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AUTOMOTIVE CLUTCH MODELS FOR REAL TIME SIMULATION

Marius BĂȚĂUȘ[*], Andrei MACIAC[*], Mircea OPREAN[*], Nicolae VASILIU[**]

[*] University "Politehnica" of Bucharest, Automotive Engineering Department
[**] University "Politehnica" of Bucharest, Fluid Power Department
Corresponding author: Marius BĂȚĂUȘ, E-mail: mvbataus@yahoo.com

One of the key issues of drivetrain real time simulations is the modeling of clutches. This paper aims to show an overview of usual clutch models with emphasis on their compatibility with real time simulation. The models are tested using a torsional model of the powertrain with parameters values typical for medium class passenger cars.

Key words: Clutch, Friction, Modeling, Real time, Simulation, xPC Target.

1. INTRODUCTION

Almost all car manufacturers and suppliers use hardware-in-the-loop (HIL) simulations for testing or calibration. HIL can be described as having the physical part of a system (for instance, part of a vehicle) as a simulation one while the electronic control unit (ECU) is either a production or a prototype one. This will make it possible to test the ECU before it is installed into a vehicle and so to eliminate bugs in the ECU in early stages of the development and also to reduce the time and costs of calibration. For HIL testing of the vehicle dynamics controllers it is needed an elastic powertrain model that includes the launch device, [15].

Internal combustion engines have a minimum stable rotational speed. The speed difference between the lowest engine operating speed and the stationary transmission input shaft speed has to be bridged by a speed converter (launch device). The main launch devices to have established themselves in motor vehicles are [11]: friction (dry or wet) clutch for manual, automated or dual clutch transmissions and torque converter for conventional fully automatic transmissions.

The friction modeling constitutes the base of all clutch models. Many examples of friction models are proposed: hyperbolic tangent model [2], Karnopp model [10], Reset-Integrator model [8], Dahl model [13], Bristle Model [8], LuGre model [1] etc. The model of friction must be chosen taking into account the purposes of use. In some cases it is desirable to have a model which provides an insight into the physical mechanisms of the friction interface. In others it suffices with a model that can predict the global, qualitative behavior of a system with friction. Furthermore, there may be limitations on the computational complexity of the model. A number of global reviews ([13]) and studies of friction models ([8], [2]) give indications regarding the accuracy, ease of implementation (for numerical simulation), computational efficiency and application area of different models. The problem of clutch modeling with different friction models is also presented ([1], [6], [5]) including some works on real time simulation ([14], [4], [9]).

To turn an offline plant & control model into a real-time model is necessary to ensure that the plant model runs with fixed step solver. This can make real-time simulation more challenging than desktop simulation. Usually some simplifications should be done but with good understanding of real-time needs simplifications can be kept small. Moreover, for a simulation to execute it in real time, the amount of time spent in calculating the solution for a given time step (execution time) together with the amount of time spent in processing inputs, outputs, and other tasks must be less than the length of that time step. It is necessary to leave sufficient safety margin to avoid an overrun when simulating in real time, [12].

This paper describes some of current available theoretical models of friction clutches that are extensively used for control problems. These include the hyperbolic tangent model, classic static model and

Karnopp model. Furthermore tests are done in order to establish those that cope with real time simulation demands.

2. FRICTION CLUTCH MODELS

The friction clutch models were developed based on existing models of friction. Some simplifications are made:

- the thermal effects are not taken into account because they have a low dynamic and are not relevant for the purpose of this paper
- the friction torques (dynamic T_{fd} and static T_{fs}) are computed using a normalized command signal *com* as a fraction of the maximum friction torque (dynamic T_{fmaxd} and static T_{fmaxs}), eq. (1) and (2).

$$T_{f\,d} = com \cdot T_{f\,max\,d} \tag{1}$$

$$T_{f\,s} = com \cdot T_{f\,max\,s} \tag{2}$$

The maximum friction torques can be computed using the usual expressions that include the friction radius, the friction coefficient and the number of friction surfaces.

When it is used in a control loop the clutch model must allow the simulation of stick-slip behavior, Stribeck effect and positive slope friction coefficients, [5], [6].

