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SIMULATION BASED OPTIMIZATION FOR CONTROLLER PARAMETERIZATION OF MACHINE TOOL AXES – ADVANCED APPLICATION

The quality of the controller parameterization of a drive system has direct influence on the obtained performance. Nevertheless, the commissioning is mainly done today by the application of basic tuning rules or not comprehensible automatisms. This often leads to not optimal performance. An alternative approach is to parameterize the typical controller cascade, which is widely used in standard industrial controllers, in one step by using the so-called "simulation-based optimization" (SBO). Significant advantages are the opportunity of defining specific restrictions concerning the desired controller parameterization and the possibility for directly processing non-linear systems without approximations. Furthermore, friction, additional filters in the cascade or controller structure extensions like the Advanced Position Control (APC) can be considered. Therefore, in the present paper, the application and the results of the SBO for different drive systems will be presented. After an introduction, the paper describes the basic principles of the simulation based optimization including the application for controller parameterization. Subsequently, the results for indirect velocity control of a test rig are discussed. After that, the findings for the commissioning of a direct position controller of an industrial servo press are presented. The paper closes with a summary and an outlook.

1. INTRODUCTION

In former research it has already been shown, that the application of simulation-based optimization (SBO) for controller parameterization offers various advantages over conventional approaches (e.g. the use of standard tuning rules like the symmetric optimum) [1],[2],[12]. This includes for example the possibility of considering non-linear systems accurately and the ability of an easy definition of optimization criteria and various constraints [5],[6],[7].

In the previous research the application of the SBO was mostly demonstrated on theoretical test systems or on relatively comprehensible mechanical systems [7]. To further

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investigate the applicability of the proposed method, it was also tested on a more complex test rig and also on a real industrial servo press.

The paper has the following structure: In chapter 0 the basics of the simulation based optimization are described. The working principle of the automatic modelling as well as the integration in the simulation structure are content of chapter 0. In chapter 0 the used test systems are described and the obtained optimization results are presented and discussed. The paper closes by summing up the results and giving an outlook in chapter 0.

2. BASICS OF SIMULATION-BASED OPTIMIZATION

SBO is a process of finding the global extreme of an objective function in a defined search space. Therefore, the simulation of the investigated system is equipped with a fitness evaluation function and is then connected to the optimizer itself. The so called fitness value is calculated by the evaluator and consists of the assigned values of the optimization criteria and the penalty values for violating constraints [3].

The optimization done in this paper uses a hybrid optimizer, a combination of the Particle-Swarm-Optimization (PSO) and the Nelder-Mead optimization (NM), to minimise the fitness value of the Simulink® model illustrated in chapter 3. Generally, the PSO is a so called metaheuristic [4], because it makes no or only a few assumptions on the investigated optimization problem. The basic principle of the PSO is the simulation of the movement of a swarm of objects (e.g. birds or fish) by trying to find the best solution. The NM algorithm (or simplex method), which was originally presented in [9], uses a geometric structure, the simplex, with n +1 points in the search space with the dimension n. So for example if n = 2 then the simplex is a triangle. By modifying the simplex according to defined rules the algorithm tries to converge the structure to the global minimum. By combining the PSO with the NM a significant performance improvement could be obtained [10].

In general, in the first step the optimizer initialises a proposed solution and sends it to the simulator, which calculates the respective fitness value and sends this back to the optimizer. It responds by sending a new, likely better solution proposal back. This cycle continues for a defined amount of runs or until the target value is reached (Fig. 1). Because of the limited space, for the description of the detailed working principle of the hybrid optimizer it is referenced to [10].

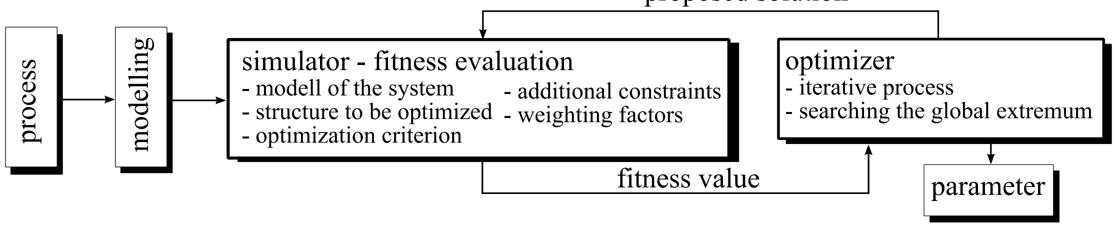

Fig. 1. Schematic loop of optimizer and simulator to find controller parameters

3. AUTOMATIC MODELLING AND USED SIMULATION STRUCTURE

As already described in the previous chapter the simulator is, with its included model of the investigated system, an essential part of the SBO. Only with an adequate representation suitable optimization results are possible. Because of the already mentioned advantages of the SBO to support also complex models with e.g. high orders or non-linear behaviour an automatic model creation for electromechanical systems was developed. Due to the limited space and the different focus of the paper only the basic function principle could be explained.

