INTEGRATED DESIGN OF A LIGHTWEIGHT POSITIONING SYSTEM

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Abstract

In this paper a new approach to the design of positioning systems is introduced. The approach aims at the design of fast and accurate systems that are lightweight compared to classical designs. The new design reduces peak power requirements and thermal effects that deteriorate performance of the whole system.

Introduction

A large class of motion systems used in precision applications (wafersteppers, scanners, pick-and-place machines for production of PCBs, wire-bonders etc.) is required to meet high performance specifications (i.e. faster and higher accuracy). The system's performance is determined by its closed-loop dynamics, which includes the mechanics of actuators, amplifiers, mechanical structure, control system and sensors. In general, the most limiting link in this loop is the mechanical structure. The traditional design approach for such motion systems starts with the mechanical design based on kinematic principles (Soemers, 2001), aimed at high servo bandwidths and repeatability. This is accomplished by designing the mechanical parts with dominant mechanical natural frequencies far beyond the required control bandwidth. For high precision systems further care is taken to reduce vibrations, e.g. by means of balance masses and vibration insulation units. In some cases passive (Mead, 1998) or active vibration damping (Fuller *et al.*, 1996) is applied, e.g. in the form of special controlled actuator-sensor pairs (Holterman, 2002). However, the traditional design has several disadvantages. When bandwidth is increased, the total mass of the system is increased also. As a consequence, the moving mass and its acceleration require higher force (current) and power of the actuators and linear power amplifiers. In practice, the ratio of the moving mass of the system to the mass of the payload is in the range of 450 up to 800, and lower values are difficult to achieve.

New Approach

This work presents a new design approach, called lightweight positioning. The research focuses on mass-reduction of the moving parts in the motion system, which allows for designing a lighter overall kinematic structure (force-path). If the mechanics are lighter, they will have lower mechanical stiffness, which causes lower dominant mechanical eigenfrequencies. This means that the internal dynamics of the structure, which deteriorate performance, are easier excited. To keep the required system performance, extra actuators and sensors are included in the design, which must improve both tracking and regulation performance. In other words, mechanical stiffness is exchanged by active control and intelligent placement of additional actuators and sensors. Applying more forces than strictly needed for the rigid body movements to be controlled is called *over-actuation*. In our approach over-actuation includes *over-sensing,* i.e. the use of more sensors than strictly needed to observe the rigid-body movement to be controlled. The new design approach incorporates three areas of expertise: control, mechanics and electromechanics. The contributions from the three different areas are all targeted at mass reduction to design an overall lighter motion system while reaching the same performance. Therefore, we call our approach an integrated design approach.

First, preliminary calculations are made to estimate the potential mass reduction if the number of actuators is increased. This is based on a semi-static analysis of the structure at hand, using setpoint specifications and desired accuracy levels. Assuming that external disturbances are small, actuator placement can be determined and also the ratio between the actuator forces to obtain minimal deflection can be calculated. The latter result can be directly used in the feedforward path of the controller. From these calculations, the peak forces for the actuators can be determined. An optimization process is used to minimize mass of the actuator design, given the set of specifications (i.e. stroke, peak force.). With these results, the overall mass reduction of the system can be determined and the optimal topology can be chosen.

Since disturbances are always present, in this case mainly due to the motion task, a feedback controller must be added. The controller in the over-actuated case is designed to reach the same level of disturbance attenuation as in the traditional case. For the rigid-body behavior of the mechanical structure, this would result in an equal bandwidth. In that case, traditional design will result in a number of modes below the tracking bandwidth, which will deteriorate performance due to the lack of damping in these modes. Fortunately, with the addition of extra actuators and sensors, it is possible to actively control a number of flexible modes, to prevent them from decreasing performance. For this purpose, an independent modal space controller is used, which enables decoupled control the lower resonance modes and design of the tracking bandwidth of the rigid-body behavior beyond these modes. The new design approach will be illustrated in the rest of this paper by the example of levitated beam. To introduce the new ideas as clear as possible, the analysis is presented in modal space.