2.1 Coulomb friction model

The most commonly friction model used is the Coulomb friction model, which, for the clutch, can be formulated as:

$$T_c = \begin{cases} T_{f\,d} \cdot sign(\omega_r) & if \quad \omega_r \neq 0 \\ T_{app} & if \quad \omega_r = 0 \end{cases} \tag{3}$$

where T_c is the torque transmitted through the clutch, ω_r is the relative speed and T_{app} is the torque applied on the plates.

In order to maintain the same causality of the model, the Coulomb friction model is often simplified as:

$$T_c = T_{f\,d} \cdot sign(\omega_r) \tag{4}$$

The Coulomb friction model is illustrated in Fig. 1.a. It can be easy implemented using general-purpose simulation software packages, as example Matlab/Simulink (Fig. 2.a).

2.2 Combined Coulomb and viscous friction model

Because the equation of motion for dynamic systems is strongly non-linear with a Coulomb friction model, for some applications, a viscous friction model can be used instead. Such a model is considerably easier to simulate, but the representation of the friction is inadequate for clutch modeling. Instead of this a combination of the viscous friction model and the Coulomb friction model could be advantageous. Such a model will have the following form:

$$T_c = \begin{cases} T_{f\,d} \cdot \min(2\,\omega_r / \omega_0, 1) & if \quad \omega_r \geq 0 \\ T_{f\,d} \cdot \max(2\,\omega_r / \omega_0, -1) & if \quad \omega_r < 0 \end{cases} \tag{5}$$

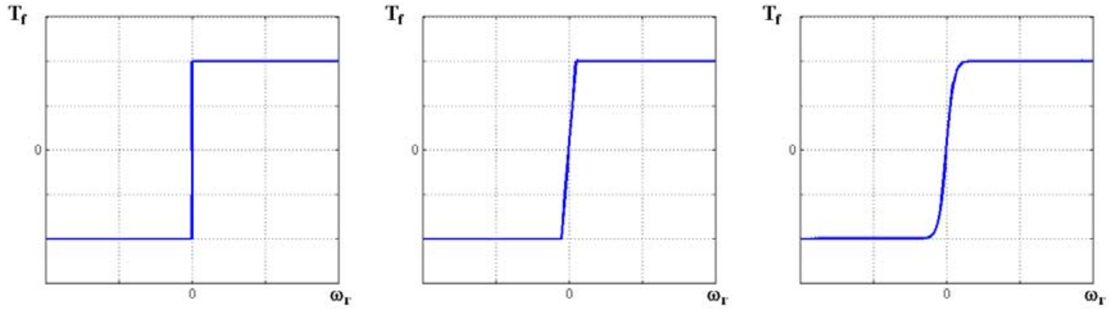
where ω_0 is a parameter that determines the speed of the transition from –1 to +1.

The combined Coulomb and viscous friction model is illustrated in Fig. 1.b. It can easily be modeled in Matlab/Simulink by using the saturation block (Fig. 2.b).

The hyperbolic tangent model is a simple friction model that employs a *tanh* function to ensure the transition through zero and limit the friction force (torque), eq. (6). This model behaves like the combined

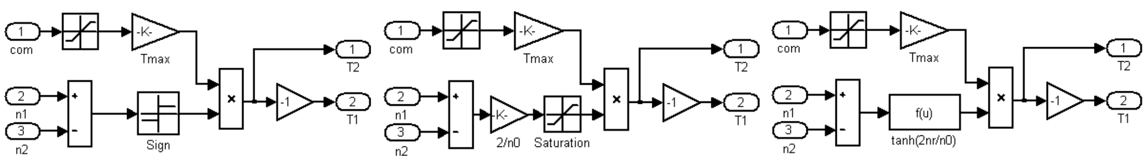
Coulomb and viscous friction model, but is more numerically stable due to the use of a perfectly continuous function, Fig. 1.c.

$$T_c = T_{f\,d} \cdot \tanh\left(2\frac{\omega_r}{\omega_0}\right) \tag{6}$$



a) b) c)