Origin is the recorded frequency response of the axis which is mainly done by the integrated measurement functions in the controller hardware. The measured frequency is first used to determine the moment of inertia J_s of the system. This is done by calculating the gradient of the measured frequency response at lower frequencies [8]. In the second step the frequency response is adjusted by removing the effects of the J_s. After that, the frequency response is separated into so called partial oscillators (PO) as shown in Fig. 2 and 3 [9],[11]. This is achieved by determining the pairs of antiresonant frequencies ω_f and the resonant frequencies ω_r for indirect controlled systems (Fig. 2) and only the ω_r for direct controlled systems (Fig. 3).

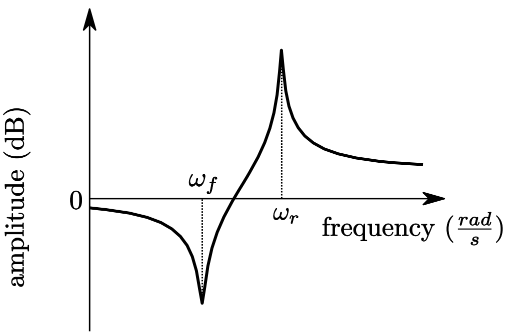

Fig. 2. Partial oscillator for indirect controlled systems

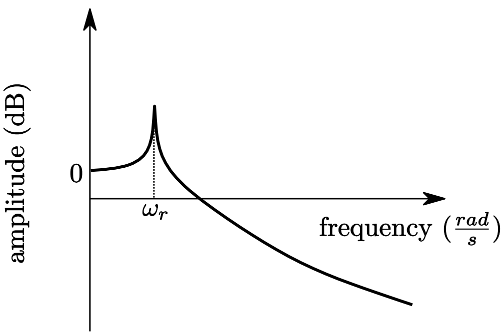

Fig. 3. Partial oscillator for direct controlled systems

The PO's could be mathematical described by the equation (1) [11]. For indirect controlled systems a is 1 and for direct controlled systems a is 0. Depending on the shape of the measured frequency response the number of used PO's is automatically determined. In a first step the damping's d_f and d_r are initialized with a default value of 0.01 (named: model, auto). In a second step the SBO is used to reduce the absolute integral between the measured frequency response without the influence of J_s and the model with the default dampings by optimizing the dampings (named: model, auto opt.).

$$G_s(s) = \frac{a*\frac{1}{\omega_f}*s^2 + \frac{2d_f}{\omega_f*s} + 1}{\frac{1}{\omega_r}*s^2 + \frac{2d_r}{\omega_r}*s + 1} \tag{1}$$

By combining the individual transfer functions of the POs and the influence of J_s a linear model of the investigated system is automatically obtained. This transfer function or system model is then integrated into a complete control loop for e.g. a velocity control as shown in Fig. 4. By including for example friction (static, dynamic) or limitations (e.g. maximum current) into the model the system becomes nonlinear. This control loop was designed in MATLAB® Simulink®, operating in a step time of 125 µs to align the simulation to the later used test rig in chapter 0.

