Design of high performance mechatronic systems using topology optimization

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1. Introduction

Developments in the semiconductor industry are driven by a desire for higher throughput, better accuracy and the transition from 300 mm to 450 mm wafers. A simple scaling of existing designs by this factor of 1.5 implies a near threefold increase in mass, leading to prohibitive increases in actuator demands and heat dissipation. Furthermore, these designs are multi-objective and multidisciplinary in nature. Aspects such as bandwidth, mass, control performance, stiffness and thermal expansion play important roles and this makes design by traditional iterative procedures involving several teams of specialists time consuming.

We develop a topology optimization approach for the conceptual design of these high performance systems. To this end, a structural design envelope is defined with the appropriate boundary conditions, such as actuation points and sensing points. Within this envelope a structure can then be designed by distributing material while simultaneously optimizing controller parameters. Objectives on the integrated design are specified in terms of the closed-loop performance of the overall system. For instance, the shape of the closed-loop disturbance sensitivity function may be optimized in the frequency domain to achieve both good rejection of disturbances as well as stability and a degree of robustness.

The integrated design approach leads to rather intricate structural designs which are unlikely to be obtained by traditional manual design methods. Combined with developments in additive manufacturing, which allows such complex designs to be produced, this provides a powerful approach to support the design of future precision devices.

2. Problem formulation

As a test case we consider a vertical positioning problem of a motion platform, such as the one depicted in Fig. 1. Actuator forces f_1 and f_2 can be exerted on the platform and the positions y_1 and y_2 are measured with respect to the fixed world. We assume that both actuator forces are equal and provided by a controller. Similarly, the measurements y_1 and y_2 are averaged to measure the net vertical position.

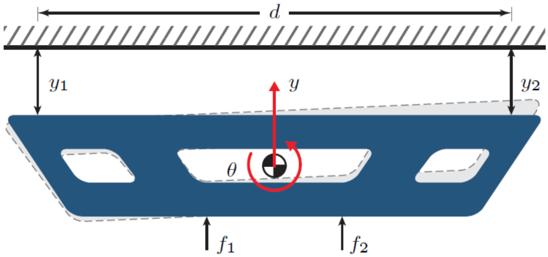

Fig. 1. Schematic representation of a motion platform that can be controlled by a combination of actuator forces (f1 and f2) and displacement sensors (y1 and y2).

The platform position is regulated to a reference value by a simplified PID controller (1). The controller is characterised by a gain k and a bandwidth ω_b, and adds integral action up to a fifth of the bandwidth, adds lead compensation between a third and three times the bandwidth and adds roll-off beyond 5 times the bandwidth. This structure is based on common rules of thumb in controller design [1].

(1) $$C(s) = k\frac{s+\frac{1}{5}\omega_b}{s}\frac{3s+\omega_b}{s+\omega_b}\frac{5\omega_b}{s+5\omega_b}$$

Given this controller structure, and the actuator and sensor locations, we consider the problem of designing the structure of the motion platform for the best attainable closed loop performance. The platform is 0.6 m long and 0.1 m tall and is represented, currently in 2D, by a design domain as shown in Fig. 2 which is discretized using finite elements. The black layer is a solid top layer to provide a useful surface. Actuator and sensor nodes are also. The finite elements in the shaded area can be varied between solid and void material. Topology optimisation is a well-known technique to systematically optimize such a structure [2]: given an optimization problem, element densities are varied on the basis of gradient information so as to optimise the objective function.

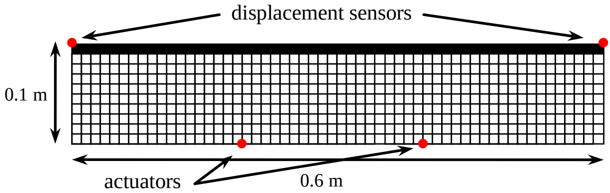

Fig. 2. The design domain for the motion platform

As objective we consider the disturbance rejection performance at 10 Hz, i.e. the amplitude of the disturbance sensitivity function $S(j\omega)$ at 10 Hz. This indicator is typical for motion systems, and the smaller this value, the better the control performance in terms of accuracy. For stability and robustness, we require the sensitivity never to exceed a value of 6 dB. This is equivalent to maintaining a safe distance from the critical point -1 in the Nyquist plot (e.g., gain, phase and modulus margins) when performing loop shaping. Summarizing, this can be formulated as:

$$\begin{array}{ll} \min & |S(10\,\mathrm{Hz})| \\ \text{subject to:} & |S(j\omega)| < 6\mathrm{dB} \text{ for } 0 \le \omega \le \infty \end{array} \tag{2}$$

3. Results

We have considered two cases. In both cases we start with a solid design in which the design domain is completely filled with solid material. For this design an initial controller is created by optimising the gain and bandwidth of (1) to achieve the best possible performance. This gave a baseline design with a certain performance, given in the first column in Table 1.

In the first case, the design is performed in an integrated fashion. To do so, the gradient of objective function (2) is computed both with respect to the element densities of the structural design and with respect to the controller parameters. Then update steps are made which improve the objective, but also satisfy the constraints. This is iterated until convergence. In the second case, the structural design is performed first with the objective of achieving the highest possible fundamental eigenfrequency. Subsequently, a controller is designed. The idea is that maximising the first eigenfrequency is often used as a design heuristic for control performance. The results are summarised in Table 1 and Fig. 3 and 4.

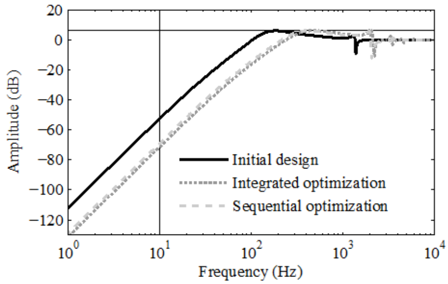

Fig. 3. The disturbance sensitivity functions for the initial, integrated and sequential designs. The horizontal line indicates the 6 dB limit.

Fig. 4. The structural designs resulting from the integrated (top) and sequential (bottom) approaches. The dark areas indicate the presence of material.

It is clear that both design approaches lead to improved designs. While both strategies lead to almost 20 dB improvement in disturbance rejection, the integrated approach leads to a significantly lower mass at 25 kg, and a higher bandwidth. However, the integrated design leads to a lower first natural frequency, indicating that the value of this frequency is not the only determining factor for performance.

Table 1. Initial and final parameters and performance metrics for the vertical positioning problem

	value		
	initial design	after integrated optimisation	after sequential optimisation
|S| at 10 Hz	-52.7 dB	-71.8 dB	-69.5 dB
mass m	41.6 kg	25.0 kg	28.3 kg
bandwidth ω_b	200 Hz	395 Hz	361 Hz
first natural frequency	1384 Hz	1980 Hz	2109 Hz

4. Discussion

We presented an approach for the integrated conceptual design of motion systems with the aim of maximising control performance. The concept was demonstrated on a simple test case and shown to give a slightly better performance than a sequential design based on an eigenfrequency heuristic, but with a significantly lower mass.

References:

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