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SIMULATION AND CONTROL OF A SERVO DRIVE WITH OSCILLATORY MECHANICS

Neugebauer, R.; Quellmalz, J.; Walther, M.; Schlegel, H.

Chemnitz University of Technology, Faculty of Mechanical Engineering, Institute for Machine Tools and Production Processes, Reichenhainer Str. 70, 09126 Chemnitz, Germany

ABSTRACT

Closed loop controlled feed drives are state of the art in the fields of machine tools and production machines. Finding optimal controller settings for a defined motion is the focus of actual research works. For this modern simulation tools can be helpful. In the following paper the simulation model of an industrial drive system with an oscillatory mechanics is discussed. The mechanics is modeled with a multiple mass approach. The verification of the simulation model is based on a test stand which represents a feed axis. A servo motor coupled with a ball screw drive moves a linear guided machine table, which can be loaded with extra masses. The verification is discussed by the step response of the closed loop current and speed control as well as the speed of the guided table. Additionally, the effect of changing the controller gain is examined based on the model and tested experimentally. Focusing on integration in virtual reality and simulation based parameter optimization the usability of the proposed simulation model is discussed.

Index Terms - servo drive, ball screw drive, multi mass oscillator, speed controller

1. INTRODUCTION

For automating machine tools and production machines high-performance control systems are used. By now modern industrial control systems of the class motion control (MC) or computerized numerical control (CNC) provide the facility of integrated commissioning and parameterization of particular drive components and control loops. Against this background finding optimal controller settings for a defined motion is the focus of actual research works and scientific publications. For providing reproducible results under constant system conditions simulation models on different complexity levels can be helpful (vgl. [1]).

The paper deals with an industrial drive system modeled in MATLAB®/Simulink®. Starting with chapter 2 the test stand is presented. The modeling of the electrical drive follows in chapter 3. First, the model of a permanent magnetic synchronous machine is introduced. After explaining the simulation of the setting element with pulse width modulation (inverter), the structure of current and speed control is presented. In chapter 4 a multiple mass approach chosen for modeling the mechanical plant is covered.

After the verification of the simulation model in chapter 5, which is based on the test stand, the paper will be summarized. An outlook is given with focus on using the model in the context of hardware-in-the-loop (HiL) coupling and optimization.

2. FEED DRIVE TEST STAND

The described simulation model is created and verified using a suitable demonstrator. For this a frequently used component of machine tools is selected. The test stand represents a feed drive as shown in figure 1. Via a ball screw, the rotational movement of the drive is converted into a translational movement of the slide. The motor is designed as a permanent magnetic synchronous machine (PMSM). In addition to the pitch and the diameter of the ball screw, the moving mass of the slide can be varied. Thus the mechanical plant can be changed. The total stroke of the axis is 1500 mm.

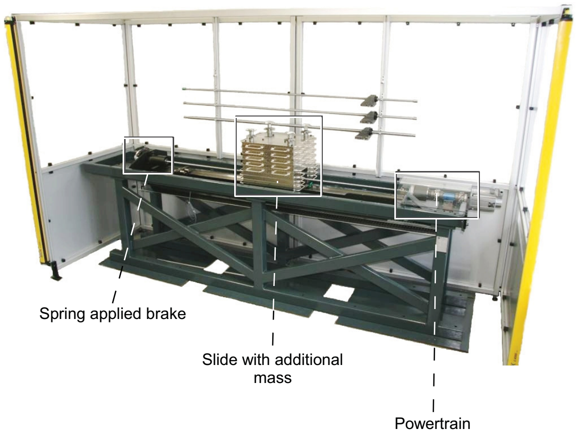

Figure 1: Feed drive test stand

For position controls either two direct (absolute and relative) or an indirect (motor encoder) measuring system can be used. The test stand is automated with SIMOTION D445 as an MC representative. SIMOTION with the integrated drive system SINAMICS S120 offers the possibility for comprehensive parameterization of the control loops

