Active Damping of Drive Train Oscillations for an Electrically Driven Vehicle

Notker Amann, Joachim Böcker, and Franz Prenner

Abstract—A problem encountered with electrically driven vehicles are 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.

Index Terms—Active damping, drive train oscillations, electrical vehicle.

I. INTRODUCTION AND PROBLEM STATEMENT

Electrical drives are of increasing interest for road vehicles, either as a 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) that was experimentally fitted to a Mercedes-Benz A-Class vehicle (Fig. 2). 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 riding comfort because the oscillations can cause, particularly at low speeds, that the whole vehicle shudders 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.

II. STATE OF THE ART

Active damping of oscillations with electrical drives has been a subject of many publications. A possible road are frequency-domain approaches like introducing filters with certain frequency characteristics in the control loop in order to compensate for the mechanical resonances [9]. Another strategy is the emulation of physical behavior of a damper by the controller [10]. State-space methods have advantages particularly for oscillating systems of higher order [6]. Some proposals assume the speed as control objective and cannot be used for torque control. The choice of method also depends on the available measured signals that are mostly motor and/or load speed. Another distinguishing feature is the consideration of nonlinearities of the system, particularly nonlinear friction or backlash [4]. Vehicle shuddering is also subject of vehicles with conventional combustion engines [8]. Good surveys can be found in [2] and [7].

The contribution of the paper is a control algorithm to improve the mechanical properties of an electric drive train. A challenging restriction is to use only existing sensors and motor controller hardware. A combined approach of a nonlinear state–space observer and a frequency-domain controller meets the requirements perfectly as shown below.

III. DRIVE TRAIN MODEL

Drive train oscillations may always appear, in principle, if there exist 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 8 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. 3) (see [4] for comparisons). The resulting functional block diagram of this model is shown in Fig. 4.

The spring $C$ represents all the elastic elements in the drive train. The backlash with total width $\pm \varphi_B$ 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. The 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 $T_{\text{load}}$.

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 $T_{\text{ref}}$ and actual air gap torque $T_{\text{mct}}$ of the motor can be approximated as a second order system with time constant $\tau$

$$
\frac{T_{\text{mct}}(s)}{T_{\text{ref}}(s)} = \frac{1}{s^2 \tau^2 + \sqrt{2}\tau s + 1}. 
$$

(1)

The input to the complete system is then the demanded torque $T_{\text{ref}}$. The values of the various parameters are either known or are determined experimentally (see Section IV).

Although, the real system has a small damping, it was neglected in the model. That simplifies the model, and, if an active damping controller is able to cope with a completely undamped system, a small additional damping will not disturb in the end. The equations for the dynamic model of the system are thus as follows:

$$
\begin{align*}
J_1 \omega_1 &= T_{\text{mct}} - T_{\text{gmr}} \\
J_2 \omega_2 &= T_{\text{load}} + T_{\text{gmr}} \\
\dot{\varphi}_1 - \dot{\varphi}_2 &= \omega_1 - \omega_2 \\
T_{\text{gmr}} &= C \varphi_1 (\varphi_1 - \varphi_2)
\end{align*}
$$

(2)

including the nonlinear backlash function $b(x)$, as depicted in Fig. 4, with total backlash 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 is just a simplification of the real system and that the artificial model elements do not necessarily have 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 (aferwards) 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.

IV. 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 $\varphi_B$ 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 $T_{\text{ref}}$ on a test track. Data of the speeds of the motor and the wheels, $\omega_1$ and $\omega_2$, respectively, and of the load torque $T_{\text{load}}$ 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.

The three parameters were identified using a function from the Matlab Identification Toolbox. Fig. 5 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 the real system quite well. Some minor differences remain due to the simplicity of the model. The resonance frequency resulting from the identified parameters is 8 Hz.
V. CONTROL DESIGN

The controller design has to take care of several objectives and constraints:

1) oscillations should be damped;
2) control must not change the stationary torque;
3) control should behave unnoticed for the driver and should not conspicuously change the driving behavior;
4) it must be robust enough to allow for considerable parameter variation and model uncertainty;
5) only existing measurement signals can be used due to cost reasons:
   a) the motor speed \( \omega_1 \), which is available from the motor control;
   b) the motor torque \( T_{\text{mot}} \), which is provided by the motor control, although only as an estimate, not a measurement;
   c) the wheel speeds \( \omega_2 \) are available from the motor data as provided by the motor control.

Linear control laws like proposed in [1]–[3], [6] have difficulties to cope with the nonlinear backlash in the system. That is why the control structure from Fig. 6 was developed.

