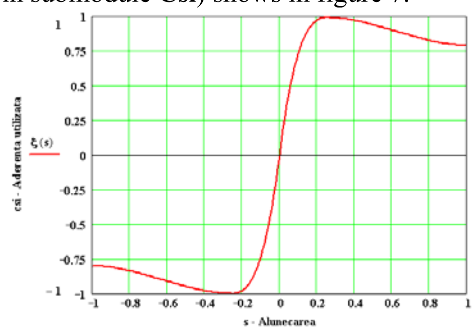

Fig. 7 *Rolling characteristic of the tire*

Submodule **Commands** serves to obtain all the commands imposed by driver (or by eventual electronic command logic) and received by the powertrain. Therefore, according to driver's intentions and vehicle kinematics, it generates positions signals (as presented in fig. 9) for three pedals (*pac* for accelerator, *pfr* for brake, *pam* for clutch) and other four signals necessary for gear shifting (*trd* for desired gear; *trc* for engaged gear; *sch* for need of shifting; *snc* for synchronization process). All these seven signals group together (multiplex) in the line **ctrl** and go to all blocks that need them (fig. 2).

The most complex submodule in submodule **Commands**, **Transmission gear**, simulates the signals generation at a gearshift and shows in figure 8.

Block **Generate_trd** simulates the driver intention to initiate a gearshift, including the neutral position. The desired gear indicates by the output signal *trd* that take integer values (for 1 to 8 and 0 for neutral).

Block **Generate_trc** produces two signals:

– *trc*, signal of currently engaged gear; becomes equal to *trd* only if the clutch is totally disengaged (pedal completely pressed – signal *pam*=0); supplementary, for a "drive" gear (i.e. not "neutral") it is necessary to wait the final of synchronization process (when *snc* fall for 1 to 0);

- *sch*, logical signal of shifting, indicates that an engaging or disengaging process was initiated, but this wasn't finished; so, if *trc* is equal to *trd*, *sch*=0, and if *trc* differs *trd*, *sch*=1.

When *sch* is high, a "drive" gear is wanted (*trd*≠0) and clutch is totally disengaged, a high value of signal *snc*, produced by the block **Generate_snc**, indicates a synchronization process in course. Also, this permits the driver shifting action (or commands a servomechanism), realized by the block **GB actuation** (in fig. 8) that generates the actuation force of the synchromesh sleeve.

The submodule **Commands** contains the blocks **Accelerator_pedal**, **Brake_pedal** and **Clutch_pedal** that resemble with the block **GB_actuation**. These simulate the actuation of the three driving pedals, but the output quantity is not a force, is a relative displacement (between 0 and 1). The blocks **Brake pedal** and **Clutch pedal** include subsystems that simulate working cylinder and control valve (foot or electro actuated), taking into account the laws of pneumatics governing their behavior.

4. Algorithm Application – Simulation of Drivetrain Behavior

The computer model was used to simulate the drivetrain behavior in different road and functional regimes, to improve dynamic and fuel consumption performances. The acceleration and the gearshifts were carried out in two different manners:

- as a **manual-automated** transmission, for the driver remaining only the task to actuate the accelerator pedal and to indicate the desired gear and the moment of shift, while the other decisions are made by the command logic;
- as a **full-automatic** transmission, the driver acting only on the accelerator pedal while all other actions are judged by the command logic.

As an example of the shortly described algorithm practical application, next figures show one of the most difficult working situations for the automated actuation system – **gear downshift**. Here are presented simulated time histories for a standstill start followed by a shift and gearing up in second gear and then by a downshift and an engine brake (shifts 0-1-2-1).

In addition to the quantities used in figures 9…11, many others can be represented as functions of time or as characteristics (one vs. other).

As a results interpretation, there is observed that synchronizing time necessary to shift from second to first gear is approximately double than for the inverse situation (from first to second).

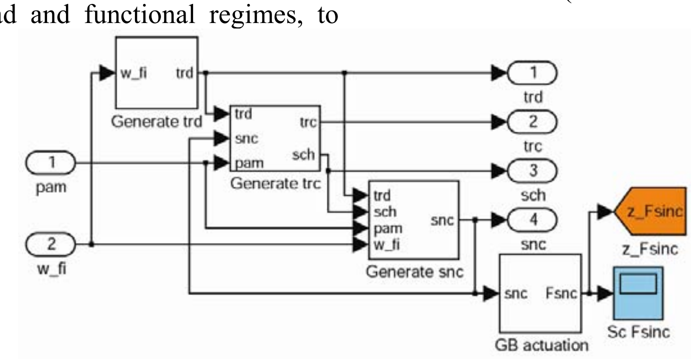

Fig. 8 *Submodule for the generation of signals related to gear shifting*

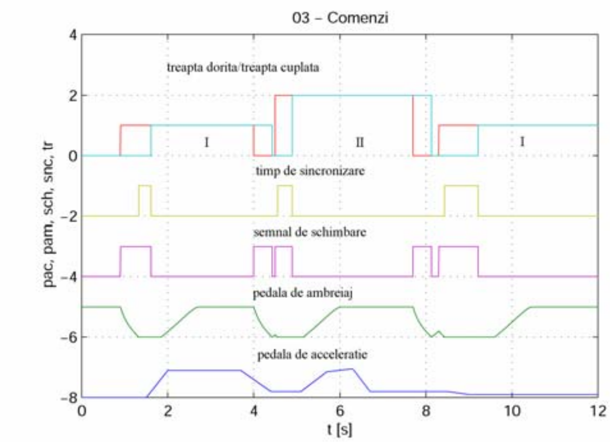

Fig. 9 *Command signals: pac (accelerator pedal); pam (clutch engagement); sch (shift); snc (synchronisation); trd and trc (desired and engaged gear)*

