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Data-based control of motion systems

Maarten Steinbuch, Jeroen van Helvoort, Wouter Aangenent, Bram de Jager and René van de Molengraft

Abstract—This paper presents recent research results for feedback control design of motion systems. Two model-free approaches are investigated, that exploit the ease of experimentation which is typical for motion systems. One approach is data-based design of a linear feedback controller which realizes desired closed-loop sensitivity and complementary sensitivity transfer functions. These transfer functions are specified via a data-based performance cost. The designer can prescribe both the controller structure and the complexity. Experimental results obtained in a direct-drive robot motion control problem confirm the effectiveness of the design. A second line of research is unfalsified control where a set of controllers is iteratively tested against measured data. Experimental results for the well-known fourth order benchmark motion system show feasibility of the approach. Finally, we implemented a nonlinear SPAN filter on the same system, which outperforms a linear feedback design.

I. INTRODUCTION

Modern high tech mechatronic devices, such as for example robotic manipulators and storage drives, constitute a major economic value. Increasing performance demands motivate a thorough analysis of limitations in the design. In this study we will address the feedback control part of the design of motion systems.

Typical for motion systems, is the relative ease of doing experiments. The performance-limiting issues are primarily due to causality. Plant models are relatively easy to acquire with high accuracy, due to the experimental conditions mentioned, and feedforward is always added for servo tasks. The primary role of the feedback is to suppress disturbances. Since most motion systems behave approximately as linear - certainly under closed-loop conditions - and constraints on signals are normally taken into account a-priori while designing the servo task, i.e. they are not a relevant issue for feedback design, controllers are typically also linear. Causality implies that with respect to disturbance suppression, feedback is always late (i.e. phase delay especially at high frequencies). Therefore, high gain controllers (learning, repetitive) are used whenever possible, for instance to suppress periodic disturbances. The Bode Sensitivity integral expresses the causality in the form of reduction of low-frequent disturbances, at the cost of amplification of (typically) high frequent signals (Waterbed effect), e.g. measurement noise. It is this fundamental limitation which is the driving force for several lines of research: (i) further exploring feedforward, including iterative learning control, (ii) disturbance- and data-based control, i.e., using the internal model principle and also the

principle of 'machine-in-the-loop' for adjusting the controller parameters on the basis of on-line measurements, and finally, (iii) non-linear control of linear motion systems. In this paper we will show some of the recent research results with respect to (ii) and (iii).

The *data-based* (DB) control field addresses techniques for control design without explicitly making use of parametric models, but merely based on measured signals. Here we refer to a few DB methods: DB LQG control (Skelton and Shi, 1994; Aangenent et al., 2005), unfalsified control (Safonov and Tsao, 1997), simultaneous perturbation stochastic approximation (Spall and Cristion, 1998), iterative feedback tuning (Hjalmarsson, 1999), disturbance-based control (Steinbuch and Norg, 1998), and virtual reference feedback tuning (Campi, Lecchini and Savaresi, 2002). In our first study on DB control (Kostić, De Jager and Steinbuch, 2004a), the motivations for the DB approach are simplified off-line design of high-performance motion controllers, and the direct supervision over the controller structure and its complexity. Here, we investigate if the control performance feasible with model-based motion controllers can also be realized with data-based controllers. The requirement is that the DB method must allow prescribing the controller structure and the complexity at the start of the feedback design. Based on the virtual reference feedback tuning approach (Campi, Lecchini and Savaresi, 2002), we derive a DB method for controller design, which enables simultaneous shaping of the closed-loop sensitivity and the complementary sensitivity transfer functions. Its practical merits will be illustrated with experimental results obtained on a benchmark direct-drive robotic system, see Section II. In Section III an alternative is shown, known as unfalsified control (Safonov and Tsao, 1997), where a set of feasible controllers is tested against data. We are proud to show for the first time experimental implementation results with this technique (Van Helvoort, De Jager and Steinbuch, 2005). For this we use the well-known fourth-order motion system. The same system is used in Section IV to show results with nonlinear control of motion systems, using the SPAN ('Split-Path Nonlinear') filter. We now have experimental evidence how to 'circumvent' Bode's gain/phase relation. We will finish with conclusions in Section V.

