MODELING OF A DUAL CLUTCH TRANSMISSION FOR REAL-TIME SIMULATION

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Lucrarea abordează modelarea unei transmisii mecanice automatizate cu două ambreiaje în scopul simulării în timp real. Este realizat un model detaliat al transmisiei care include şi sistemul hidraulic de control al ambreiajelor şi sunt descrise amănunțit modelele principalelor componente dezvoltate special pentru simularea în timp real.

Se realizează modelul global al sistemului de propulsie care include motorul, transmisia, unitățile de control ale acestora, vehiculul şi conducătorul şi este demonstrată capacitatea acestuia de a rula în timp real folosind o platformă de timp real dSPACE.

This paper deals with the modeling of an automated mechanical transmission with dual clutch for real-time simulation. A detailed model of the transmission that includes the clutches hydraulic control system is developed and the models of key components developed especially for real-time simulation are presented in detail.

The global model of the powertrain system that include the engine, the transmission, the control units for engine and transmissions, the vehicle and the driver is build and the capacity to be executed in real-time is demonstrated using a dSPACE real-time platform.

Keywords: vehicle dynamics, transmission, dual clutch, synchronizer, real-time simulation

1. Introduction

Over the past decade the automobile manufacturers have profoundly redefined the way their products are designed, developed and constructed. The aim was to condense the time required from concept to production by using new design and calibration tools.

Today almost all automobile manufacturers and suppliers relay on hardware-in-the-loop (HiL) simulations for testing or calibration. HiL simulation can be described as having the physical part of a system (for instance, part of a vehicle) as a simulation while another part (usual the control system) is either a production or a prototype one.

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Hardware-in-the-loop (HiL) test benches are indispensable for the development of modern vehicle dynamics controllers (VDCs) [1], [2]. For this purpose, a model of the powertrain able to run in real time is necessary.

A simulation can be executed in real time if the amount of time spent by calculating the solution for a given time step is less than the length of that time step. Varying the step size is not an option for real-time simulation, so a fixed-step solver (implicit or explicit) must be used. This can make real-time simulation more challenging than desktop simulation.

In order to be able to use fix step solver is necessary to: use submodels compatible with real time, simplify the model (reduction of the number of states and especially elimination of the ones with higher dynamic) and to use adequate parameters for real time in order to limit the system dynamic.

Some useful tools for model simplification and parameter tuning are: examination of the step sizes during the simulation, linear analysis, activity index and state count. They allow the user to easy identify time consuming submodels, the states with high dynamics, the simulation time at which high dynamics appear and to decide the step size that can be used. Details on how to convert an offline model in a real-time one are presented in [3], [4], [5].

The dual clutch transmission (DCT) concept is a step forward in bridging the gap between automatic transmissions (AT) and other types of two pedal transmissions. It eliminates the torque interruption of the automated manual transmissions (AMT), improves significantly the efficiency compared to an AT or a CVT (continuously variable transmission) and allows rapid and cost effective customization.

This paper aims to present typical modeling of DCT's and real-time simulation issues involved in these applications. Models of key components (clutches, synchronizers, hydraulic pistons etc.) adapted for real-time are described and a realistic plant model including clutch control hydraulics and mechanics is developed.

The models are implemented using the 1D multi-domain simulation platform LMS Imagine.Lab AMESim (which will be referred as AMESim). This platform is particularly suited for powertrain and hydraulic applications [4], [5], [6], [7]. The AMESim RT (real-time) option enables the export of a model to a real-time environment such as dSPACE or xPC for use in HIL simulation. A dSPACE platform was used in order to evaluate the benchmark problems and to demonstrate the real-time performance of the complex powertrain model.

2. Transmission configuration and model description

In recent years a considerable number of DCT architectures was produced in series and many others architectures are proposed [8], [9]. For this study, one of the most complex architectures is chosen in order to demonstrate the capacity to build detailed models of those configurations. Fig. 1 shows the layout of the Volkswagen DSG 02E transmission. This transmission has two outputs shafts that combine the two partial transmissions (the first output shaft for gears 1, 2, 3 and 4; the second output shaft for gears 5, 6 and reverse).