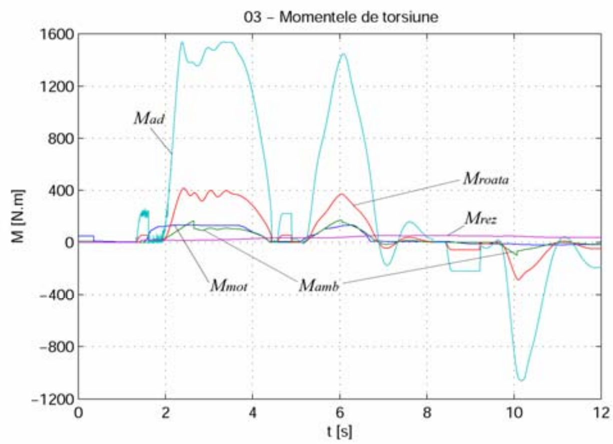

Fig. 10 *Real torsional moments: Mmot (engine); Mamb (clutch); Mroata (axle shaft); Mad (driving wheels friction); Mrez (global resistance)*

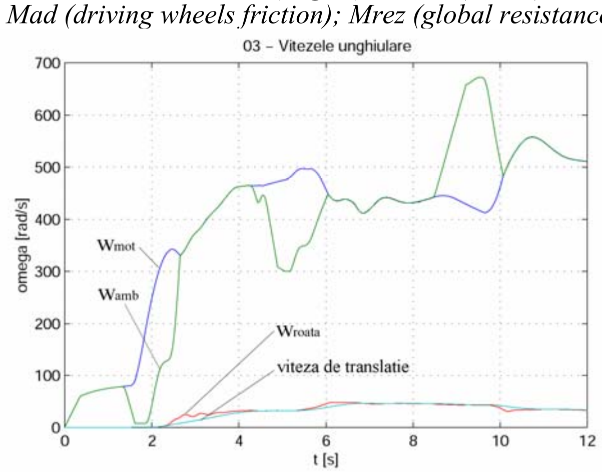

Fig. 11 *Real angular speeds: wmot (engine); wamb (clutch disk); wroata (driving wheels); angular speed equivalent to the vehicle body translational speed*

5. Conclusions

This article shows the way chosen by the authors to simulate vehicle longitudinal dynamics. Here were presented only aspects regarding a tractor-semitrailer combination with a servo-actuated transmission, but other vehicles types (car, all-terrain vehicles or agricultural and industrial tractors) were also studied. To solve the models, Matlab-Simulink and original FORTRAN or PASCAL programs were designed. The obtained results were confronted with many experimental data that permitted to calibrate or to improve the models.

The computer model shortly presented here can be exploited in many useful ways, not only to better understand how a complex mechanical (and not only) system behaves, but also to obtain important values for kinematics, stress level, frictional mechanical work, energetic consumptions (for vehicle subsystems or actuation servomechanisms) or durations of different processes. In addition, such models can facilitate the correct setting of drivetrain parameters and the design optimization of the system.

6. References

1. Seitz, N., Preda, I. *Microprocessor system for gear shift control of DAC truck tractor*. Contract no. 63/1990, Universitatea "Transilvania" din Braşov.
2. Preda, I. *Studiul solicitărilor din transmisia autoturismelor de teren, în vederea optimizării acesteia* (*Transmission strains study of all-wheel drive vehicles, for optimizing purposes*). Doctoral thesis, Transilvania University of Brasov, 1993.
3. Preda, I., Piso, P., Coandă, S. *Simularea dinamicii longitudinale a vehiculului în timp real* (*Real time simulation of vehicle longitudinal dynamics*). In: Bulletin of C.A.R. Conference, vol. II, p.122-127, Piteşti, 1994.
4. Ciolan, Gh., Preda, I., Pereş, Gh. *Cutii de viteze pentru automobile* (*Transmissions for motor vehicles*). Editura Didactică şi Pedagogică, Bucureşti, 1998.
5. Preda I., Ciolan Gh., Dogariu M., Todor I. *Computer model to simulate the vehicle longitudinal dynamics*. The Xth International Automotive Congress "CONAT 2004", paper 1060, Braşov, 2004.

Contribuţii la simularea dinamicii longitudinale a autovehiculelor

Rezumat: *Lucrarea prezintă unele modele Matlab-Simulink imaginate şi folosite de autori pentru a simula modul de funcţionare al subansamblurilor sistemului de propulsie. S-a studiat comportamentul unui autotren cu semiremorcă în diferite condiţii de deplasare: pornire din loc, schimbarea treptelor de viteze, frânarea cu motorul, trecerea peste obstacole, modificări ale aderenţei pneurilor. Suplimentar, submodelele concepute pentru cilindrii de acţionare, electrovalve sau dispozitive electronice de control au permis simularea funcţionării sistemelor automate sau servo ale autovehiculului.*

Cuvinte cheie: *dinamica autovehiculelor, sistem de propulsie, sistem de acţionare, logică de comandă, model pentru calculator.*

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