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TIME-BASED PARAMETER IDENTIFICATION AND CONTROLLER DESIGN FOR MOTION CONTROL SYSTEMS

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ABSTRACT

Today, a cascaded system of position loop, velocity loop and current loop is standard in industrial motion controllers. Each controller has to be designed according to its subordinated system behavior.

Usually, the controller commissioning is realized in the frequency domain with the open-loop frequency response. In contrast to that, several tuning rules in the time domain are applicable, which require a model of the plant.

The paper presents a method for the identification of plant parameters in the time domain. The approach is based on the auto relay feedback experiment by Åström/ Hägglund and a modified technique of gradual pole compensation. In addition to a theoretical description, the paper presents the implementation as an automatic application in the motion control system SIMOTION. Finally, the velocity controller is adjusted with various tuning rules. Furthermore, the identification results as well as the achievable controller performance on a test rig will be presented.

Index Terms – Identification, Parametric Models, Controller Design, Motion Control

1. INTRODUCTION

The identification of controlled systems in servo drives is an important field in controller engineering. The derived models are primarily used for controller tuning. Several tuning algorithms (e.g. symmetrical optimum) have been published, which require exact model parameters as a one main criterion to be efficient. The model order is another important criterion for the accuracy of the tuning rules [1]. According to [2] the velocity controller (PI-Structure) can be tuned based on order reduced parametric models. The tuning of the velocity controller can be even carried out for oscillatory mechanical systems, because standard velocity controller structures are not able to consider higher order models.

In addition, various limitations can be defined in servo drives based on the identified models [3]. In [4] online monitoring functions like detection of variation in the moment of inertia or friction moments have been proposed.

Nowadays, the identification of the velocity controlled system is often carried out in the frequency domain. However, in the area of low frequencies, the detection of mechanical parameters is restricted, because the measurement is performed in the closed loop. The resulting errors of the magnitude and phase response are estimated in [5].

In addition, standard identification techniques (e.g. step response) in the time domain have been developed. These methods demand high measurement accuracy and are limited, when the expected time constants are in the range of the sample time.

The intention of the paper is to establish a new identification method in the time domain, which is suitable for electrical servo drives. It is based on the relay feedback experiment and the technique of gradual pole compensation. The order reduced parametric model that is derived, will be the basis for several tuning rules, which were carried out.

The paper is divided in 5 sections. Subsequent to the introduction the identification method is discussed theoretically in chapter 2. In Chapter 3, several tuning rules will be introduced. Chapter 4 describes the experimental set-up. The results of the experiments are presented in chapter 5. The conclusions are given in chapter 6.

2. IDENTIFICATION METHOD

The relay feedback experiment by Åström and Hägglund [6] is used as an automatic excitation for various plants in the presented identification method. The results are parametric models of the closed current loop G_{CuL} and of a simplified mechanical system G_{mech} which is mainly characterized by the moment of inertia J. A simplified structure of the closed velocity loop is shown in Figure 1 including the velocity controller transfer function G_{VC} and a nonlinear relation of the friction moment M_{Fric}.

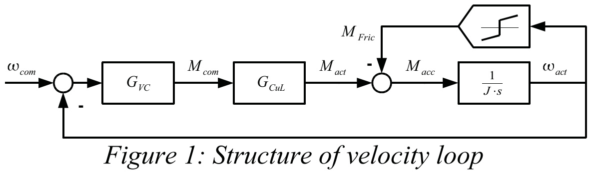

Figure 1: Structure of velocity loop

For the illustrated closed velocity loop the effective acceleration torque M_{acc} can be described by the difference of the actual drive torque M_{act} and the friction moment:

$$M_{acc} = M_{act} - M_{Fric} \tag{1}$$

In case the mechanical system is regarded as a single mass system, the angular momentum can be written as:

$$M_{acc} = J \cdot \dot{\omega}_{act} \tag{2}$$

Equation (2) can not be applied to identify the moment of inertia, because the acceleration torque M_{acc} can not be measured. Consequently, an alternative solution method is required. Therefor, the following approach is proposed.