Fig. 1 The friction force variation as function of relative velocity for: a) Coulomb friction model; b) combined Coulomb and viscous friction model; c) tanh friction model



a) b) c)

Fig. 2 Simulink top level diagram of simple clutch models: a) Coulomb friction model; b) combined Coulomb and viscous friction model; c) tanh friction model

2.3 Classic static model with switch

Well known for its simplicity, the classic static model with switch is based on the equality of acceleration in the phase of locked state. In this mode, the equilibrium of the two moving masses (inertias) leads to the friction force (torques). In motion, the force is only computed from the dynamic friction force. This model must include the inertia of the input and output parts. The model is described in table 2. As shown in Fig. 3.b it allows the modeling of stick-slip behavior.

Table 1 The description of classic static friction model

State	Slip (unlocked)	Stick (locked)
Torque capacities	$T_{f\,d} = com \cdot T_{f\,max\,d}$	$T_{f\,s} = com \cdot T_{f\,max\,s}$
State equations	$I_1 \cdot \dot{\omega}_1 = T_1 - b_1 \cdot \omega_1 - T_c$ $I_2 \cdot \dot{\omega}_2 = T_c - b_2 \cdot \omega_2 - T_2$	$(I_1 + I_2) \cdot \dot{\omega} = T_1 - (b_1 + b_2) \cdot \dot{\omega} - T_2$
Friction torque	$T_c = \operatorname{sgn}(\omega_1 - \omega_2) \cdot T_{f\,d}$	$T_c = \dfrac{I_2 \cdot T_1 - I_1 \cdot T_2 - (I_2 \cdot b_1 - I_1 \cdot b_2) \cdot \omega}{I_1 + I_2}$
From slip to stick	$\omega_1 = \omega_2$ and $\lvert T_c \rvert \le T_{f\,s}$	
From stick to slip		$\lvert T_c \rvert > T_{f\,s}$

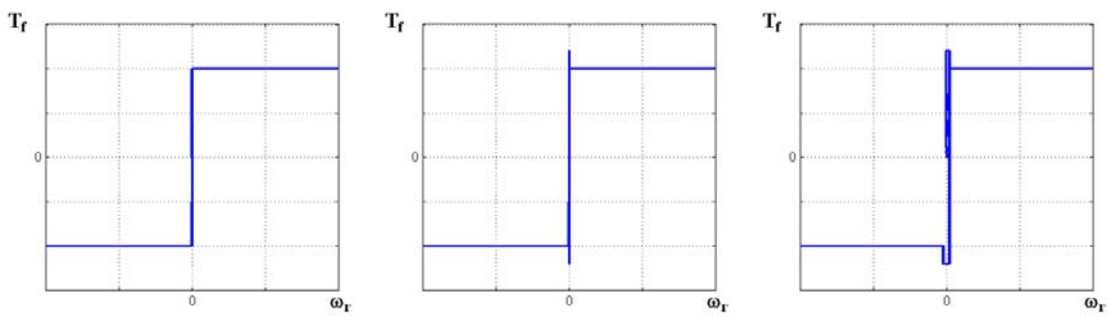
The notations in table 1 are: ω is the angular speed, I is the moment of inertia and T is the torque. The subscripts used are: *1* for the input (engine) and *2* for the output (transmission).

The method employed to implement this model in Simulink is to use two different dynamic models and switch between them at the appropriate times, [17]. The switching between two dynamic models is performed with care to ensure that the initialized states of the new model match the state values immediately prior to the switch. The simulation can use one subsystem while the clutch is slipping and the other when it is locked by employing the enabled subsystems feature of Simulink. A diagram of the Simulink model is shown in Fig. 4.a.