II. VIRTUAL REFERENCE FEEDBACK TUNING

A. Method

The system shown in Fig. 1 illustrates a standard feedback system, with controller C and plant P. Here we assume a SISO one degree-of-freedom structure, although this is not essential for the method we will develop. For simplicity of

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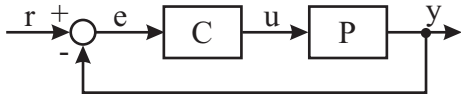


Fig. 1. Feedback system

notation we will omit the Laplace argument s . The desired control performance is specified via the desired closed-loop sensitivity S_o and complementary sensitivity functions T_o , by which the designer specifies the desired dynamics of the closed-loop system, e.g., a minimum bandwidth requirement, integral control, and a level of error reduction and specification of the maximum closed-loop bandwidth for achieving robustness against resonances and noise at higher frequencies. For design purposes it is not strictly necessary that S_o and T_o are complementary.

The objective is to design a stabilizing controller C which closely realizes S_o and T_o . As objective function we now define:

$$J(\theta) = |T_o - C(\theta)S_oP| \quad (1)$$

where the symbol θ denotes the vector of controller parameters to be designed. In order to minimize this objective function we need the plant model P . However, if we apply Eq. (1) to the measurable input data $u(t)$, and by using $y = Pu$, we obtain the data-based objective function:

$$J(\theta) = \sum_t T_o u - C(\theta)S_o y \quad (2)$$

where the sum is taken over the data sequence of inputs $u(t)$ and outputs $y(t)$. The operators in Eq. (2) are taken as discrete-time filter operators. For more details we refer to (Kostic, de Jager and Steinbuch, 2004a).

B. Results

The direct-drive robot with three revolute joints (RRR robot), shown in Fig. 2, is the subject of our case study. We refer to (Kostić, De Jager and Steinbuch, 2004b) for the kinematic and dynamic models. Due to direct-drive actuation, the robot dynamics is highly nonlinear and coupled, which impedes motion control of high performance. Their effects are reduced via a nonlinear compensation based on the robot rigid-body dynamic model. The robot dynamics that are not covered by this compensation are handled by feedback controllers, which we typically design using H_∞ control theory. For this paper, the feedback controllers were designed using the DB method presented in the previous section. Here, we will illustrate the DB feedback design for the 1st robot joint only. The designs for the other joints were carried-out in a similar way.

The prescribed structure of the controller was the product of one integrator (thus directly enforced in the controller

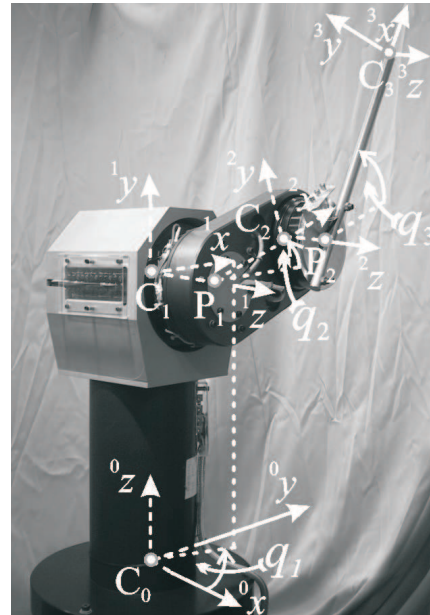


Fig. 2. The RRR robot

structure) and two notch filters; the notches were based on our experience of resonances at 28 Hz and 98 Hz in the position measurements from the 1st robot joint; we used a 12th order FIR filter as the basis function, with a total of 13 tuning parameters. The parameters were computed by minimization of Eq. 2. The Bode plot of the resulting controller is shown in Fig. 3. By inspection of the plot, one notices that the integral action was achieved, and that effects of the enforced notches are present in the controller. Apart from the enforced ones, several other notch effects show up. Induced by the resonances in the plant dynamics, these effects were created by the tuning part of the controller.

The Bode plot of the achieved sensitivity transfer functions is shown in Fig. 4, together with the corresponding desired transfer function response. The achieved transfer function was computed based on the plant FRF data. Similarities between the plots in the frequency ranges of interest are in agreement with our criteria, the peaking in the sensitivity is below 6 dB, and the controller passed the stability test. Therefore, our requirements for the quality of the design have been met.