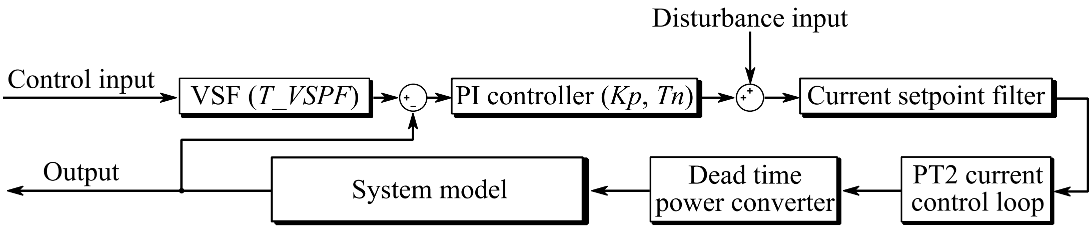

Fig. 4. Modelled control loop with filter, controller and controlled system

The simulation was accomplished as well as in the time domain as also in the frequency domain as shown in Fig. 5 [7]. Thereby, the discrete fitness values of three individual control loops (frequency domain in open and closed loop, time domain) are calculated and evaluated considering also the respective penalties for violating constraints simultaneously. Closed loop variants are further subdivided into a setpoint response and a disturbance response each. The fitness values of the sub-model are multiplied with an assigned weighting factor to ensure a balanced impact on the results and subsequently cumulated to compose the global fitness value. In the frequency domain, a pseudorandom bit stream (PRBS) signal is used as input to the simulation, to determine the frequency responses [13]. This paper uses a 13-bit-signal, alternating from -1 to +1, providing 8192 discrete states. Therewith a frequency range up to 2000 Hz can be investigated.

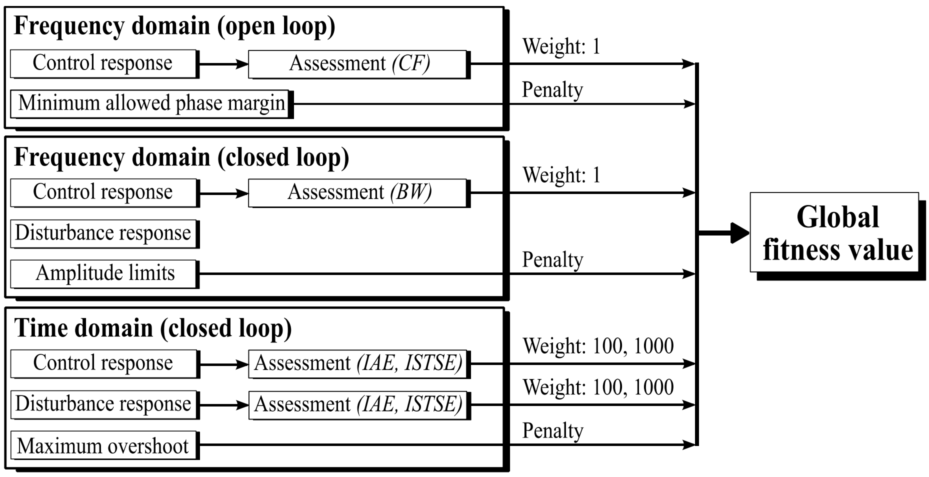

Fig. 5. Simulation model for the calculation of the fitness value [7] (CF = crossover frequency, BW = bandwidth, IAE = integral of absolut error, ISTSE = integral of squared time multiplied by squared error)

4. TEST SYSTEM DESCRIPTION AND OPTIMIZATION RESULTS

The proposed method of tuning a complete cascaded control loop by the usage of the SBO should be presented on two different applications. In chapter 4.1 this is done by parameterizing the velocity controller, the velocity setpoint filters (VSPF) and the current setpoint filters (CSPF). The parameterization of the position controller, the feed forward, the velocity controller and the VSPF of an industrial servo press is presented in chapter 4.2.

4.1. TEST RIG

The first application of the SBO was to parameterize the velocity controller of the test rig shown in Fig. 6. It is equipped with a SIEMENS motion controller SIMOTION D445® and SINAMICS® drives. The motion controller is sampled with 500 μs and the drive components with 125 μs [5].

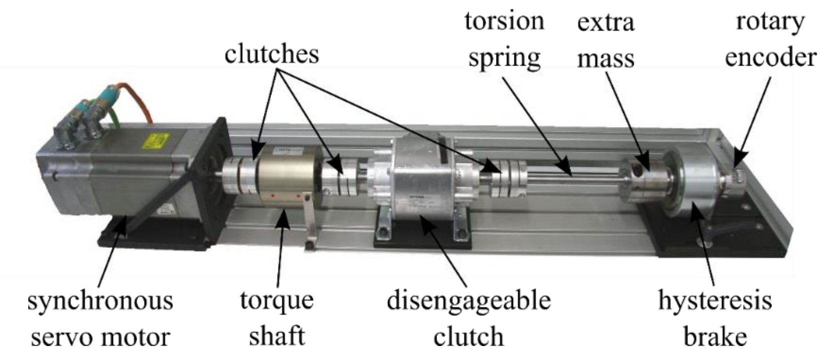

Fig. 6. Used test rig with known mechnical parameters

By closing the disengageable clutch the system characteristic is a three mass system. For forming the core of the simulation, this experimental set-up has been modelled and optimized according to the described automatism in chapter 3 resulting in a model with two POs. The identified parameters are listed in Table 1.

