ACTIVE SUPPRESSION OF TORSIONAL OSCILLATIONS

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Abstract: A problem with electrically driven vehicles is resonances in the drive train caused by elasticity and gear play. Disadvantageous effects caused by this are noticeable vibrations and high mechanical stresses due to torque oscillations. The oscillations can be damped using a control structure consisting of a nonlinear observer to estimate the torque in the gear and a controller, which computes a damping torque signal that is added to the driver's demand. The control algorithm was implemented in the existing motor control unit without any additional hardware cost. The controller was successfully tested in a test vehicle. The resonances can essentially be eliminated. The controller copes satisfactorily with the backlash problem.

Keywords: Active damping, drive train oscillation, control

1. INTRODUCTION

Electrical drives are of increasing interest for road vehicles, either as sole propulsion engine that is fed by a battery or a fuel cell, or in combination with a combustion engine as a hybrid drive. This paper, however, deals with a pure electrical drive system (Fig. 1). The induction motor of 50 kW rated power is fed by a voltage source converter with a Zebra battery supply allowing 140 km operation range.

The paper focuses on the particular problem of drive train oscillations. Torsional oscillations in drive trains of electrically driven vehicles are a common but annoying problem. However, the problem is more awkward with electrical drives than with combustion engines. Unlike drive trains with combustion engines, which are usually equipped with dampers due to the high torque pulsation of the engine (at engine firing frequency), electrically driven trains are rather weakly damped. Dampers seem not to be necessary due to the smooth torque of electric motors. But, not surprisingly, any unevenness of the driver's demand or, more evident, of the load from the disturbances of the road-wheel-contact will then excite mechanical resonances (at low driveline oscillation frequency). Typically, the motor controller does not provide any damping of the oscillations, because it is usually designed in a way to produce the demanded torque independently of the mechanical state of the system.

Drive train oscillations are undesired due to the high material stress but also due to poor comfort because the oscillations can cause, particularly at low speeds, the whole vehicle to shudder quite unpleasantly. Corrective measures are necessary in order to improve the riding comfort and to lessen mechanical stress.

In this paper, a control scheme for active oscillation damping is described. The controller determines a small corrective signal, which is added to the torque demanded by the driver such that the oscillations in the drive train are reduced. A special problem the control design had to deal with was the backlash of the gear and of the cardan shafts that introduces nonlinearities in the control loop. If the controller were not able to cope with that, the backlash could, of course, be minimized using more precise, but also more expensive mechanical components. So it is appreciated that the developed controller can indeed operate sufficiently with a considerable backlash.

The control approach was realized without any additional hardware cost within the existing motor control unit (MCU). The computational effort is moderate. Results of vehicle tests are shown.

2. DRIVE TRAIN MODEL

Drive train oscillations may always appear, in principle, if there exists any elasticity and inertia. Elastic elements are the shafts, the gear, the motor supports and, last but not least, the tires. Typical values of the most crucial first eigenfrequency lie in the range of 5 to 50 Hz depending on the drive train design. It was about 7 Hz with the test vehicle.

For the purpose of controller design, a model of the drive train is required first. A rather simple model, which is suited to the crucial phenomena, is used here. The model consists of a two-mass system with a spring and backlash, Fig. 1. The resulting functional block diagram of this model is shown in Fig. 2.

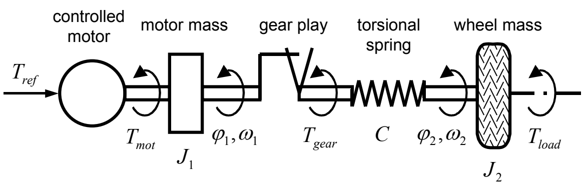

Fig. 1. Drive train model.

The spring C represents all the elastic elements in the drive train. The backlash stands for the total gear play (transmission gear and cardan shaft). The inertia J_1 includes the rotor of the motor and parts of the gear. Inertia J_2 includes everything from the gear until the wheels. It does not include the mass of the vehicle which acts indirectly over the load torque T_{load}.

The motor is considered as a closed-loop control system with the torque as control objective. Torque control of induction motors is an established and well performing technique so that the closed-loop behavior between the torque reference T_{ref} and actual air gap torque T_{mot} of the motor can be approximated as a second order delay system. The input to the complete system is then the demanded torque T_{ref}.

