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WORKS UPON IDENTIFICATION OF CLUTCH FRICTION CHARACTERISTICS FOR A PASSENGER CAR

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Abstract: *the goal of this paper is to describe the activities done at the drivability team in dapem rtr, in order to identify the clutch friction characteristics while slipping. A reverse engineering approach was used, and a first-order dynamic model of vehicle was put in place, in order to estimate the degree of trust for the clutch friction characteristic – for an x84 project car. In order to be able to identify the unknown friction parameters, a methodology was designed and put in practice, including measurements with more devices than usually set. Initial simulations with the determined friction parameter showed an acceptable match with experimental data, for additional tests. The present work was, therefore, put in place just to identify some important friction characteristics, namely friction torque as a function of pedal position. Further work must be done for refining and improving the model, in order to catch the additional, second-order dynamics of drivetrain.*

INTRODUCTION

Clutch friction has been modeled by numerous authors, and the literature is rich in various clutch models. One would emphasize on the classical Coulomb friction model, perhaps generally accepted as the entry point of a clutch model, as part of an automotive powertrain subsystem. Also, viscous friction is often considered, especially in wet clutch domain – hardly the case of a manual transmission type, dry clutch. Stribeck effect – whether present or not – is also an issue, and it usually applies at dry clutches.

Modeling the clutch in simulation field had fallen to an old paradigm, constituted by two main modeling and simulation directions:

- (a) model(s) based on a single set of dynamic equations – sometimes referred to as “static friction model” (Deur et al, 2006);
- (b) model(s) based on (at least) two sets of dynamic equations – often considered “stick – slip models” or, sometimes, switching models (Karnopp et al., 1985).

Often the type (a) models are more easily to be simulated by common computer simulation languages, and, historically, they have been the first choice in the automotive simulation domain. The problems associated with these models are usually numerical (stiff numerical systems of differential equations (ODEs or DAEs)), as the steep grade that simulates the locked clutch state is prone to facilitate numerical instabilities. The so-called “classical model” (referred by Deur et al (2006), for instance) consists of a steep line around origin, that simulates clutch stiction state, and choosing the derivative of this straight-line approximation is an application – dependent process (a clutch model that works fine in one environment can be unsatisfactory in another case).

On the other hand, a true “stick-slip” model assumes switching the equations, when the stick conditions are supposed to be met. This can be potentially tricky for some simulation languages not especially designed for switching equations during simulation. AMESim, for instance, falls into this category – but largely compensates at its embedded numerical methods, that make type (a) models pretty accurate.

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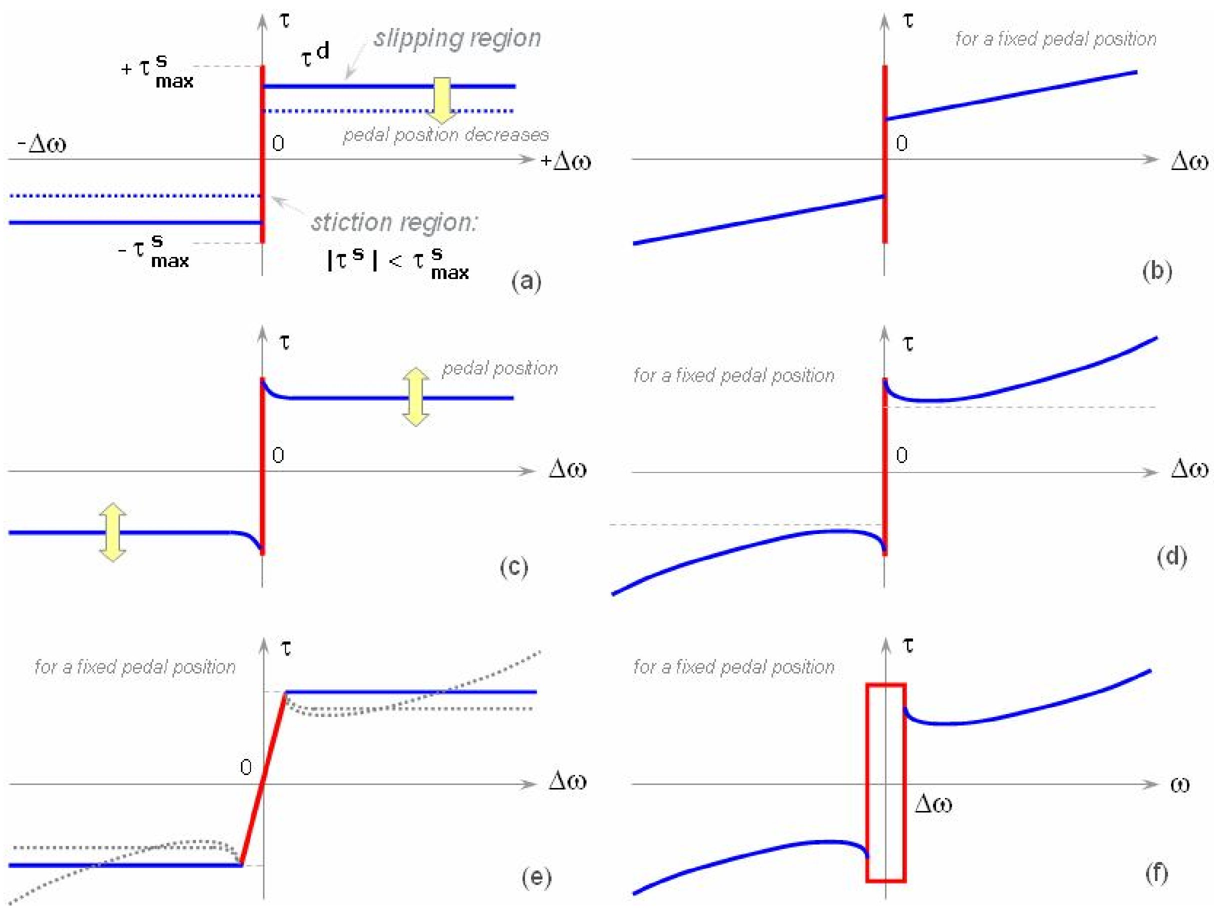

Figure 1. Clutch models (friction clutch in function of speed difference).
(a) Coulomb friction; (b) viscous friction;
(c) Stribeck effect near stiction area (near origin), (d) Coulomb, viscous and Stribeck effects added;
(e) common approximation of the stiction mode (state) for computer codes that do not allow causality change at runtime (slight slipping allowed in simulation), (f) Karnopp model: around origin, there is a domain when, once entered, it is supposed to have stick state.