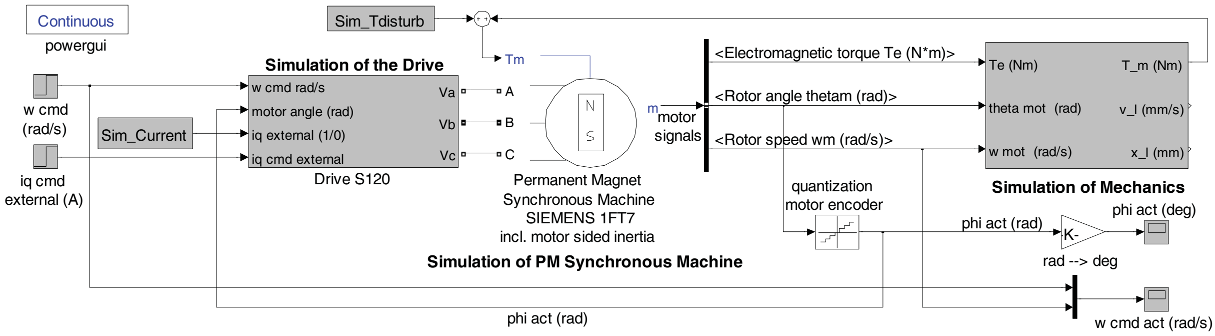

Figure 2: Model structure

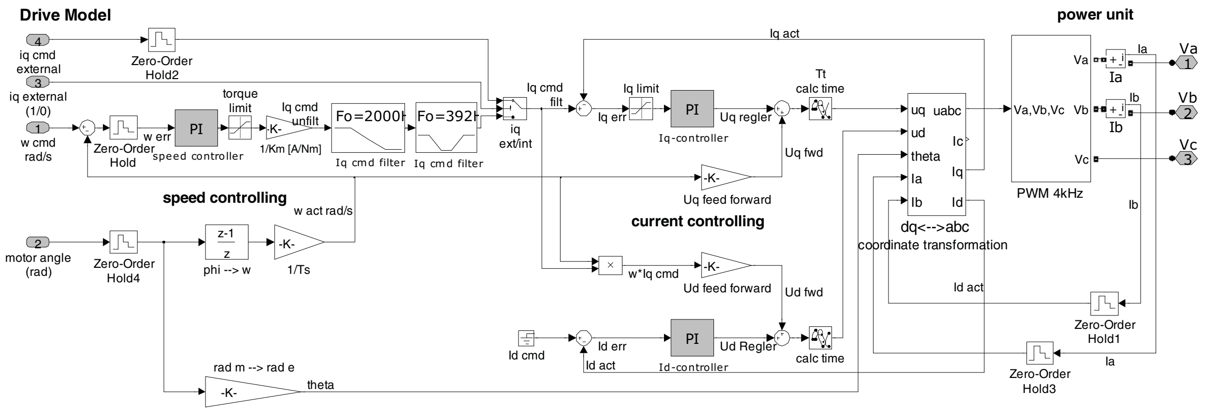

Figure 3: Drive model

beginning with the position control down to the level of current control.

3. ELECTRICAL DRIVE MODELING

3.1. Overview

The model is completely implemented in MATLAB®/Simulink®. A goal of design has been high closeness to industrial drives with respect to function, parameterization and not at least signal measurements. This approach was chosen because of three longer-term objectives:

- integration into HiL simulation/Virtual Reality focusing on virtual commissioning, as proposed in [2]
- usage in the context of simulation-based optimization, as in [3]
- simulative commissioning/tuning of extended controller structures, e.g. published in [4] or also Advanced Position Control (APC), compare [5]

Of course for either case, some adequate simplifications of the model may be done. The top-level structure of the model is seen in figure 2. It is comprised of the drive, the motor and the mechanics, each of them explained in the following.

3.2. Permanent magnetic synchronous machine

For the simulation of the PMSM an element from the SimPowerSystems®-Toolbox of Simulink® is used. It can be parameterized widely by data sheet values of the motor (see Table 1) and offers all interfacing signals needed, like voltages, currents and torque as well as position and speed signals. The element integrates the coordinate transformation between 3-phase values and d/q-coordinates. For interfacing to mechanics the PMSM-element has single-mass acceleration integrated, which can be fed back from outside either by speed or load torque input; the latter one is used here.