2.1. Relay Feedback Parameterization

The velocity controller G_{VC} in Figure 1 has to be substituted with a relay with hysteresis [6], [7] represented by Equation (5). The friction moment in Equation (1) is not considered. Instead the following approach is applied:

$$M_{Fric} = f(\omega) \tag{3}$$

M_{Fric} can be considered as constant at selectable operation points for the command velocity ω_{com}:

$$M_{Fric} = f(\omega_{op}) = const. \tag{4}$$

Hence, the relay output M_{com} in Figure 2 is defined:

$$M_{com} = \begin{cases} 0 & \omega_{err} \le -\omega_{Hyst} \\ 0;\, 2 \cdot M_{Fric} & -\omega_{Hyst} \le \omega_{err} \le \omega_{Hyst} \\ 2 \cdot M_{Fric} & \omega_{Hyst} \le \omega_{err} \end{cases} \tag{5}$$

In the static case, the acceleration torque yields to:

$$M_{acc} = \pm\, M_{Fric} \tag{6}$$

2.2. Identification of the Closed Current Loop

For the closed current loop a first order lag plus time delay model (FOLPD) is identified by using the relay feedback combined with the method of gradual pole compensation, published in [8], [9]. Using this method, the model parameters are automatically adjusted according to the time behaviour of system (Equation (9)). The method is adapted for the identification of the closed current loop as shown in Figure 2.

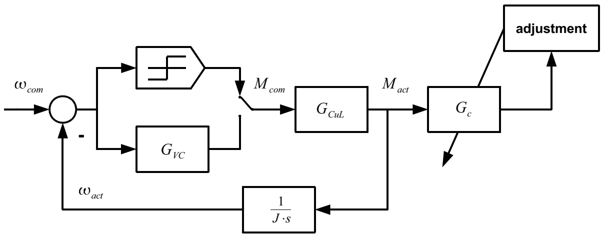

Figure 2: Scheme of gradual pole compensation

Considering the model of the closed current loop, given by [3]:

$$G_{CuL}(s) = \frac{M_{act}(s)}{M_{com}(s)} = \frac{1}{T_{cur} \cdot s + 1} \cdot e^{-sT_d} \tag{7}$$

The model parameters can be estimated by using the compensator G_c:

$$G_c(s) = \frac{X_c(s)}{M_{act}(s)} = \frac{T^*_{cur} \cdot s + 1}{s} \tag{8}$$

and the proposed adjustment strategy [8] for the gradual pole compensation (Figure 3).

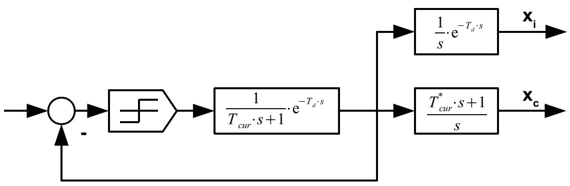

Figure 3: Criterion for compensator adjustment

The compensator time constant is adjusted according to the magnitude ratio resulting in the following equation.

$$T^*_{cur(n+1)} = T^*_{cur(n)} \cdot \frac{\hat{x}_i}{\hat{x}_c} \tag{9}$$

As Figure 3 illustrates, the performance of the method is based on the powerful criterion. Hence, the criterion has been proven to be very fast and highly efficient. This has also been approved for time constants which are smaller than the sample time of the controller. The dead time of the closed current loop (T_d) can be determined by the time behaviour of $x_c(t)$ and $M_{com}(t)$. The value of the dead time is not required in the presented identification method, as shown in Equation (14).

2.3. Identification of the Moment of Inertia

Achieving the closed current loop model, the identification structure (Figure 4) has to be changed.

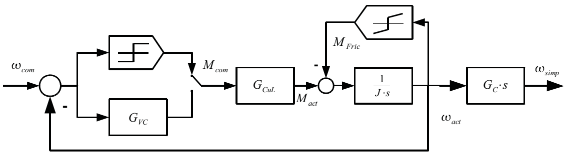

Figure 4: Identification of the moment of inertia

Based on the Equations (4-6), the relay output M_{com} can be used for calculation. The relation to the actual torque M_{act} has been established by Equation (7). The forward path of the loop becomes a first order lag plus integral plus time delay model (FOLIPD):

$$G_{CuL}(s)\cdot G_{mech}(s)=\frac{\omega_{act}(s)}{M_{com}(s)}=\frac{e^{-sT_d}}{(T_{cur}\cdot s+1)\cdot J\cdot s} \tag{10}$$

The influence of the delay on the actual velocity ω_{act} can be eliminated by using the compensator (equation (8)).