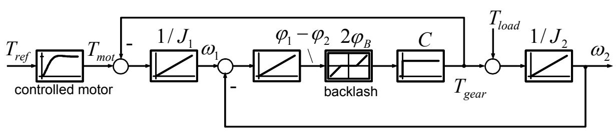

Fig. 2. Block diagram of the drive train model

The equations for the dynamic model of the system are thus as follows:

$$\begin{aligned} J_1\dot{\omega}_1 &= T_{mot} - T_{gear} \\ J_2\dot{\omega}_2 &= T_{load} + T_{gear} \\ \dot{\varphi}_1 - \dot{\varphi}_2 &= \omega_1 - \omega_2 \\ T_{gear} &= C\, b(\varphi_1 - \varphi_2) \end{aligned} \tag{1}$$

where $b(x)$ is the backlash function with total width $\pm\varphi_B$. This simple model is deliberately chosen for controller design. Its simplicity allows good control design and the few parameters are easy to identify. However, one must be aware that the model really is just a simplification of the real system and that the artificial elements do not have necessarily a one-to-one correspondence in the real system. Consequently, the identified parameters may possibly vary depending on the operating point, and the model describes the dynamic behavior of the real system only to a certain degree. The controller must thus be robust and be able to deal with these constraints.

The underpinning idea why to use this simple model is that even a complex model will never exactly agree with the reality. Therefore, it is better to design a good robust control for a simple model, which covers, because of its robustness, also the actual behavior of the more complex real system than to optimize a controller for a complex, but nonetheless fictitious model. Note that for controller validation (afterwards) a more complex model is used which includes several additional real world effects. This allows to assess the robustness of the controller in a fairly realistic way.

3. SYSTEM IDENTIFICATION

While the inertia J_1 of the rotor of the motor is quite precisely known, the values of inertia J_2, backlash width φ_B and the spring constant C are rather imprecisely or even not at all known. Though the backlash and the spring constant may be measured at standstill, it is, however, not clear that such statically measured values would fit best for the dynamic behavior. Therefore, these parameters were subject of an identification procedure. The system (a specially equipped test vehicle) was stimulated by a pseudo random binary signal (PRBS) as torque demand T_{ref} on a test track. Data of the speeds of the motor and the wheels, ω_1 and ω_2, respectively, and of the load torque T_{load} were recorded. The load torque was measured by a set of particular measurement wheels, with which the test vehicle was equipped. Such a torque measurement is not available in standard vehicles.

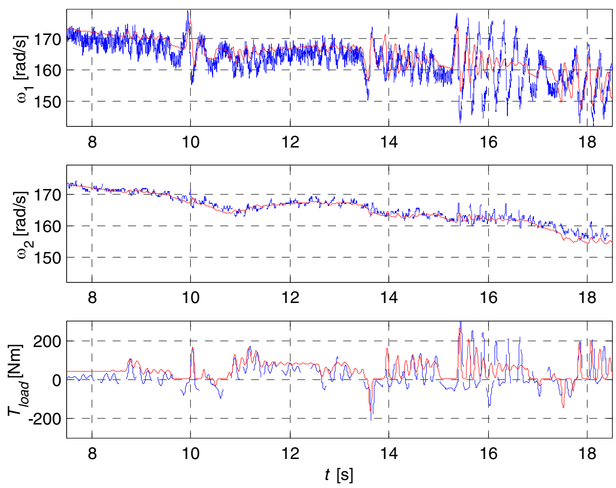

Fig. 3. Comparison of PRBS responses for a vehicle speed of about 15km/h. Blue: Measured data. Red: Simulated model with identified parameters.

The three parameters were identified using a function from the Matlab Identification Toolbox. Fig. 3 shows as a result of this identification a comparison between the measured data and a simulation of the model using the identified parameters. The comparison shows that the model represents the behavior of real system quite well. Some minor differences remain due to the simplicity of the model.

4. CONTROL DESIGN

The controller design has to take care of several objectives and constraints: The main objective is the damping of the oscillations. However, the control is not allowed to change the stationary torque. The control should behave unnoticeably for the driver and should not conspicuously change the overall dynamics of the drive. Because a quite simple model was applied for the control design, the control must be robust enough to allow for considerable parameter variation and model uncertainty.

It was desired not to introduce additional sensors due to cost reasons, but to use only existing measurement signals: These are the motor speed ω_1, which is available from the motor control, the motor torque T_{mot}, which is provided by the motor control, although it is only as an estimate, not a measurement, and the wheel speeds ω_2 that are provided by the anti blocking systems over CAN bus. However, the wheel speed resolution is low and the data are available only above a minimum speed and with some time delay

Linear control laws have difficulties to cope with the nonlinear backlash in the system. That is why the control structure from Fig. 4. was applied.