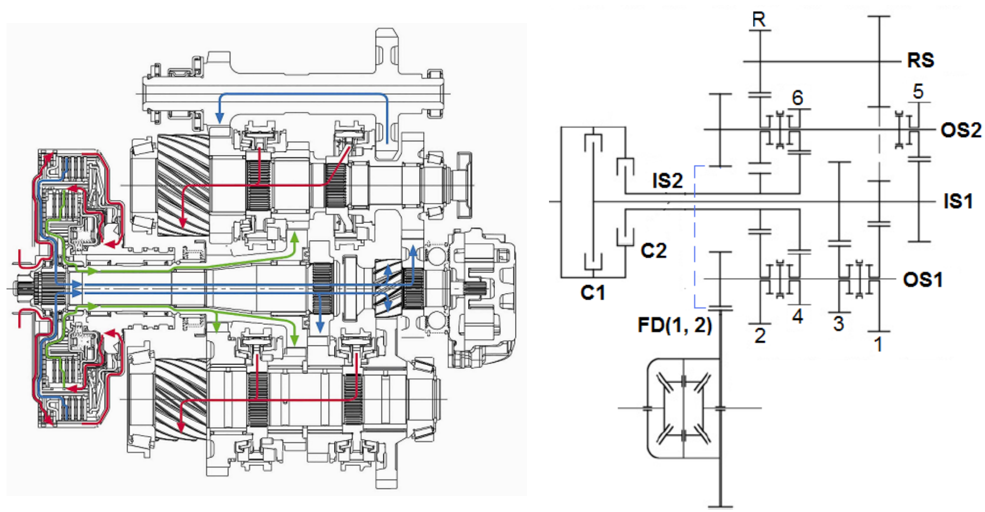


Fig. 1. Cross-section and layout of DSG 02E.

The transmission is modeled as a 4 DOF (degree of freedom) system. This complexity level is similar or higher than that of other models used for gearshift dynamics and control [10], [11], [12]. The physical mechanical model includes: inertias, clutches, gear sets, synchronizers and final drive (fig. 2).

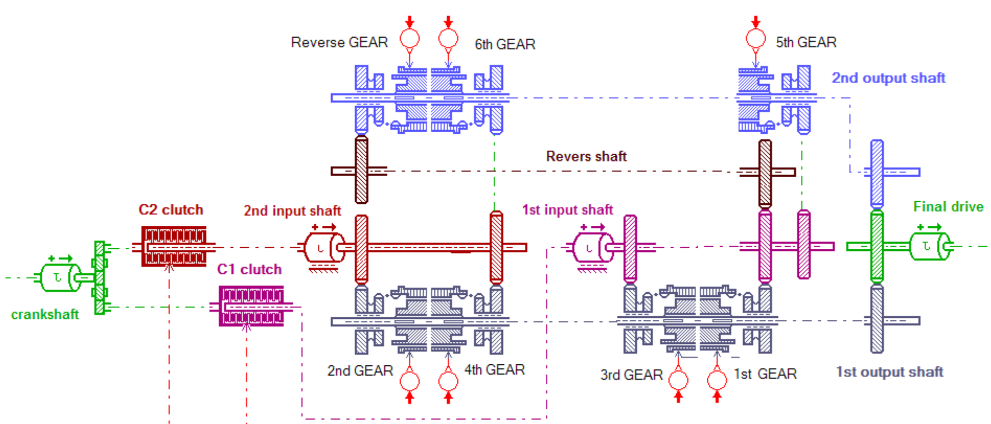


Fig. 2. AMESim sketch of DSG 02E.

The AMESim sketch is close to the technical plan of the gearbox and this facilitates the recognition of the different elements. It includes 2 clutches, 2 input shafts, 2 outputs shafts and the final drive. All the connected inertias are reduced at the corresponding shaft.

The gears submodels include efficiency and for the idle gears a submodel with losses is used. With a proper setting of the inertia friction parameters, the bearing and the lubricating losses can be introduced. The stiffness of the shafts and gear pairs are considered inside the synchronizers models that are particularly feet for real time simulation.

3. Hydraulic control system configuration and model description

The hydraulic actuation of the clutches is also modeled. The new hydraulic control systems of the transmission clutches are based on direct acting solenoids with high flow instead of pilot solenoids controlling flow control valves which have higher time delays [12], [13].