Case study; a levitated beam

Initial investigations pointed out that the most dominant vibration modes in structures are often torsion and bending modes, since these modes possess the lowest eigenfrequencies. A simple but relevant benchmark to verify the new ideas in practice, is to perform a motion task with a simple beam with flexible (bending) modes. The positioning task is to move the entire beam over a certain trajectory *r(t)* in z-direction (figure 1.a). The stroke is limited to a few mm, comparable with the specification in high-precision short-stroke units. The Lorentz actuators are used, since they are known as perfect force actuators. The motion task must be performed as fast as possible with as little as possible residual vibrations (less then 10 μm). In this case, the performance criterium is defined over the complete length of the beam, and not in a single point. For small deflections this system can be modelled as a Bernoulli-Euler beam, for which the equations of motion for the system are given by (1):

$$\rho A \frac{\partial^2 w(x,t)}{\partial t^2} + EI \frac{\partial^4 w(x,t)}{\partial x^4} = f(x,t) \tag{1}$$

Here *w(x, t)* is the deflection of the beam in z-direction along the position x. The density of the used material is ρ, A is the area of the uniform cross-section, E is the Young's modulus of the material and I is the second moment of area of the cross-section and $f(x,t)$ represents the external distributed force applied to the beam. In theory, the system

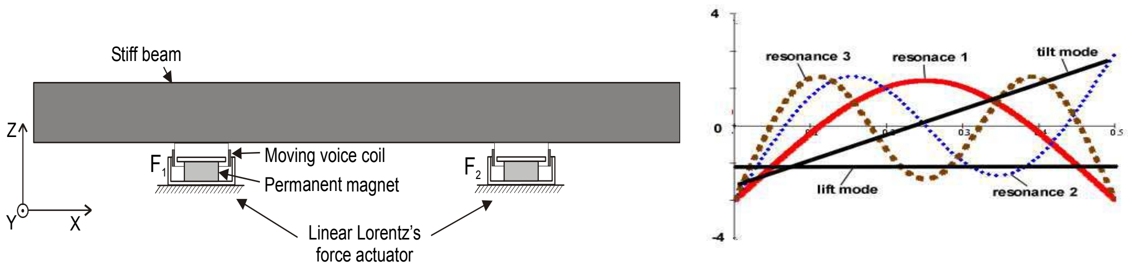

Figure 1. a) Topology of the benchmark positioning system - a levitated beam. b) first five eigenmodes of the beam.

contains an infinite number of resonance modes (see figure 1.b), given by the set of eigenvalues $\lambda_r (= \omega_r^2)$ and corresponding eigenfunctions $\phi_r (r \in \{1,2,\ldots\})$. The expansion theorem (Meirovitch 1967) relates the displacement in any point x of the construction as a linear combination of the modal coordinates (2):

$$w(x,t) = \sum_{r=1}^{\infty} \phi_r(x) \eta_r(t) \tag{2}$$

The system can be converted into a finite element description (3) (K. Bathe 1996). Since damping is hard to describe in a model, we assume modal damping structure. From (3), we can create a finite set of decoupled system equations (4), where η_r is the modal coordinate of the r^{th} mode, and ω_r is the eigenfrequency of that mode. The right-hand side of (4) represents the force input by a discrete set of n_i actuators with forces F_i at positions x_i:

$$M\ddot{q}(t) + B\dot{q}(t) + Kq(t) = f(x,t) \tag{3}$$

$$\ddot{\eta}_r(t) + 2\varsigma_r \omega_r \dot{\eta}_r(t) + \omega_r^{\ 2} \eta_r(t) = \sum_{i=1}^{n_i} \phi_r(x_i) F_i \qquad r \in \{1,2,\ldots n\} \tag{4}$$

In our setup, a beam with length of 500 mm, width of 20 mm, height of 5 mm is used. The system possesses two rigid-body modes ($\omega_r = 0$): a tilt mode and the desired lift mode. In a classical design, two actuators F_1 and F_2 would be used to drive the system (see figure 1.a).