MEASUREMENT SETUP

Figure 2 shows the measurement setup used for experiments. Apart from the classical signals achieved by CAN network – such as engine speed, engine torque (estimated by ECU), current gear ratio or vehicle speed –, additional signals were added to the measurement process. These are:

- (a) accelerometer, in order to measure instantaneous vehicle acceleration;
- (b) powertrain case displacement sensor;
- (c) angular speed sensor for transmission input shaft;
- (d) torquemeter – a device that measures actual torque applied at one driving wheel;
- (e) pedal displacement and force sensors.

Several breakaway evolutions have been made and measured, and we were mainly focused on cross-correlations between certain important vehicle dynamics variables, such as vehicle acceleration. That is the reason we used 3 signals that determine, directly or indirectly, vehicle acceleration: an accelerometer (for direct measure, though unfiltered), the CAN vehicle speed information, and the torquemeter signal (for wheel torque measurement, from which one can fairly easily estimate the vehicle acceleration, or some uncertain parameters like vehicle mass, for instance).

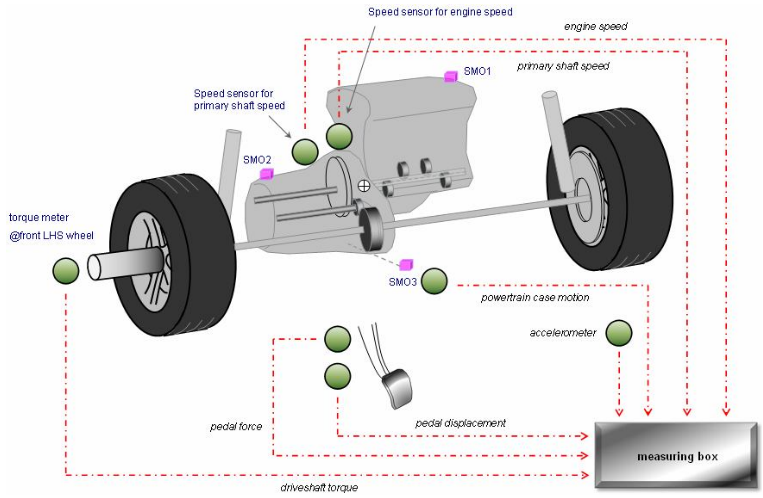

Figure 2. Signals measured for torque characteristics experiment.

FRICTION CHARACTERISTICS MAP FORMAT

We were looking for a nonlinear, non-physical clutch friction model, from several reasons:

. (a) *linearity*: it seemed likely that the friction characteristics of our clutch – to be determined – is hardly linear in pedal position, thus a linear model of type p(t)*Tcmax was ruled out;

. (b) *degree of physical representation*: we had neither friction coefficient, nor any normal force (on clutch disc) characteristics or information whatsoever, therefore it seemed fairly logical to adopt a simpler, non-physical model, i.e. attempting to link the instantaneous clutch torque directly to the maximum (dynamic) torque, rather than expressing it in function of friction coefficient, normal force, average radius etc.

. (c) a step-by-step method was chosen, i.e. firstly determining the allure of the clutch torque shape, and only then to get into details, if possible (μ, N etc.).

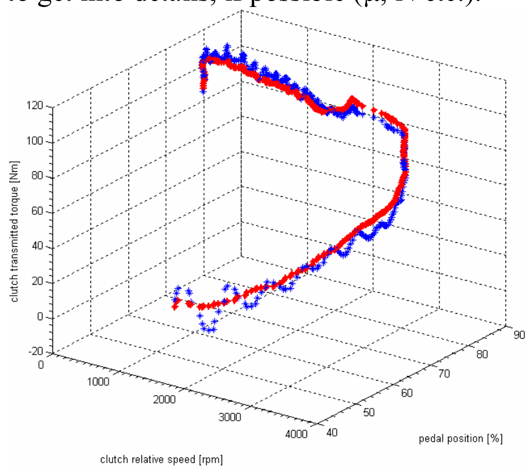

Figure 3. Typical breakaway clutch evolution – in terms of clutch relative speed and pedal position → in order to determine friction characteristics shape and expression.

Thus we were attempting to find a mathematical – actually, empirical function – of the instantaneous clutch torque in function of these 2 variables, while slipping (it was the breakaway friction characteristics we aimed to find, in order to simulate takeoff and, sometimes later, gearshift maneuvers).

Starting from various maneuvers like that depicted in Figure 3 above, we attempted to determine a *surface* of torque in function of these 2 inputs (pedal position, clutch relative speed). While assembling, on the same plot, about 5 or 6 measurements, it was not clear that all evolutions (curves) belong to a single surface, though – however, their distance to a unique surface was rather small. This would mean, in our point of view, that:

. (a) either there were some errors in the measurement process, and, actually, there is a single surface;

. (b) or there are multiple surfaces, and that means the clutch slipping torque is not only a function of pedal and relative speed (i.e., clutch slip), but also in function of other, perhaps less important, drivetrain variables, such as clutch friction discs temperature, gradient of clutch engagement, even the average value of engine speed etc.

For this initial level of analysis – and for the purpose of creating an algorithm, a test procedure – we have considered, for the moment, a dependency of pedal position and slip. Moreover, it was soon clear that the clutch torque was clearly mainly a function of pedal position, and only secondarily a function of clutch halves relative speed (or, clutch slip). Therefore, a search for an empirical function of pedal position was straight forward, and, for an initial analysis, considered for simulation – in order to evaluate its degree of acceptance.