Fig. 3 represents a simplified clutch hydraulic control system including one clutch piston and the proportional control valve. The clutch pack is modeled as a gap and end-stop. The line pressure is a function of a command signal.

The complexity of the model is adequate for the study of transmission clutch control [12], [14].

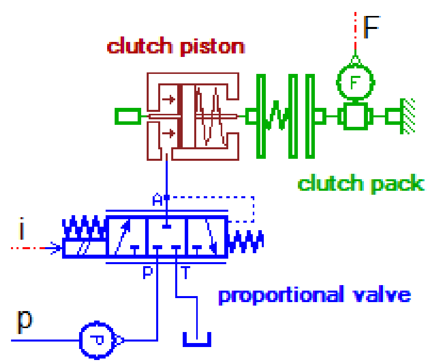


Fig. 3. Simplified hydraulic system for clutch control.

3. Component models

The models used for the components (referred as submodels) must be compatible with real time simulation. For the mechanical transmissions the most critical submodels are those of the clutch and the synchronizer. The modeling of the hydraulic control circuit is also difficult since the usual hydraulic component submodels are not adapted for real time simulation [4], [15].

The following convention is applied to all AMESim submodels:

- The input variables are represented with an arrow toward the icon and

the output variables with an arrow starting from the icon;
- For variables which have a direction associated with them, a positive sign is in the direction of the arrow.

Clutch model

The friction modeling constitutes the base of all clutch models. Many examples of friction models are proposed but not all are real-time compatible when used for clutch modeling [16].

The clutch models studied in [16] are those based on the following friction representations: Coulomb, combined (Coulomb and viscous friction), hyperbolic tangent, classic with switch and Karnopp. It was shown that the most adequate for real-time application is the Karnopp model.

In a successive step the Reset-Integrator friction model was also tested using the same procedure and RT platform and produced a maximum task execution time reduced with 14-22% when compared with the Karnopp model.

A multidisc clutch model which computes the friction torque base on the normal force that acts on the clutch pack and use a Reset Integrator friction representation is chosen for this application. Stick-slip behavior and Stribeck effect are introduced.

Synchronizer model

The synchronizers cause discontinuities in the passage from synchronization to engaged (coupled) gear: the interaction between elements, initially due to friction is then realize by the contact of the dog-teeth. A number of modeling techniques for the synchronizer are available.

The simplest use only the friction torque between the sleeve and the idle gear both for synchronization of the two velocities and locking of the gear [17]. The model with 2 DOF can be satisfactory for fuel consumption studies but is not suitable for comfort ones. In order to transmit the maximum torque it needs a friction torque more than 10 times greater than the real one. This will produce an extremely reduced synchronization time.

Using a coupling logic that increase the friction torque when the velocities are matched it was possible to use this model also for comfort studies [18]. However, this solution cannot be simulated with a fixed step solver using usual step dimensions for real-time applications.

Models with variable structures are also used. These models have 2 DOF for uncoupled state and 1 DOF for the coupled state [10]. The drawback of the model is that it forms an integrated part with the rest of the system. Therefore, the transmission model equations have to be tailor-made for each configuration. Variations of this model are widely used since they allow efficient simulations.

Complex synchronizer models are also available. With such a model, one can perform detailed dynamic analysis of the synchromesh mechanism during any phase of the synchronizing process. Nevertheless, this model is too complex, difficult to be parameterized and increases the simulation time. For example, a model proposed in [17] has 5 DOF (3 rotations – for the blocking ring and the two coupled elements – and 2 translations – for the sleeve and blocking ring) and 15 state variables.

A special submodel was developed for the synchronizer, fig. 4. The new model is build using hybrid modeling techniques (using continuous models triggered by discrete events) and ensures three phases: disengaged (no torque is transmitted), synchronization (the synchronizer is similar with a clutch) and engaged (the synchronizer is similar with a shaft).

The submodel has 4 ports: one for the command and 3 for the mechanical connection to the shaft and the idle gear. The engagement and disengagement are commanded using as input a command signal *com* or a force applied to the sleeve F_1. The port number is used as index for the torque T and the angular speed ω passed at every connection port.

