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Effect of Thermal Expansion in a Dry Clutch on Launch Control

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Abstract: Dry clutch model with thermal dynamics is added to a driveline model of a heavy-duty truck equipped with an automated manual transmission. The model captures driveline oscillations and can be used to simulate how different clutch-control strategies affect vehicle performance, drivability and comfort. Parameters are estimated to fit a heavy-duty truck and the complete model is validated with respect to shuffle, speed trajectory, clutch torque and clutch lock-up/break-apart behavior. The model shows good agreement with data. Furthermore the model is used to study the effect of thermal expansion in the clutch on launch control. It is shown that the effect of thermal expansion, even for moderate temperatures, is significant in launch control applications**.**

Keywords: Automated manual transmission, driveline model, slipping phase, dynamic clutch**.**

1. INTRODUCTION

The demands are increasing on comfort, performance, and fuel efficiency in vehicles lead to more complex transmission solutions. One such solution is the Automated Manual Transmission (AMT). It works just like an ordinary manual transmission but the clutch and gear selection are computer controlled. In this way high efficiency can be accomplished with increased comfort and performance. To be able to control and fully utilize an AMT it is of great importance to have knowledge about how torque is transmitted in the clutch. The transmitted torque in a slipping dry clutch is therefore studied in experiments with a heavy duty truck (HDT). It is shown that material expansion with temperature can explain torque variations up to 700 Nm for the same clutch actuator position. A dynamic clutch temperature model that can describe the torque variations is developed. The dynamic model is validated in experiments, and shown to reduce the error in transmitted torque from 7 % to 3 % of the maximum engine torque compared to a static model. The clutch model is extended with lock-up/break-a-part dynamics and an extra state describing wear. The former is done using a state machine and the latter using a slow random walk for a parameter corresponding to the clutch disc thickness. An observability analysis shows that the augmented model is fully or partially observable depending on the mode of operation. In particular, by measuring the actuator position the temperature states are observable, both during slipping of the clutch and when it is fully closed. An Extended Kalman Filter (EKF) was developed and evaluated on measurement data. The estimated states converged from poor initial values, enabling prediction of the translation of the torque transmissibility curve. The computational complexity of the EKF is low and it is thus suitable for real-time applications. The clutch model is also integrated into a driveline model capable of capturing vehicle shuffle (longitudinal speed oscillations). Parameters are estimated to fit an HDT and the complete model shows good agreement with data. It is used to show that the effect of thermal expansion, even for moderate temperatures, is significant in launch control applications. An alternative use of the driveline model is also investigated here. It is found that the amplitude discretization in production road-slope sensors can excite vehicle shuffle dynamics in the model, which is not present in the real vehicle. To overcome this problem road-slope information is analyzed and it is shown that a third-order butter worth low-pass filter can attenuate the vehicle shuffle while the shape of the road profile is maintained.

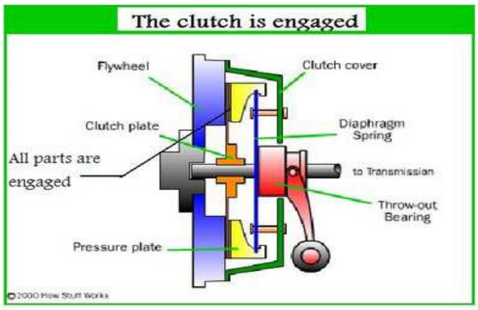

Fig. No.1- Dry single-plate clutch

A sketch of a dry single-plate clutch is found in Fig. 1, while in-depth explanations are found in for example Mashadi and Crolla [2012] and Vasca et al. [2011]. In clutch-modeling literature a wide range of models are proposed. The most simple models have a clutch torque that is assumed to be a controllable input, see for example Dolcini et al. [2008], Garofalo et al. [2002]. These models rely on the assumption that there is perfect knowledge of how the clutch behaves. More advanced models include sub models for slipping and sticking torques. For example a Lu Gre model is used in Dolcini et al. [2005] and a Karnopp model in Bataus et al. [2011]. The former is a one state model that captures stick-slip behavior, varying breakaway force, Stribeck effect, and viscous friction. The latter simply applies a dead-zone around zero speed to ease the simulation of stick-slip behavior. Models for the transmittable clutch torque during slipping commonly use a function with the following structure