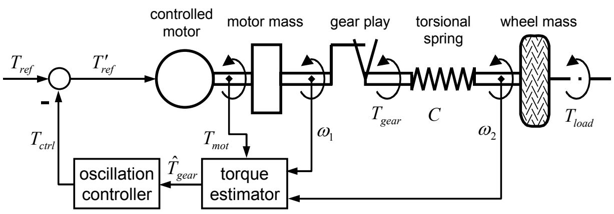

Fig. 4. Control structure

The control system consists of two parts: One block has the task to estimate the gear torque T_{gear} because knowing this quantity will enable a rather effective damping control. This torque estimator is built as an optimal Kalman filter that includes a nonlinear state space model for the system dynamics between T_{mot} and ω_2. The nonlinear backlash can be considered in this observer without problems. The complete observer is of third order. The delay of the wheel speed signal ω_2, which is caused by the CAN bus data transfer, is easily considered within the observer introducing the delay N.

$$\begin{aligned}
&\hat{\omega}_1(k+1) = \hat{\omega}_1(k) + \frac{t_s}{J_1}\left(T_{mot}(k) - \hat{T}_{gear}(k)\right) + L_{\omega_1} e(k) \\
&\hat{\omega}_2(k+1) = \hat{\omega}_2(k) + \frac{t_s}{J_1}\hat{T}_{gear}(k) + L_{\omega_2} e(k) \\
&\Delta\hat{\varphi}(k+1) = \Delta\hat{\varphi}(k) + \\
&\frac{t_s}{2}\left(\hat{\omega}_2(k+1) - \hat{\omega}_1(k+1) + \hat{\omega}_2(k) - \hat{\omega}_1(k)\right) + L_{\Delta\varphi} e(k) \\
&\hat{T}_{gear}(k) = C\, b\left(\Delta\hat{\varphi}(k)\right)
\end{aligned} \tag{2a}$$

$$e(k) = \begin{bmatrix} \omega_1(k) \\ \omega_2(k-N) \end{bmatrix} - \begin{bmatrix} \hat{\omega}_1(k) \\ \hat{\omega}_2(k-N) \end{bmatrix} \tag{2b}$$

$L_{\omega 1}$, $L_{\omega 2}$, $L_{\Delta\varphi}$ are the Kalman gains, which come from a linear quadratic Gaussian (LQG) design procedure. Compared with the model equations (1), the load torque T_{load} was omitted in the observer equation because it is not available as measurement in a standard vehicle. However, the feedback of the observation error will compensate for the missing term. The second part of the control system, the oscillation controller

$$G_c(z) = \frac{T_{ctrl}(z)}{\hat{T}_{gear}(z)} \tag{3}$$

is a linear controller of third order. This controller calculates a corrective torque T_{ctrl} depending on the estimate $\hat{T}_{gear}$ that is provided by the observer. Because the oscillation controller must not change the stationary torque, its stationary response must be zero, $G_c(1) = 0$, i.e. one of the controller zeros has to be at $z = 1$. The remaining zeros and poles were designed via root locus, Fig. 5.

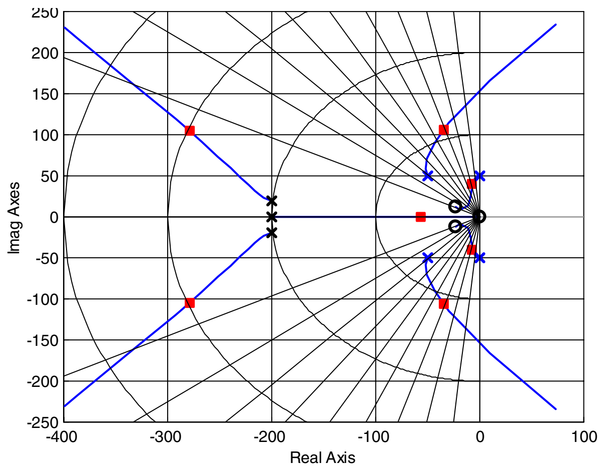

Fig. 5. Root locus design of the controller

For the control design, a linearized system without blacklash was assumed. The system has two poles (marked as crosses) on the imaginary axis, which reflects the undamped oscillation. Two other poles are due to the approximation of the induction motor as second order system. One zero (marked as circles) of the controller is at the origin, as mentioned above. Placing two more zeros close to the origin and close to the real axis in the left half plane ensures that the poles on the imaginary axis are drawn towards them with satisfactory damping depending on the controller gain. High gain makes good damping with less robustness, low gain makes moderate damping with good robustness. The controller has three zeros and therefore three poles need to be placed appropriately. They are placed on (or close to) the real axis in the far left of the left half plane so that they do not disturb the root locus behaviour close to the origin.