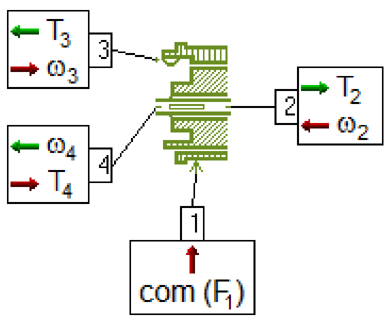


Fig. 4. Synchronizer icon with input and output variables.

The relative speed ω_r is used for switching between synchronization and engaged phase and for the computation of the transmitted torque in these states.

$$\omega_r = \omega_2 + \omega_3 \tag{1}$$

The equations for the disengaged phase are:

$$T_3 = 0Nm\,; \qquad \frac{d\theta_r}{dt} = 0rad/s \tag{2}$$

In synchronization phase the transmitted torque is calculated from the synchronization torque T_S using a hyperbolic tangent friction model and the angular deformation θ_r is computed using this torque.

$$T_3 = T_S \cdot \tanh\left(\frac{2\omega_r}{\omega_0}\right); \qquad \theta_r = \frac{T_3}{k} \tag{3}$$

where ω_0 is a parameter that determines the speed of the transition from -1 to +1 and k is the stiffness of the elastic element.

When engaged, the synchronizer acts as an angular spring-damper having the stiffness k and the damping coefficient b. The initial angular deformation is set from the synchronization phase.

$$\frac{d\theta_r}{dt} = \omega_r; \qquad T_3 = k \cdot \theta_r + b \cdot \omega_r \tag{4}$$

The stiffness is that of the shaft and gear.

Independent of the state, the rest of the output variables are computed as follows.

$$\omega_4 = \omega_2; \qquad T_2 = T_3 + T_4 \tag{5}$$

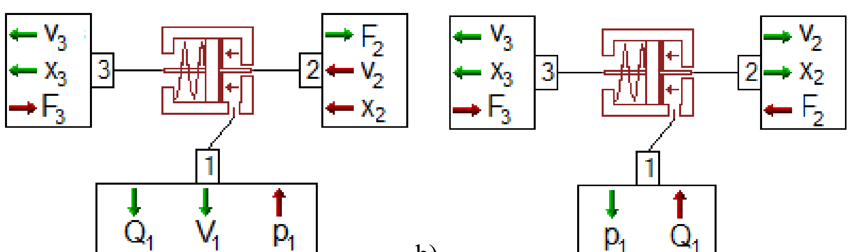
The synchronization torque is considered constant when a command signal is used or is computed from the synchronizer geometry when the force on the sleeve is used. For a single cone synchronizer with the angle α, the friction radius r_s and the friction coefficient μ it is:

$$T_S = \frac{\mu \cdot r_s}{\sin(\alpha)} \cdot F_1 \tag{6}$$

When compared to a complex model it produces an error of only 3% for the coupling time. The speed is dramatically improved, for 0.8 simulated seconds the computing time is reduced from 4.018 s to 0.036 s.

Piston model

Since the usual hydraulic component models are not adapted for real-time simulation a special piston model is used. This piston model includes the velocity integration to suppress the need of the mass model [15].



a) b)

Fig. 5.a) The standard HCD piston model (BAP017); b) The real time piston model (BAP10RT)

The standard model BAP017 (fig. 5.a) uses the piston velocity v_2 (obtained from the mass model) to compute the flow rate Q_1 and the pressure p_1 to compute the spool force.

$$Q_1 = A_S \cdot v_2; \qquad \sum F = A_S \cdot p_1 \tag{7}$$

where A_S is the piston area.

The real time model BAP10RT (fig. 5.b) use the flow rate (usually computed inside an orifice model) to compute the velocity and then the displacement. The pressure results from the force balance.

$$v_2 = \frac{dx_2}{dt} = \frac{Q_1}{A_S}; \qquad p_1 = \frac{\sum F}{A_S} \tag{8}$$

Pressure control valve model

The spool-type pressure regulator is modeled in detail using dedicated submodels for real-time applications and the resulted model is encapsulated in a supercomponent, fig. 6.