Preliminary Calculations

In this preliminary calculation we try to estimate the positioning performance, independent on the exact type of setpoint, i.e. only based on the maximum force that can be applied. The system should perform the motion task as fast as possible, and for this reason the applied force in the feedforward will resemble a bang-bang controller. However, this is the worst-case trajectory that can be applied to the system in terms of excitation of the parasitic dynamics, but it achieves the fastest setup-time. The resulting trajectory is a piecewise second order polynomial, when applied to a simple mass system (figure 2.a). In this case, the steering force will excite the flexible modes. Dependent on the switch time, a certain level of rest vibrations will be present (figure 2b, 2c).

Since the switch time can vary in the motion task at hand, it is hard to assess performance on the basis of rest excitations. Furthermore, by applying higher order setpoints or input shaping these effects can be reduced. In all cases, the level of residual vibration depends strongly on the setpoint. The modal excitation after applying the first step is given by (5). It consists of an oscillatory part and a steady-state part. The oscillatory part is due to the change in input force and the free response of the mode. Input shaping or setpoint design can reduce this oscillatory part. The steady-state part is directly related to the instantaneous value of the actuator force. The reduction of this modal excitation during the motion task can only be reduced if the input force is lowered, which will result in a slower setpoint. Since this steady-state excitation is not setpoint dependent, it is used as a performance measure for actuator placement:

$$\eta_r(t) = \frac{\phi_r^T(p_i)}{\omega_r^2} F\left(1 - \cos(\omega_r t)\right) \tag{5}$$

The force vector F is pre-multiplied with the transposed r-th column of $\Phi(p_i)$, the mode-shape values at the actuator positions. Summing up all the steady-state contributions at a spatial point, is exactly the same as determining the static deflections for a simply supported beam under a uniform distributed force. The actuation forces are then regarded

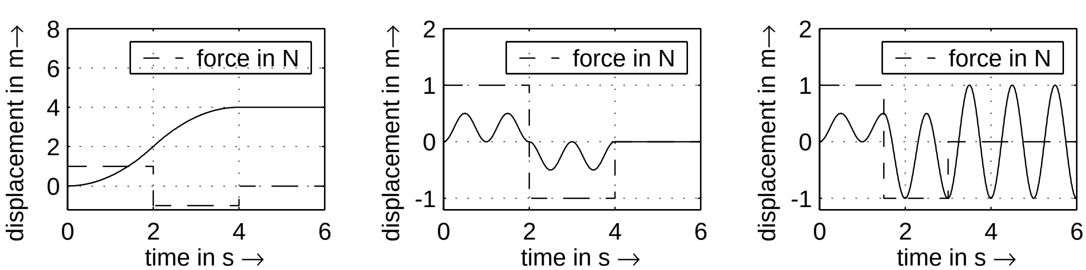

Figure 2. Bang-bang control on a single resonance system: a) rigid body mode behavior. The excitation of the vibration is dependent on choice of the switch-time in relation to the phase of the vibration: b) best-case situation. c) worst-case situation.

as reaction forces to a distributed load per length on the beam, as can be seen from (1) by setting $f(x,t) = \rho A \ddot{w}(t)$, where the acceleration $\ddot{w}(t)$ is kept constant. This distributed load is a result of the inertia forces of the beam, which are proportional to the height of a homogeneous beam.

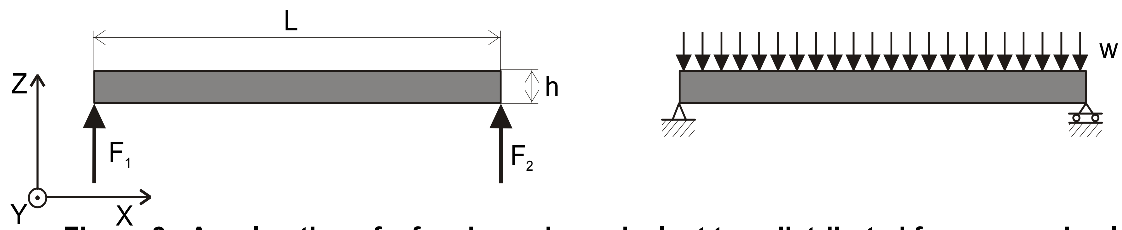

Figure 3. Acceleration of a free beam is equivalent to a distributed force on a simply supported beam.

