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The CONAT International Congress is a traditional scientific event initiated by TRANSILVANIA University since 1965. The XI-th Edition is organised by SIAR (the Society of Romanian Automotive Engineers), Transilvania University of Brasov, Romania - Department of Automotive Engineering and SAE International, under the patronage of FISITA and EAEC.

At the previous 10 editions of the Congress, valuable scientific papers were presented, by teaching staff of the technical universities, as well as by specialists working in the design and research departments, manufacturing companies or in the operation division of automotive transport companies.

The Congress intends to facilitate a profitable exchange of information among specialists in automotive engineering, and to discuss the new challenges that are facing the automotive industry over the future.
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REAL TIME SIMULATION OF DRIVETRAIN LAUNCH DEVICES

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KEYWORDS - real time, launch devices, clutch, torque converter, xPC Target

ABSTRACT – To test and calibrate the transmission control software, or even the entire ECU, it is used a hardware-in-the-loop (HIL) simulation. For this purpose it is needed a hardware model able to run in real time. The model includes the engine, the transmission and there control units, the vehicle and the driver too. One of the key issues of drivetrain real time simulations is the modeling of launch devices. This paper aims to show an overview of usual launch devices models with emphasis on their compatibility with real time simulation.

NOMENCLATOR

\( c_m \quad \) normalized clutch command
\( i \quad \) parameter for Strubeck effect
\( J \quad \) moments of inertia
\( D \quad \) torque converter active diameter
\( T \quad \) torque
\( T_c \quad \) torque transmitted through the clutch
\( T_{fmaxd} \quad \) max. dynamic friction torque
\( T_{fmaxs} \quad \) max. static friction torque
\( T_{fd} \quad \) current dynamic clutch capacity
\( T_{fs} \quad \) current static clutch capacity
\( \omega \quad \) angular speeds
\( \omega_r \quad \) relative angular speed
\( \omega_0 \quad \) parameter that determines the speed of the transition from -1 to +1
\( \omega_s \quad \) parameter for Strubeck effect
\( \lambda \quad \) torque converter performance factor
\( \nu \quad \) torque converter speed ratio
\( \mu \quad \) torque converter torque ratio
\( \rho \quad \) oil density

Subscripts
1 input (engine / impeller)
2 output (transmission / turbine)

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 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.

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. These devices are named launch devices, moving off or power take-up elements. The main launch devices to have established themselves in motor vehicles are, [10]: friction (dry or wet) clutch is standard for manual, automated or dual clutch transmissions and torque converter is standard for conventional fully automatic transmissions.

The friction modeling constitutes the base of the clutch model. Many examples of friction models are proposed: hyperbolic tangent model [2], Karnopp model [9], Reset-Integrator model [6], Dahl model [12], Bristle Model [6], 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 ([12]) and studies of friction models ([6], [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], [5]) including some works on real time simulation ([13], [4]).

For the torque converter two types of models are proposed: static models and dynamic models. Almost all available studies are done with static models.

This paper describes some of current available theoretical models of launch devices. These include the hyperbolic tangent model, classic static model, Karnopp model for the friction clutch and static and dynamic models for the torque converter. Furthermore tests are done in order to establish those that cope with real time simulation demands.

**REAL TIME SIMULATION DEMANDS**

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 in real time, the amount of time spent calculating the solution for a given time step (execution time) together with the amount of time spent processing inputs, outputs, and other tasks the 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, figure 1, [11].

![Figure 1. The constrains of the step size for real time simulation](image)

To move from desktop simulation to real-time simulation on the chosen real-time hardware, the following items can be adjusted: solver type, number of solver iterations, step size, model size and fidelity. The challenge is to find appropriate settings that provide accurate results (results sufficiently close to the results obtained from desktop simulation) while permitting real-time simulation. Recommendations for this process are given in [11], [4].

**FRICTION CLUTCH MODELS**

The friction clutch models were developed based on existing models of friction. Some simplifications are made: the friction torque is computed using a normalized command signal
and the thermal effects are not taken into account because have a low dynamic and are not relevant for the purpose of this paper.

HYPERBOLIC TANGENT MODEL

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), figure 3.a. 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. In table 1 the simplified Coulomb, combined Coulomb and viscous friction and hyperbolic tangent clutch models are described.

The problem with the combined Coulomb–viscous friction model and hyperbolic tangent friction model is that they assume zero friction force at zero sliding speed. This means that friction force exists only when there is a motion. These models, therefore, cannot be expected to accurately predict limit cycling or other effects associated with sticking phenomena. The hyperbolic tangent model can be easy implemented using general-purpose simulation software packages. A clutch model base of hyperbolic tangent implemented in Simulink is presented in figure 4.a.

CLASIC STATIC MODEL WITH SWICH

Well known for its simplicity, this model is based on the equality of acceleration in lock phase. 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. As shown in figure 3.c it permits the modeling of stick-slip behavior.

The method employed to implement this model in Simulink is to use two different dynamic models and switch between them at the appropriate times. 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 figure 4.b.

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, [9]. 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, figure 3.f.

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 figure 4.a.