$M_{trans,k}= \mathrm{sgn}(\Delta w)\, \mu R_e F_N$ ……………………………….(1)

Where, Δw is the clutch slip (speed), μ the friction coefficient, *Re* the effective radius and F_N the clamping (normal) force. In these models FN is often either given as input or a static nonlinear function of clutch position, x, i.e. FN = FN(*x*), see for example Vasca et al. [2011], Glielmo and Vasca [2000]. Furthermore a graph with speed dependency of the normal force is shown in Hong et al.[2012]. In Dolcini et al. [2010] that speed dependency is said to be due to centrifugal forces acting on the springs in the clutch. Mashadi and Crolla [2012], Moon et al. [2004] reports of hysteresis in the diaphragm spring, that could lead to hysteresis in the normal force. In Mattiazzo et al. [2002] a temperature and wear dependency of the normal force/bearing position characteristics is shown.

Concerning the other model components it is generally recognized that μ can depend on temperature, slip speed, and wear and that Re can depend on temperature and wear as well, see Velardocchia et al. [1999]. In Vasca et al. [2011] a slip speed dependency of μ Re is shown, this was especially pronounced for slip speeds below ~100 RPM.

It can be difficult to separate which parameter in (1) that is the reason for a change in $M_{trans,\ k}$. Therefore the clutch torque is often studied as a lumped model. In Velardocchia et al. [1999] $M_{trans,k}$ is seen to decrease withtemperature. However there are large variations with wear. In Ercole et al. [2000]$M_{trans,k}$ initially decreases with temperature for low temperatures and then increases for medium and high temperatures. Similarly there are variations with wear and in addition temperature-torque hysteresis are reported. In Velardocchia et al. [2000], Wikdahl and _Agren [1990] and Myklebust and Eriksson [2012b] temperature models are established, but only Myklebust and Eriksson [2012b] includes the effect of the temperature on M_{trans}.

Here yet another driveline model with focus on the clutch and the control of it is presented and validated. The contribution lies in that here the thermal dynamics of the clutch are included in the model. Particularly the significance to launch performance of including the thermal part is shown.

2. DRIVELINE MODEL

In order to evaluate the quality of a certain launch control, a longitudinal model of the heavy-duty truck in question is required. The model has to capture important dynamics in the driveline and how they make the truck shuffle. One such model is found in Myklebust and Eriksson [2012a]. It is used here with one modification, the clutch model is replaced with the more advanced model from Myklebust and Eriksson [2012b]. An overview of the model is seen in Figure 2. There the different parts of the model, Internal Combustion Engine (ICE), clutch, gearbox, propeller shaft, final drive, drive shafts and vehicle dynamics, can be seen as well as where the exibilities are located. Next, a quick review of the model equations are given.

If not otherwise stated the nomenclature follows this system: θ=angle, ω=θ, v=velocity, r=radius, T=temperature, M=torque, F=force, P=power, c=damping or vehicle dynamics coefficient, k=spring coefficient, b=viscous friction coefficient, J=inertia, i=gear ratio, and x=clutch piston position. These quantities are often equipped with subscripts, e=engine, fw=flywheel, c=clutch transmission side, t=transmission, p=propeller shaft, f=finaldrive, d=drive shaft, w=wheel, i=gear number, and amb.=ambient.

Internal Combustion Engine
The ICE produces the engine torque, Me, that is given as model input. Note that this is the net (brake) torque of the ICE, e.g. Me = 0 with open clutch will keep the engine speed constant.