III. UNFALSIFIED CONTROL

A. Method

In this section, unfalsified control theory is applied to determine which control parameter sets in a specified control structure are able to meet a given performance specification, using only measured input/output data. The concept of unfalsified control was introduced by Safonov and Tsao (1997) as “a framework for determining control laws whose

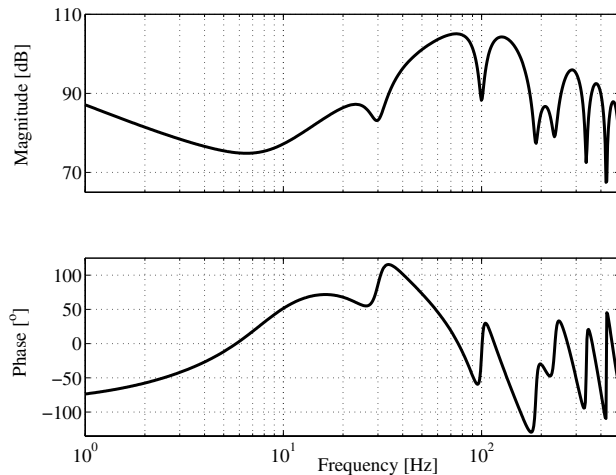


Fig. 3. the data-based controller FRF

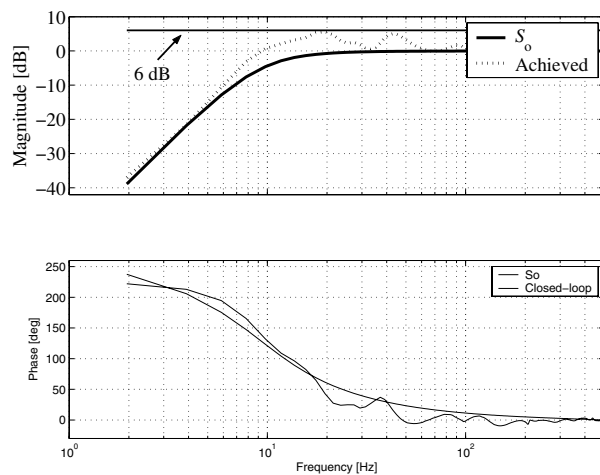


Fig. 4. desired and realized sensitivity

ability to meet given performance specification is at least not invalidated (i.e., not falsified) by the experimental data.” This data-driven model-free control approach recursively falsifies control parameter sets that fail to satisfy a performance specification, given measured data. The only assumption is that at least one controller from the original controller pool satisfies the performance specification at all times.

In early works, the controller parameter space was gridded, resulting in a finite set of candidate controllers. A trade-off has to be made between the number of candidate controllers and the computational load. The restriction to a finite set was lifted by Cabral and Safonov (2003) by employing an ellipsoid to describe the region containing all unfalsified controllers. New measurement data defines another ellipsoid, and the intersection of both ellipsoids specifies the region containing the unfalsified controller parameter sets including

the information of the new measurement data. This intersection is approximated by an outer-bounding ellipsoid, to ensure that no unfalsified parameter set is wrongly falsified and that an ellipsoidal unfalsified region is maintained. However, the *minimum-volume* outer-bounding ellipsoid can only be computed efficiently in a few specific cases, for instance if the old ellipsoid is exactly sliced in half, as is employed in (Cabral and Safonov, 2003).

In (van Helvoort, de Jager and Steinbuch, 2005), it is shown that if new measurement data defines two parallel half-spaces, the minimum-volume outer-bounding ellipsoidal approximation of the intersection can be computed analytically. Hence, the resulting algorithm is fast and can be implemented in real-time, as is shown in an experiment on a motion system.

B. Results

Consider the performance specification

$$-\Delta(t_k) \leq G_m(s)r(t_k) - y_2(t_k) \leq \Delta(t_k) \quad (3)$$

with G_m the reference model = desired closed-loop dynamics, r the reference signal, y_2 the angular position of the load mass and $\Delta(t_k)$ a threshold. With the controller structure

$$r(t_k) = w(u(t_k), y(t_k))^T \theta \quad (4)$$

the performance specification defines two parallel half-spaces in the controller parameter space $\theta \in \mathbb{R}^p$, if $w(u(t_k), y(t_k))$ is causally-left-invertible for $u(t_k)$.

The algorithm, as presented in (van Helvoort, de Jager and Steinbuch, 2005), is implemented using a dSpace DS1102 controller board at a sampling rate of 1.0 kHz. The average turnaround time is 0.57 ms. The experimental setup is a dual rotary 4th order motion system, as shown in Fig. 5. It consists of a load which is connected to a motor by a thin, flexible bar.