CLUTCH MODEL

Figure 4 shows our approach to clutch simulation. From the 3 possible models, we chose to implement, for this initial analysis, the simplest 2 models, because we were mainly interested to characterize clutch friction parameters while slipping. A second importance was assigned to the issue of accurately simulating the clutch engagement process – hence, the transition between slipping and sticking states. Should the model be taken forward, such a transition will be considered in detail (and, perhaps, a true stick/slip model will be envisaged, i.e. a model that switches drivetrain equations in the (assumed instantaneous) transition between these 2 states).

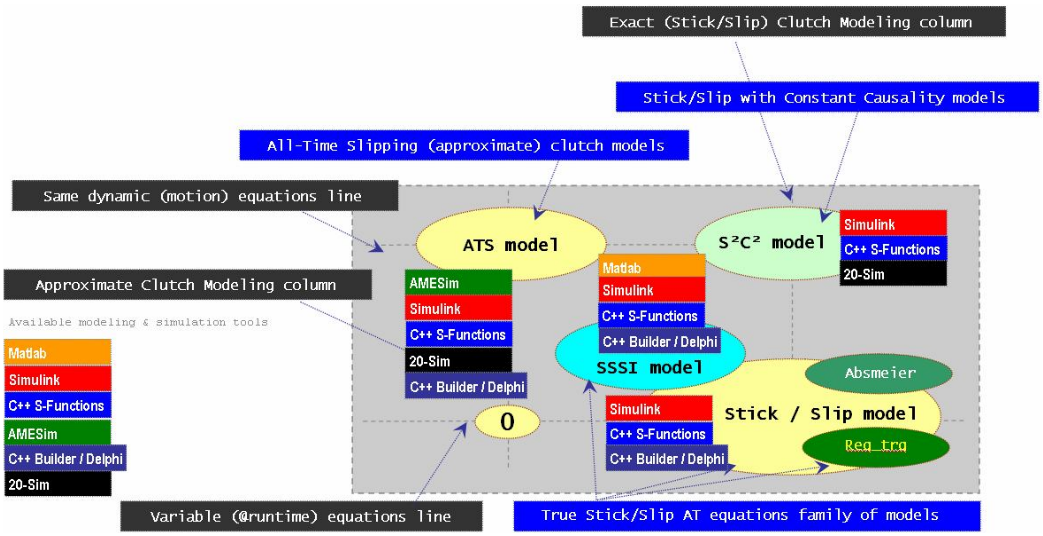

Figure 4. Analysis of available tools (simulation languages available in our team) for clutch simulation – the “clutch simulation matrix”, in our vision.

For the moment, though, we chose 2 models:

. (a) an approximate model with same expression (“equation”) of clutch torque regardless of state;

. (b) an approximate model with switched expression of transmitted clutch torque (one for slipping state, one for the approximated stiction state).

The 1st model, which we called ATS (from *all-time slipping*), assumes a short amount of slippage even when the clutch is supposed to be locked (in stiction state). However, instead of assuming a “classical” shape of the clutch torque in this *approximated stiction phase* (state) – Figure 5, (a) –, we felt that a sharper reaction around origin would have been envisaged – otherwise numerical instabilities appeared regularly in the vicinity of clutch stiction event. Figure 5, (b) shows our model, which proved better in simulations.

When dealing with an approximate model – that estimates stiction with a “slight slipping” clutch model –, the numerical simulation algorithm – linked directly to the simulation environment – is important. We used Simulink as simulation software, and it appeared that choosing the right parameters for the approximate part of the clutch model was important. Should we have chosen AMESim, which has alleged more robust integration algorithms (and selects the most appropriate integration method while running, unlike Simulink), the appearance of numerical oscillations would have been less likely to occur. However, Matlab / Simulink language has many advantages (its ability to switch submodels at runtime, far better than that of AMESim, and also its larger openness for alternate languages, such as Matlab code or C code embedded in Simulink models), and, for the present analysis, it was considered as not only sufficient, but the right tool to use.

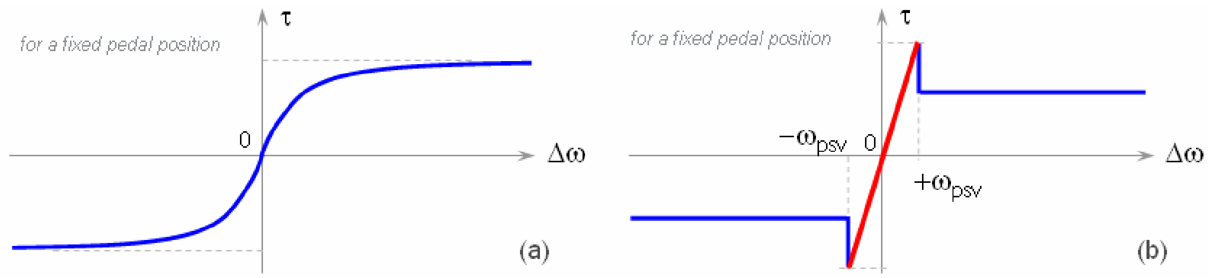

Figure 5. Clutch models we used for simulation.
(a) tanh or arctan – based models, (b) "sharp pre-slip velocity" model.

