Mechatronics 24 (2014) 307–317

Contents lists available at ScienceDirect

Mechatronics

journal homepage: www.elsevier.com/locate/mechatronics

Model-based spatial feedforward for over-actuated motion systems

M.J.C. Ronde [a,*], M.G.E. Schneiders [a], E.J.G.J. Kikken [a,b,1], M.J.G. van de Molengraft [a], M. Steinbuch [a]

[a] Eindhoven University of Technology, Dept. of Mechanical Engineering, Control Systems Technology Group, P.O. Box 513, 5600 MB Eindhoven, The Netherlands
[b] ASML Research Mechatronics, P.O. Box 324, 5500 AH Veldhoven, The Netherlands

ARTICLE INFO

Article history:
Received 31 January 2013
Revised 22 July 2013
Accepted 20 September 2013
Available online 7 November 2013

Keywords:
Feedforward design
Flexible systems
Lightweight motion systems
Over-actuation

ABSTRACT

In high-performance motion systems, e.g. wafer-stages and pick-and-place machines, there is an increasing demand for higher throughput and accuracy. The rigid-body design paradigm aims at very stiff designs, which lead in an evolutionary way to increasingly heavier systems. Such systems require more and more power, such that this paradigm rapidly approaches the boundary of its scalability. An alternative paradigm is to design a lightweight machine with over-actuation and over-sensing, to deal with the resulting flexibilities. This paper presents a spatial feedforward method for over-actuated flexible motions systems, which aims at reducing the vibrations over the complete flexible structure during motion. The proposed method is experimentally validated on an industrial prototype and compared to mass feedforward and the standard input shaping technique.

1. Introduction

In the semiconductor industry higher throughput and higher accuracy are desired to keep up with Moore's law [1] and stay ahead of competition. More aggressive motion profiles (i.e. higher accelerations) and a design with higher stiffness (i.e. higher mass) are required to obtain the desired higher throughput, while maintaining desired accuracy. This will require larger forces, which put stricter demands on actuators, amplifiers and cooling, which is expected to become infeasible in the near future. For a general overview of the control of high-performance motion systems see [2,3].

The next generation of advanced motion systems is expected to be lightweight, which results in significant internal flexibilities. An example of the mode shapes of such systems is shown in Fig. 1. This has several consequences for control design:

1. resonances in the region of interest, i.e. close to or even below the objective bandwidth, and
2. transfer between sensor output y and performance location z becomes dynamical due to the limited stiffness, i.e. there is no geometrical transformation possible anymore to analyze/control the performance.

Common feedforward methods, which do not take the flexible dynamics into account, will not lead to satisfactory results for lightweight systems. Lightweight systems are typically over-actuated [4–6], i.e. the system contains more actuators than rigid-body degrees of freedom. The additional actuators provide extra design freedom, which is not exploited by the current feedforward design methods. Therefore, the goal is to exploit this design freedom to obtain a higher performance than traditionally designed systems with rigid-body feedforward.

Snap-feedforward [7,8] is a common method to compensate for the compliance of the low frequent contribution of flexible modes in feedforward, i.e. the deformation due to compliance during motion. However, this method only guarantees local performance, i.e. at the sensor location only.

Data-based tuning of the feedforward parameters [9] can improve performance, but still suffers from the same drawbacks as snap-feedforward. Also similar work with a more generic structure, such as [10,11], only guarantees performance at the sensor location.

For the class of lightweight systems, local performance at the sensor y is generally not sufficient, since this does not provide any guarantees at the performance location z, i.e. the location where the tool operates [12,13].

A common method to reduce vibrations in motion systems is input shaping, where the objective is to remove the eigenfrequencies of the flexible structure from the input signal. This is typically done by convolving the input signal with an input shaper [14–20]. Such methods aim to increase the performance after setup-time, i.e. the residual vibrations after the point-to-point motion are attenuated. However, these methods obtain global performance, i.e. at any point of the structure.

If positive shapers are considered [21], the shaped input signal satisfies the same bounds as the original input signal at the cost of extra delay, which may be undesired in the intended application.

* Corresponding author. Tel.: +31 402472798.
E-mail addresses: M.J.C.Ronde@tue.nl (M.J.C. Ronde), M.J.G.v.d.Molengraft@tue.nl (M.J.G. van de Molengraft), M.Steinbuch@tue.nl (M. Steinbuch).

[1] MSc student at ASML Research Mechatronics.

0957-4158/$ - see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.mechatronics.2013.09.010

Negative shapers can reduce this delay [16,22], however there are no guarantees provided on the bound of the shaped input signal. For MIMO systems this delay can be reduced [21], if the input signal is known in advance. However, in this paper the setpoint trajectory is assumed not to be known a priori. Also, application of input shaping changes the setpoint trajectory which is typically not desired in many high-performance motion applications.

Learning based approaches [23–25], such as Iterative Learning Control (ILC), require a measurement of the performance variable during the learning process, which is not available in the considered class of motion systems. Furthermore, ILC requires a new learning sequence for every new setpoint trajectory. In [26,27] ILC compensation for residual vibration prevents excitation of modes by using an actuation and observation window. In this method only local performance is guaranteed.