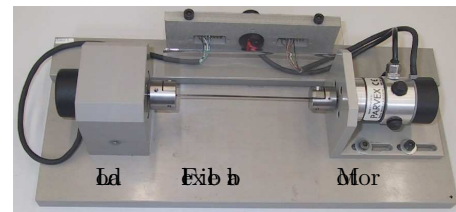


Fig. 5. Photo of the dual rotary 4th order motion system.

The control parameter set is updated once it is falsified. In Fig. 6, the tracking error $G_m r - y_2$ is shown, together with the thresholds $-\Delta$ and $+\Delta$. In this preliminary experiment, a 1 Hz sinusoid is applied as a reference signal input and the controller parameters are randomly initialized. The current controller parameter set is unfalsified if the tracking error is within the thresholds $-\Delta$ and $+\Delta$. It can be seen that at $t \approx 1$ second and repeatedly in the interval $t \in (23, 38)$, the currently implemented controller parameter set is falsified. A new controller parameter set is selected at

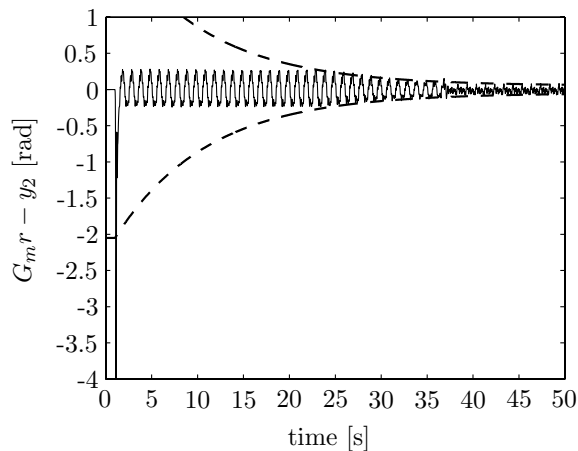


Fig. 6. Tracking error during the experiment, together with the thresholds $+\Delta(t_k)$ and $-\Delta(t_k)$. As long as the error is within the bounds, the current controller parameter set is unfalsified.

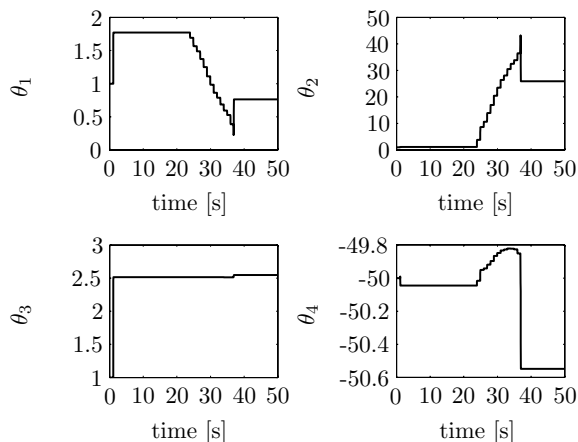


Fig. 7. Values of controller parameters during the experiment as a function of time. The values change if the current control parameter set is falsified.

these time instances, as can be seen in Fig. 7, which shows the implemented controller parameters as a function of time.

In Fig. 8 a Bode plot is shown of the open loop of the system, with the unfalsified controller obtained after 50 seconds. A bandwidth of around 10 Hz is achieved, with a phase margin of 40° (at 10 Hz) and a gain margin of 9 dB (at 42 Hz). However, it can also be seen that the resonance at 48 Hz peaks through the 0 dB line, causing only 15° phase margin at 46 Hz. The controlled system is stable but the sensitivity is bad at the resonance peak. However, due to the sinusoidal reference signal of 1 Hz, this frequency-range is not excited and hence it will not result in a large tracking error. Results with a more exciting reference signal (e.g., a block signal) are still subject of research, as well as incorporation of an unfalsified friction compensator.

IV. NONLINEAR CONTROL OF LINEAR MOTION SYSTEMS

A. Introduction

There has been considerable interest in literature on the topic of fundamental limitations in feedback systems with known plant dynamics. This has its origin in the seminal

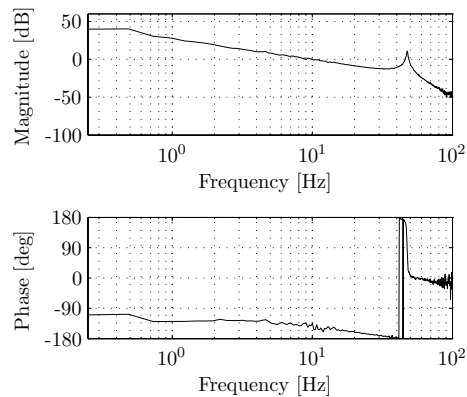


Fig. 8. Bode plot of the open loop transfer with the unfalsified controller obtained after 50 s.