Therefore, the cost of using a less robust integration scheme was the selection of model depicted in Figure 5, (b).
On the other hand, the 2nd clutch model assumed the existence of an additional input variable to the clutch model, namely the *required clutch torque* (by the drivetrain system). For a two-DOF powertrain model (such as ours), the computing of such a required torque was fairly simple to do. However, for larger models, with additional dynamics (trends for the future), such computations are no longer that easy, since the drivetrain might change equations at runtime (during the simulation), thus triggering a modification of the "required torque" information to be passed towards the clutch model.
Our simulations use these three models below:

. (a) "classical" approximate models ("ATS models") use either hyperbolic tangent function, or arctangent function, and they proved acceptable for particular cases, but prone to oscillations (numerical instabilities) near stiction point:

$$\tau_c^d = \tau_c^{cap} \cdot \tanh \frac{\omega_e - \omega_{is}}{\xi} \tag{1}$$

$$\tau_c^d = \frac{2}{\pi} \cdot \tau_c^{cap} \cdot \arctan \frac{\omega_e - \omega_{is}}{\xi} \tag{2}$$

. (b) "sharp PSV approximate model" – our proposed model, for this case:

$$\tau_c^d = \begin{cases} \tau_c^{cap} \cdot \dfrac{|\Delta\omega|}{\Delta\omega_{psv}} \cdot \operatorname{sgn}(\Delta\omega), & |\Delta\omega| < \Delta\omega_{psv} \\ \tau_c^{cap} \cdot \operatorname{sgn}(\Delta\omega), & |\Delta\omega| \geq \Delta\omega_{psv} \end{cases} \tag{3}$$

$$\Delta\omega = \omega_e - \omega_{is}$$

. (c) "S^2C^2 model" – stick / slip with constant causality: alternate model we equally used, with pretty much the same outcome, with respect to simulation accuracy, speed and numerical (in)stabilities:

$$\tau_c^d = \begin{cases} \tau_c^{cap} \cdot, & \textit{sticking} \\ \tau_c^{cap} \cdot \operatorname{sgn}(\Delta\omega), & \textit{slipping} \end{cases} \tag{4}$$

Referring to eq. (4), the conditions for transitions from slipping to sticking or vice versa are summarized, schematically in Figure 6 below.

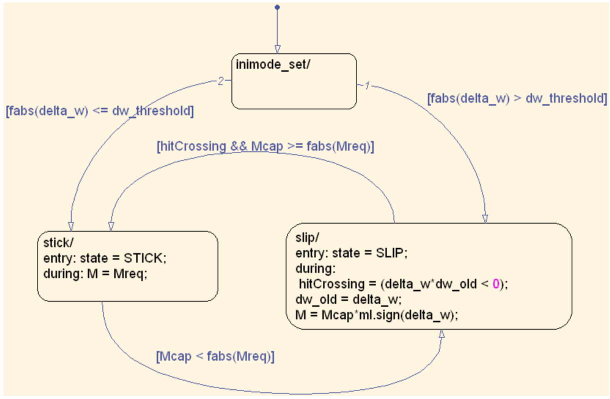

Figure 6. Schematic of transitions from slipping to stiction state, for the stick/slip model

Figure 7 shows the used models, for this 1st level simulation (rigid drivetrain, clutch torsional spring – damper overlook etc.). There were 2 hypotheses regarding capable clutch torque ($\Box_c^{cap}$):

. (a) *external* capable torque expression (external to clutch model block, in the Simulink model), and

. (b) *internal* expression (embedded in the clutch model).

While earlier models used the external expression – using a lookup table in function of pedal position – and model equations (1) and (2), the latter models used the internal capable torque expression (with the empirical formula in function of pedal position), as well as the model defined by equation (3) above.The function was eventually found to be (with c_j the experimental coefficients):

$$\tau_c^{cap} = \begin{cases} \begin{pmatrix} c_1 & c_2 & c_3 & c_4 & c_5 & c_6 & c_7 \end{pmatrix} \cdot \begin{pmatrix} p_p^6 & p_p^5 & p_p^4 & p_p^3 & p_p^2 & p_p & 1 \end{pmatrix}^t, & p_p \in \left[p_p^{\min}, p_p^{\max} \right] \\ 0, & p_p < p_p^{\min} \\ \tau_c^{cap,\max}, & p_p > p_p^{\max} \end{cases} \tag{5}$$

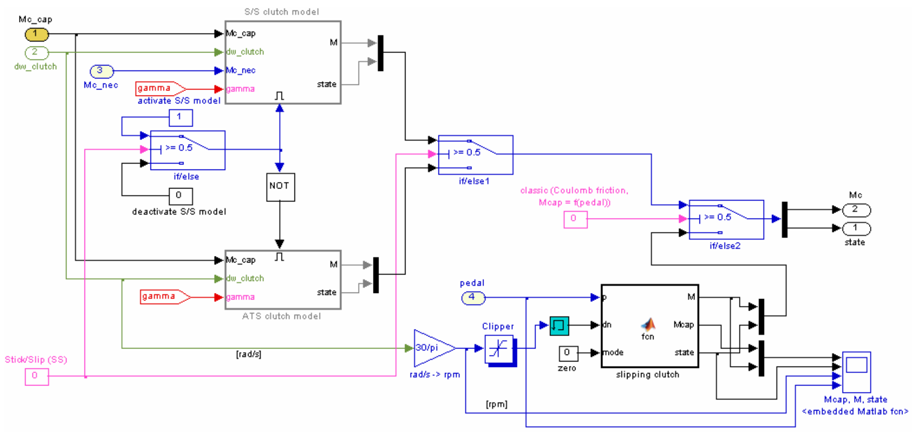

Figure 7. Clutch model, in the 2 hypotheses cited above.
The "slipping clutch" block is the embedded Matlab function – type representation of model (b) in the list above (eqn. (3)).

FIRST – ORDER MODEL FOR VEHICLE DYNAMICS ANALYSIS

A very basic first-order model was designed in Simulink, and the following hypotheses were employed:

. (a) basic, torque – source engine block: although quite basic, it was the most straight-forward option, since we disposed of the effective engine torque estimation, given by a CAN – accessed variable, in our data acquisition system (it is the engine software variable that estimates engine instantaneous effective torque);

. (b) a basic, single – state vehicle dynamics model: our goal being to determine clutch friction characteristics, drivetrain oscillations were of less interest for now; instead, average breakaway performance was targeted. Finding the appropriate clutch slipping formula (with respect to pedal position, at least) that suits parameter setup of the model, and follows experimental data sufficiently accurately (in an average sense, without drivetrain oscillations), was the target. Should such a simple, empirical friction characteristics (in function of either pedal only, or also in function of clutch slip) respond to our request (to match the experimental data rather accurately) be found, then a second, more elaborate, drivetrain and vehicle model will be employed. Should it fail to accurately simulate, then it would follow that such a simplistic approach for clutch friction characteristic while slipping was wrong, and additional parameters (such as friction discs temperature etc.) should be measured and evaluated.

