Self-tuning speed control for servo drives with imperfect mechanical load

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Abstract — A self-tuning PI-speed control for drives featuring elasticity is presented including automatic identification of total inertia, estimation of the Frequency Response Function for the mechanical plant and optimal design of PI-speed controller. The approach allows automatic controller commissioning for drives already mounted in the mechanical environment of the industrial plant because position, speed and torque of the motor are kept within specified limits during the identification process. The control performance is compared to the results of an analytic design method based on a parametric two-mass model.

I. INTRODUCTION

Parameters for high performance speed and position controllers of industrial servo drives have to be designed with respect to a number of drive parameters. These are bandwidth of current control loop, structure and parameters of the mechanical plant as well as the accuracy of measurements. Because of these various influences adjustment of controller parameters is often a complicated and time consuming task, especially for drives featuring elasticity, friction and torque ripple. Consideration of these mechanical imperfections improves the performance of speed and position controllers and reduces the commissioning time. The PI controller has its limits for elastic drive systems, if compared to more sophisticated controllers featuring e.g. active vibration damping [1], [5]. But nevertheless the cascaded structure of PI-speed- and P-position control is still the standard concept in most industrial drives and therefore automatic tuning methods are investigated here.

A general automatic approach leading to optimal results for any kind of drive application seems to be illusory, but software tools sustaining the application engineer in experimental modeling, controller design, systematic documentation and preservation of commissioning experiences allow a significant reduction of this commissioning effort [2].

In this paper an approach is presented featuring a high degree of automation regarding the commissioning of a PI-speed controller with command signal filter for drives already mounted in the mechanical environment of an industrial machine. The current control loop of the motor is approximated as first order lag featuring a high bandwidth compared to the dominant resonance frequencies of the mechanic. Furthermore it is assumed that just motor position and motor currents are measurable.

In section II an automatic procedure for experimental identification of the total drive inertia is presented. The result is used for initial and automatic commissioning of a provisory PI speed controller with low bandwidth. The provisory controller is needed in order to perform the experiments for estimation of the Frequency Response Function (FRF) of the mechanical transfer elements as presented in section III. In section IV an analytic design method for a PI-speed controller and a two-mass model of the mechanic is presented based on the method of double ratios. The parameters of the mechanic are extracted from the FRF by an identification method presented in [1]. An alternative PI-controller design method based on numerical optimization in frequency domain is presented in section V. The latter method utilizes the measured FRF of the mechanic and does not require parameter identification. The complete commissioning approach as well as the two design methods are validated by measurements performed on a rapid controller prototyping system for a mechanical drive set-up.

II. INITIAL COMMISSIONING

The initial commissioning and all further experiments for identification are performed in a fully automatic procedure and are based on the signal flowchart depicted in Fig. 1. The forward loop consists of two alternative speed controllers, a saturation block and a low pass filter for the torque reference and of the mechanical sub-system.

Figure 1: Structure for identification and controller commissioning

RELAY: two-state relay controller with hysteresis;
PRBS: generator for pseudo random binary sequence;
PI-SC: PI-speed controller;
FOL: first order low pass filter;
CCL: approximation of closed current control loop.

Commissioning of current controllers is assumed to be already completed and the bandwidth of the torque producing current control loop is known. The current controlled motor is considered to behave like a first order lag with time constant
which also includes all delay times possibly existing in the speed control loop. The motor speed is measured by differentiation of the position signal generated by an incremental encoder or resolver and fed back to close the speed control loop. A signal generator is available to generate speed- or torque command signals. The drive system is set to different operation modes during commissioning by adjustment of the switches S1 and S2 depicted in Fig. 1.

For experimental identification of the frequency response function (see section III) the drive has to be operated temporarily at constant speed in order to avoid the effect of non linear friction not included in the identification model. This is achieved by a robust provisional PI-speed controller. To design this intermediate controller automatically the total inertia $J_x$ of the drive system has to be known.

Experimental identification of $J_x$, e.g. with a Least-Squares method, is only possible with measurement data containing sufficient acceleration of the drive. Even though speed and position control is not working yet, speed and position have to be limited during the experiment - even if external load torque acts on the drive mechanism. For automatic conduction of identification experiments the following quantities are required and have to be set by the operator:

- motor inertia $J_{motor}$
- maximum torque $M_{max}$
- maximum speed $\omega_{max}$
- maximum position $\varepsilon_{max}$

The identification experiment is performed under control of the two-state relay speed controller with hysteresis with a superimposed P-position controller with a constant reference of $\varepsilon^* = 0$ as depicted in Fig. 2a. The speed controller output passes a first order lag filter featuring an adjustable time constant $T_F$. The current control loop and the filter are approximated by a single first order lag with the time constant $T_x = T_E + T_F$.