Table 1: Parameters PMSM

parameter	value [unit]
nominal torque T_N	2.9 [Nm]
nominal current I_N	3.4 [A]
nominal speed n_N	6000 [rpm]
stator phase resistance R_S	0.38 [Ω]
stator inductance (direct L_d, quadrature L_q)	$4.1 \cdot 10^{-3}$ [H]
torque constant	1.0 [Nm/A]
pole pairs	5

3.3. Setting element

The drive itself is subdivided into the setting element and the controllers as in figure 3. The setting element is confined to the pulse width modulation representing the motor sided power unit. So only the inverter is simulated and the switching operation of the electrical components is idealized. The rectifier and DC link are neglected assuming constant DC link voltage.

3.4. Current and speed control

Current and speed control are implemented as the classic cascade structure. These are calculated at discrete sample rates equal to the industrial drive.

Current control – the inner control loop – realizes the well-known field-oriented current control used for PMSM [6, p. 824]. For this the 3-phase voltage/current system has to be transformed into two components by the transformation angle (theta in the model), determined by the rotor angle and pole pairs. The quadrature-axis current Iq is proportional to the torque and therefore its command value is the manipulated variable of the super-ordinated speed controller. The current controllers in both axis (quadrature and direct) are implemented as PI-controllers with the same parameter values and some anticipatory control.

The speed controller, also a PI-controller, has the requested torque as output. Therefore it has to be converted to the current command value by the motors torque constant. In the compared industrial drive system there are also current command value filters implemented. The single low pass filter is used to neglect frequencies out of influence for the drive, due to the sample time and the PWM frequency respectively. Other types of filters (bandstop, other 2nd order filters) are used in the automatic controller commissioning to enhance the frequency response of the speed controllers plant, so that a better control dynamic can be achieved through e.g. damping mechanical eigenfrequencies, allowing for higher controller gain values.

Table 2: Parameters drive

parameter	**value [unit]**
cycle time	$125 \cdot 10^{-6}$ [s]
PWM frequency	$4 \cdot 10^{3}$ [Hz]
current control	
controller gain K_{P_i}	10.724 [V/A]
integral time T_{I_i}	$2 \cdot 10^{-3}$ [s]
1st current command value filter - lowpass	
frequency	$2 \cdot 10^{3}$ [Hz]
damping	0.707
2nd current command value filter - bandstop	
frequency	392 [Hz]
damping	0.25
speed control	
controller gain K_{P_n}	2.0 [Nm·s/rad]
integral time T_{I_n}	$20 \cdot 10^{-3}$ [s]

4. MECHANICAL MODEL

For modeling the mechanic system, the technical design of the test stand has been analyzed. The main mechanic element is the feed axis' ball screw drive. The screw has the major resiliences of the whole system in torsion and axial direction. Another significant resilience is the torque measurement shaft. For security reasons a brake is mounted to the screw opposite to the motor. The brakes rotor and the motor have nearly the same moment of inertia. This means the combination motor – screw – brake is a two mass system alone i.e. without the nut and the guided tables mass. This system is obviously a pure torsion oscillator. The linearly guided table connected to the motor by the ball screw is of course another oscillator. So at least a 3-mass approach has to be used here. According to [1] the torsion and axial stiffness of the screw can be combined via the screw pitch and modeled as a single one. This brings up the problem, that the spring stiffness between table and motor is lower (because of the additional axial resilience) than between the brake and the motor, although the brake is mechanically mounted in series to the table/nut. This issue is not completely resolved here, but organizing the system as a parallel structure rather than a 3-mass series system showed up with good results as seen in chapter 5. As a result, the mechanical system is modeled as seen in the following figure. For convenience the oscillatory mechanic system is completely implemented as translatory system on load side.

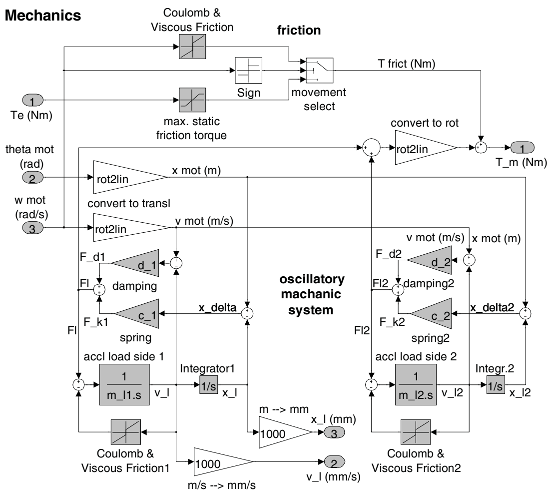

Figure 4: Mechanic model in MATLAB®/Simulink®

The parameters of the oscillatory mechanics are broken down to two triples of spring stiffness, damping factor and load sided mass each, as seen in figure 5. The motor sided moments of inertia are included in the PMSM described in section 3.2 as follows:

$$J_m = J_{mot} + J_{cl} + \frac{J_{sd}}{4} \,. \tag{1}$$

The load sided masses and moments of inertia are combined and reduced accordingly:

$$m_{l1} = \frac{J_{sd}}{2} \cdot \frac{4\pi^2}{h_{sd}^2} + m_{sn} + m_{sl} + m_{load} \tag{2}$$

$$m_{l2} = \left(\frac{J_{sd}}{4} + J_{br} \right) \cdot \frac{4\pi^2}{h_{sd}^2} \tag{3}$$

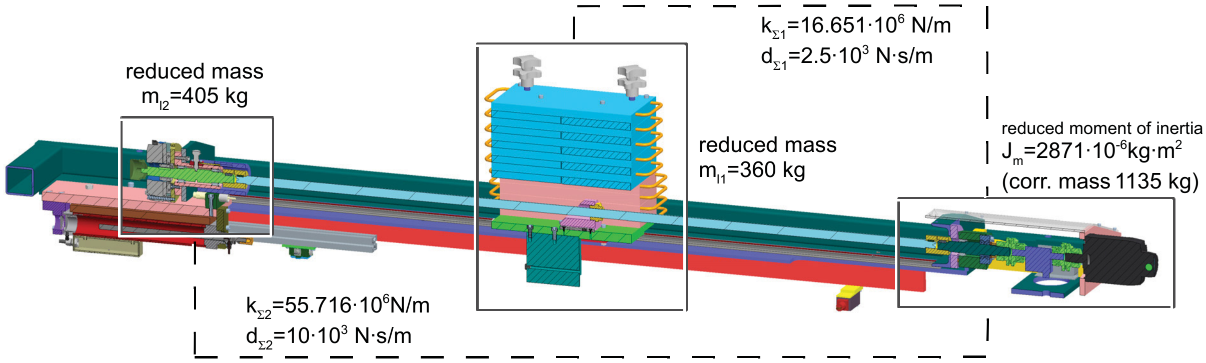

Figure 5: Test stand mechanics and modelled reduced parameters

The stiffness parameters can also be calculated from the technical construction (see Table 3) and/or are to be tuned, so that the model meets the eigenfrequencies of the test stand. In the case here good results were achieved with the values obtained by the following formulas:

$$\frac{1}{k_{\Sigma 1}} = \frac{1}{1.5} \cdot \frac{h_{sd}^2}{4\pi^2} \cdot \left(\frac{1}{k_{sctor}} + \frac{1}{k_{ms}} \right) + \frac{1}{k_{sn}} + \frac{1}{k_{scax}} \qquad (4)$$

$$\frac{1}{k_{\Sigma 2}} = \frac{h_{sd}^2}{4\pi^2} \cdot \frac{1}{k_{sctor}} \qquad (5)$$

The factor of value 1.5 in formula (4) is used to meet a frequency of the oscillation of the table speed. It can be argued, that the table is not at the end of the screw (like the brake), so the oscillator is stiffer. Consequential, a factor to the screw stiffness's only could be applied.