Before explaining the single modification to an otherwise quite basic 1-dimensional vehicle model, it must be pointed out that *the test vehicle was weighted*, in the laboratory, with and without "virtual occupants" (equivalent bags were used to simulate the 2 persons that participated to the test). Also, the official weight of the test vehicle was compared with database chart, and the results were close.

The vehicle model has two states:

- (a) vehicle in motion → here we compute its longitudinal acceleration. Obviously, rolling resistance and aerodynamic drag are counted here for; slope was added, but let at 0 (flat road tests only). The acceleration resistance (inertial resistance) was considered the excedentary force, and vehicle state was integrated from this force.
- (b) vehicle stopped (the initial state of the vehicle model) → until the driving force reaches the value of resistive force at vehicle wheels level (rolling resistance), or if speed is less than an arbitrarily set threshold, the vehicle state is forced to 0. At the moment the driving force exceeds this rolling resistance, vehicle switches its mode to "running mode", thus the normal model.

There were 2 tests made for rolling resistance estimation:

- (a) constant rolling resistance coefficient – its parameter being evaluated experimentally;
- (b) a linear formula of vehicle speed – these parameters being determined by an additional test: with transmission in neutral position, vehicle speed while freely decelerating was measured, on a flat road and straight line. Then, using various linear parameters, we found an acceptable match for the following expression of rolling resistance:

In order to simulate a breakaway process, no gearbox model should have been necessary. However, we attempted to check the clutch friction model (characteristics) in a 1 → 2 upshift too, in order to evaluate whether it manages to describe engine and transmission input shaft speeds with sufficient precision (w.r.t. measured data). Should our simplified, clutch characteristics – type model be sufficiently accurate for takeoff (clutching period) but insufficiently accurate for the 1st upshift, some model update might be necessary. For the purpose of this analysis – without bringing second-order dynamics (like oscillations etc.) into discussion – a variable gear, sudden gear shifting transmission (gearbox) model was considered.

Transmission Simulink model has been derived from a bond graph, following the procedure from (Karnopp et al., 1990). Interested readers might also read Cruceru et al. (2008), for instance, for a related transmission modeling tehnique. It has a transformer (TF) element with variable modulation, at its core. Therefore, it assumes the shift takes place instantaneously in the mechanical part of the transmission (clutch model is elsewhere, it imposes a torque, regardless of clutch mode (state) etc.). however, there is supposed to be a certain delay between opening of the clutch, on one hand, and engaging the synchronizer into the 2nd gear. This time delay had been chosen after having studied our test driver's behavior. Thus, it might be inappropriate for other drivers etc. Nevertheless, the *shape* of engine, transmission and vehicle speed evolutions are of our interest, rather than some slight delays between simulation and measured variables (unlike in automated (AMT) or classic automatic (AT) transmissions, in a manual transmission the simulation cannot be always accurate, because of the subjective – and unique – driver, that cannot be simulated perfectly).

Figure 8 shows the transmission model, while Figure 9 depicts our overall simulation model, with some emphasis to a so-called "coherence section" (between various measured signals, like acceleration, LHS front wheel torque, vehicle, engine and primary shaft speeds etc.).

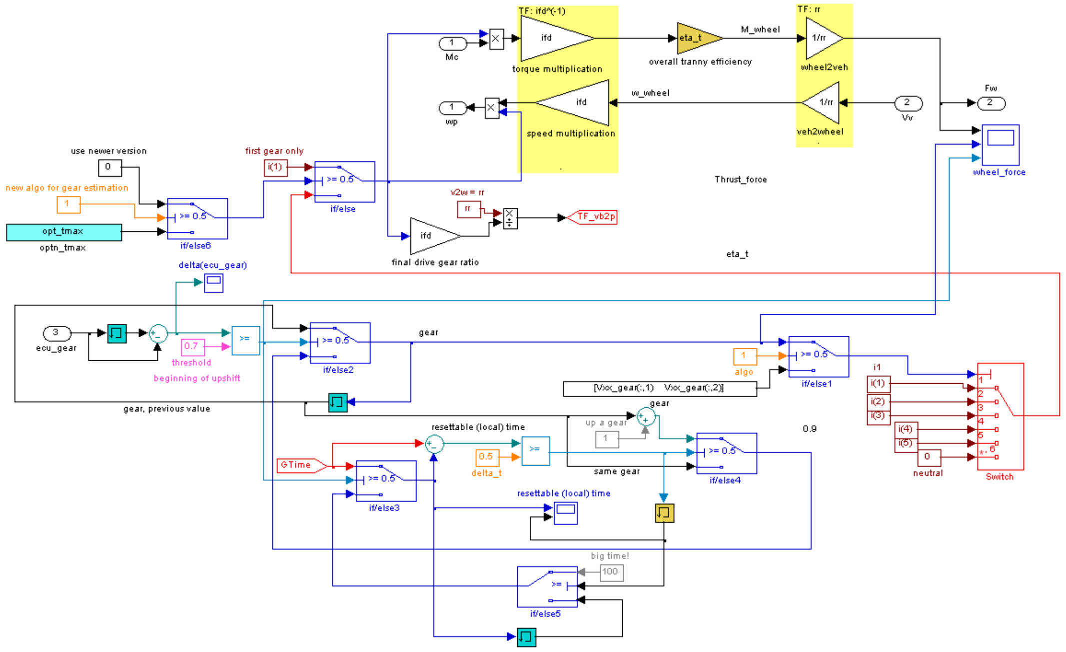

Figure 8. Transmission model used in our simulation.

For the moment, we assumed rigid axle and driveshafts (which is hardly the case in a breakaway analysis for drivability purposes, but it was considered enough for our targets).