1) Choice of free parameters for autotuning

The strategy for setting the free controller parameters $a, b, T_F$ and $K_{pos}$ for the initial identification experiment is described in the subsequent text.

The width $a$ of the symmetrical controller hysteresis and its gain $b$ as well as the time constant $T_F$ have to be adjusted so that a stable limit cycle is initiated within the non linear speed control loop, i.e. a stationary oscillation of the state variables $\omega_p(t)$, $\varepsilon_M(t)$ and $m_M$ exists. On the one hand the limit cycle should not stress the mechanical system too much and furthermore the drive position must be kept within a specified range. On the other hand a sufficient acceleration of the mechanic is needed for proper identification of the drives total inertia.

Both requirements can be checked roughly prior to the start of the experiment by calculation of the estimated frequency $\omega_D$ and amplitude $A_0$ of the harmonic limit cycle according to the theory of harmonic balance [3], [4]. For the following analysis the position controller is omitted and just the switching speed controller, the first order lag filter and the mechanical system approximated by the total inertia of a one-mass model is considered. Then the nonlinear standard control loop depicted in Fig. 2b results, consisting of a linear part described by the frequency response

$$G_L(j\omega) = \frac{1}{1 + j\omega T_x}$$

and the non linear describing function

$$N(A) = \frac{4b}{\pi A} \sqrt{1 - \frac{(a)^2}{A^2}} - \frac{4ab}{\pi A^2}.$$  

The latter varies with the Amplitude $A$ of the limit cycle. For validity of the prediction with the method of Harmonic Balance, $G_L(j\omega)$ must feature low-pass characteristic to a sufficient degree which can be always forced by a suitable choice of $T_F$.

A necessary condition for the existence of a limit cycle is the occurrence of an intersection point between the Nyquist curve of $G_L(j\omega)$ and the negative reciprocal of the describing function in the complex s-plane as expressed by (3). The solutions $A_0$ und $\omega_D$ of (3) belong to such an intersection point and characterize amplitude and frequency of the limit cycle.

$$G_L(j\omega_D) = -N(A_0)^{-1}$$

Analytical calculations show that (3) has a unique pair of real solutions which is given by

$$\omega_D = \frac{1}{T_x} \left( u - \frac{1}{3} \right), A_0 = a \left( \frac{K}{\omega_D J_x} \right)^{\frac{1}{2}}$$

with the abbreviations

$$K = \frac{4b}{\pi a}, u = \frac{3}{4} \left( D - \frac{1}{27} + jD \right)$$

and

$$D = \frac{1}{27} + \left( \frac{KT_x}{2J_x} \right)^{\frac{1}{2}}$$

1. This approximation is valid below certain frequencies.
2. Note that Harmonic Balance is an approximate method even if plant and controller are known exactly.
Because the actual value of $J_2$ is not known, it is not possible to predict the parameters of the limit cycle exactly. But since $\omega_0$, as well as $A_0$ decrease monotonously with rising inertia, it is possible to calculate upper limits for both values by inserting an arbitrary value $J_1 < J_2$ into (4). The fundamental oscillations of the motor speed and the motor position are then described by (6) and (7) respectively.

$$\omega_M(t) = A_0 \sin(\omega_D t)$$  

(6)

$$\varepsilon_M(t) = B_0 \cos(\omega_D t), \quad B_0 = \frac{A_0}{\omega_D}$$  

(7)

$\varepsilon_M(t)$ alternates periodically with amplitude $B_0$. This peak cannot be predicted in advance but if an upper limit $J_2 > J_1$ for the inertia is known, an upper bound for $B_0$ can be estimated by the inequality

$$|B_0| \leq a \left( \frac{1}{K} \cdot J_2 + T_Z \right).$$  

(8)

To cope with external load torque that possibly acts on the drive, a position controller is superimposed which prevents the drive position drifts away from the medium set point chosen for the experiment. If, e.g. for technological reasons, the drive may only move in one direction, a ramp can be set as position reference in order to perform the identification experiment with a constant speed offset. Design of position controller is performed approximating the closed speed control loop result in

$$G(s) = \frac{1}{1 + sT_Ew}.$$  

(9)

An upper limit of the time constant $T_Ew$ can be estimated by

$$T_Ew < \frac{1}{2bJ_2}.$$  

(10)

The required controller gain $K_{pos}$ is calculated by the method of double ratios which is applied to the closed position control loop resulting in

$$K_{pos} = \frac{1}{\alpha T_Ew},$$  

(11)

where the design parameter $\alpha$ (typically: $2 \leq \alpha \leq 4$) is a measure for the damping of the control loop.