For this static situation the maximum deflection is now a performance measure for the accuracy of the motion system, and can be easily calculated. By adding actuators (supports in the static case) and placing them in the optimal positions, the height of the beam can be calculated that keeps the maximum deflection constant. Furthermore, the force distribution between the actuators can be calculated, which can be used in the feedforward controller (C_{ff}, figure 9), as proposed in (Schneiders *et al.*, 2004). The mass of the beam as function of the number of actuators is presented in figure 6. For more actuators, the peak force for each actuator decreases even more, since this smaller actuator force is divided over more actuators. The consequences of this effect on the actuator mass will be presented in the next section.

Actuator optimization

The actuators in our laboratory setup belong to the category of short stroke Lorentz force actuators, also called voice or air coil actuators(see figure 4). They consist of a coil (usually the moving part) and a magnetic circuit: an iron core with magnet (usually the static part)

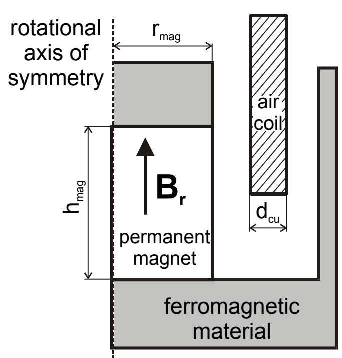

Figure 4. Topology of the air coil actuators in axisymmetrical plane.

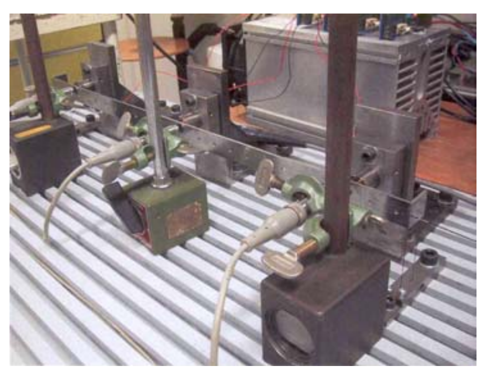
Figure 5. Flexible beam with three actuators.

The actuators are designed for application in the lightweight positioning system, and therefore, they are optimized for a minimal mass at certain force level. The optimization requires an electromagnetic and a thermal mathematical models of the actuator. Although the nature of the magnetic field as well as the heat transfer in the actuator are essentially three dimensional, the models can be reduced to two dimensions. In this case, three mathematical models are created:

- A simple equivalent magnetic circuit model, used in the optimization approach, includes the calculation of the required force by the Lorentz formula taking into account the convection heat transfer from the coil to the surrounding air.
- A finite element (FE) axisymmetric magneto static model of the actuator to verify the results of the optimization.
- A FE axisymetric thermal model also to verify the design by precise modeling.

A linear mathematical model of the actuator is used to formulate a constrained single objective optimization problem. The total mass of the actuator M_{act} is used as the objective function, defined on the nonempty feasible variable space constrained by the continuous force F_{act} as the equality constraint and double-sided inequality constraints, which limit the design parameters within the certain intervals as follow:

$$\text{Minimize} \qquad M_{act}\left\{d_{cu}, r_{mag}, h_{mag}\right\}, \bar{r} = \left\{d_{cu}, r_{mag}, h_{mag}\right\} \tag{6}$$

$$\text{Subject to: } F_{act}\left\{d_{cu}, r_{mag}, h_{mag}\right\} - F_{req} = 0,\; 0 \le d_{cu} \le 0.03,\; 0 \le r_{mag} \le 0.03,\; 0 \le h_{mag} \le 0.03 \tag{7}$$