Table 1. Identified and optimized parameter for the indirect velocity model of the test rig

Parameter		Identified and optimized	Parameter		Identified and optimized
PO1	ω_f	94.38 Hz	PO2	ω_f	335.58 Hz
	d_f	0.1900		d_f	0.0574
	ω_r	164.80 Hz		ω_r	405.77 Hz
	d_r	0.2107		d_r	0.0347
	a	1		a	1

To demonstrate the quality of the automatic determined model, the measured frequency response of the test system and the two stages of the system models are presented in Fig. 7. The expected behaviour of the three mass systems is comprehensible. It can be stated, that the model with the optimized dampings (model, auto opt.) has a very good accordance to the measured system.

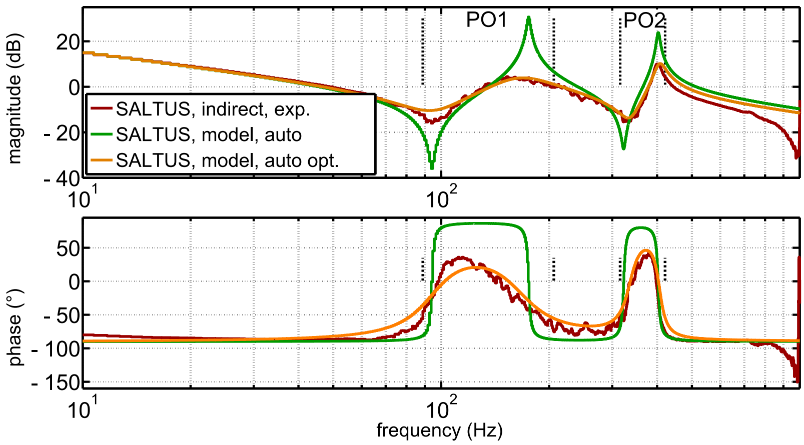

Fig. 7. Measured frequency response (velocity loop) and the two stages of modelleling of the test rig

The goal of the optimization was to parameterize the velocity control loop including the velocity setpoint filters (VSPF) and the current setpoint filters (CSPF) step by step (Table 2). In run No.1 only K_p, T_n and T_{VSPF} were optimized. For the CSPF the predefined configuration in the controller was used. Then in run No. 2 und 3 also the dampings of the CPFS were included into the optimization. Run No. 4 represents the settings obtained from the automatic tuning algorithm included into the controller. This automatism is not able to configure the VSPF and therefore it is not activated.

Table 2. Identified and optimized parameter for the indirect velocity model of the test rig (optimized values are red)

No.	Velocity controller		Velocity setpoint filter (VSPF)	Current setpoint filters (CSPF)				Remarks
	K_P $\left(\frac{Nms}{rad}\right)$	T_n (s)	T_{VSPF} (s)	ω_{FCN} (Hz)	D_{FCN}	ω_{FCD} (Hz)	D_{FCD}	
1	0.3000	0.01383	0.000125	164.80	0.0	164.80	0.25	3 CSPF active
				405.77	0.0	405.77	0.25	
2	0.4586	0.02484	0.000125	164.80	0.357	164.80	1.0	3 CSPF active, 1st damping's optimized
				405.77	0.0	405.77	0.25	
3	1.2500	0.02213	0.000125	164.80	0.153	164.80	0.372	3 CSPF active, 1st and 2nd damping's optimized
				405.77	0.084	405.77	1.0	
4	1.335	0.00170	not used	171.86	0.0	171.86	0.25	3 CSPF active, setting from automatic tuning of the controller
				406.25	0.0	406.25	0.25	

In Fig. 8 and 9 the corresponding step responses (simulation and measurement) are presented. It could be stated that there is a very good match between the simulation and the experimental results for runs No. 1 and 2. By optimizing the dampings of the first CSPF the raise time and the overshoot could be reduced by comparable settling time.