As a result from (4)-(11) the initial choices for the free parameters are suggested as follows:

- $T_F = 20T_Ei$
- $a = 0, 1\omega_{max}$
- $b = M_{max}$
- $K_{pos} = 1/(\alpha T_Ew)$ with $\alpha = 4$

Before starting the experiment by closing the speed and position control loop, the parameters $\omega_D$, $A_0$, and $B_0$ are estimated. If no information about the total inertia is available, the known motor inertia $J_{motor}$ is inserted into (4) to calculate upper limit values for $\omega_D$ and $A_0$. Furthermore $B_0$ is calcu-
tion (13) in discrete time domain using Euler method with sampling time $T$.

$$\frac{d}{dt} \omega_M(t) = m_M(t) - m_L(t)$$

$$= m_M(t) - (M_{L0} + M_{FC} \cdot \text{sgn} (\omega_M(t)) + \mu_{FV} \cdot \omega_M(t))$$

$$m_M(kT) = \frac{J_L}{T} (\omega_M(kT) - \omega_M((k-1)T)) + m_L(kT)$$

$$m_L(kT) = M_{L0} + M_{FC} \cdot \text{sgn} (\omega_M(kT)) + \mu_{FV} \cdot \omega_M(kT)$$

With the data depicted in Fig. 3 equation (13) is solved for the unknown parameters listed in Table I using a Least Squares algorithm.

$$J,_{11} = 0.0127 \text{kgm}^2, J, = 0.0128 \text{kgm}^2$$

$$M, = 0.0213 \text{Nm}, M, = 0.009 \text{Nm}$$

$$M_{FC} = 0.0681 \text{Nm}, M_{FC} = 0.237 \text{Nm}$$

$$\mu_{FV} = 0.0238 \text{Nms/rad}, \mu_{FV} = 0.001 \text{Nms/rad}$$

The results shown in the right column of Table II are in good agreement with the reference parameters gained in a more detailed analysis. It turns out that for identification of inertia the data from Fig. 3a are already sufficient, while for accurate identification of the load characteristic longer acceleration and a wider speed range is required during the experiment as it is the case in Fig. 3b.

$$J_L = J_L + J_M, V_J = \frac{J_L}{J_M}, d_a = \frac{\omega_0 - D_s}{2C_s}, \omega_0 = \frac{C_s}{(J_L + J_M)}$$

Then the linear part of the standard control loop in Fig. 2b has the transfer function

$$G_L(s) = \frac{1 + 2d_a \left( \frac{s}{\omega_0} \right) + [1 + V_J] \left( \frac{s}{\omega_0} \right)^2}{1 + sT_2}$$

For a graphical analysis with the Harmonic Balance the Nyquist curve resulting from (15) is plotted in the complex s-plane together with the negative reciprocal of the describing function (2). The example in Fig. 6a shows that in general two intersection points between the curves are possible, both characterizing a stable limit cycle [4].

$$a = 5 \text{rad/s}, T_s = 11 \text{ms}$$

$$a = 20 \text{rad/s}, T_s = 11 \text{ms}$$
The point \( f_1 \) located on the circular part of the Nyquist curve belongs to a frequency near the resonance frequency of the mechanic. This becomes evident looking at Fig. 7, where \( f_1 \) is marked within the frequency response functions for the linear part of the control loop \( G_L(j\omega) \). In this case the structural resonance dominates the oscillation, which stresses the mechanic and makes identification of the one-mass model difficult, since the measurement data do not fit to the reduced order model. The set point \( f_1 \) should therefore be avoided for the initial identification experiment.