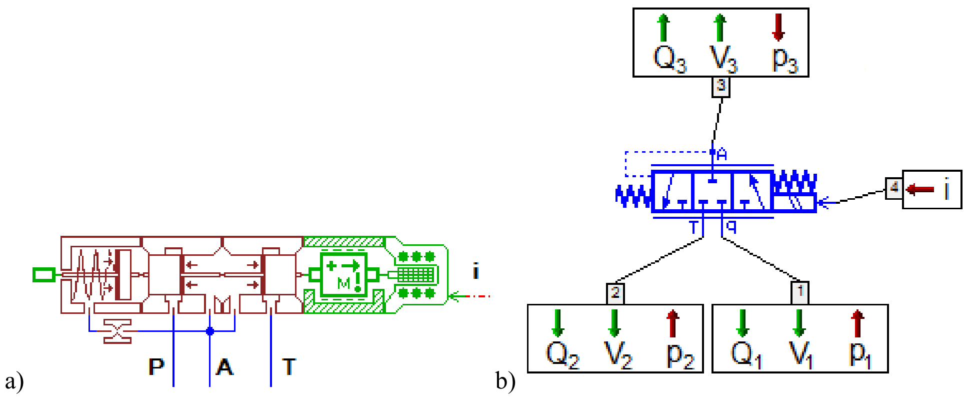


Fig. 6. Proportional pressure control valve: a) AMESim model; b) supercomponent icon.

To control the pressure valve, a linear variable reluctance machine is used. Because the steady-state value of the electromagnetic force F_e is approximately proportional to the coil current i, and considering the servovalve current driver and electromagnetic coil dynamics, a first-order lag has been introduced to model the current-controlled electrical machine.

$$\dot{F}_e = -\frac{1}{\tau_e} \cdot F_e + \frac{1}{\tau_e} \cdot k_f \cdot |i| \tag{9}$$

where k_f is the electro-mechanical constant; τ_e is the solenoid time constant.

The friction and the displacement limitations (end-stops) are introduced using a special mass model developed for real-time applications. To avoid algebraic loops the velocity used to compute the friction forces is obtained by applying a first order lag (with the time constant τ) to the velocity at the ports v_p.

$$\dot{v} = \frac{v_p - v}{\tau} \tag{10}$$

The model employs elastic end-stops for both directions.

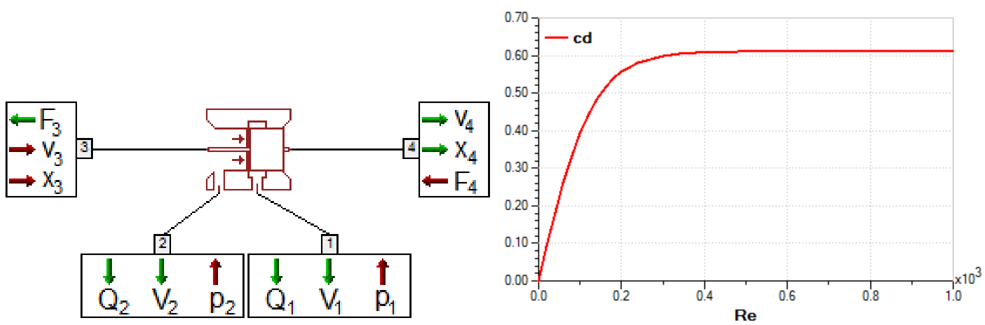
The spool is modeled using the BAO11RT (fig. 7.a) and BAO12RT submodels. These represent the one-dimensional motion of an annular section valve with sharp edges in two different causalities (variables associated with ports 3 and 4 are interchanged).

Inside these models, the following variation of the flow coefficient is considered.

$$C_d = C_{d\_\max} \cdot \tanh\left(\frac{2 \cdot R_e}{R_{et}}\right) \tag{11}$$

where R_e, R_{et} are the Reynolds number and the transition Reynolds number.

Using values characteristic for this type of annular orifice ($C_{d\_max} = 0.61$ and $R_{et} = 260$ [19]), the flow coefficient variation from fig. 7.b results.



a) b)

Fig. 7. a) BAO11RT icon with input and output variables; b) Orifice flow coefficient variation.