work of Bode (Bode, 1945), more recent work is contained in (Seron *et al.*, 1997). Most of this work is restricted to linear feedback loops. Part of the limitations is inherently linked to the plant and thus hold irrespective of how the input is generated, be it via linear, nonlinear or time-varying feedback. Other limitations are a consequence of the plant acting in combination with linear time invariant (LTI) feedback control. This naturally rises the question if these latter limitations can be ameliorated by using nonlinear or time-varying in stead of LTI feedback.

Being aware of the difficulties and bad effects normally associated with the presence of nonlinearities, it may appear a step backwards to intentionally introduce nonlinearities into an otherwise essentially linear system. A problem that arises is to be able to predict the systems response for various inputs. In many cases however, there may be good reasons to intentionally introduce nonlinear elements in the feedback loop. Indeed, in literature several examples have been presented showing that nonlinear control can, in certain circumstances, outperform linear time invariant feedback control for known plants. In (Feuer *et al.*, 1997), it is shown that a simple PI controller whose integrator is switched on and off depending on the size of the error, performs better than its linear time invariant counterpart. Also, based on experience, several nonlinear ‘tricks’ are used in industry to obtain better performance of an LTI feedback system (Heertjes and Steinbuch, 2004). A more systematic strategy for nonlinear control of an LTI plant is reset control. Reset control action resembles a number of popular nonlinear control strategies such as relay control (Tsytkin, 1984), sliding mode control (Decarlo, 1988) and switching control (Branicky, 1988).

The motivation for these and other types of nonlinear control for linear systems is the fact that linear controllers have the inherent disadvantage that their gain and phase characteristics are related (Bode, 1945). Specifically, the need to optimize the open-loop high frequency gain often competes with required high levels of low frequency loop gains and phase margin bounds. Typically, in motion systems the open-loop frequency response is required to have sufficient bandwidth

and large low frequency gain to obtain a fast response and good settling behavior or tracking. On the other hand, at high frequencies, the loop gain needs to be small to suppress residual vibration and sensor noise. This performance trade off is defined by Bode's gain/phase relation, which limits how fast the open-loop gain can cross unity gain while maintaining closed-loop stability, whereas in the ideal case these characteristics would be designed independently of one another.

This section shows one example of nonlinear control of a linear system and presents experimental results obtained on the dual rotary 4th order motion system of Fig. 5, see also (Aangenent, Van De Molengraft and Steinbuch, 2005) for details. Since a common test of servo performance is the step response, the goal is to show that the introduction of nonlinear elements in an essential linear motion system can improve the step response with respect to the combination of overshoot and settling time when compared to standard linear feedback. To the authors knowledge, no experimental results of this kind obtained on motion systems have been presented in literature yet. In section IV-B, the SPAN (split-path nonlinear) filter is discussed, and experimental results are presented in section IV-C.

B. Nonlinear control example

In this section, results of nonlinear control is presented for the SPAN filter (Foster *et al.*, 1966). The control setup is depicted in Fig. 9.

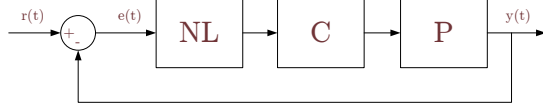


Fig. 9. Control scheme.

In this figure, NL denotes the nonlinear control element, C denotes the linear controller and P denotes the linear system dynamics. In order to guarantee zero settling error, a linear controller consisting of a proportional part, a lead/lag filter and an integrator is applied to the system.

The SPAN filter is an attempt to obtain a filter which has independent gain and phase characteristics. In Fig. 10, a block scheme of a SPAN filter is depicted. This filter processes the input in two paths and multiplies the output of the two branches. The path containing the *sign* element

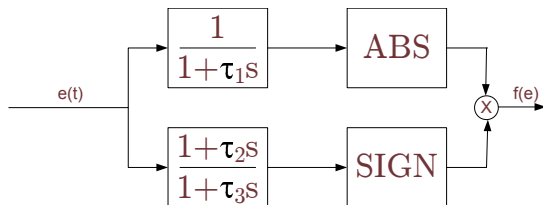


Fig. 10. Block diagram of a SPAN filter.

controls the sign of the signal and destroys all magnitude

information, while the *absolute value* element destroys all sign information, and therefore controls the magnitude information. With this filter, it seems that the 'phase' and the magnitude can be independently chosen (although it is of course now a nonlinear filter of which definitions of phase and magnitude are unclear).