2.4 Karnopp model

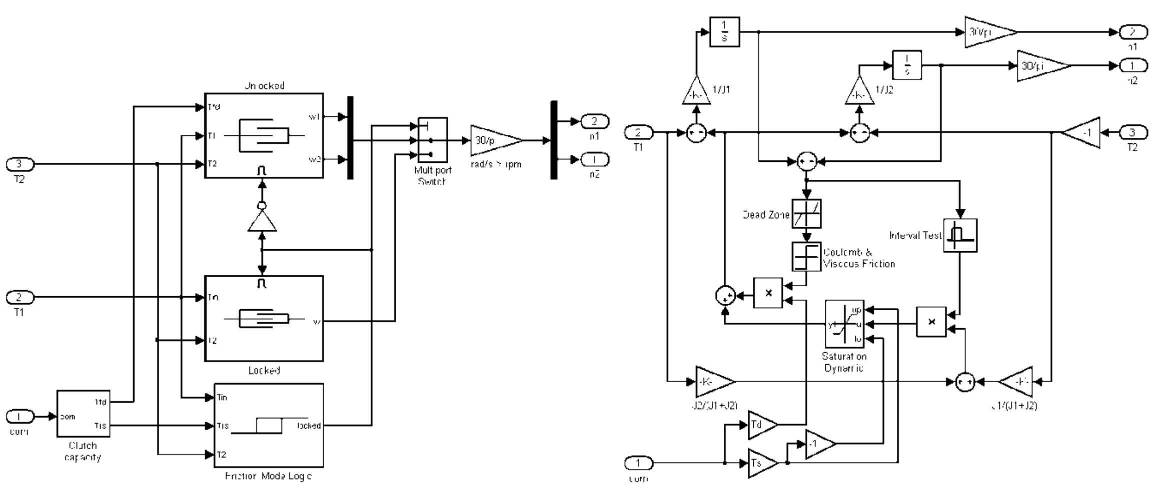
The Karnopp model (proposed for simulation purposes) was developed to overcome the problems with zero velocity detection and to avoid switching between different state equations for sticking and sliding, [10]. The model defines a zero velocity interval, $|v| < DV$. For velocities within this interval the internal state of the system (the velocity) may change and be non-zero but the output of the block is maintained at zero by a dead-zone. Depending on if $|v| < DV$ or not, the friction force is either a saturated version of the external force or an arbitrary static function of velocity. The interval $\pm DV$ can be quite coarse and still promote so called stick-slip behavior, Fig. 3.c.

The drawback with the model is that it forms an integrated part with the rest of the system. The external force is an input to the model and this force is not always explicitly given. The model equations therefore have to be tailor-made for each configuration. Variations of the Karnopp model are widely used since they allow efficient simulations. The zero velocity interval does, however, not agree with real friction. A Karnopp clutch model implemented in Simulink is presented in Fig. 4.b.



a) b) c)

Fig. 3 The friction force variation as function of relative velocity for: a) classic switch model; b) classic switch model with stick-slip effect; c) Karnopp model



a) b)

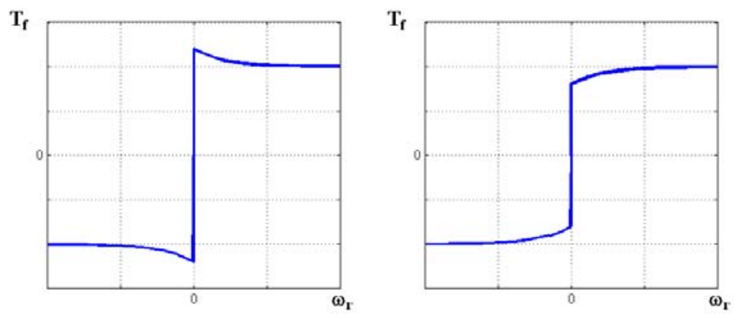
Fig. 4 Simulink top level diagram of complex clutch models: a) classic switch model; b) Karnopp model;

2.5 Stribeck effect

The Stribeck effect is a friction phenomenon that arises from the use of fluid lubrication and gives rise to decreasing friction with increasing velocity at low velocity, Fig. 5.a. Therefore, is recommended to include this effect in the wet clutches models. It is possible to implement the Stribeck effect, for example as used in [2] by using the following equation:

$$T_f = T_{fd} + \left(T_{fs} - T_{fd}\right) \cdot e^{-\left(|\omega_r|/\omega_s\right)^i} \tag{6}$$

This effect was implemented in the classic static model with switch and the Karnopp model. This equation also enables the modeling of positive slope friction coefficients, Fig. 5.b.