The pressure loss Δp on the orifice and the average pressure p_m are used to compute the flow through the orifice Q_1.

$$\Delta p = p_1 - p_2 \ ; \qquad p_m = \frac{p_1 + p_2}{2} \tag{12}$$

$$Q_1 = C_d \cdot A \cdot \sqrt{\frac{2 \cdot |\Delta p|}{\rho}} \cdot sign(\Delta p) \cdot \frac{\rho(p_m)}{\rho(p_0)} \tag{13}$$

The flow at the port 2 is corrected with the contribution due to the spool displacement x_3.

$$Q_2 = Q_1 + \frac{dx_3}{dt} \cdot \frac{\pi}{4} \cdot \left(d_s^{\ 2} - d_r^{\ 2}\right) \cdot \frac{\rho(p_m)}{\rho(p_0)} \tag{14}$$

where d_s and d_r are the diameters of the spool and the rod.

In fig. 8 simulated pressure vs. current steady-state characteristics of the proportional pressure control valve is shown. A good correlation with experimental data given in [14] is obtained.

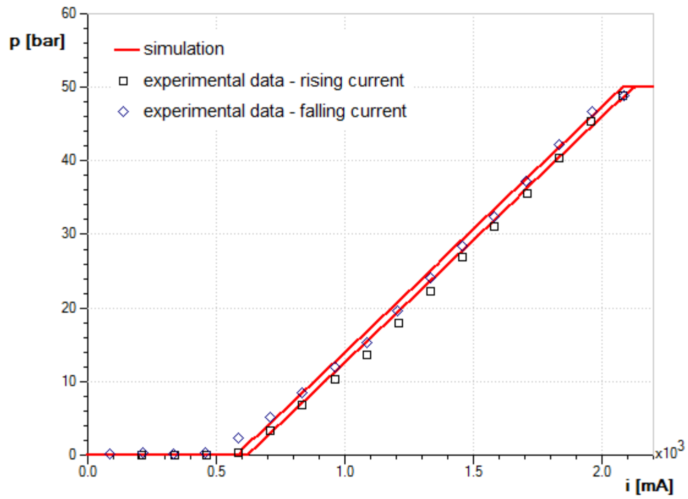


Fig. 8. Servovalve pressure vs. current steady-state characteristics.

4. Global model

Gearshift dynamics can only be simulated if the input and output torques of the transmission represent a real-life vehicle maneuver. Therefore, at least the engine and the longitudinal dynamics of the vehicle have to be modeled beside the transmission. Fig. 9 shows the schematic structure of the powertrain global model.

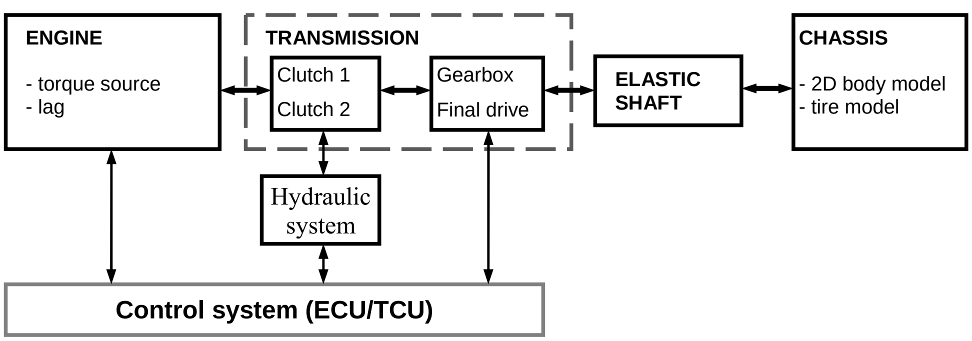


Fig. 9. Schematic structure of the powertrain global model.

The engine is modeled as a torque source using a look-up table and then applying a variable lag to the torque. Therefore, there is a delay between the ECU torque request and the delivered engine torque. This delay depends on the engine configuration (naturally aspirated or turbocharged) and on the torque request variation, increase or decrease. It does not include the rotary inertia, so this was added on the input of the clutches to take into account the complete engine dynamic. The model can compute the fuel consumption from user defined maps.

