Self-tuning Integral Force Feedback

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SELF-TUNING INTEGRAL FORCE FEEDBACK

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ABSTRACT

A self-tuning procedure is proposed for an active structural element with collocated sensing and actuation (a so-called 'Smart Disc'). The procedure aims at optimal active damping by means of Integral Force Feedback control. In case the behavior of the structure to be damped may be described by a single dominant vibration mode, self-tuning is realized in two fairly simple steps: (1) recursive estimation of three system parameters and (2) determination of the optimal controller parameters. However, if the assumption concerning a single dominant vibration mode does not hold, both steps in the self-tuning procedure may easily fail.

1. INTRODUCTION

High-precision machines typically suffer from small but annoying, badly damped vibrations. In case such a vibration problem can not be solved by passive means, a solution may be found in active vibration control. In this respect, research at the Drebbel Institute at the University of Twente is aimed at the development of a 'Smart Disc', which is envisioned as an active structural element, consisting of a piezoelectric position actuator collocated with a piezoelectric force sensor and control and amplifier electronics. By inserting Smart Discs at appropriate locations in a high-precision machine frame, the effective damping as well as the effective stiffness of the frame may be improved [1,2,3].

It is well known that active damping by means of a position actuator collocated with a force sensor can be realized by applying Integral Force Feedback (IFF) control. The most appealing property of IFF is the robust stability of the control scheme. A drawback however is that, as the damping of the mechanical structure increases, the effective low-frequency stiffness of the structure decreases. Consequently, there is a trade-off between 'active damping augmentation' and 'effective stiffness loss'.

A Smart Disc is intended to be easily employable within a machine frame. This implies that it should ideally be self-tuning: it should be able to automatically achieve its optimal controller settings. The aim of the present paper is to propose a self-tuning procedure for an IFF-controlled Smart Disc. To this end, it is first assumed that a Smart Disc has to damp a single dominant vibration mode. Subsequently it is examined to what extent the procedure, based on the single mode assumption, is also applicable within structures that suffer from additional vibration modes.

The paper is organized as follows. Section 2 introduces the typical IFF 'damping versus stiffness' trade-off, along with a criterion for 'optimal damping'. Section 3 is devoted to the analysis of a structure suffering from a single vibration mode to be damped. This analysis will lead to two expressions, relating the optimal IFF-settings to the parameters of this simple, idealized structure. Section 4 discusses to what extent these IFF-settings may still be considered optimal within a structure that also suffers from a second vibration mode. In this respect, section also briefly touches upon the issue of parameter estimation, as this is crucial for obtaining the optimal IFF-settings. Section 5 finally presents conclusions.

2. BACKGROUND

In this section the typical IFF 'damping versus stiffness' trade-off is introduced within the context of a structure suffering from a single vibration mode, which is to be damped by a single Smart Disc. Subsequently a criterion is introduced for determining the optimal IFF-settings with respect to the damping-stiffness trade-off.

2.1 Smart Disc Concept

Consider the model in Fig. 1. In consists of a simple, idealized structure supported by a Smart Disc. In practice, a structure will exhibit an infinite number of modes, but for the present analysis, it is assumed that the behavior of the structure may be approximated by a single dominant vibration mode, i.e., the lowest vibration mode. Due to the finite 'frame stiffness' k_f, any disturbance source within or outside the structure (F_d in the model) may cause vibrations (of the mass m in the model). The task of the Smart Disc is to counteract these vibrations. In terms of the model of Fig. 1 this means that the Smart Disc should ideally keep the mass at a fixed position ($x_m = 0$), despite the fact that $F_d \neq 0$.

The main part of the Smart Disc is the piezoelectric stack k_s. The stack comprises a force sensor (indicated by F_s) and a position actuator (modeled by x_a). As piezoelectric materials should not be exposed to tensile forces, the Smart Disc has been equipped with a preload element k_p.