$$\omega_{simp}(s)=\omega_{act}(s)\cdot G_c(s)\cdot s=\frac{M_{com}(s)\cdot(T_{cur}^{*}\cdot s+1)}{(T_{cur}\cdot s+1)\cdot J\cdot s}\cdot e^{-sT_d} \tag{11}$$

In the case of $T_{cur}^{*}=T_{cur}$:

$$\omega_{Simp}(s)=M_{com}(s)\cdot\frac{1}{J\cdot s}\cdot e^{-sT_d} \tag{12}$$

The further derivation is carried out in the time domain. Especially, the time behaviour of ω_{simp} is of interest:

Fehler! Es ist nicht möglich, durch die Bearbeitung von Feldfunktionen Objekte zu erstellen. *(13)*

The dead time does not have to be considered, because only the magnitude ratio is significant.

$$\omega_{simp}(t)=\frac{1}{J}\cdot\int M_{com}\cdot dt \tag{14}$$

The closed loop with relay controller (Figure 4) achieves oscillation with the time period (T_{Per}). The oscillation at the operation point can be expressed as a sum for the sampled system.

$$\omega_{simp}(t)=\frac{1}{J}\cdot\left[\sum_{t=0}^{\frac{T_{Per}}{2}}M_{com}\cdot t-\sum_{t=\frac{T_{Per}}{2}}^{T_{Per}}M_{com}\cdot t\right] \tag{15}$$

The structure, shown in Figure 3, is used for the calculation of the moment of inertia. A half-cycle is sufficient for the magnitude of ω_{simp}. Finally, the resulting formula is:

$$J=\frac{M_{com}\cdot T_{Per}}{2\cdot\hat{\omega}_{simp}} \tag{16}$$

3. TUNING RULES

According to [10], "the most direct way to set up controller parameters is the use of tuning rules". In this book, various tuning rules for a wide range of parametric models are put together. Some of them will be introduced as follows. The choice is reduced to PI controller rules for parametric model. The structure of the controller is:

$$G_{VC}(s)=K_P\cdot\left(1+\frac{1}{T_N\cdot s}\right) \tag{17}$$

Table 1: PI controller tuning rules for FOLIPD model

Rule	K_P	T_N
Shinskey I	$\frac{0.556\cdot J}{(T_d+T_{cur})}$	$3.7\cdot(T_d+T_{cur})$
Shinskey II	$\frac{0.952\cdot J}{(T_d+T_{cur})}$	$4\cdot(T_d+T_{cur})$
McMillan	1	$3.33T_d\left(1+\left(\frac{T_{cur}}{T_d}\right)^{0.65}\right)$
Poulin	2	$x_1\cdot(T_d+T_{cur})$
Sym. Optimum [3]	$\frac{J}{2\cdot(T_d+T_{cur})}$	$4\cdot(T_d+T_{cur})$
Samal	$\frac{\pi}{4}\cdot\frac{J}{(T_d+T_{cur})}$	$3.3\cdot(T_d+T_{cur})$

The extracted tuning rules differ in their suitability for electrical servo drives and will be benchmarked with various criteria in chapter 5.

4. EXPERIMENTAL SET-UP

The presented approach has been verified on an experimental rig, as shown in Figure 5. It is equipped with the SIEMENS motion controller SIMOTION

1 $J\cdot\frac{T_{cur}}{T_d^2}\cdot\left\{\frac{1.477^2}{1+\left(\frac{T_{cur}}{T_d}\right)^{0.65}}\right\}^2$

2 $\frac{J\cdot x_2}{T_d+T_{cur}}\cdot\sqrt{\frac{T_{cur}^2}{x_1\cdot(T_d+T_{cur})^2}+1}$ (x_1, x_2 from [10])

D445 and SINAMICS drives. The motion controller is sampled with 500 µs and the drive components with 125 µs. There are two mechanical configurations (System 1 & System 2).

The experimental set-up contains of a two-mass-system and a three-mass-system, whereas the third mass can be connected by a clutch. The basic parameters are the moment of inertia J, the resonance frequency f_0 and antiresonance frequency f_N. The preset values of the parameters for the experimental set-up are shown in Table 2

Table 2: Characterization of the test rig

Configuration	J [kgmm²]	f_0 [Hz]	f_N [Hz]
Two-mass-system (System 1)	1355	422	333
Three-mass-system (System 2)	2763	184 411	106 350

For the application of the tuning rules and the monitoring functions, which are the aim of the identification, order-reduced parametric models are sufficient. Consequently, it is not necessary to take a two or three mass system as basis for the mechanical system. A single-mass-system satisfies the requirements for the identification. Therefor, only moment of inertia (Equation (16)) is of interest for the calculation.