Clutch
The explanation of the clutch model is split into three parts, the friction and temperature dynamics, the mode changes between locked and slipping, and the torsional springs. An overview of the studied clutch

Clutch Friction The natural output from the actuator is the clamping force, FN. However FN is not measurable, therefore it is directly recalculated into a transmittable torque, M_{trans} = k _FN, that is used as actuator output. The shape of the torque transmissibility curve is described by a third order polynomial
The clutch disc temperature Td, clutch body (flywheel and pressure plate) temperature Tb, and the clutch housing temperature, The, have been modeled in order to explain the torque drift, see, due to temperature. An electrical analogy of the model is found in and below are the equations.

$$(mcp)bTb = k_{ICE2b}(T_{coolant} - Tb) + k_{b2h}(Th - Tb) + k_{d2b}(Td - Tb) + k_P P \quad \text{……….(3)}$$

$$(mcp)hTh = k_{b2h}(Tb - Th) + k_{h2amb}(Tamb - Th) \quad \text{….(4)}$$

$$(mcp)dTd = k_{d2b}(Tb - Td) + (1 - kP)P \quad \text{……….(5)}$$

where,

$$P = M_{trans;k}, \Delta w = M_{trans;k}(we - wc) \quad \text{…..(6)}$$

The temperature model is connected to the transmitted torque through a change of the position x_{cor}, corresponding to the expansion of parts in the clutch. The expansion of the clutch body and disc as a function of temperatures is assumed linear.

$$\Delta x_0 = (k_{exp,1} + k_{exp,2})(T_b - T_{ref}) + k_{exp,2}(T_d - T_b) \quad \text{.(7)}$$

$$x_{cor} = x - \Delta x_0 \quad \text{…….(8)}$$

The transmitted torque can now be calculated as,

$$M_{trans,k} = M_{ref}(x_{cor}) \quad \text{………………..(9)}$$

Note that x_{cor} increases with temperature which in turn makes $M_{trans;k}$ increase with temperature. The k in the subscript stands for kinetic because the friction is modeled as coulomb friction with stick-slip behavior. Define $k\mu$ as the ratio of the static friction coefficient over the kinetic. Then the maximum transmittable torque when sticking is:

$$M_{trans;s} = k\_M_{trans;k} \quad \text{…….(10)}$$

Lock-Up/Break-Apart Logic The clutch model has two modes, locked and slipping mode. While in locked mode, the clutch behaves as one rigid body, whereas during slipping the clutch consists of two bodies where each one has an angular velocity and position. The equations are: Conditions for switching from slipping to locked mode

3. PARAMETER ESTIMATION

The driveline model and the clutch model have both been validated separately in their respective paper. However they have not been validated when put together and the lock-up/break-a-part detection has not been validated before. In order to do that a number of launches have been recorded. However these experiments have been carried out in a different truck than those used in the previous papers. Therefore the parameters have to be estimated. The driveline parameters have been estimated using a launch where the clutch has been closed rapidly (clutch is slipping less than 0.1 s). This gives negligible clutch dynamics and large shuffle oscillations, which is appropriate when estimating the exibilities and damping coefficients. The result can be seen in Fig. 4. Since it is difficult to do open-loop simulation of a system under feedback, the model also utilizes feedback. In the experiments the combustion engine is under speed control, therefore also the modeled engine is put under speed control. A PI controller with feed forward of the measured engine torque is used. However since the engine model is simply an inertia, the PI-controller performs better than the real controller. Therefore the measured engine speed has been used as reference, in order to capture the imperfections in the speed control and the inherent shuffle. The other driveline speeds follow the measurements well. The model is leading somewhat in the start due to sensor dynamics. There is some difference in the engine reported torque and the modeled engine torque, although it is hard to draw any conclusions from this, since the torque signal is inexact during transients. The parameterization seems good. Next the clutch model needs to be parameterized. The experiment conducted to do this has consisted of ramping the clutch position back and forth while the truck has been kept stationary using the parking brake. The resulting data can be seen in Fig. 5. The torque drift due to temperature can clearly be seen. By applying (3)-(8) to the data, using the same parameters as in Myklebust and Eriksson [2012b], Fig. 6 is attained. There the ramps have converged to one curve and consequently these parameters work here too. The 3rd degree polynomial, (2), has been fitted to this curve using the least square method.