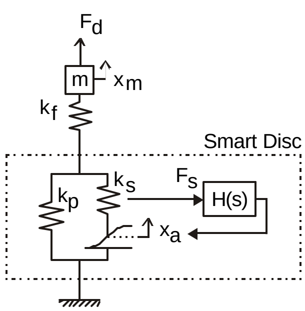

Fig. 1 Mechanical structure supported by a Smart Disc

The behavior of the mechanical structure of Fig. 1 is described by the following transfer functions [4]:

$$x_m = \frac{1}{k_m}\frac{w_e^2}{s^2 + w_e^2} F_d + d_a \frac{w_e^2}{s^2 + w_e^2} x_a \qquad \text{(1a)}$$

$$F_s = d_s \frac{w_e^2}{s^2 + w_e^2} F_d + c_{hf} \frac{s^2 + w_a^2}{s^2 + w_e^2} x_a \qquad \text{(1b)}$$

The relation between the parameters appearing in Eq. 1 (mechanical parameter k_m, actuator-related parameter d_a, sensor-related parameter d_s, high-frequent actuator-sensor feedthrough c_{hf}, resonance frequency w_e and anti-resonance frequency w_a) and the physical parameters is not that important here. It is however worth mentioning that the main effect of an increasing preload stiffness k_p (in the context of vibration control) is an upward shift of w_a towards w_e in the transfer function from the piezo-actuator to the piezo-sensor.

Bode plots of the transfer functions are depicted in Fig. 2. Here k_m, d_a and d_s have been normalized to 1, the resonance frequency $w_e = 2p\,100$ (rad/s), the 'frequency ratio' $r = w_a / w_e = 0.5$ and the relative damping has been set to 0.5%, a typical value for structural damping in high-precision machine frames [5,6].

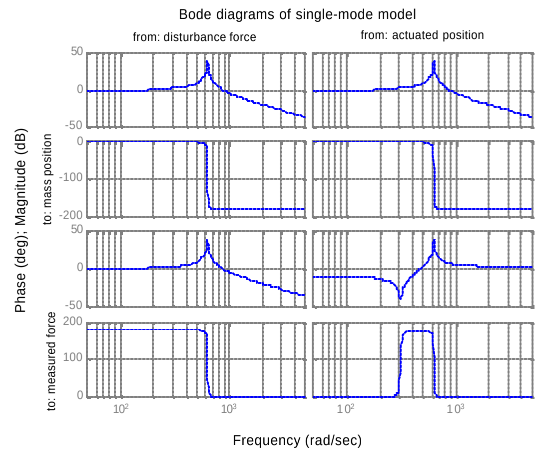

Fig. 2 Bode diagrams of single-mode model

From Fig. 2 it is obvious that the main problem within the mechanical structure of Fig. 1 is the badly damped resonance. By choosing a suitable control law $H(s)$ that closes the loop between F_s and x_a within the Smart Disc, the resonance may be damped. To that end, Integral Force Feedback is a suitable choice.

2.2 Integral Force Feedback

Integral Force Feedback is a well-known method for adding active damping by means of a collocated actuator-sensor pair [5]. Figure 3 shows the effect of 'pure' IFF (i.e., the loop to be closed from actuator to sensor is augmented with a pure integrator):

$$H(s) = \frac{g}{s}. \qquad (2)$$

Due to the fact that the actuator and sensor are collocated, the zeros and poles (even those corresponding to unmodeled structural dynamics) alternate along the imaginary axis. As a consequence all poles are drawn into the left half of the s-plane, which implies that IFF damps (and thus stabilizes) all resonances. The robust stability of the control scheme, briefly touched upon here, due to its simplicity, is the main virtue of IFF.

When looking at the frequency response of interest (i.e. the transfer function from the disturbance source to the 'response quantity of interest', Fig. 4)