5. EXPERIMENTAL RESULTS

5.1. Identification results

The closed current loop (Equation (7)) under relay feedback and the identified model are shown in the following time plot. The identified model has been calculated in the sample time of the motion controller (500 µs). Even for a lag time which is smaller than the sample time, the reaction curves of the real values and the modelled values are nearly identical. Hence, the performance of the chosen adjustment strategy is proven (Figure 5).

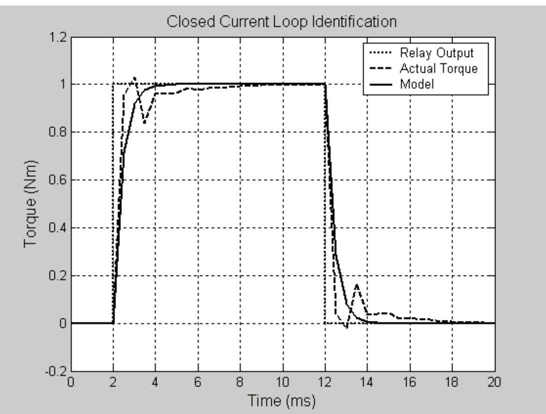

Figure 5: Time behavior of closed curred loop

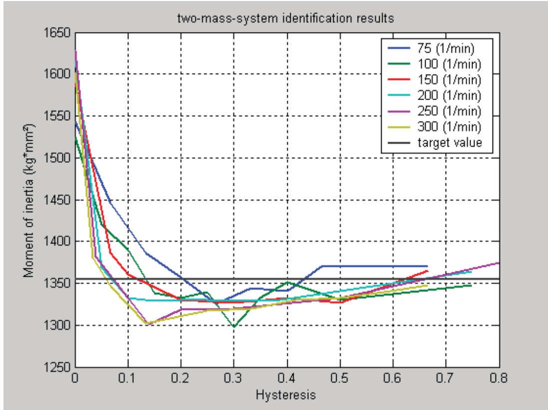

Figure 6: Results for the moment of inertia

According to Equation (5), the hysteresis of the relay is a free selectable parameter. Therefor, a compromise between the magnitude of the relay oscillation and the linearization error for the friction moment (Equation (4)) has to be found. The moment of inertia for the two-mass-system at different operation points is plotted against the hysteresis (Figure 6). The hysteresis is displayed in percent of the velocity.

The graph demonstrates that the value of the moment of inertia for the two-mass-system can be identified with sufficient accuracy. A variance of less than 4% can be achieved over the whole range of the chosen hysteresis. Comparing the achieved moments of inertia to other investigations ([11], [12]), the experiments have shown an improvement of the accuracy. Simultaneously, a small value of the velocity (operation point) is sufficient.

Consequently the FOLIPD model (Equation (10)) can be written with the identified parameters.

Table 3: Identified parameters for controller design

Model $G_m(s)$	J [kgmm²]	T_d [ms]	T_{cur} [ms]
$\frac{e^{-sT_d}}{J \cdot (T_{cur} \cdot s + 1) \cdot s}$	1340	0.25	0.4

5.2. Controller design

Using the parametric model parameters and the tuning rules for PI controller (Table 1) the achievable phase margin and gain margin can be calculated.

The typical phase margin for set point response is a range of 70°…40° and for disturbance response a range of 50°…20° [3]. As it is recognizable in Figure 7, all listed tuning rules for FOLIPD models can be classified as proper for a good disturbance response.

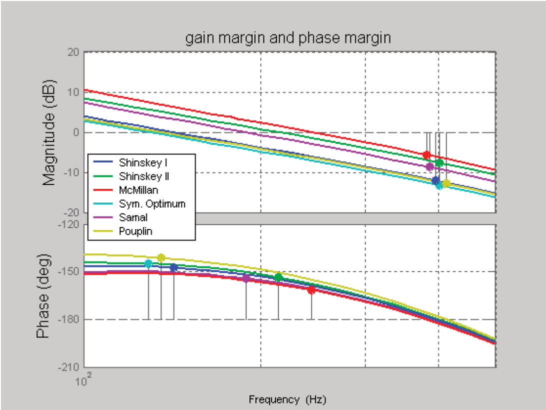

Figure 7: Gain and phase margin

The results of the Symmetrical Optimum, Pouplin and Shinskey I are nearly identical. Hence, only the Symmetrical Optimum will be displayed in the following step responses. To show the variety of the other tuning rules, Samal and McMillan will be demonstrated as well. The behavior on disturbance steps is shown in the following plot. In addition the internal tuning rule of the drive system is listed as "automatic".