4. MODEL VALIDATION

A number of launches have been performed in different gears in order to validate the complete model. Here two launches are shown, one in third gear, and one in sixth gear. In third gear there is some drift in vehicle speed with the consequence that the clutch locks up earlier in the model compared to the measurement. During the slipping phase the clutch torque has been modeled correctly, however after lock up the torque decreases due to the engine speed controller. When the engine speed reaches a set value the controller is switched off and torque is used as model input, since then the measurement is no longer under feedback. When the controller is switched off the drift naturally returns. Nevertheless the oscillation in the driveline are captured with respect to amplitude and frequency, although the attenuation in the model is a bit too high. In sixth gear the driveline speeds andclutch torque matches the measurement very well. As a result it is easy to see that the lock-up/break-apart logic makes mode switches at the correct time points. This is a further addition to the base-line model in Myklebust and Eriksson [2012a]. However when the clutch torque goes from positive to negative and vice versa there is some oscillations seen in the measurement due to backlash. These oscillations are naturally not captured in the model since the backlash is not modeled.

In conclusion both validations look fine and the model is suitable for investigating different clutch control strategies during launch and their effect on vehicle shuffle and performance.

5. THERMAL EFFECT ON LAUNCHING

This section highlights possible problems that can arise in clutch control during launch when not considering the thermal effects. Here two controllers taking requested torque as input are studied. This is a natural choice of input since it is common to use torque based driveline control, Heintz et al. [2001]. When so the driver intention over time can be interpreted as a reference torque trajectory. The first controller is the simplest possible, an open-loop controller consisting of an inversion of (2), the torque transmissibility curve at 60 _C. The second controller uses the first controller as a feed forward part but in addition it has a PI-controller in order to utilize the engine torque for feedback. For both controllers the clutch is fully open when the torque request is zero and fully closed when there is no slip in the clutch. The clutch actuator is fast and exact and therefore it is modeled with a rate limiter of 82 mm/s on x. The ICE directly gives the requested torque and has in addition a PI idle-speed controller in order to not stall the engine if the requested torque is too low. The simulations will be evaluated mainly from a comfort perspective and a measurement relating to comfort is jerk (time derivative of the vehicle acceleration). According to Zeng et al. [2013] the maximum jerk and minimum (negative) jerk are important comfort measures.A requested torque trajectory has been chosen as follows; starts out at zero torque until 1.5 s where it is ramped up to 3 % of maximum torque at 2.5 s. It is kept there until 4 s when it is ramped further up to 4 % at 5 s and kept there for the rest of the simulation. This trajectory is used as input to the simulation model together with no braking, no slope, fourth gear and 500 RPM as idle speed. Simulations are run for both controllers in cold,
$[T_b; T_h; T_d]=[60; 50; 60]\dot{c}$, and warm, $[T_b; T_h; T_d]=[120; 110; 120]\dot{c}$. The maximum jerk levels can be seen to rise more than a factor of five when the clutch is warm. This is due to that the clutch controller overshoots the kiss point at 1.5 s. The feedback controller also get a large negative jerk when it tries to compensate for the excessive torque due to incorrect kiss point. In the open loop case no such compensation is present and naturally the truck receives a completely different speed trajectory although the driver input is the same. Furthermore the jerk is larger when the clutch locks-up in this case. The open loop controller for the cold case has a drift in torque due to heating of the clutch. Only the closed-loop controller for the cold case manages to follow the desired trajectory.

6. CONCLUSION

A driveline model for vehicle shuffle and a clutch model including thermal effects have been merged together in order to simulate how different clutch-control strategies affect vehicle shuffle and performance. Parameters have been estimated to fit a heavy-duty truck and the complete model has been successfully validated, including the lockup/ break-apart logic. The complete driveline model has been used to show the profound effect of thermal phenomenon in the clutch on launch control, even for moderate temperatures. The launch control example showcases the importance of incorporating a thermal model of the clutch in launch control applications.

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