$$H_{F_d}^{x_m}(jw) = \frac{x_m(jw)}{F_d(jw)} \qquad (3)$$

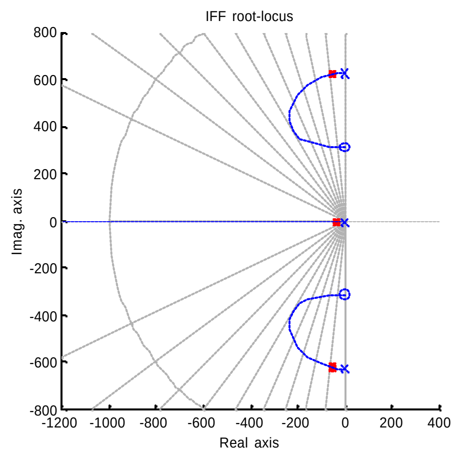

Fig. 3 Root-locus of the IFF-controlled structure

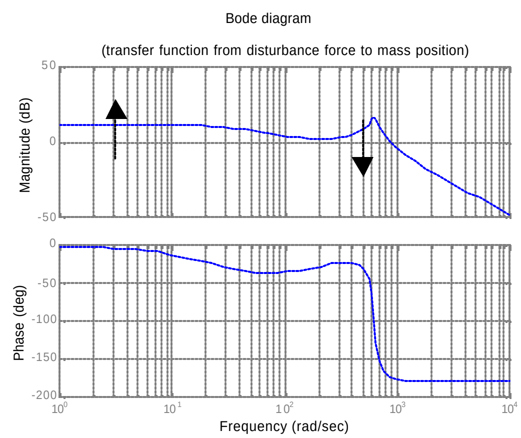

Fig. 4 The trade-off between stiffness and damping

an other consequence of the simplicity of the control scheme is observed. By means of the arrows it is indicated that the increase of damping (i.e., the lowering of the resonance peak), is accompanied by an unwanted side effect: a decrease of the effective stiffness (i.e. an increase of the low-frequency response). Consequently, when applying IFF, there is a trade-off between 'active damping augmentation' and 'effective stiffness loss'. The optimal IFF-settings with respect to this trade-off obviously depend on the situation at hand: what does the frequency spectrum of the disturbance source look like, and what is the frequency range of interest (for proper operation of the high-precision machine that is considered)?

2.3 General Optimal IFF Control Criterion

As the aim of the present paper is to come up with a general self-tuning procedure for an IFF-controlled Smart Disc, at this point two plausible assumptions need to be made. These assumptions will lead to a well-defined 'general situation at hand' and subsequently to the introduction of a general criterion that can be used to determine the 'optimal' IFF-settings with respect to the damping-stiffness trade-off.

Assumption 1: disturbance spectrum is flat (white noise)
As acoustics are often a dominant disturbance source in a high-precision machine environment, the disturbance source may well be characterized by a flat spectrum within a broad frequency range (band-limited white noise). As a consequence of this assumption, the optimal IFF-controller settings depend solely on the transfer function from the disturbance source to the response quantity of interest (Eq. 3).

Assumption 2: optimum implies minimum residual power
Furthermore, for many applications the frequency range of interest is not limited to a narrow band; in general, a machine should function properly in a broad frequency range. In this case, an appropriate measure for the vibration control performance is the residual power present in the response quantity of interest. Minimization of this power implies that the optimal trade-off between damping and stiffness has been achieved.

The cost to be minimized can now be defined as follows:

$$J = s^2 = \int_{w=0}^{\infty} \left| H_{F_d}^{x_m}(jw) \right|^2 dw \qquad (4)$$

In case of a normalized flat disturbance spectrum for F_d, the integrand represents the Power Spectral Density of the response quantity of interest, and the cost J indeed represents the (residual) power present in the response quantity of interest.

At this point it should be noted that the (frequency domain) cost to be minimized can also be given an appropriate interpretation in the time domain. The square root (s) of this cost may be regarded as a measure of the accuracy at which the response quantity of interest is kept at its ideal value ($x_m = 0$). Minimum cost J thus truly implies 'optimal damping'.

Furthermore, in case of a normally distributed signal, 99% of the values over time are known be within the [$-3s$, $3s$]-range. Therefore the $3s$-value that can be calculated from the cost J will be used as the criterion for optimality in the sequel of the paper.

3. IFF TUNING FOR THE SINGLE-MODE MODEL

In this section, the problem as set up in the preceding section, will be analyzed. Instead of applying 'pure' Integral Force Feedback, the IFF control law is given by

$$x_a = -\frac{g}{s+a} F_s \ . \qquad (5)$$

The frequency response of interest can be shown to be (initial structural damping is neglected again):

$$H_{F_d}^{x_m}(s) = \frac{w_e^2}{k_m}\frac{s + a + c_{hf}g}{s^3 + (a + c_{hf}g)s^2 + w_e^2 s + w_e^2(a + r^2 c_{hf}g)} \,. \quad (6)$$

Normalizing Eq. 6 with respect to w_e and c_{hf} results in

$$H_{F_d}^{x_m}(w_e s_n) = k_m^{-1}\frac{s_n + a_n + g_n}{s_n^3 + (a_n + g_n)s_n^2 + s_n + (a_n + r^2 g_n)} \quad (7)$$

with

$$s_n = s/w_e\,,\ \ g_n = c_{hf}\,g/w_e\,,\ a_n = a/w_e\,. \quad (8)$$

It should be noted that the parameters d_a and d_s (Eq. 1) do not appear explicitly in Eq. 6, and that the mechanical parameter k_m only appears as a gain. This implies that, in order to determine the optimum setting for the IFF-parameters a and g, only knowledge of the transfer function from the piezo-actuator to the piezo-sensor is needed. In turn, this implies that IFF-control within a Smart Disc may be truly *self-tuning*: a Smart Disc ‘only’ needs to determine the three parameters (w_e, w_a and c_{hf}) that appear in the collocated transfer function from the actuator to the sensor it is equipped with (compare Eq. 1):