The chassis model includes a 2D dynamic vehicle (body) model, front and rear suspension models and tires models.

The vehicle is a 2D dynamic submodel of car with mass transfer due to the car pitching. The model has 3 DOF: pitch rotation, longitudinal and vertical translations.

The suspension submodel is used to introduce damping, stiffness and spindle mass in 3 planar degrees of freedom: longitudinal and vertical translation and self rotation. Road slope angle is taken into account to consider gravity field influence on spindle mass.

A tire submodel generates the contact force at the tire/road interface and rolling resistance for all situations: braking, accelerating or stopped even with fast dynamics (up to 40Hz). When the vehicle is moving, the force is modeled with the classical Pacejka formula [7] depending on the longitudinal slip and the vertical force of the tire. When it is stopped, the force is modeled by damper-spring behavior.

A Simulink interface bloc corresponding to the TCU is added for the real-time model. The AMESim generated C code is used by Simulink and employing the Real Time Workshop module the code is compiled and load on the RT platform.

The number of states resulted in this global model is 46 from which 43 corresponds to the powertrain AMESim model and 3 to the TCU Simulink model.

5. Simulation results

Because the model was build from the start for real time simulation only, small modifications of parameters are needed to ensure the simulation with a fixed step solver. These modifications usually consist of adding more damping in the system. The target value for the step size is 0.5 ms, a usual value for powertrain real-time simulation [1].

The model is tested offline for accuracy. The results of the model with parameters tuned for real-time obtained with an Euler solver with 0.5 ms step size are compared with those of the reference model obtained with the standard variable step solver, fig. 10.

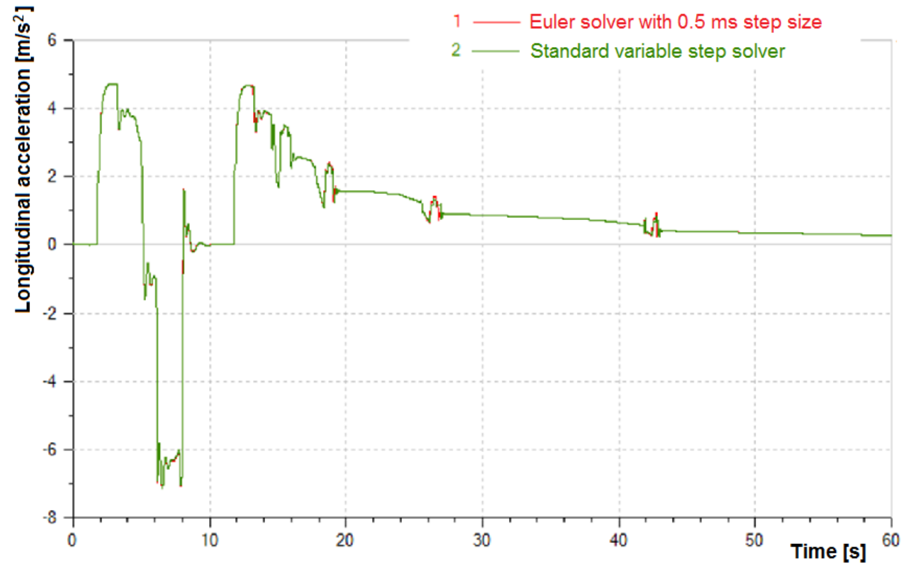


Fig. 10. The vehicle horizontal acceleration obtained with different solvers.

The coupled model AMESim-Simulink is tested for the TCU shift logic. The test cycle consist of two launches from standstill at wide open throttle and at half engine load followed by brake to zero velocity. Fig. 11 shows the results of this test.