The controller was validated with an extended model of the vehicle dynamics, using a Simulink environment before tests in the real vehicle were tried. The test of the controller software was done in a two-step procedure. For a first test of the implemented C code modules, they were substituted for the original Simulink modules in this simulation environment so that their output could be directly compared with that of the original simulation modules. As a second test, the C code modules were run in their final embedded environment together with vehicle and motor control on the target MCU hardware in an off-line mode: The responses of the controller to artificially excited inputs were recorded and were again compared with the simulation results.

After that preparation, the commissioning of the controller in the vehicle was performed without problems in a very short time because functional faults and coding bugs were already eliminated.

5. TEST RESULTS

Fig. 6 shows measurements without the oscillation control. The measurements are taken on a test track with moderate speed. The driver accelerates unevenly which excites the drive train oscillations as it can particularly be seen from the motor speed ω_1 and the wheel torque T_{load}. The oscillations are poorly damped. After each excitation it takes a few seconds for fading away. It is evident that these oscillations cause high mechanical stress and are felt as shuddering by the driver. The large difference between wheel torque T_{load} and motor torque T_{mot} in the last few seconds of the plot is due to the mechanical braking.

Now, Fig. 7 shows a situation with the oscillation control turned on. Because the accelerator pedal was manually operated by the driver, the situation is not exactly the same as in Fig. 6, but fairly similar. The oscillations are just about gone. The torque and the motor speed are very smooth. Only when the torque demand crosses zero so that the gear backlash is traversed, a small flicker can be seen in the motor speed.

A Fourier analysis of the measured wheel torque without and with the proposed control scheme is shown in Fig. 8. Without control, a clear resonance peak can be seen at about 7 Hz, while that is completely gone with the active suppression.

The figures show very impressively the effectiveness of the proposed control. Obviously, the control can also cope well with the backlash nonlinearity, which is an advantage compared with other linear controllers.

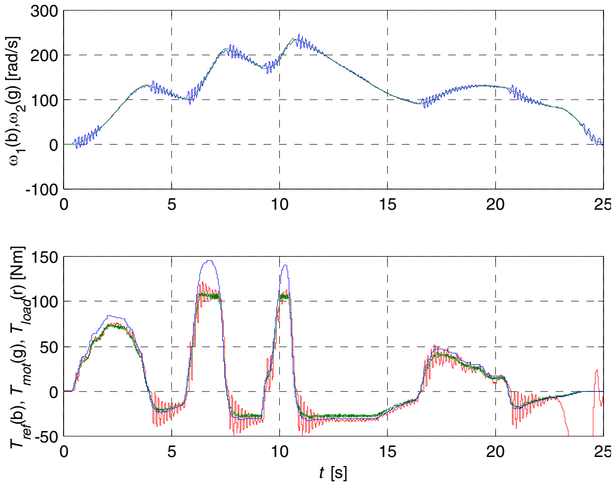

Fig. 6. Measurement data of vehicle test at moderate speed without oscillation control

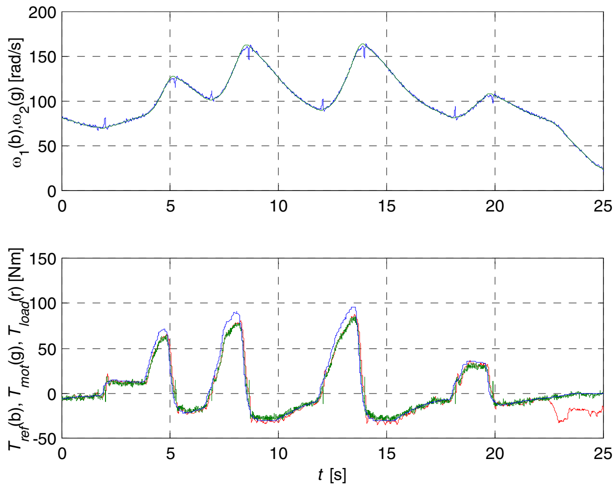

Fig. 7. Measurement data of vehicle test at moderate speed with oscillation control turned on