$$H_{x_a}^{F_d}(s) = c_{hf}\frac{s^2 + w_a^2}{s^2 + w_e^2} \quad (9)$$

Furthermore, from Eq. 7 it can be concluded that the normalized optimal IFF-parameter settings a_n and g_n do not depend on w_e (as w_e does not appear explicitly in the expression). In turn, this implies that the original IFF-parameters a and g depend linearly on w_e. In the same way, g is easily seen to depend linearly on c_{hf} also (with Eq. 8).

The dependence of a_n and g_n on the frequency ratio r however is not that straightforward. Optimization by simulation with respect to the 3s-criterion for various values of r resulted, as expected, in various optimal IFF-parameters. The transfer function of interest (Eq. 6) of the optimally IFF-controlled system however appeared to be independent of the value of r (see Fig. 5).

Close inspection of the optimal transfer function of interest, revealed that for the optimum normalized IFF-settings, the following relations hold (at least within eight digits):

$$a_n + g_n = \tfrac{1}{3}\sqrt{3}\ , \quad (10a)$$

$$a_n + r^2 g_n = \tfrac{1}{9}\sqrt{3}\ . \quad (10b)$$

From this the optimal IFF-settings can be derived:

$$a = a_n w_e = \frac{1}{9}\sqrt{3}\left(3 + \frac{2}{r^2 - 1}\right)w_e\ , \quad (11a)$$

$$g = g_n\frac{w_e}{c_{hf}} = \frac{2}{9}\sqrt{3}\,\frac{1}{1 - r^2}\,\frac{w_e}{c_{hf}}\ . \quad (11b)$$

The optimal criterion value appeared to be $3s^{opt}$ = 84.62, which is 23% of the original value $3s^{\zeta=0.5\%}$ = 375.99 (for the response depicted in Fig. 2). IFF thus improves the performance by a factor 4.

Though it may seem now that the value of the frequency ratio is of no importance for the damping result that may be achieved, for reasons of controllability it is obvious that r should preferably be as small as possible. In this respect it is worth mentioning that by (partly) compensating the low-frequency feedthrough in the collocated transfer function (Eq. 9), w_a can be shifted down, while not affecting w_e. A full down-shift of w_a to 0, may not be advisable in practice, because of possible non-linearities in the system (for instance hysteresis in the piezo material [6]).

In order to conclude this section on optimal IFF-parameter settings for a structure with a single vibration mode, the effect of sub-optimal IFF-parameter tuning is summarized in Fig. 6. From these plots it becomes clear that a change of 50% in the value for the IFF pole (a) hardly affects the damping performance (corresponding

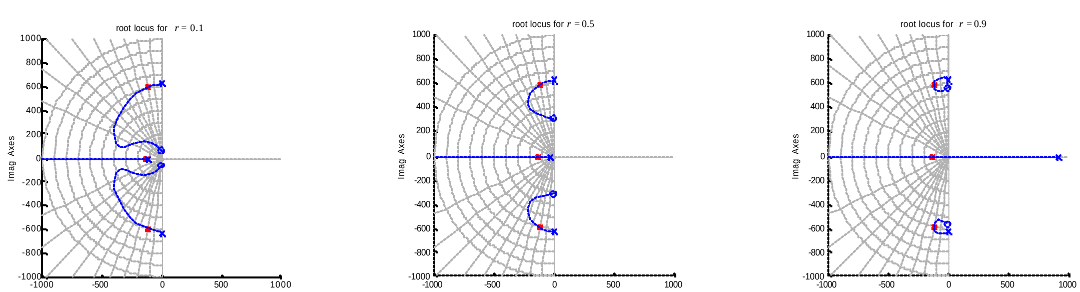

Fig. 5 The optimal locations of the closed-loop poles are the same for various frequency ratios r

to the fact that the root-locus shape hardly changes). Erroneous tuning of the IFF gain (g) has more impact on the performance (corresponding to the fact that the pole moves 'fast' along the root-locus). However, a change of 25% with respect to the nominal optimal gain (according to Eq. 11) results in a 3σ-value of 87, which is still very close to the optimum value presented before (84.62).