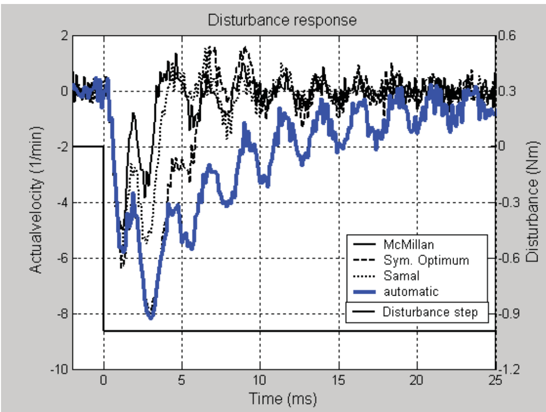

Figure 8: Disturbance response

The introduced tuning rules show a better disturbance response, compared to the automatic tuning of the drive system. As expected from the open loop stability calculation (Figure 7), McMillan and Samal have the smallest settling times.

The validation of the setpoint response is divided in two time plots. In a first step the original structure of the PI-controller is used.

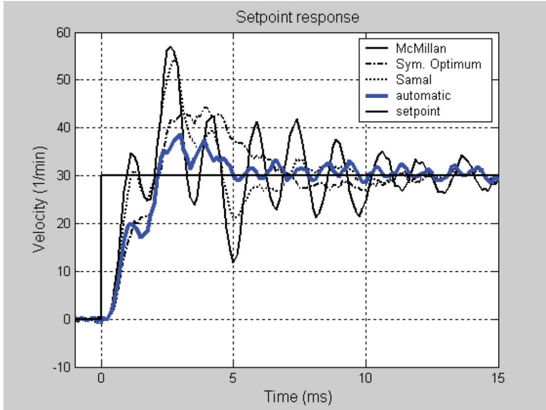

Figure 9: Setpoint response

As known from the Symmetrical Optimum, all setpoint responses show an overshoot up to 80%. Consequently an additional setpoint filter has to be used in the command value branch. The filter is described by the following equation:

$$G_{Filter}(s) = \frac{1}{T_F \cdot s + 1} \qquad (T_F = T_N) \qquad (18)$$

The filter parameter was set equal to T_N for all experiments.

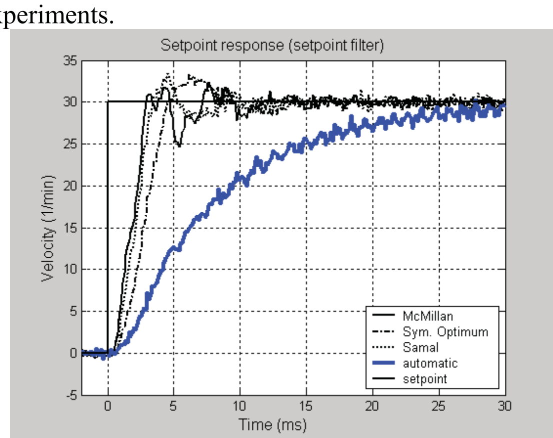

Figure 10: Setpoint response with velocity filter

By using the proposed filter, the overshoot is reduced to a range of 0-10%. The settling time for all approaches is about 10ms. For the automatic tuning algorithm, another tuning rule for the filter has to be found.

6. CONCLUSIONS

In this paper a new identification method of parametric models for velocity loop parameters in the time domain has been presented. As an excitation, the auto relay feedback experiment has been used and has been combined with the method of gradual pole compensation. The model parameters are identified by applying a criterion, which compares the magnitudes of two signals. A high accuracy of the model parameters has been achieved. The advantages of the approach are a less a priori knowledge, the possibility of a simultaneous identification of various parameters and a low excitation of the mechanical system.

The presented algorithm has successfully been implemented as an automatic tool in the motion control system SIMOTION. Based on the identification results the PI velocity controller is designed by using various tuning rules. The achievable results for setpoint and disturbance responses have been compared.

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