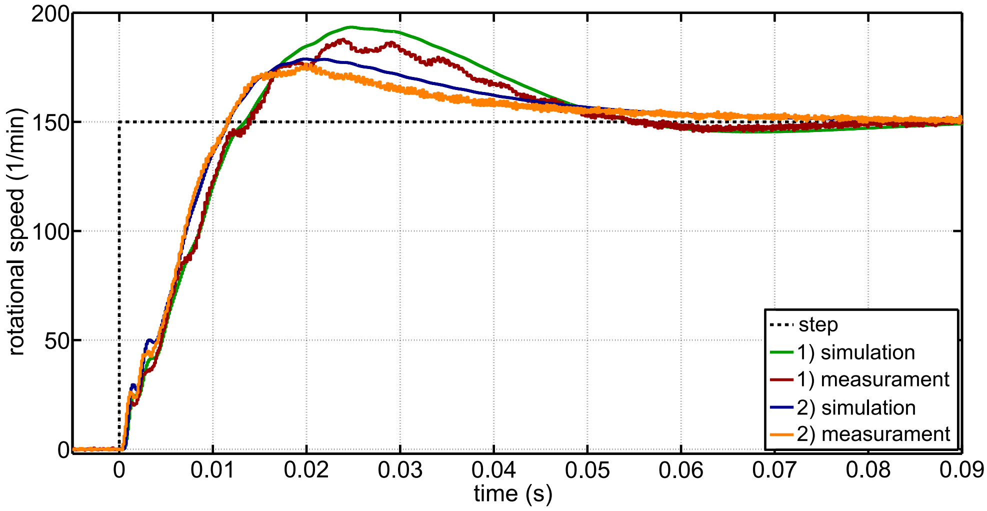

Fig. 8. Step responses (measurement and simulation) of the velocity loop for settings 1) and 2)

When runs No. 3 and 4 are compared, it is recognizable that the simulation also represent the behaviour of test rig in a sufficient way. Furthermore, the raise and the settling time are even smaller and setting No. 3 and 4 are showing the same dynamics.

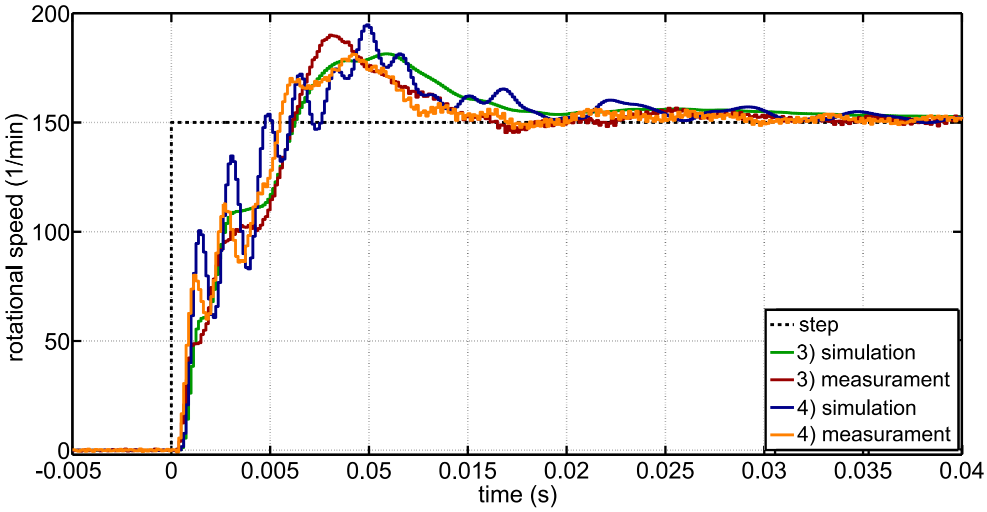

Fig. 9. Step responses (measurement and simulation) of the velocity loop for settings 3) and 4)

Thus, by only comparing the time behaviour the differences between the optimized and the obtained one from the automatic tuning of the controller, are not obvious. Because of that we must also compare the frequency responses as shown in Fig. 10 and 11. It could be noted, that the simulation represent the characteristics of the test rig in an acceptable way, but there are also some differences recognisable. However, in all cases the simulation estimates the system behaviour as an upper border and therefore the stability is guaranteed. As already mentioned runs No. 3 and 4 show comparable dynamics in the time domain but in the frequency domain the setting for run No. 3 has a higher stability margin. Especially the better damping between 400 Hz and 800 Hz is a great advantage.