4. COMMENTS ON THE TUNING PROCEDURE

In this section it is examined to what extent the IFF-settings according to Eq. 11 may still be considered optimal within a structure that also suffers from a second vibration mode. In this respect, however, first the issue of parameter estimation will be discussed, as this is crucial for obtaining the optimal IFF-settings.

4.1 Parameter Estimation

The processor that is used for parameter estimation within a Smart Disc, will be the same as the processor on which the IFF-controller has to be implemented. As the IFF-control law is simple, the parameter estimation procedure should also be kept simple. The estimator should thus preferably be realized with low memory and low computing power demands. An on-line, recursive estimator is favorable in this respect.

Furthermore, due to the nature of the problem that has been set up, continuous time parameters (w_a, w_e) have to be estimated from discrete time data. In this respect parameter estimation approaches can be divided into indirect and direct techniques [7].

The main advantage of an *indirect* approach is that it directly relates the discrete time input and output signals, which makes it easy to build an estimator for its parameters. A drawback however is that the relation between the discrete time parameters and their continuous time counterpart is rather complicated, which makes it difficult to reliably calculate the continuous time parameters. Another well-known disadvantage is that the choice of the sample frequency is rather critical.

For a *direct* parameter estimation approach, a discrete time approximation of continuous time operators is used. This results in a discrete time approximate of the continuous time model which contains the original continuous time parameters (see [8] for instance). Advantages of this approach are the direct estimation of continuous time process parameters and the fact that there is no upper sample frequency constraint. Drawbacks are the more difficult implementation and strict constraints on the excitation signal.

Both direct and indirect approaches however do work, especially in the simple context as set up in this paper, i.e., for a structure that suffers from a single vibration mode. As soon as other, unmodeled dynamics come into play, it is difficult to ensure that the parameter estimates convergence to the vibration mode of interest. Experiments have shown that convergence can be ensured, by correct filtering of signals used by the estimator. In the experimental set-up, the system exhibited a second interfering mode at 140% of the dominant vibration mode. When the cut-off frequency of the low-pass filter was set in between these modes (at 125%) the estimation accuracy was within 1%. When the cut-off frequency was set beyond the second mode (at 150%) the estimated parameters still converged to the correct vibration mode, but the accuracy decreased to 5% [6]. For true self-tuning IFF within a Smart Disc, the correct estimator-filter settings should ideally also be determined automatically in some sense. This problem however has not been addressed yet at the Drebbel Institute.

4.2 Impact of a Second Vibration Mode

In this section the effect of an additional mode being present in the structure is examined. Simulations have been performed for the system as described before (Fig. 2), augmented with a second vibration mode at 200, 400 or 800 Hz. The influence of the second mode (in terms of the 'low-frequency contribution', see the upper plots in Fig. 7 for an illustration) was gradually increased.

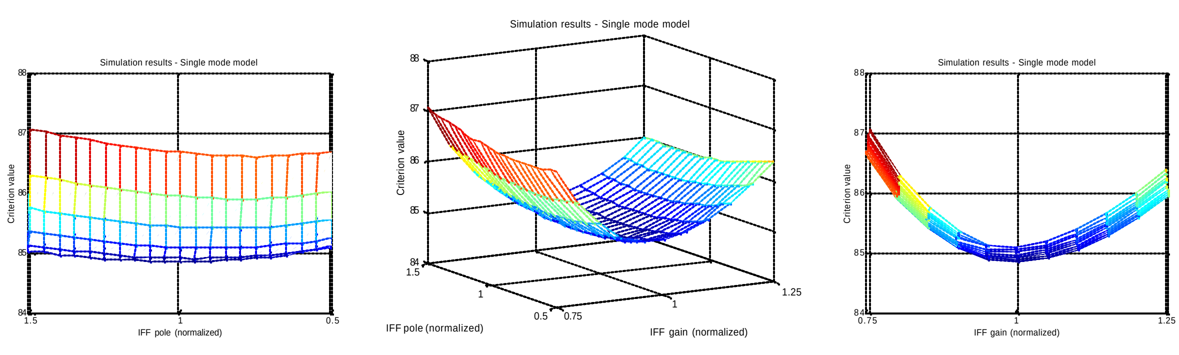

Fig. 6 The effect of sub-optimal IFF-parameter setting on the damping performance (in terms of the 3 *s*-value)

For each case the $3s$-value as obtained by IFF according to Eq. 11 was calculated and compared with the optimal $3s$-value for the particular case. The value of the second resonance frequency appeared to have no significant influence on the results, which are gathered in Table 1.