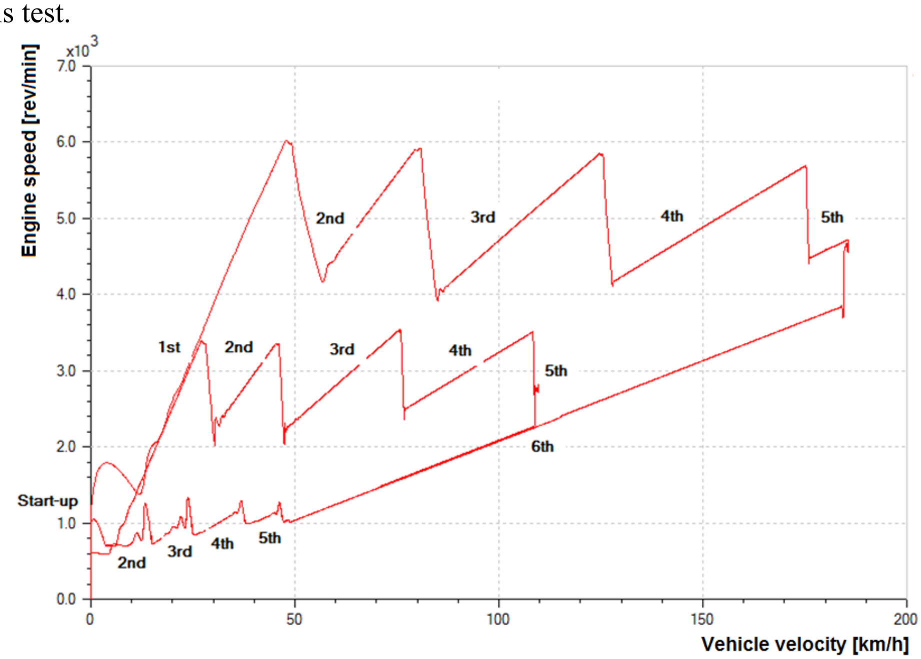


Fig. 10. Testing of the TCU shift logic.

Finally, the model was simulated in real time with a sample rate of 2 kHz on the dSPACE RT platform equipped with ds1006 processor board (2.6 GHz). The results show a turnaround time of maximum 0.035 ms and no overruns. The model is fast enough to allow the use of complex control software for the transmission. Fig. 11 shows the result of a wide open throttle start-up as view in the dedicated Control Desk interface.

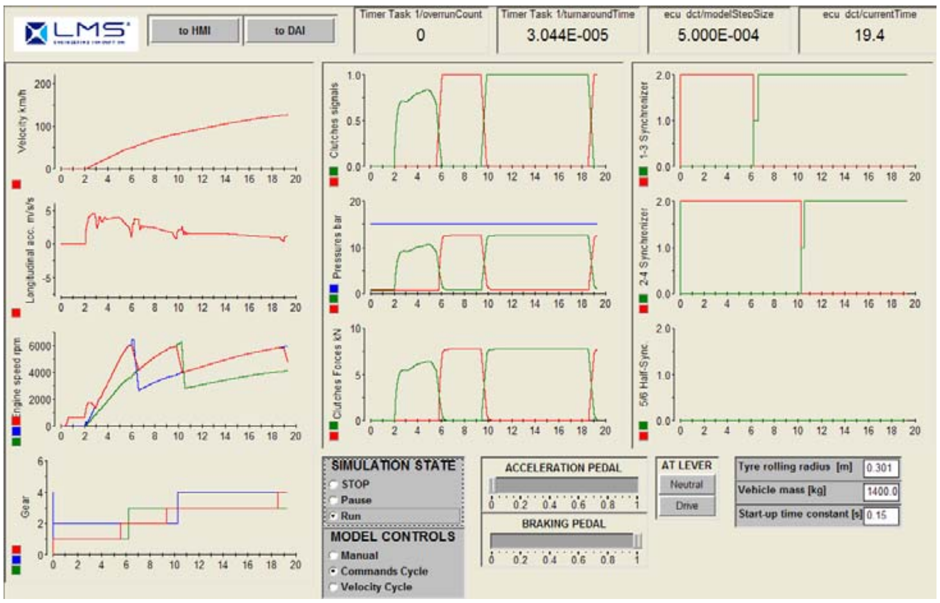


Fig. 11. Real-time simulation results (Control Desk interface).

6. Conclusions

It was demonstrated that is possible to simulate in real time high-fidelity models of DCT equipped powertrains coupled with complex vehicle dynamics models. The model includes also the hydraulic control of the clutches beside the mechanical part.

These models are extremely useful for HiL simulations both for fuel consumption and comfort issues. It is possible to test and calibrate the TCU but also other control units that interact with the transmission control for example the ECU.

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