The SPAN filter can be used as a phase lead filter that does not cause magnitude amplification. In the control scheme in Fig. 9, the SPAN filter takes the place of the nonlinear element, and the integrator and the lead filter are still used. It is now possible to increase the cut-off frequency of the integrator while keeping the closed loop stable by applying a lead filter in the sign path of the SPAN filter. In the absolute value path, a low-pass filter is used to attenuate higher frequencies. In Fig. 11, the describing function of the SPAN filter is depicted. This describing function is independent of the amplitude of the input but depends only on the frequency. As can be seen, within the describing function theory, this filter is able to obtain phase lead while attenuating the magnitude, something which is not possible with any linear filter.

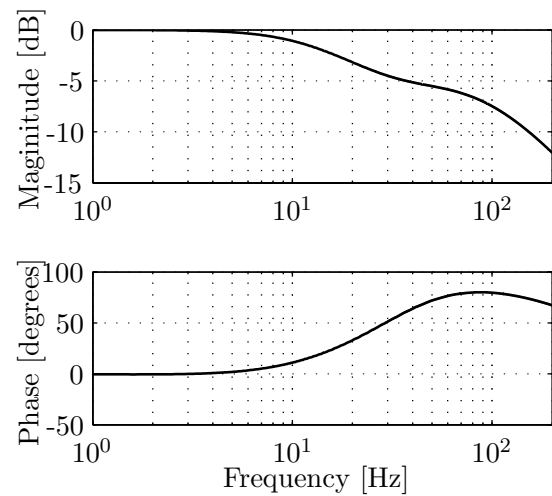


Fig. 11. Frequency dependent describing function of the SPAN filter.

With the SPAN filter in the loop, the cut-off frequency of the integrator can be increased without destabilizing the closed-loop system.

In the design of the filter, the cut-off frequency of the integrator is increased to 18.5 Hz, the cut-off frequency of the low-pass filter is set to 11.14 Hz, the zero of the lead filter is placed at 2.12 Hz, and the pole is located at 38.19 Hz. The gain of the SPAN filter was tuned to obtain a step response without overshoot and a reasonable settling time and is set to 0.15.

C. Experimental results

In Fig. 12, the measured step responses of the linear controlled system and of the system with the SPAN filter are shown. This filter is able to obtain approximately the same settling time, while avoiding overshoot completely. This is a response that cannot be obtained using a linear controller. A

drawback of the SPAN filter is the tedious tuning. Since it is a nonlinear filter, superposition does not hold and, therefore, the tuning procedure for every parameter is based on trial-and-error. A big advantage of the SPAN filter is the fact that its performance is independent of the amplitude of the input.

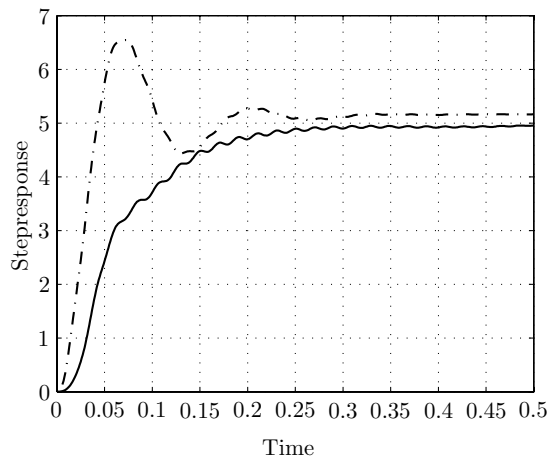


Fig. 12. Measured stepresponse of the linear system (dashed) and with the SPAN filter (solid).

V. CONCLUSIONS

For motion systems in particular, experimentation is relatively easy. The use of data to further improve feedback designs is hence a useful research item for this class of systems. In the paper both data-based approaches as well as nonlinear control design have been shown. Experiments show feasibility and stimulate further work in this area. With respect to the data-based methods one could argue that implicitly there is a plant model used. This holds especially for the virtual reference method. It is not clear how this works out for the unfalsified control. Further research is necessary. The nonlinear control method shown here generates more questions than answers. However, it is well known that in industrial practice people use many sorts of nonlinear control action to improve performance for specific inputs or requirements. Interesting options for research are to generate a systematic design procedure for classes of such nonlinear elements.

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