Table 1 Comparison of optimal IFF-performance and performance of IFF with settings according to Eq. 11

Low-frequency contribution of second mode	1%	5%	10%	25%	50%
Optimal IFF performance	23%	23%	24%	28%	37%
Performance with IFF settings as in Eq. 11	23%	23%	24%	29%	48%

From Table 1 it can be seen that as long as the low-frequency contribution of the second mode is relatively low (25% or less) the IFF-settings as provided by Eq. 11, may reasonably be considered optimal. In case the low-frequency contribution of the second mode is increased to 50%, the settings are far from optimal. However, one also sees that the optimal performance, with respect of damping of the lowest vibration mode, gets worse (from 23% to 37%). This is a direct consequence of the assumption concerning the dominance of the lowest vibration mode being violated.

5. CONCLUSION

A self-tuning procedure is proposed for an active structural element with collocated sensing and actuation (a so-called ‘Smart Disc’). The procedure aims at optimal active damping by means of Integral Force Feedback control. In case the behavior of the structure to be damped may be described by a single dominant vibration mode, self-tuning is realized in two fairly simple steps: (1) recursive estimation of three system parameters describing the frequency response from Smart Disc actuator to sensor (Eq. 9), and (2) setting the IFF controller parameters, according to Eq. 11.

If the assumption concerning a single dominant vibration mode does not hold, both steps in the self-tuning procedure may easily fail, first of all because parameter estimation becomes difficult, and secondly because the settings as given by Eq. 11 are not optimal any more.

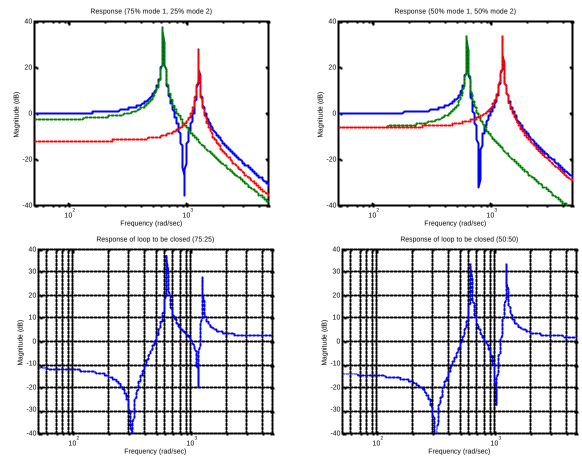

Fig. 7 Effect of additional mode at $w_{e2} = 2\pi 200$ (rad/sec) Low-frequency contribution 25% (left) and 50% (right) Upper plots: x_m/F_d (jw), lower plots: F_s/x_a (jw) (no IFF)

REFERENCES

[1] Holterman, J., T.J.A. de Vries, and M.P. Koster: Experiment to evaluate the feasibility of the Smart Disc concept. In: *Proc. 6th UK Mechatronics Forum International Conference ‘Mechatronics ’98’*, Skövde, Sweden, pp. 217-222 (1998).

[2] Van Schothorst, G.: Active Vibration Control using Piezoelectric Smart Discs. In: Mathematics and Control in Smart Structures, *Proc. SPIE Smart Structures and Materials*, **Vol. 3667**, pp. 637-648 (1999).

[3] Holterman, J. and T.J.A. de Vries: Active structural elements within a general vibration control framework. In: *Proc. 1st IFAC Conference on Mechatronic Systems ‘Mechatronics 2000’*, Darmstadt, Germany, pp. 997-1002 (2000).

[4] Verdaasdonk, B.W.: *Controller design for Smart Disc in 3D-experiments*, M.Sc. Thesis, 015R2000, Control Laboratory, Electrical Engineering Department, University of Twente, Enschede, The Netherlands (2000).

[5] Preumont, A.: *Vibration Control of Active Structures, An Introduction*, Kluwer Academic Publishers, Dordrecht, The Netherlands (1997).

[6] Jansen, B.S.H.: *Smart Disc tuning and application in an ASML wafer stepper*, M.Sc. Thesis, 016R2000, Control Laboratory, Electrical Engineering Department, University of Twente, Enschede, The Netherlands (2000).

[7] Söderström, T., H. Fan, B. Carlsson, and S. Bigi: Least squares parameter estimation of continuous time ARX models from discrete-time data. In: *IEEE Transactions on Automatic Control*, **Vol. 42**, pp. 659-673 (1997).

[8] Söderström, T., and M. Mossberg: Performance evaluation of methods for identifying continuous-time autoregressive processes. In: *Automatica*, **Vol. 36**, pp. 53-59 (2000).