The point \( f_2 \) in Fig. 6a and Fig. 7 belongs to the second possible limit cycle which would be more suitable for initial identification. Which of the two oscillations will appear during the experiment is not foreseeable in the concrete case. The unwanted intersection point with the circular shaped part of the Nyquist curve can be avoided by shifting the negative reciprocal of the describing function along the negative imaginary axis by rising the hysteresis width \( a \) (see Fig. 6b). As \( f_3 \) in Fig. 7 shows, the frequency \( \omega_0 \) of the according limit cycle has decreased compared to \( f_2 \). Resulting from these considerations the rules for the initial choice of parameters for relay and filter are similar to the approach suggested for stiff systems in subsection 1. For drives with elasticity an additional goal must be the minimization of \( \omega_0 \) choosing a sufficiently high hysteresis width \( a \) to keep away from unknown structural resonance frequencies of the mechanic.

The measurement data depicted in Fig. 8 were acquired from a drive embedded in a mechanical set-up presented in [5]. The time series in Fig. 8b shows that besides the limit cycle with \( A_0 = 20,40\, \text{rad/s} \) and \( \omega_0 = 1,08\, \text{Hz} \) a weakly damped eigenfrequency of \( f_0 = 44\, \text{Hz} \) can be observed.

Prior to identification of the one-mass model the measurement data have to be processed by a digital low-pass filter in order to remove the information about elasticity which is not contained in the identification model depicted in Fig. 3.

![Figure 7: Bode plot of \( G_L(j\omega) \)](image)

Intersection points with the negative reciprocal of the describing function are marked

Table III summarizes the identification results which are in good agreement with theory.

<table>
<thead>
<tr>
<th>Parameters of one-mass model</th>
<th>Parameters of limit cycle: observed (calculated by (4))</th>
</tr>
</thead>
<tbody>
<tr>
<td>( J_L = 0,0818, \text{kgm}^2 )</td>
<td>( A_0 = 20,40, \text{rad/s} ) ( (20,04, \text{rad/s}) )</td>
</tr>
<tr>
<td>( M_L = 0,047, \text{Nm} )</td>
<td>( \omega_D = 1,08, \text{Hz} ) ( (0,86, \text{Hz}) )</td>
</tr>
<tr>
<td>( M_{FC} = 0,09, \text{Nm} )</td>
<td>( B_0 = 0,65, \text{rad} ) ( (0,80, \text{rad}) )</td>
</tr>
<tr>
<td>( \mu_F = 0,075, \text{Nms/\text{rad}} )</td>
<td></td>
</tr>
</tbody>
</table>

Table III: Identification results for one-mass model approximation for the experimental data depicted in Fig. 8.

III. ESTIMATION OF FREQUENCY RESPONSE FUNCTIONS

In the next step of commissioning the Frequency Response Function (FRF) of the mechanic is estimated to gain a more specific model of the plant. Data for experimental identification are generated by setting the drive to alternating constant speed levels by means of a provisional PI-speed controller. The controller parameters are calculated by the "Symmetric Optimum" making use of the total inertia \( J_L \) identified by the method suggested in section II.

\[
K_P = \frac{J_L}{\alpha T_N}, \quad T_N = \alpha^2 T_L, \quad (16)
\]

Setting the design parameter \( \alpha = 4 \) leads to a controller with low bandwidth which is advantageous for the estimation of the FRF.

Within these alternating set-points system excitation is performed by adding an artificial pseudo random noise signal (PRBS) to the speed controllers output [6]. The speed offset is required because a non linear friction characteristic would falsify the measurement data, if the sign of the speed signal would alternate during the experiment.

\( N \) samples of the motor torque reference \( m_M^*(t) \), the excitation signal \( \xi(t) \) and the motor speed \( \omega_M(t) \) are acquired with a sample time of \( T = 250\, \mu\text{s} \). The respective average values are
subtracted from each signal as depicted in (17), where \( k \) denotes the sampling instant \( k \cdot T \).

\[
\begin{align*}
  u(k) &= m_d^a(k) - m_d^f(k), \quad x(k) = \omega_d^a(k) - \omega_d^f(k) \\
  z(k) &= z(k) - \hat{z}(k)
\end{align*}
\]

(17)

Because the measurement is conducted in closed loop, the input signal \( u(k) \) is correlated with the measurement noise \( r(k) \) included in the output signal \( x(k) \). In this case for identification of the exact plant model utilization of the excitation signal \( z(t) \) is required for estimation [7]. The plant’s frequency response is calculated by

\[
\hat{G}_p(j\omega_v) = \frac{\hat{S}_{xz}(j\omega_v)}{\hat{S}_{xx}(j\omega_v)},
\]

with \( v = 0, 1, 2, ..., N-1 \) and \( \omega_v = \frac{2\pi}{NT} \) (18)