The proposed method in this paper can be considered as a special case of static input–output decoupling [28,29]. Typically, static decoupling aims at diagonalization of the plant, by pre- and post-multiplying the plant with a static matrix, to allow for decentralized control. However, the proposed method aims at independent control of the rigid-body modes and preventing the excitation of flexible modes. This is achieved by pre-multiplying the plant with a static matrix in the feedforward path. Therefore, there is less freedom compared to standard decoupling techniques, since only an input transformation is applied. Hence, the standard decoupling techniques cannot be applied for the problem considered in this paper.

The proposed method, called **spatial feedforward**, exploits the freedom of over-actuation explicitly. This design freedom is used to prevent excitation of the performance-relevant flexible modes. Compared to existing methods, the proposed method does not introduce extra delay in the input signal.

The techniques in this paper aim at obtaining *global performance*, i.e. at any point of the flexible structure, in contrast to earlier work [30,31] where *local (inferential) performance* is obtained.

Compared to earlier work on spatial feedforward [32], this paper provides the extension to multiple modes, including conditions for the existence of the solution. The contributions of this paper are to provide a feedforward method which has the following properties:

1. explicit use over-actuation, and
2. no additional delays introduced, and
3. prevent the excitation of multiple flexible modes, and
4. performance guarantee over the whole structure, i.e. global performance, and
5. independent of the setpoint trajectory.

The outline of this paper is as follows. In Section 2 the problem is formulated. Subsequently, spatial feedforward is introduced in Section 3. The conditions for the existence of solutions are formulated in Section 4. In Section 5 a method to compute partial solutions is provided. In Section 6 input shaping, which is used as benchmark method, is briefly discussed. In Section 7 and 8 the experimental validation and conclusions are presented respectively.

2. Problem formulation

Consider a system with proportionally damped modes. Such systems can be written in the following modal description [33]:

$$G(s) = C_m\left[Is^2 + 2Z\Omega s + \Omega^2\right]^{-1}B_m = [\,C_b C_{int}\,]\begin{bmatrix}\Theta^{(b)}(s) & \\ & \Theta^{(int)}(s)\end{bmatrix}\begin{bmatrix}B_b\\ B_{int}\end{bmatrix}, \quad (1)$$

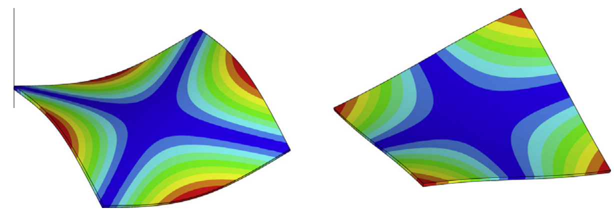

Fig. 1. Mode shapes of a plate as an example for the problems faced in advanced motion systems. The stage is typically measured at the edges, while processing takes place at a different location, i.e. a good performance at the sensors does not guarantee good performance at the location where processing takes place due to the different dynamics.

with Z and Ω diagonal, due to proportional damping. Therefore, the matrices $\Theta^{(b)}(s)$ and $\Theta^{(int)}(s)$ are diagonal and contain the second order transfer functions of the body modes and internal modes respectively. Furthermore B_{mi}, i.e. the i-th row of B_m, is associated with the i-th mode only.

The plant $G(s)$ has n_u inputs and n_y outputs and is controlled using the control structure shown in Fig. 2.

The goal is to find a static input transformation $T_{u,ff}$, such that the body-modes are independently controllable and the flexible modes are not excited by the feedforward, i.e. the flexible modes are uncontrollable.

The static transformation matrices $T_{u,fb}$ and T_y are used to decouple the system as $G_d = T_yG\,T_{u,fb}$, to allow for decentralized feedback control. The motion $m(t)$ represents the pose of a motion system. The mapping between the sensors $y(t)$ and the measured rigid-body motion $m(t)$ is given by:

$$m(t) = T_y y(t), \quad (2)$$

where $m(t)$ typically has dimension n_b.

Remark 1. The choice of T_y is not unique, i.e. scaling or linear combination of translations/rotations can be taken. In Section 7.4 a choice will be made based on physical interpretation.

Definition 2 (*Body mode*). The body modes are defined as the set of rigid-body and suspension modes. The number of body modes is denoted by n_b.

Definition 3 (*Suspension mode*). A suspension modes has, by design, a significantly lower resonance frequency than the internal modes, i.e. the structural stiffness of the suspension system to the fixed world is much smaller than the body stiffness.

Definition 4 (*Internal mode*). The undesired flexible modes are called the internal modes, i.e. without the suspension modes. The number of internal modes is denoted by n_r.

Definition 5. The number of internal modes to be suppressed by spatial feedforward is denoted by n_m.

Corollary 6. *A single mode of a system in the form of (1) is controllable if and only if $b_{mi} \neq 0$*

Proof

Consider a single mode in modal from (1). The controllability of this system can be tested by [34,35],

$$\text{rank}[B \quad AB] = \text{rank}\begin{bmatrix}0 & b_{mi}\\ b_{mi} & -2\zeta_i\omega_i b_{mi}\end{bmatrix},$$

which has clearly full row rank if and only if $b_{mi} \neq 0$. □