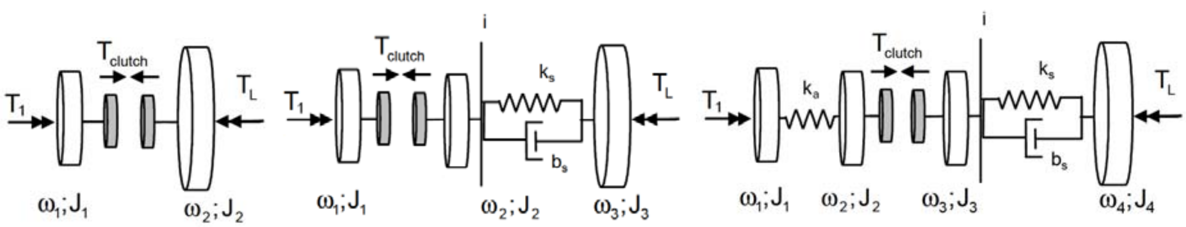


a) b)

Fig. 5 The friction force variation as function of relative velocity for: a) model with Stribeck effect; b) positive slope friction coefficients

3. SIMULATION RESULTS

To realize the tests it is used a torsional model of the powertrain. Usual models applied in control problems have a limited number of degrees of freedom (DOF). The most employed torsional models (Fig. 6) are with: 2DOF ([7]), 3DOF ([6], [16]) and 4DOF ([5]). When choosing a model for real-time applications it is important to consider the limits in terms of step size and CPU calculation capacity. The current typical sampling rates are between 0.5-1 ms, [15]. A model with an increased number of DOF includes small inertias and high stiffness which, in turn, generate high dynamics that cannot be resolved with usual step size and will produce overruns. In order to run a more complex model it is possible to accelerate the calculation by using multi-rate and reduce the scheduler overhead, [12], [9]. Because the real-time simulation is conducted at a single rate a 3DOF model is considered sufficient. This model contains the following simplification: the vehicle mass is transformed in the equivalent rotational inertia and all the elements after the gearbox are transformed in their engine-side equivalent. Parameters used are within a range of values typical for medium class passenger cars. No damping and no viscous friction is added in order to better show the clutch models behavior and limits.



a) b) c)

Fig. 6 Simplified models of powertrain: a) 2DOF model, b) 3DOF model, c) 4DOF model

The test models are implemented in Simulink using the developed clutch models encapsulated in subsystems (submodel). Because some clutch models include the inertia of the input and output parts, two structure of the powertrain model result as shown in Fig. 7. The number of states resulted is 4 for all the models except the classic static model which has 5 states.

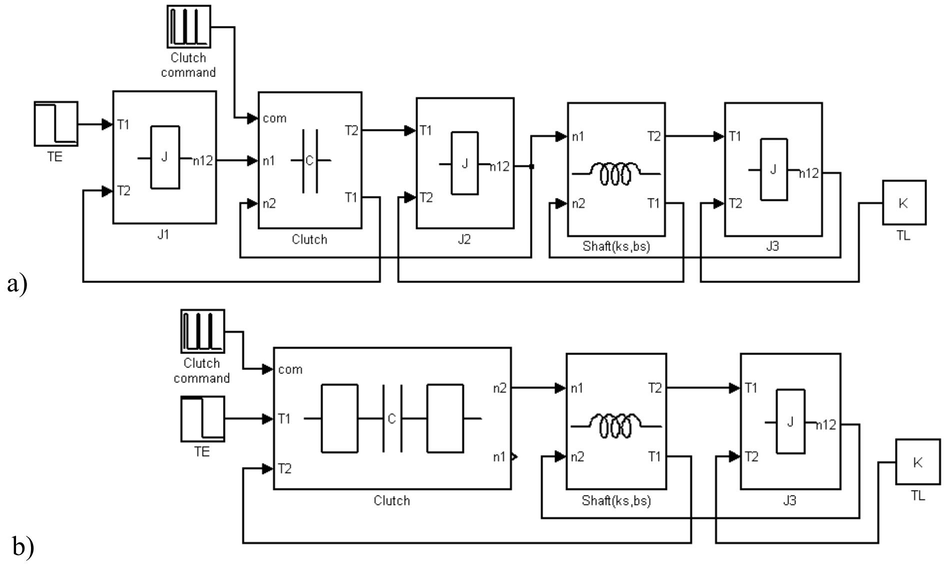


Fig. 7 Simulink top level diagram of a 3DOF powertrain model for: a) Coulomb, combined Coulomb-viscous friction and hyperbolic tangent clutch models; b) classic static model with switch and Karnopp model