Table 3: Mechanical parameters

parameter	value [unit]
screw pitch h_{sd}	$10 \cdot 10^{-3}$ [m]
mass and moment of inertia	
slide m_{sl}	50 [kg]
load mass m_{load}	300 [kg]
screw nut m_{sn}	1.0 [kg]
motor J_{mot}	$1190 \cdot 10^{-6}$ [kg·m^2]
clutch + measurement shaft J_{cl}	$1670 \cdot 10^{-6}$ [kg·m^2]
screw drive J_{sd}	$45 \cdot 10^{-6}$ [kg·m^2]
brake J_{br}	$1014 \cdot 10^{-6}$ [kg·m^2]
stiffness parameters	
screw axial k_{sdax}	$15 \cdot 10^{6}$ [N/m]
screw torsion k_{sdtor}	141.13 [Nm/rad]
clutch k_{cl}	$20 \cdot 10^{3}$ [Nm/rad]
measurement shaft k_{ms}	$5.4 \cdot 10^{3}$ [Nm/rad]
screw nut k_{sn}	$200 \cdot 10^{6}$ [N/m]
damping factors	
$d_{\Sigma1}$	$2.5 \cdot 10^{3}$ [Ns/m]
$d_{\Sigma2}$	$10 \cdot 10^{3}$ [Ns/m]
friction	
static friction	0.875 [Nm]
sliding friction	0.8 [Nm]
viscous friction	0.022 [Nm·s /rad]

Damping factors are even harder to get from data sheets or construction values. So they were tuned to fit the decay of the resulting oscillations to experimental results.

Friction is only implemented on motor side as combination of viscous, sliding and static friction. They were also tuned by experimental results. Tests were made on distribution of friction to load (guided table) and motor side, but no major differences in behavior could be detected for the examined test case. Static friction is included by principle, but no dedicated examinations were made for this, since the experiments all have significant offset to static friction. Also some problems with solving the model in relation to static friction showed up during tests, which may be avoided through implementing Stribeck friction curve in the future.

5. VERIFICATION

5.1. Model verification

The verification shall be shown in two steps. At first only the current control loop is closed and a command step with offset is applied. The responses of both – the test bench and the model – are shown in figure 6. The characteristic of the curves is different and the simulation model shows more overshot. However, rise and settle times are in good accordance. Since the current offset is higher than static friction a low speed before the step can be seen. After the step a clear acceleration is seen, which agrees good between simulation and test bench. In the speed curves the mechanical oscillations are hardly to see, but some accordance can also be seen here.

The second step is to close the speed control loop and apply a step with offset, too. The comparison can be made by looking to figure 7. For the controlled variable, which is the motor speed, very good accordance is achieved in terms of characteristic, overshot as well as rise and settling times. The mechanical oscillations – the influence of which is observable here – show quite good accordance. The guided tables speed is shown in figure 7 with identical scaling to the motor speed. The effect of the elastic coupling is seen in the strong oscillation with much more "overshot" compared to the motor speed.