The control system consists of two parts: one block has the task to estimate the gear torque \( T_{\text{gears}} \) 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 dynamics between \( T_{\text{mot}} \) and \( \omega_2 \) according to Fig. 4. The nonlinear backlash can be considered in this observer without problems. Also, the time delay of the wheel speed signal \( \omega_2 \), which is caused by the CAN bus data transfer, can easily be taken into account in the observer. The observer is of third order and consists of the following set of discrete-time equations (compare the continuous-time equations of Section III):

\[
\begin{align*}
\dot{\omega}_1(k + 1) &= \frac{t_r}{J_1} \left( T_{\text{mot}}(k) - \hat{T}_{\text{gears}}(k) \right) + \frac{t_e}{J_1} \left( T_{\text{gears}}(k) - \hat{T}_{\text{gears}}(k) \right) + L_{\omega 1} \epsilon(k) \\
\dot{\omega}_2(k + 1) &= \frac{t_r}{J_1} \left( \hat{T}_{\text{gears}}(k) + L_{\omega 2} \epsilon(k) \right) \\
\Delta \hat{\omega}(k) &= \Delta \hat{\omega}(k) + t_r \left[ \left( \dot{\omega}_2(k + 1) - \dot{\omega}_1(k + 1) \right) + \dot{\omega}_1(k) \right] \\
\hat{T}_{\text{gears}}(k) &= C b \left( \Delta \hat{\omega}(k) \right) \\
\epsilon(k) &= \left[ \begin{array}{c} \omega_1(k) \\ \omega_2(k - N) \end{array} \right] - \left[ \begin{array}{c} \omega_1(k) \\ \omega_2(k - N) \end{array} \right] .
\end{align*}
\]

Because the wheel speed \( \omega_2 \) is not available below a minimum speed, the estimator can operate in both modes: with and without the wheel speed. The two modes require, of course, two different sets of Kalman gains \( L_{\omega 1}, L_{\omega 2}, L_{\Delta \omega} \).

The second part of the control system, the oscillation controller \( G_c(z) \), is a linear controller of third order. This controller calculates a corrective torque \( T_{\text{ctrl}} \) depending on the estimate \( \hat{T}_{\text{gears}} \) that is provided by the observer

\[
G_c(z) = \frac{T_{\text{ctrl}}(z)}{\hat{T}_{\text{gears}}(z)} = \frac{b_3 z^3 + b_2 z^2 + b_1 z + b_0}{z^3 + a_2 z^2 + a_1 z + a_0} .
\]

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.

Both the torque estimator and the oscillation controller were implemented as \( C \) code modules in the motor control unit with a sampling time \( t_s \), which is a multiple of the sampling time of the actual motor control. The modules were thoroughly tested before the first vehicle tests.
The controller functionality was validated with a model of the vehicle dynamics, similar to [5], using a Simulink environment before tests in the real vehicle were tried (see Section VI). 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 the 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.

VI. VEHICLE TEST RESULTS

Fig. 7 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 and the wheel torque . 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 and motor torque in the last few seconds of the plot is due to the mechanical braking.

Now, Fig. 8 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. 7, 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. 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.

VII. CONCLUSION

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.

REFERENCES


Parameter Identification and Vibration Control in Modular Manipulators

Yangmin Li, Yugang Liu, Xiaoping Liu, and Zhaoyang Peng

Abstract—The joint parameters of modular manipulators are prerequisite data for effective dynamic control. A method for identifying these parameters using fuzzy logic was devised to study modular redundant robots. Experimental modal analysis and finite element modeling were exploited to model the dynamics. The joint parameters of a nine degrees-of-freedom (9-DOF) modular robot have been identified. In addition, active vibration control based on a neural network and a genetic algorithm were investigated. Ideal control simulation results for a reduced dynamic model of the 9-DOF modular robot were then derived.

Index Terms—Genetic algorithm, modular robot, neural network, parameter identification, vibration control.

I. INTRODUCTION

The development of a robot requires that it be able to adopt as many configurations as possible using limited links, so as to allow the construction of new types of robot without redesign and remanufacturing. The modular robot concept can be traced back to the 1970s [1]. Along with the development of modular robots, the current trend is toward the design of self-organizing, self-reconfigurable, self-assembling, and self-repairable modular robots [2], [3]. A snake-like Polybot was developed several years ago, which has had wide application in pipe exploration and biomedical engineering [4]. A hormone-inspired control strategy was proposed to control the CONRO self-reconfigurable robots that form different types of robots to suit various tasks [5]. Dynamics and control are becoming the next key issues after design. A robot’s joint parameters (stiffness and