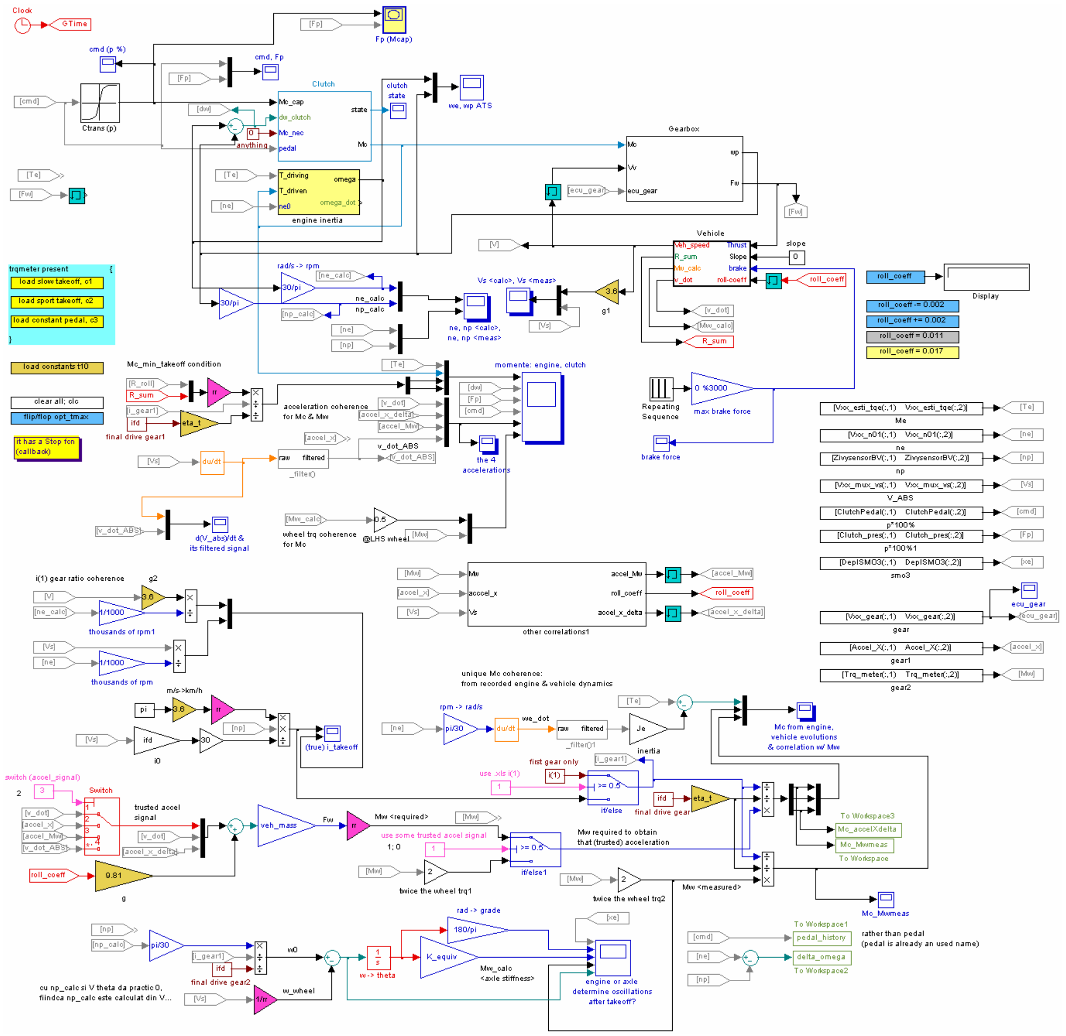

Figure 9. Simulation model, with some emphasis on a coherence analysis (lower section of model).

Figures 10 – 13 show some typical simulation results. Discussions are presented in next section.

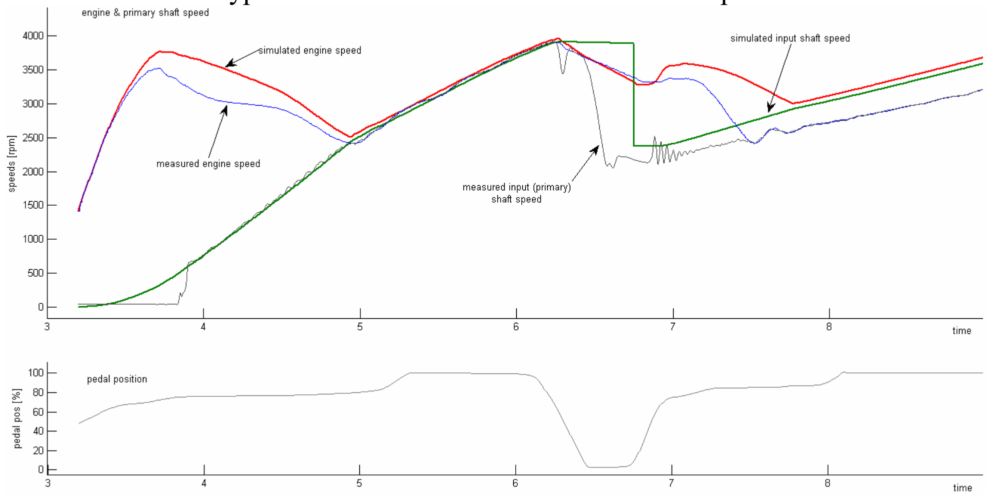

Figure 10. Simulated vs. measured engine and primary (input) transmission shaft, respectively. It yields that our calculated overall gear ratio in 2nd gear is larger than the real one (parameters uncertainty). Only the clutch – related behavior was of concern. First gear behavior, including takeoff (slipping clutch) is considered to be acceptable.