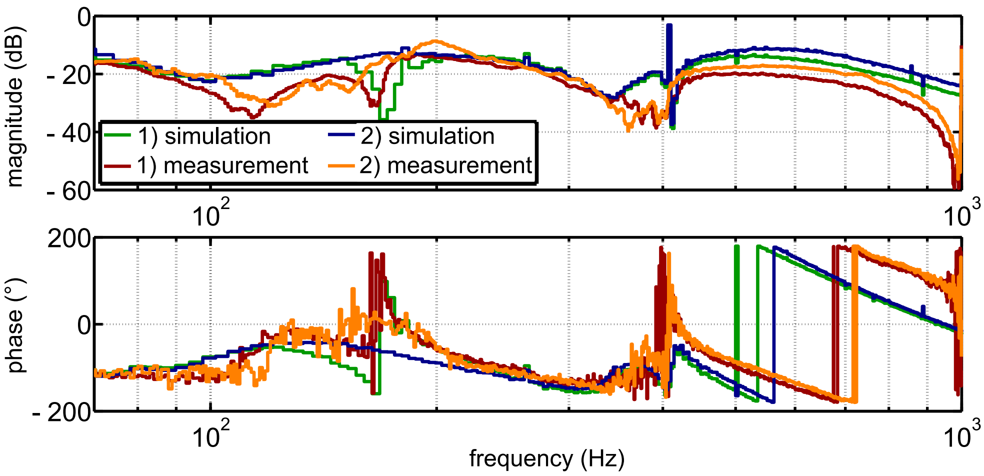

Fig. 10. Measured and simulated frequency responses of the velocity loop for settings 1) and 2) of the test rig

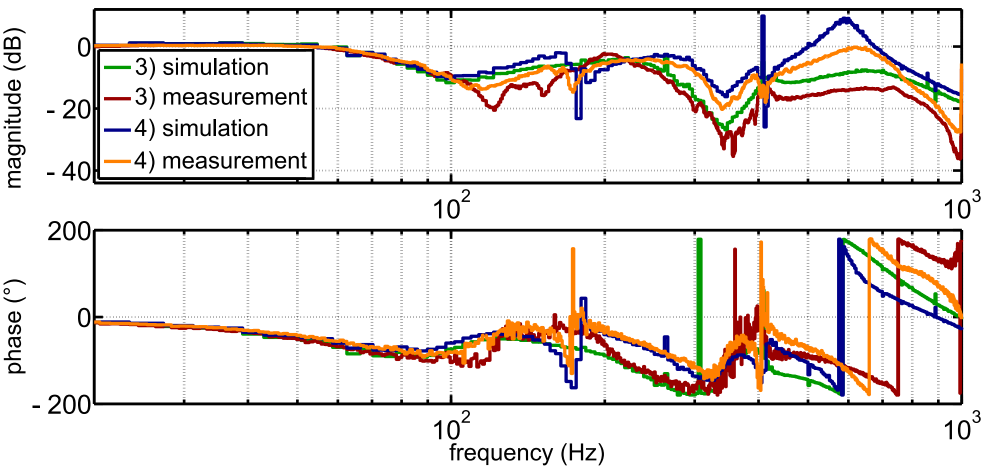

Fig. 11. Measured and simulated frequency responses of the velocity loop for settings 3) and 4) of the test rig

4.2. SERVO PRESS

The second application of the SBO is the parameterization of the position controller, the feed forward, the velocity controller and the VSPF of an industrial servo press as illustrated in Figure 12. It is equipped with a SIEMENS motion controller SIMOTION

D445-2® and SINAMICS® drives. The both servo motors can generate a tapped force of about 800 kN with a velocity of 50 mm/s. Furthermore, it is equipped with four linear position measuring systems each at the four guides of the slides of the press. On basis of the both fronts measuring systems the position of the slide is directly controlled.