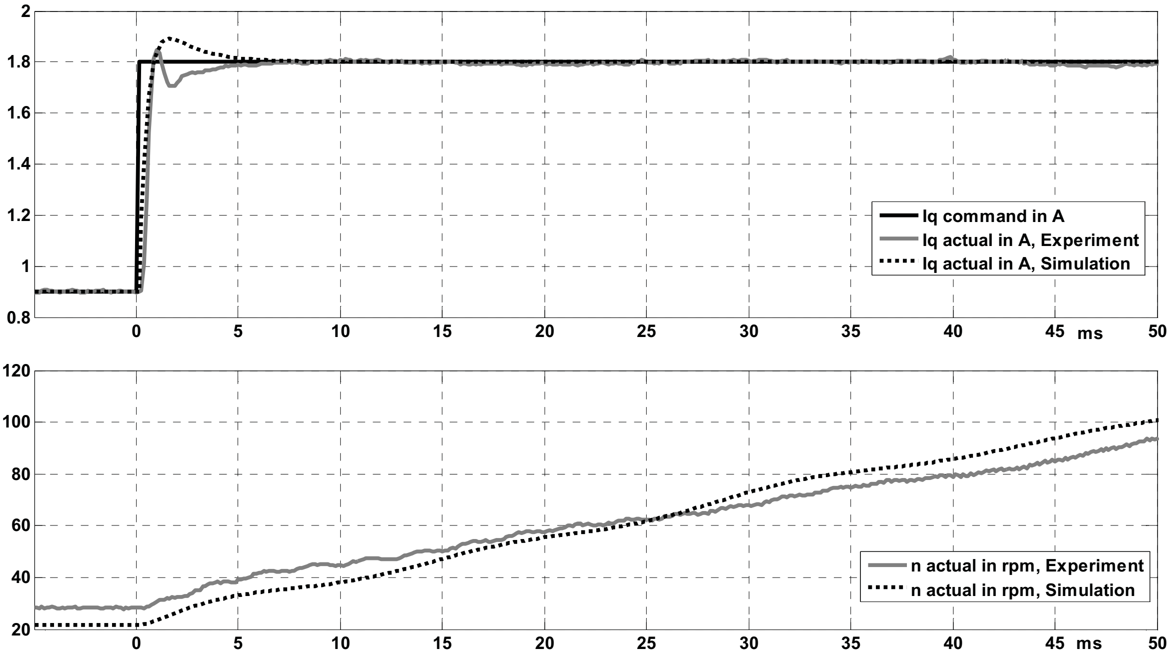

Figure 6: Command current step, test stand compared to simulation

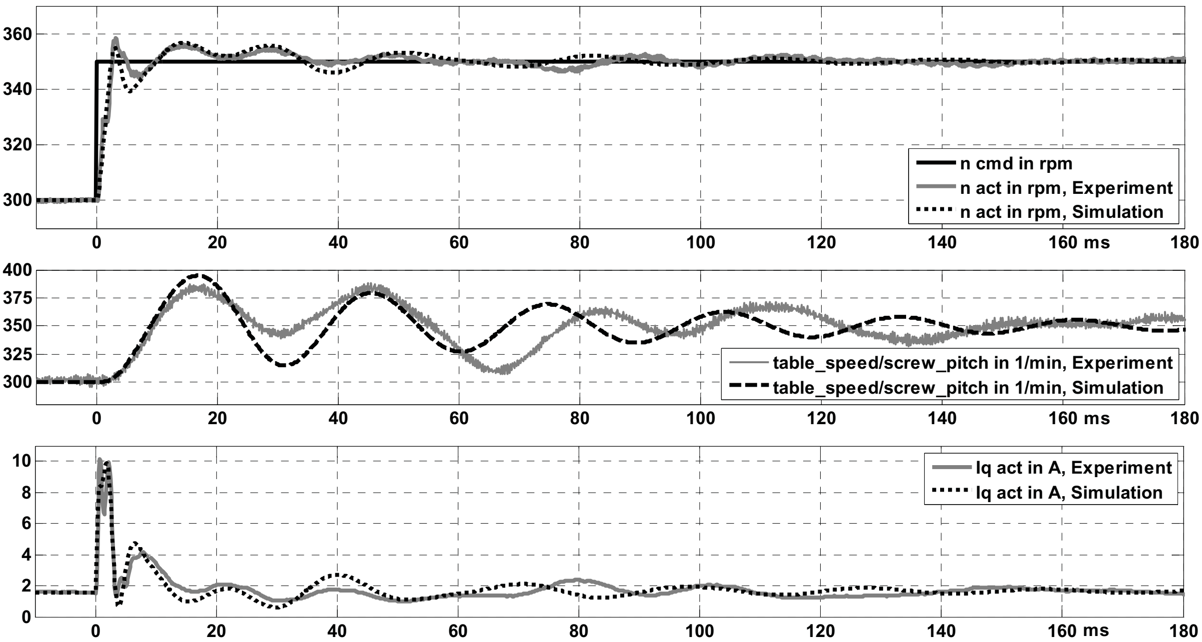

Figure 7: Command speed step, test stand compared to simulation

Overshot and decay of the table speed can be modeled quite well. Due to the implemented structure of the mechanical model, only one frequency is seen here. At the test bench, there is of course a mix of frequencies. Tests were made using a serially structured mechanical model (motor – brake – table), but the achieved accordance was even worse.

The static current values – being owed to friction parameters – agree very well. Also the duration and value of the first current peak and first oscillations agree. Differences in oscillations some time after the step are seen as a result of the reduced model order.