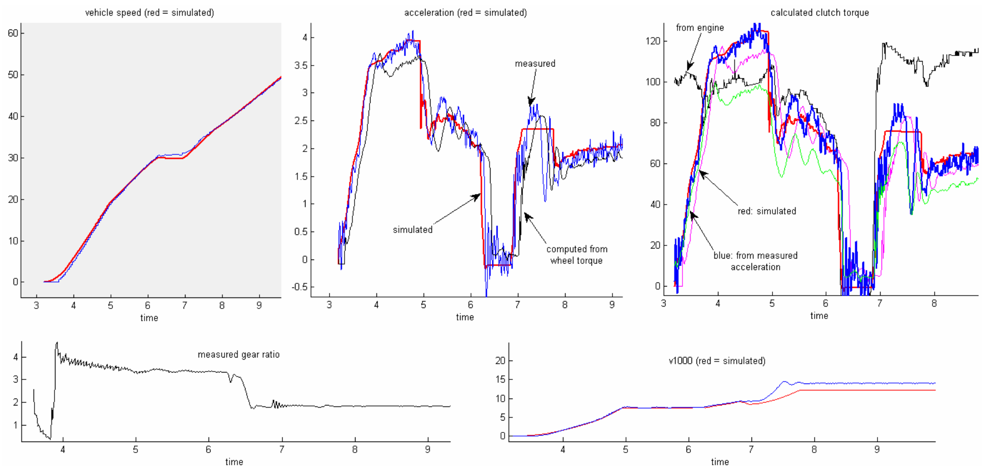

Figure 11. Additional results, simulation vs. sporty breakaway measurements (test #1). Coherence calculations are shown at the right-hand side of graph (accelerations, estimated clutch torque and v1000 parameter, respectively).

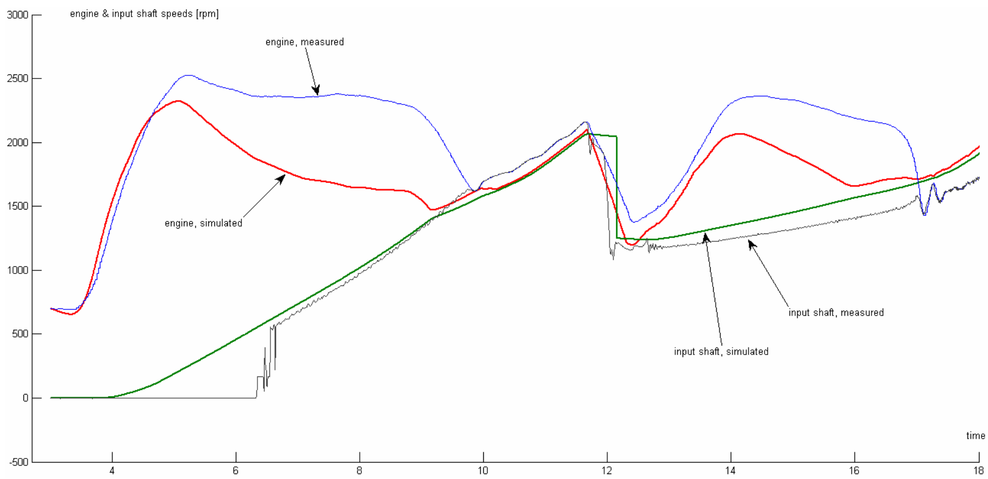

Figure 12. Same parameters, slow acceleration test.

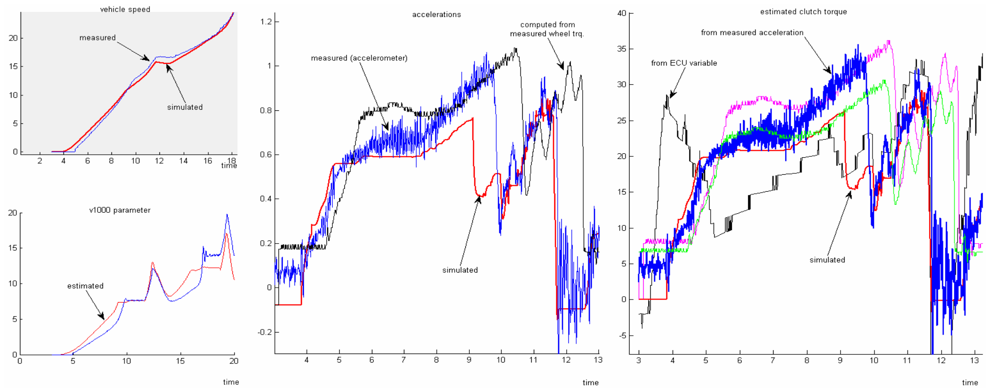

Figure 13. Slow acceleration test, coherence – correlation variables.

COHERENCE TESTS AND DISCUSSIONS

From the results above, as well as from some other, less significant, results (constant clutch pedal breakaway test), one can withdraw some discussions.
First of all, it must be said that, under 500 rpm, our input shaft speed sensor didn't provide accurate output – as it can clearly be seen on plots in Figure 10 and Figure 12. Therefore, comparison analysis could be done only in regions with gearbox primary shaft speed above 500 rpm.
Two types of coherence tests were made:

- (a) *type 1*: offset of certain signals with respect one to another;
- (b) *type 2*: value of signals → these will indirectly affect initial estimation of drivetrain parameters (engine flywheel inertia, transmission ratios, wheel roll radius, vehicle mass).