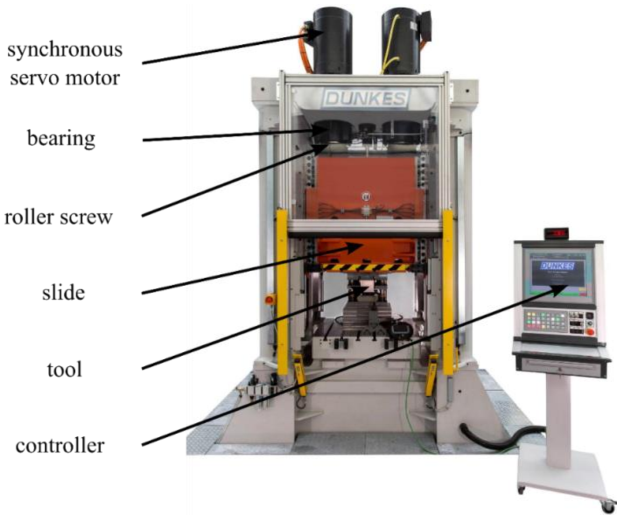

Fig. 12. Components of the investigated servo press

The measurement of the frequency response of the drives was performed with the slide mounted to the press and therefore the drives were mechanically connected. To be able to record the systems frequency response a pseudorandom bit stream (PRBS) signal is used as input for the drives. Thereby, it must be guaranteed that the signal is send to both drives at the same time. As optimization criteria in the time domain the absolute integral between the setpoint value and the actual value of the position was used. The goal was to reduce this value to a minimum.

In Table 3 the first two of eight identified and optimized POs are presented. In Fig. 13 the corresponding frequency responses of the measurement are illustrated. It could be noted, that the automatic generated model represents the behavior of the press in the amplitude as well as in the phase plot in a sufficient way.

Table 3. Identified and optimized parameter for the direct position model of servo press (only the first two partial oscillators)

Parameter		Identified and optimized	Parameter		Identified and optimized
PO1	ω_f	67.96 Hz	PO2	ω_f	183.36 Hz
	d_f	0.1362		d_f	0.1382
	ω_r	96.05 Hz		ω_r	208.72 Hz
	d_r	0.0490		d_r	0.0323
	a	0		a	0

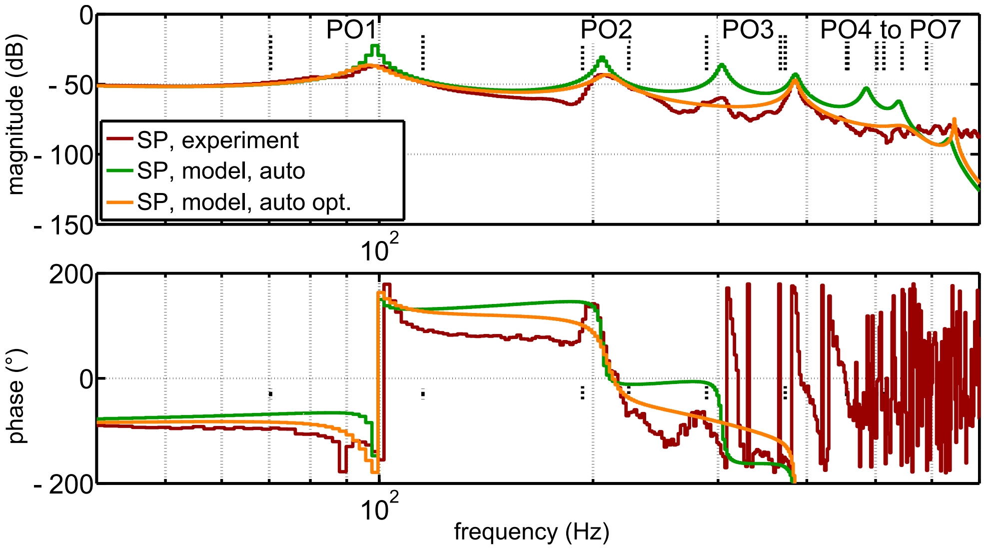

Fig. 13. Frequency response of the servo press (position) and the two stages of model identification

The results of the optimization of the position controller, the feed forward, the velocity controller and the VSPF are shown in Table 4 as run No. 6. To be able to compare the results also the settings determined from the commissioning are listed as run No. 5. By comparing No. 5 with 6 it could be noted, that the optimized setting is more conservative because the gain of the position and the velocity controller are smaller. Furthermore, also T_n is five times larger than the original setting.

Table 4. Identified and optimized parameter for the direct position model of the servo press (optimized values are red)

No.	Position controller	Feed forward	Velocity controller		Velocity setpoint filter (VSPF)	Remarks
	$K_V\ \left(\frac{1}{s}\right)$	$T_{sym}\ (s)$	$K_P\ \left(\frac{Nms}{rad}\right)$	T_n (s)	T_{VSPF} (s)	
5	100	0.002	650	0.0050	--	setting determined from the start-up engineer
6	54.46	0.000125	638.12	0.02636	0.000125	

Nevertheless, by comparing the time behaviour of the position of the slide in Fig. 14 and 15 the differences are not that significant. It could be noted, that due to the selected optimization criterion in the time domain the contouring error was reduced but resulted in a higher overshoot in the lower position of the slide.