In (18) \( \hat{S}_{xz}(j\omega_v) \) and \( \hat{S}_{xx}(j\omega_v) \) denote the cross spectral density functions calculated by the discrete Fourier transformation of the cross correlation functions of \( z(k) \) and \( x(k) \) as well as \( z(k) \) and \( u(k) \). Because the torque reference is utilized for identification instead of the actual torque, \( \hat{G}_p(j\omega_v) \) includes the dynamic of the closed current control loop as well as the delay times resulting from sampling, numerical differentiation of measured position for speed calculation and data transfer. The latter are summarized by the total delay time \( T_c \). To get \( \hat{G}_m(j\omega_v) \), the FRF of the mechanical system, \( \hat{G}_p(j\omega_v) \) has to be corrected as described by (19).

\[
\begin{align*}
  \hat{G}_m(j\omega_v) &= \hat{G}_p(j\omega_v) \cdot G_M^{-1}(j\omega_v) \\
  G_M^{-1}(j\omega_v) &= (1 + j\omega_v T_c) e^{j\omega_v T_c}
\end{align*}
\]

(19)

To minimize the influence of disturbance signals several experiments are conducted and the resulting FRFs are averaged. Some rules for conduction of the identification experiments are listed below:

* Bandwidth of provisional PI-controller
  As already mentioned the bandwidth of the provisional PI-control loop should be low since the controller serves only for linearization of the drive in a set point. Beyond this the analysis of the statistical error of the measured FRF given by

\[
\begin{align*}
  \sigma^2(\hat{G}_p) &= \frac{1}{|G_p(\omega)|^2} \cdot \left[ \hat{S}_{xz}(\omega) \cdot \frac{1 + G_c(\omega) G_P(\omega)}{\hat{S}_{xx}(\omega)} \right]^2 \quad \text{and} \\
  \sigma^2(\hat{\omega}_p) &= \frac{1}{|G_p(\omega)|^2} \cdot \sigma^2(\hat{\omega}_p)
\end{align*}
\]

(20)

shows that the variances \( \sigma^2(\hat{G}_p) \) and \( \sigma^2(\hat{\omega}_p) \), related to the controller \( G_c(\omega) \), decrease with the bandwidth of the control loop\(^1\). This affects mainly the low frequency part of the FRF.

* Period length of PRBS
  The PRBS is generated by the output stage of a shift register alternating between the values 1 and -1. Its input is generated by feeding the result of a logical combination of several register stages back to the first stage. As a result the signal is periodic with a period length of \( N_p = 2^N - 1 \) depending on the number of register stages \( n \). On the one hand \( N_p \) should be sufficiently high in order not to loose the pseudo random character of the signal on the other hand it should not be too high because this results in longer time periods without signal alternations, i.e. low excitation. As a trade-off \( n = 11 \) is chosen.

* Cycle time of PRBS
  The cycle time \( \lambda = \mu T \) \((\mu = 1, 2, ... )\) of the PRBS specifies the minimum time interval between an alternation of the output signal. This means that if \( \lambda \) is small, the bandwidth of PRBS is high and vice versa. On the one hand the whole frequency range of the mechanic up to the Nyquist frequency is of interest, on the other hand bandwidth of excitation is limited by the limited bandwidth \((1/T_c)\) of the current control loop. For the choice of \( \lambda \) the following formula is suggested:

\[
\omega_{av} = \frac{2\pi}{3\lambda} \quad \implies \quad \lambda = \frac{2\pi}{3T_c}
\]

(21)

* Amplitude of PRBS
  According to (20) the variance of estimation error decreases with increasing power spectral density of the excitation signal. As a consequence the amplitude \( a_{PRBS} \) of excitation signal should be high. Because of the deviations from the speed set-point which result from the disturbance through the PRBS and in order not to stress the mechanic too much, the the amplitude is limited. The value \( a_{PRBS} \) should therefore be chosen by an iterative procedure starting with a low values derived from the rated drive torque.

* Reference speed for linearization
  The value for the alternating, piece wise constant reference speed \( \omega^* \) depends on the friction and on amplitude and cycle time of the PRBS and should be so high that the speed signal does not change its sign.

* Measurement time
  With regard to disturbance rejection the measurement time \( T_{meas} \) should be as long as possible. Limitations result from the amount of memory available for realtime data acquisition. Furthermore physical position constraints, as e.g. present in case of linear movements, restrict the measurement time.

* Number of experiments for averaging
  The number of experiments \( N_{Exp} \) should be as high as possible but is limited by the amount of physical stress allowed.