The models are tested first using variable step solvers. From the step size point of view an appreciation can be made regarding the possibility to use a fix step solver. It was discovered that the Coulomb friction model is not adequate for such highly oscillating system as the powertrain. The combined Coulomb and viscous friction model eliminates the difficulty in determining the friction force at zero sliding speed both at start up and at direction change. The hyperbolic tangent model behaves like the other combined model, but is probably more numerically stable. As a result of this test only the hyperbolic tangent, classic and Karnopp models are further tested for real time application.

In the next step the models are run using different fix step solvers with a 1 ms step size and the results are compared with that obtained with variable step solvers. We determine three situations: the model is adequate (Fig. 8.a), the model diverge (Fig. 8.b) and the model produce results with numerical noise (Fig. 9).

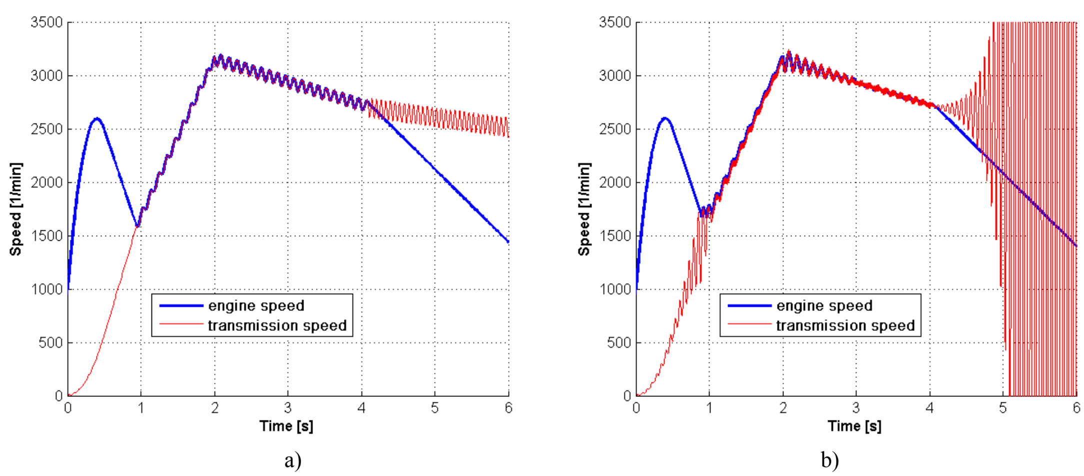


Fig. 8 Offline simulation results with fix step solver: a) good results; b) divergent simulation

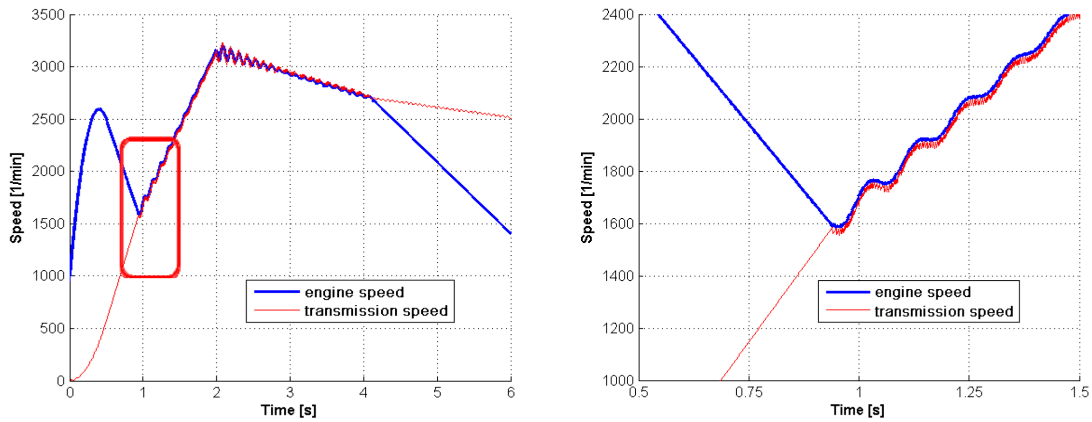


Fig. 9 Offline simulation results with numerical noise (see the left detail)