where d_{cu} is the thickness of the air coil, r_{mag}, h_{mag} are the radius and height of the permanent magnet, respectively, and finally F_{req} is the required force of the actuator. These parameters are then used to derive other sizes of the actuator as the dimensions of the ferromagnetic core, air coil etc. The formulated constrained optimization problem can be transformed into an unconstrained problem by using the augmented Lagrangian penalty in the following form:

$$F_{ALAG}\left\{\bar{r},\bar{s},\bar{v},\bar{u}\right\} = M_{act}\left\{\bar{r}\right\} + \sum_{j=1}^{1} v_j h_j\left\{\bar{r}\right\} + \sum_{i=1}^{6} u_i\left[g_i\left\{\bar{r}\right\} + s_i^2\right] + \left[\sum_{j=1}^{1} a_j h_j^2\left\{\bar{r}\right\} + \sum_{i=1}^{6} b_i\left[g_i\left\{\bar{r}\right\} + s_i^2\right]^2\right] \tag{8}$$

where $\bar{r}$ is the vector of the design variables, v_i and u_i are the Lagrangian multipliers associated with the equality h_j and inequality g_i constraints, respectively, a_j and b_i are the penalty parameters and s_i is the additional variable introduced to convert inequalities into equality constrains (Bazaraa, 1993). The augmented Lagrangian algorithm with the proper stop criterion, multipliers update formulas, form and rate of penalty coefficients increase is used to find the unconstrained minimum of F_{ALAG}, which is the optimal solution and minimum of the constrained problem defined by eq. 6 and 7 (for more details on the search algorithm see (Makarovic, 2003).

Results

This optimization approach guaranties that the find minimum converges to the required force value, but does not ensure the uniqueness of the solution. Therefore the optimization procedure is repeated with different starting points. Additionally, the simplified model of the actuator used in the optimization introduces inaccuracies;

therefore, the obtained results are verified by precise finite element (FE) magneto-static and thermal models. The criterions for magneto static FE simulations are:

- The relative error of the calculated actuator force should not be higher than 10%.
- The magnetic field density should be in the area of the knee of the steel BH-curve.

The thermal FE simulation should confirm:

- The maximum temperature of the coil does not exceed 160°C.
- The temperature of the magnet does not decrease the value of the magnetization significantly (maximum temperature 80°C).

The FE models show that relative error of the required force, when comparing FE and simple models, does not exceed 7%. The maximum magnetic density in the ferromagnetic core is 2.1-2.3 T (the knee of BH curve).

From the thermal point of view the temperature of the coils does not exceed 150°C. The thermal FE simulations confirm that the maximum temperatures of the magnets are not higher than 75°C.

Overall mass reduction of the system

To determine the overall mass reduction of the system, the mass of one actuator and also the total mass of all actuators and dynamics as function of the number of the actuators are in figure 6 (see also table 1). It can be clearly seen that the benefits of the mass reduction are the highest for a system with three actuators. For this reason, three actuators are also applied to the prototype (figure 6).

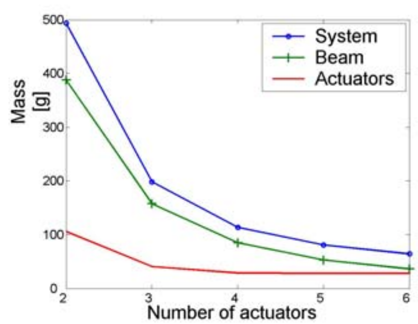

Figure 6. Mass of the whole system, beam and actuators as the function of the number of actuators applied to the system.

N_{act}	F_{req} [N]	M_{beam} [g]	M_{act} [g]	$M_{act\ tot}$ [g]
2	7	387	53.1	106.2
3	1.7	157	13.6	40.8
4	0.9	85	7.2	28.8
5	0.65	53	5.6	27.5
6	0.53	36	4.7	28.2

Table 1. Masses of beams and actuators.