---

1. For low frequencies the approximation

\[
|1 + G_c(\omega) G_P(\omega)| = |1 + G_c(\omega)| = |G_c(\omega)| \quad \text{holds.}
\]
for the respective mechanic.

In Fig. 9 the estimated FRFs for a drive with elasticity is depicted featuring one dominant eigenfrequency at 44Hz and several further resonance peaks at higher frequencies.

IV. PARAMETER IDENTIFICATION AND ANALYTIC CONTROLLER DESIGN

For the FRF depicted in Fig. 9 a two-mass model approximation holds very good for frequencies below $f = 200Hz$. Identification of model parameters for the structure given in (14) can therefore be performed. A numerical optimization method presented in [3] is utilized for identification. The result of this frequency response analysis is depicted in Fig. 9.

![Figure 9: Results of parameter Identification](image)

The parameters are calculated by solving (23) for the unique positive real solution $T_N$ and inserting it into (24).

In Fig. 12a and b the Bode plots of the open and the closed speed control loop are depicted resulting from the analytic design with (23) and (24) setting $a = 2$. The according plant parameters are given in Fig. 9. Fig. 11a depicts the responses of the control loop to a reference and a load step.

V. ITERATIVE OPTIMIZATION OF PI-SPEED CONTROLLER

In some cases the mechanic cannot be modelled as two-mass system because it contains no single dominant eigenfrequency. Usually it is too much effort to find a parametric model for higher order systems within a self-tuning approach. For such plants the gain and phase characteristics of the non parametric FRF of the mechanical plant can directly be utilized within the controller design process. The main benefit of this approach is the possibility to consider both, multiple resonances and the phase shift caused by delay times or additional filters.

The approach for controller design is to find parameters of a PI-speed controller ($K_p, T_N$) and of a torque reference filter (time constant $T_F$) by numeric optimization of a performance criterion $Z$, as defined in (25), including the controller specifications in frequency domain. The specifications are set by the operator in terms of the phase margins $\phi_1^*$ and $\phi_2^*$ at the lowest two frequencies $f_c1$ and $f_c2$ for those the open loop gain crosses the 0dB-axis (see Fig. 12). By these two values the damping of the control loop is specified. Furthermore $f_c1^*$ itself is set in order to specify a desired bandwidth. In detail $Z$ is composed of the weighted sum of three parts:

$$Z = k_1(\phi_1^* - \phi_1) + k_2(\phi_2^* - \phi_2) + k_3(f_c1^* - f_c1)$$

The weighting factors are initially set to $k_1 = 1$ and may be tuned after the first optimization run. If the specification(s) of

1. The damping of the mechanic has been neglected for derivation of the design equations.
one or more parameters are not met. During optimization the values \( f_{c1}, f_{c2}, \phi_1 \) and \( \phi_2 \) are detected automatically and the stability of the control loop is monitored by the Nyquist criterion. In case of an unstable control loop a penalty function is added to the performance criterion in order to keep the algorithm away from unstable parameter regions. A first set of controller parameters for initialization of the optimization algorithm is calculated by (16) by iterative reduction of \( \alpha \) until the stability limit is reached. Starting with the initial values optimal parameters are searched using a simplex optimization method. For the results depicted in Fig. 12c and d the specifications \( \phi_1^* = 45^\circ \) and \( \phi_2^* = 30^\circ \) and \( f_{c1}^* = 12 \) Hz were given which are nearly reached by the algorithm.

The step responses in Fig. 11 and the Bode plots in Fig. 12 show that despite the different approaches the two design methods give very similar results. Compared to the analytic design the numeric design features a slightly higher bandwidth at the expenses of lower damping.

VI. CONCLUSIONS

An approach was presented comprising the complete commissioning process of a PI-speed controller for drives featuring elasticity. It includes a strategy for the automatic step-wise identification of the mechanic under consideration of mechanical limit values specified by the operator (section II).

In section III a method for measurement of the Frequency Response Function (FRF) of the mechanic is presented, considering test signal design and noise signal rejection. The FRF is utilized for an analytic and a numeric controller design method presented in sections IV and V respectively.

The analytic method requires approximation of the mechanic by a two-mass model and identification of according model parameters. The numeric method requires just the non parametric FRF without any further identification and is therefore a more general method. The prize for this is that more effort has to be spent for formulation of the design specifications (which is very easy for the analytic approach) because contradictory specifications cannot always be met by the algorithm.

REFERENCES