5.2. Parameter influence

The desired usage of the model requires its ability to reproduce the effects of changing parameters, especially the controller tuning. Exemplarily this shall be shown by detuning the speed controller gain. Figure 8 shows the step responses for halving this parameter. It can be seen, that – compared to figure 7 – the rise time increases and the current peak is lowered. In contrast, the curves of the table speed are nearly identical, the excitement of the relevant oscillation is nearly the same. But the fact of most interest is, that these effects can be predicted by the

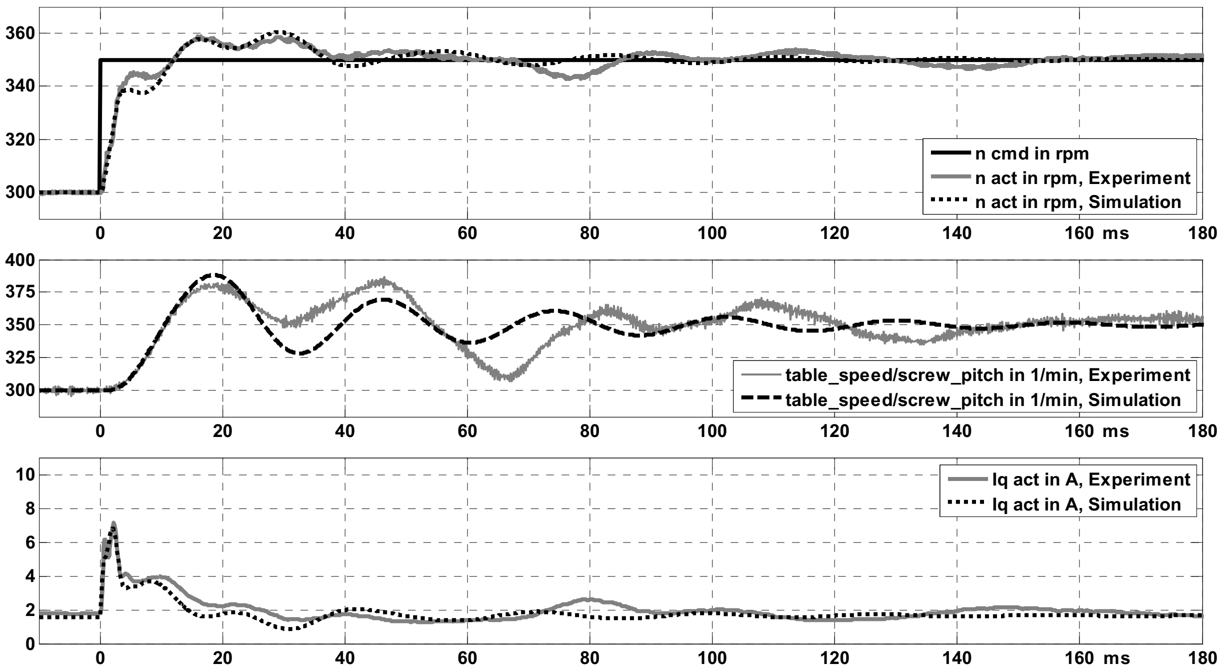

Figure 8: Command speed step, speed controller gain halved, test stand compared to simulation

simulation model quite well. The accordance is as good as without detuning the controller gain.

6. SUMMARY AND OUTLOOK

In the paper at hand a detailed model of an electrical servo drive with oscillatory mechanics is presented. Deriving the model and tuning its parameters to a feed drive test stand is shown. It has been pointed out, that because of the design of the test bench problems have been encountered by using a standard serial mass system known well and described in the literature for modeling the mechanics. A different approach has been used, and – as discussed in the paper – results close to experiments have been achieved.

Some oscillatory effects cannot be reconstructed by a reduced order model. By increasing the order of the mechanical model possibly a better accordance can be attained. But it has to be checked, whether this is necessary for the desired application, having the number of parameters to be tuned in mind, too.

The influence of a single (controller) parameter is shown and verified in the paper. Of course, to use the model for tuning controller parameters, it would be good to check other parameters, too. For robustness considerations, also variation of plant parameters (especially mechanical parameters) is of great importance.

For the desired usage of the model, e.g. using it in a HiL environment and for simulation based optimization, the model quality is seen as satisfactory. Some simplifications may even be adequate. Of course, this has to be proved in future research. Another objective will be commissioning and/or tuning of extended controller structures, e.g. Advanced Position Control (APC) by using the model. Besides, the superordinated position control can be implemented and examined with respect to the closed-loop speed control behavior and usage of direct and indirect position measurement systems. Furthermore, the model shall be adapted for different mechanic systems, like a flexible robot structure (see [7]) or linked linear drives, see [8].

7. ACKNOWLEDGEMENTS

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