Control design

Traditional design

Traditionally, two actuators would be used to drive the beam system, and also two position measurements would be used for feedback (figure 7.a). In this case collocated control is proposed for each actuator/sensor pair, which enables SISO control design. Since the motion task is regarded as the largest disturbance, actuator placement as proposed in the preliminary calculations is ideal. However, it is not possible to add any damping to the first resonance mode, since this mode is uncontrollable. This problem is

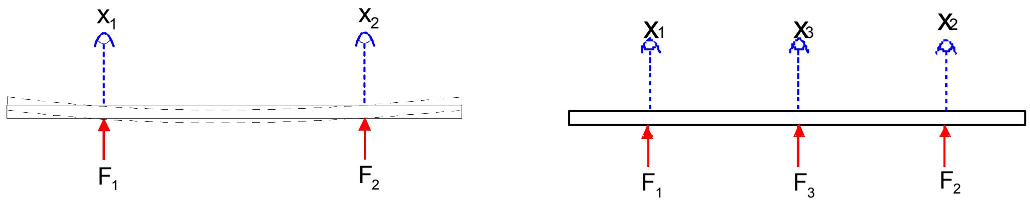

Figure 7. Actuator and sensor locations: a) Traditional design. Optimal actuator locations from the preliminary calculations are [0.23 0.77] (length of the beam normalized to 1) b) Over-actuated design. Optimal locations are [0.14 0.50 0.86].

always present in traditional designs (Schneiders *et al.*, 2003). It is hard to present a guideline to determine an optimal control bandwidth. In (Koster, 1987) optimal bandwidth for a single-resonance system is determined based on an equal level of damping for all modes of the closed-loop system. However, this is not possible for this system, since the first mode is uncontrollable. Moreover, badly damped higher modes can cause instability, since the phase lead cannot be continued for higher frequencies due to noise and actuator saturation. There should be a safe amplitude margin at the point where the phase of the open loop is crossing –180 degrees. This is often hard to design between resonance modes. Therefore, the bandwidth is traditionally designed far below the first resonance, which is located at 107 Hz for this beam of 5 mm thickness. In our case bandwidth is tuned around 20 Hz (figure 8.a), to create enough amplitude margin for robust stability. For this purpose, two controllers are tuned based on frequency-domain based loop-shaping. The controllers consist of a proportional gain with lead-lag element and a first order low-pass filter.

Over-actuated design

In the over-actuated design we allow for one extra actuator and one extra sensor. The beam thickness is now decreased to 2mm, which results in a first resonance at 43 Hz. If only feedback control is considered, it is common to use criteria based on the maximization of a certain level of controllability and observability to determine actuator and sensor positions. Then, input/ouput selection techniques can be used, which are