STRIBECK EFFECT

The Striebeck 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, figure 3.d. Therefore, is recommended to include this effect in the wet clutches models. It is possible to implement the Striebeck effect, for example as used in [2] by using the following equation:

\[
T_f = T_{fd} + (T_{fs} - T_{fd}) \cdot e^{-\left(\frac{|\omega_r|}{\omega_s}\right)^i}
\]

This effect was implemented for the all three developed models. This equation also enable the modeling of positive slope friction coefficients, figure 3.e.
Figure 4. Simulink top level diagram studied clutch models

TORQUE CONVERTER MODELS

For the torque converter we can use either static or dynamic models. The static models employ steady-state performance curves. These are introduced in lookup tables and convert the speed ratio into a torque multiplication ratio and a capacity factor (or performance factor), which is used in conjunction with engine speed to calculate the pump torque. These models are valid in steady-state conditions. However, since the fluid dynamics processes inside the torque converter are substantially faster than the typical time constants of the vehicle longitudinal dynamics, the fluid dynamic effects may often be neglected.

Most of the static models use the capacity factor ([7], [16]) and, moreover, they use only the traction zone of this (for $0 \leq \gamma < 1$) making the model invalid in engine break regime. The transition between traction and engine break is difficult due to the presence of a discontinuity. A solution to this problem is the use of a hyperbolic tangent function to ensure a continuous and smooth transition, [15].

Figure 5. Torque converter performance curves
a) Torque ratio vs. speed ratio; b) Capacity factor vs. speed ratio; c) Performance factor $\lambda$ vs. speed ratio
Another solution is to use the performance factor $\lambda$ instead of the capacity factor. The variation of this parameter is more appropriate for numerical simulation, figure 5.c. Instead of $\lambda$ it is possible to use the variation of the impeller torque at 2000 rpm in function of the speed factor (so called $M_{P2000}$ curve). The two approaches are similar due to the proportionality relation between $\lambda$ and $M_{P2000}$. The models are described in table 2.

<table>
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<th>Table 2. The equation of torque converter static models</th>
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<tr>
<td>For capacity factor $\gamma = \omega_2$</td>
</tr>
<tr>
<td>$K = K(\gamma)$</td>
</tr>
<tr>
<td>$T_1 = \frac{\omega_1^2}{K^2}$</td>
</tr>
<tr>
<td>$T_2 = \mu \cdot T_1$</td>
</tr>
<tr>
<td>For performance factor $\lambda = \lambda(\gamma)$</td>
</tr>
<tr>
<td>$\lambda = \lambda(\gamma)$</td>
</tr>
<tr>
<td>$T_1 = \lambda \cdot \rho \cdot \omega_1^2 \cdot D^5$</td>
</tr>
<tr>
<td>$\mu = \mu(\gamma)$</td>
</tr>
</tbody>
</table>

Accurate torque converter dynamic models can extend the frequency range up to 50Hz. A detailed dynamic model is presented in [8]. This model has 18 parameters, most of them giving the torque converter internal geometry (e.g. inlet and outlet angles of impeller, turbine and stator) making this model difficult to be parameterized.

**SIMULATION RESULTS**

For tests a torsional model of the powertrain is used. In order to reduce the complexity of the model the following simplification are made: the vehicle mass is transformed in the equivalent rotational inertia and all the elements after the gearbox are transformed in their engine-side equivalent. The use of this reduced transmission model (figure 5) is a common practice for clutch studies, [5], [14]. 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.

![Figure 5. Simplified model of powertrain](image)

The models are tested first using variable step solvers. From the step size appreciation can be made regarding the possibility to use a fix step solver. Then the models are run using different fix step solvers with a 1ms step size and the results are compared with that obtained with variable step solvers. We determine three situations: the model is not adequate (figure 6.a), the model produce results with numerical noise (figure 6.b), the model is adequate. If the results are unacceptable, the parameters of friction model are modified in order to allow simulation with fix step solvers and small errors compared with the original model (figure 6.c).

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 4.

**Figure 6. Offline simulation results**

Table 4. Task execution times (TET) for different clutch models

<table>
<thead>
<tr>
<th></th>
<th>No stick-slip / No Stribeck</th>
<th>stick-slip / No Stribeck</th>
<th>stick-slip / Stribeck</th>
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<td></td>
<td>Tanh</td>
<td>Clasic</td>
<td>Karnopp Clasic</td>
</tr>
<tr>
<td>Average TET [s]</td>
<td>3.998 x10^-5</td>
<td>3.815 x10^-5</td>
<td>2.976 x10^-5</td>
</tr>
<tr>
<td>Maximum TET [s]</td>
<td>7.133 x10^-5</td>
<td>8.900 x10^-5</td>
<td>8.000 x10^-5</td>
</tr>
</tbody>
</table>

The torque converter models are first tested offline using LMS Imagine.lab AMESim simulation software where all the models discussed are implemented. In the reduced transmission model the clutch was replaced with the torque converter. Results show that the dynamic torque converter model reproduces the damping effect of real torque converter and is also able to run with fix step solver using a 1 ms step size. As expected the static models give similar results but the model based on capacity factor cannot be simulated with the fix step solver using a 1 ms step size. Real time simulation test was done using a Simulink model based on performance factor $\lambda$.

**CONCLUSIONS**

A number of drivetrain launch devices models proposed in the technical literature and implemented in commercial simulation software are tested. 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 a solution exists when a torque converter is employed.

All the models are efficient and provide a good degree of accuracy when they are well tuned. The hyperbolic tangent model seems the most efficient but he 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 bought 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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