If the results are unacceptable, the parameters of friction model are modified in order to allow simulation with fix step solvers while maintaining small errors compared with the original model (Fig. 10).

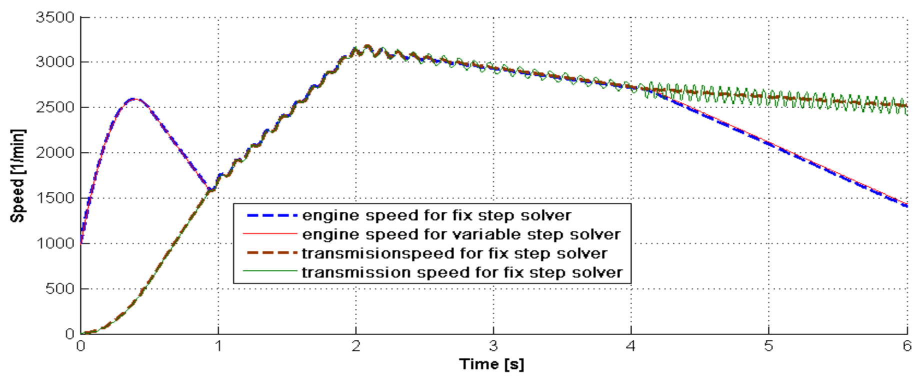


Fig. 10 Verification of the fix step solver results against those obtained with variable step solver

The models retained are then tested on the real time platform. For each case we construct a complex model formed by 6 drivetrain models with different command inputs. The efficiency of the models is evaluated on an Intel Pentium III (801MHz) that was running xPC Target from MathWorks. The results are centralized in table 2.

Table 2 Task execution times (TET) for different clutch models

	No stick-slip / No Stribeck			stick-slip / No Stribeck		stick-slip / Stribeck	
	Tanh	Clasic	Karnopp	Clasic	Karnopp	Clasic	Karnopp
Average TET [s]	3.998×10^{-5}	3.815×10^{-5}	2.976×10^{-5}	3.761×10^{-5}	2.966×10^{-5}	4.993×10^{-5}	3.940×10^{-5}
Minimum TET [s]	4.067×10^{-5}	3.500×10^{-5}	2.633×10^{-5}	3.604×10^{-5}	2.633×10^{-5}	4.667×10^{-5}	3.633×10^{-5}
Maximum TET [s]	7.133×10^{-5}	8.900×10^{-5}	8.000×10^{-5}	9.133×10^{-5}	7.600×10^{-5}	10.233×10^{-5}	8.567×10^{-5}

4. CONCLUSIONS

A number of drivetrain clutch models proposed in the technical literature and implemented in commercial simulation software are tested using the same simulation platform (Malab/Simulink). It was showed that the hyperbolic tangent, the classic and the Karnopp friction models can be use to create clutch models able to run in real time when used as drivetrain launch devices. Also the Coulomb model was discarded for this application.

All the models are efficient and provide a good degree of accuracy when they are well tuned. The hyperbolic tangent model seems to be the most efficient but it has the lowest accuracy. When tuned for real time the model introduce an artificial damping. The Karnopp model is more efficient that the classic model. It produces a maximum TET reduced with 10-17 % in all the situations. When the static friction is bigger that the dynamic friction the models have a different behavior: for the classic model the maximum TET increases with 2.5% while for the Karnopp model the maximum TET decreases with 5%. The introduction of the Stribeck effect increases the TET of both models (15% for the classic model and 7% for the Karnopp model).

More research is necessary to test the models behavior and limits when damping and close loop control are added.

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