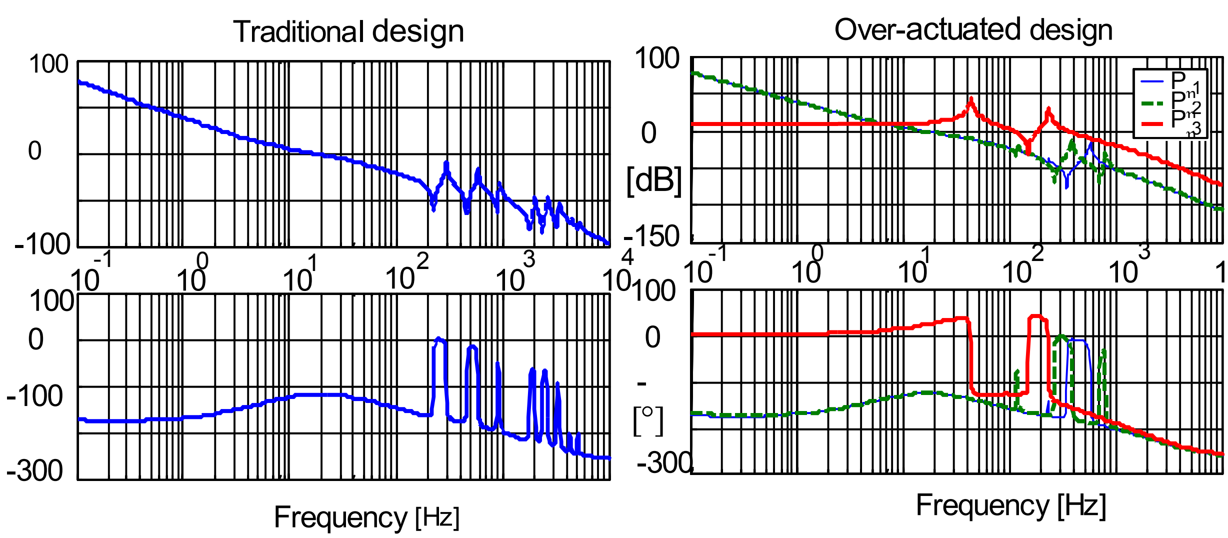

Figure 8. Open-loop responses a) for the traditional and b) the over-actuated cases.

extensively dealt with in literature ((Leleu (2000), (Hac 1993), (Moore 1981)). However, in this system the motion task itself is the largest disturbance. With little or no external disturbances, actuator placement as given by the preliminary calculations is considered to be optimal (figure 7.b). The feedforward path C_{ff} (figure 9) can be tuned to minimize excitation of resonances: the optimal ratio for the three forces is [0.33 0.34 0.33].

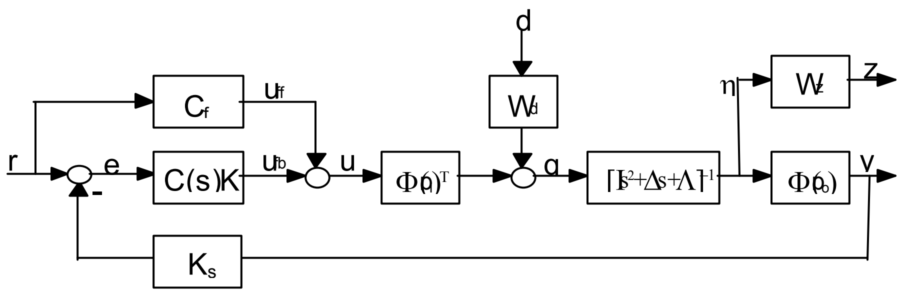

Figure 9. Control scheme for over-actuated control.

Actuator placement for the case of three actuators, as studied in the preliminary calculations (see figure 8.b), results in zero controllability of the fifth mode (third resonance mode). In general the lowest resonance will limit the performance in closed-loop. This mode is still controllable in the proposed configuration, although not optimally. Further feedback design is now focused on independent modal space control (IMSC) of the lowest modes, which are all controllable. Independent modal-space control relies on decoupled control design for a limited set of modes (Meirovitch 1983). The decoupling into modal space K_s and back from modal space K_a is based on inversion of the expansion theorem (2), see figure 9. Consequently, the actual (dynamic) controller C(s) is diagonal. In this case, the two rigid-body modes and the first resonance are now individually controlled. For this propose, all actively controlled modes should be controllable and observable. Sensor placement can be based on the maximization of some observability measure. However, higher order modes are not decoupled and show up in the three decoupled loops (figure 8.b). This effect can cause instabilities and is known as spillover. Therefore, minimal observability of the higher modes, especially the fifth, is beneficial for controller design aimed at high bandwidth. By choosing the same locations for sensor placement as for the actuator placement, this goal is achieved. With this design, the bandwidth of the two rigid-body modes is be placed at 20 Hz; this is close to the first resonance mode at 43 Hz (see figure 8.b). Regulation performance of this mode can now be separately tuned by a third loop (see figure 8.b, thick line).

Conclusions

Only considering steering (feedforward) without disturbances, a mass-reduction of 60% of the beam can be achieved, claiming the same maximum deflection for a given class of setpoint profiles. The mass of the moving coils and the total mass of the actuators are reduced by 69% and 71%, respectively, compared to the system designed by the traditional approach. Also the required total continuous and peak forces for a given acceleration profile are reduced by 63% and 54% respectively.

In the over-actuated case, it is possible to design a tracking bandwidth equal to the traditional design, with additional control over the actively controlled modes. However, total system performance cannot be completely verified by only the open-loop properties of the control loops. A more detailed analysis is needed to enable comparison of different control strategies.

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