Because of some restrictions during the experiments at the SSP the frequency responses could not be recorded. By comparing the simulated responses it could be concluded, that the setting No. 6 has significant higher stability margin than setting No. 5.

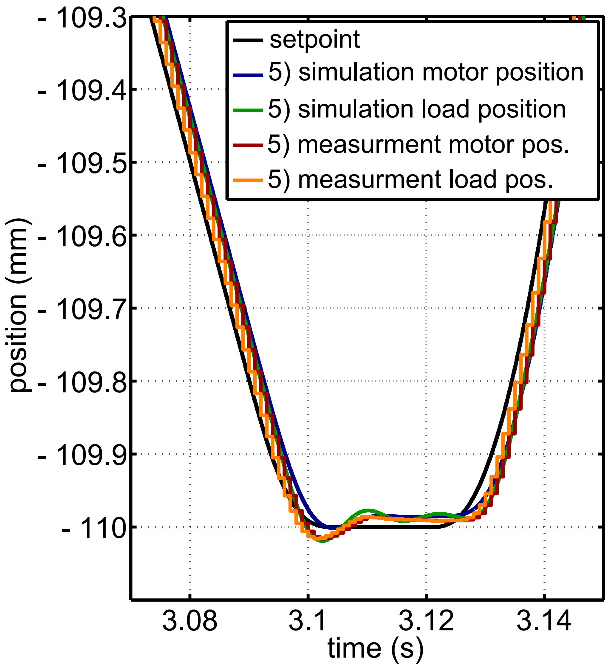

Fig. 14. Comparison of the slide position on basis of the motor or the direct position sensors for setting 5)

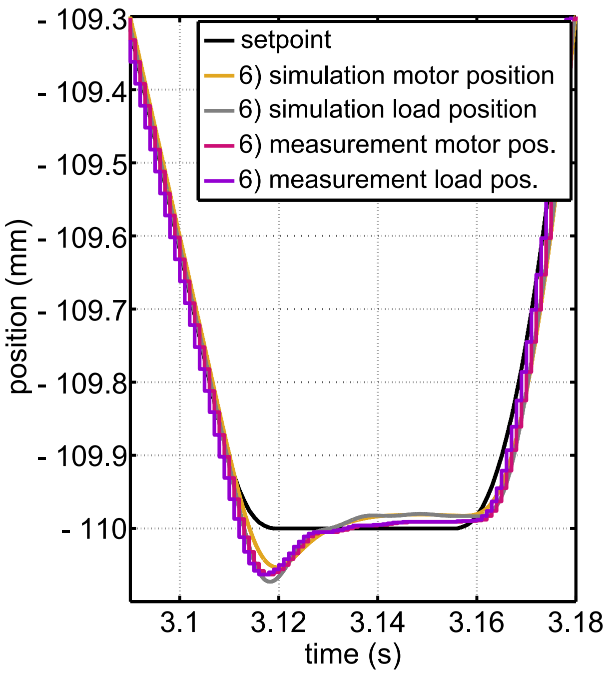

Fig. 15. Comparison of the slide position on basis of the motor or the direct position sensors for setting 6)

5. CONCLUSION

The SBO is a promising approach to parameterize controller cascades in standard industrial controllers. It allows the automatic tuning of complex controller structures and non-linear systems in one step. By simulating the time and frequency domain in parallel good system dynamics as well as sufficient stability margins could be defined and fulfilled.

In this paper the application of the SBO for controller parameterization was demonstrated on two different applications. The first one deals with the adjustment of the velocity controller of a test rig configured as a three mass system. The results showed, that the module of automatic model generation based on the measured frequency response delivered a good model accuracy. Furthermore, the optimized controller parameters and filter settings resulted in a satisfying system behaviour. By including the dampings of the CSPF into the optimization the same system dynamics could be obtained as the settings from the integrated auto-tuning methods showed but with higher stability margins.

It could be summarized that the proposed procedure of automatic model acquisition and controller parameterization by using the SBO is very promising. It allows even unexperienced users to commission complex systems satisfactorily. Beyond, the necessary time for commissioning could be reduced significantly. For example the complete start-up of the SSP could be done in half a day.

Still, future research is necessary to improve the individual modules of the proposed method for different kinds of application. Perhaps, it could be used as a base for developing new tuning rules.

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