There were some slight offset – type miscorrelations between some important measured signals, such as *wheel torque signal* (output from the torquemeter device), *acceleration signal* (from the accelerometer, the non-filtered signal in figures above) and *CAN signal* (providing *average* – computed from the 4 wheels – vehicle speed). Thus, a coherence / correlation test had to be done in order to estimate whether these errors were local or they belong to the measurement setup (in this case, an offset must be added / withdrawn from the respective signal). Typically, from our observations it yielded that the measured wheel torque was offset w.r.t. acceleration signal, but that is quite normal, since the measurements team used two laptops: one for "main" input signals and one for the torquemeter. It is possible that the 2 laptops were slightly de-synchronized. We had to analyze measured data in order to estimate which of the 3 measured signals was the most trustful one, and, in order to do so , we had to look at the pecise moments of shifts (not only the 1-2 upshift). It turned out that the accelerometer signal was quite accurate as far as offset is concerned (no offset), thus it became the most trustful signal for the co-called "event moments" – those time values when the vehicle begins to accelerate from standstill, or begins a gear shift etc. These moments were important to our analysis, since we had to match our simple, empirical, clutch friction function with the measured variables, and clutch slipping time had to be matched fairly accurately.
From the same analysis it turned out that, despite its slight lack of synchronization, the wheel torque signal was very robust, from the signal quality point of view (the torquemeter device output signal is already filtered). Also, the measured wheel torque signal was quite robust, as far as type 2 criterion (see above) is concerned (its output yields credible acceleration values, see for instance Figure 11 and Figure 13).
On the other hand, however, the measured acceleration signal failed to be 100% accurate w.r.t. its output (thus, not synchronization, which was good), because, integrating its output yielded slightly different speed, w.r.t. the speed signal received on CAN. It therefore had to be affected with a 0.95 coefficient.
Vehicle speed from CAN was of less importance, though, since it failed to match other signals, in terms of timing (type 1 criterion) and even absolute value of its signal. Also, since it was registered with a smaller refresh rate, numerical derivation of the CAN – measured vehicle speed was difficult, and inaccurate. Therefore, we excluded CAN vehicle speed signal from our final analysis, and relied

on other 2 basic important measured variables: vehicle acceleration and LHS front wheel torque, respectively.
After these slight “custom adjustments” (offset of a signal and multiply by a 0.95 coefficient of another), all the tests were coherent – within an acceptable margin –, and we had confidence in these signals, in order to go forward.
Another class of coherence tests allowed slight affectation of some important parameters, like engine inertia moment and gear ratios. These were supposed to have clear values – from some Excel data files –, but it turned out that the equivalent engine inertia moment was less than its “catalogue value”, and 2^{nd} gear ratio was also inexact (Figure 11 and Figure 13 show, in the “v1000” subplot, that the computed “v1000” parameter is smaller than the one computed from measured variables). It then turned out that, in the future, more accurate, up-to-date “Excel data” must be provided by our colleagues in other departments.
The shape of engine and input shaft speeds were analyzed.

CONCLUSIONS AND OUTCOME

A first step in the complete procedure of fully characterizing the clutch of Renault passenger cars has been allegedly done. With a simple empirical function of pedal position only, the dry friction clutch of this class of Renault vehicles showed fairly acceptable simulation results.
As stated in the beginning of this paper, we were mainly interested in the “level one” vehicle behavior simulation, since we had to characterize the clutch friction parameters. This, in itself, turned to be rather complex, since we had no functional clutch parameter whatsoever (μ friction coefficient, normal force in function of pedal position or anything else etc.). Therefore, starting from scratch, only with a set of tests, we managed to obtain an empirical function that describes clutch friction torque, while in its slipping state, a function that is supposed to be used in further simulations of more specific phenomena related to clutches equipping Dacia and Renault passenger cars.
The shape of engine and input shaft speeds showed decent accuracy w.r.t. measured data also in shifts, see Figure 10 and Figure 12. This induced the conviction that, although a simple empirical function, it is a credible starting point for further, more elaborate (and sophisticated) simulation models, that are clearly necessary in the future – if we take a look at Figure 10, around t = 7 sec. there are small amplitude, high frequency oscillations on the primary shaft of the gearbox, but these oscillations do not appear also in the engine speed signal. Thus, it is concluded that these oscillations are produced by the clutch spring – damper unit, and hence will be included in the next – level clutch model, in the future. Also, at the end of the gear shift, there is a significant drivetrain oscillation, and this is due to additional important drivetrain dynamics, neglected for now (from the reasons explained above): driveshaft (axle) stiffness, tire characteristics, and, perhaps also important, engine oscillations on its mounts. All these important dynamics, together with a slightly refined clutch model (more tests required, though), are eligible for further work.

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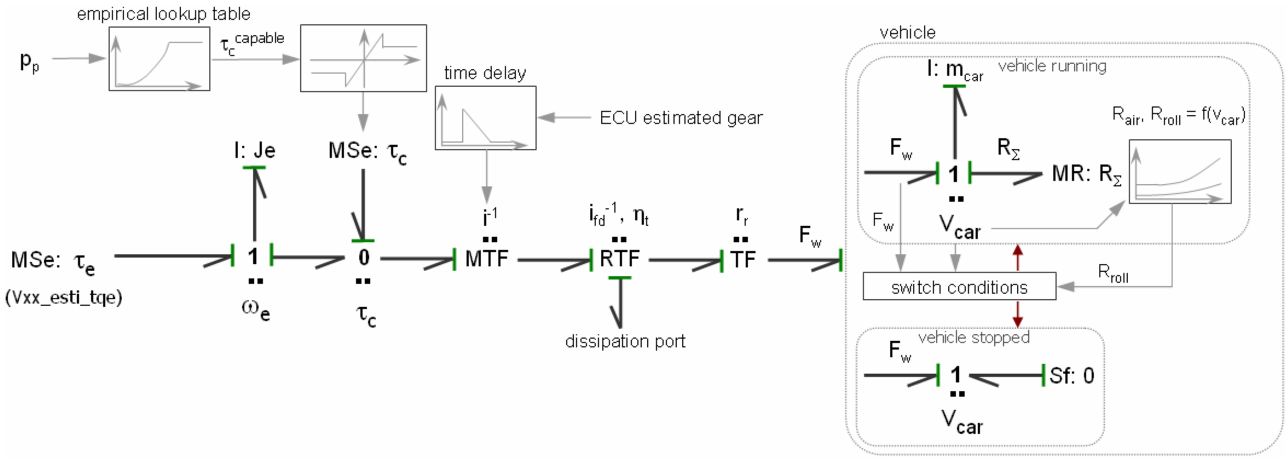
APENDIX



Bond graph model used for deriving equations used in simulations.