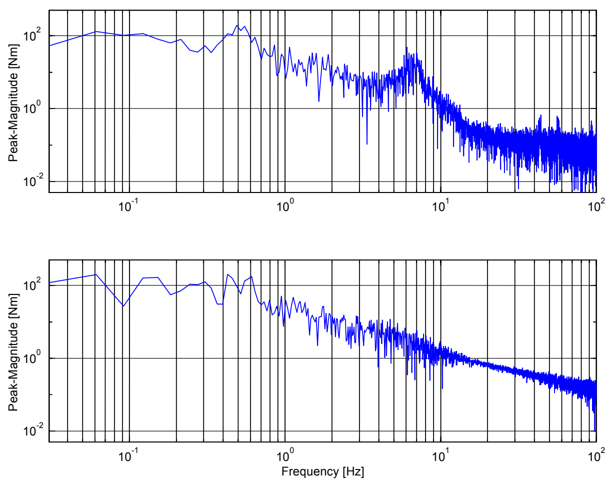

Fig. 8. Fourier analysis of measured wheel torque (top: without control, bottom: with control)

6. OUTLOOK

Since parameters of the drive train such as the elasticity and the gear play may vary during the lifetime of the vehicle, an adaptation of the observer and controller parameters is advisable in order to ensure optimal damping and robustness (see chapter 4). Thus, the design specification for the controller need to be tuned accordingly, which is beyond an adaptive control scheme. In a Collaborative Research Centre of the German Research Council self-optimizing systems are investigated, which are considered as consequent post-development of adaptive controllers, combining behavior-based and model-based schemes. In cases of varying requirements, it is necessary to re-adjust the specification for the adaptation. Within this framework, *self-optimization* of a mechatronic system means the endogenous adaptation of a target vector as response to altered environmental conditions. This can lead to an adaptation of controller structures, system behaviours and parameters.

A framework aiding the structuring and providing the means to accomplish self-optimization in complex mechatronic systems is called *operator controller module* (OCM), which is illustrated in Fig. 9. The *cognitive operator*, at the highest level of the OCM, employs physical-technical models of the plant for optimizing controller structures and parameters as well as set points. These proposals will be transferred to the second level, called *reflective operator*, which handles a state machine for

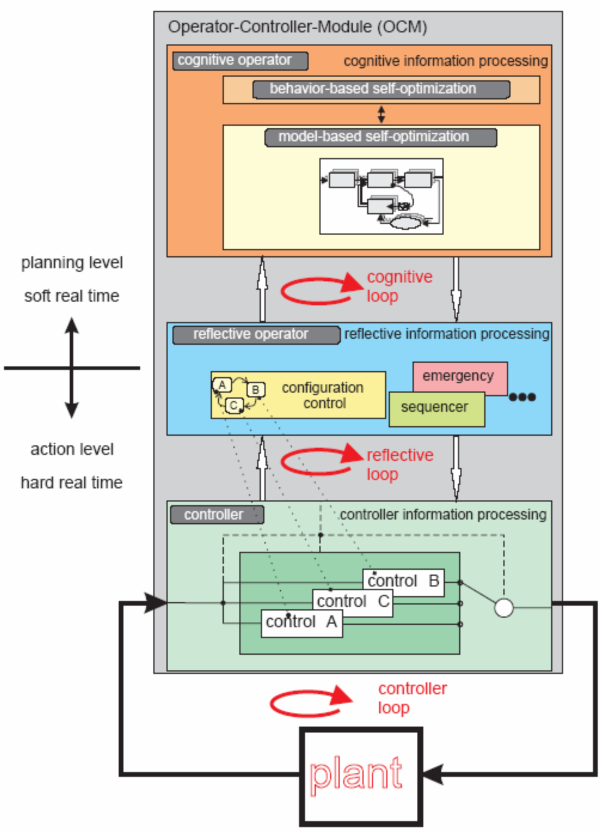

Fig. 9. Operator Controller Module

activating the new controller structure and evaluates the optimization targets by an objective monitoring. The optimized controller structures and parameters can be downloaded to the third level, the controller, which is directly linked to the plant.

The model quality of the plant is important for the quality of the optimization result. Nevertheless, due to the aforementioned structuring, the model also has to consider influences of other subsystems, e.g. OCMs with their own optimization.

Such concepts of self-optimization enable autonomous systems with inherent intelligence.

7. CONCLUSIONS

The proposed active damping control scheme can very effectively reduce the drive train oscillations. In comparison with other linear control structures, the controller copes with nonlinear backlash effects that are included in the torque observer. Mechanical stress is diminished and riding comfort is increased. The control algorithm is implemented in the existing motor control unit without any additional hardware cost. The algorithmic effort is moderate.

Damping of a mechanical system by control electronics is a good example of mechatronic design, where a well designed system turns out a better performance than it was initially intended by the design of the components.

The outlook shows an intended implementation for structured online optimization of a mechatronic system, considering variable objectives.

8. ACKNOWLEDGMENT

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