# INTRODUCTION TO AN INTEGRATED DESIGN FOR MOTION SYSTEMS USING OVER-ACTUATION

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**Keywords:** Motion systems, Mechatronic design, vibration control, overactuation.

## Abstract

In this paper an introduction and motivation is given towards an integrated design approach for motion systems using *overactuation*. Looking to motion systems and their history, the current status of mechatronic motion systems is discussed. One of the main aspects that limit performance in mechanical systems is the presence of vibrations. An overview is presented of several passive and active methods to solve vibration problems. Active vibration control can be regarded as a form of *overactuation* to improve system performance. It is expected that an integrated overactuated design approach will be advantageous over traditional vibration control solutions, which often make use of adaptations after the mechanical design has been completed. Before a framework for an integrated design approach can be posed, different strategies of overactuation must be investigated. To study the closed-loop characteristics of resonances in mechanical systems with more actuators more closely, some first explorations of a dual-input single-output motion system are made.

## 1 Introduction

The goal of a motion system is to let a certain part of the system (the end-effector) perform a motion task. This can be the tracking of a certain setpoint or the execution of a point-to-point motion. Although point-to-point motion is the main task in practice, often trajectory-control is used to limit speed, acceleration and jerk of the moving parts. Still, mechanic resonances are likely to be excited for fast motion tasks. Examples of high-accuracy motion systems are component mounters, wirebounders and the stages in waferscanners. Trends in technology are always heading towards higher accuracies, higher densities and faster systems but costs may not rise drastically. For positioning systems this means not only accuracy must increase, but also throughput is an issue. Unfortunately these two demands are conflicting.

From a historical point of view, motion systems stem from industrial positioning systems, which are traditionally cam-driven. The design was done by mechanical engineers only. To make these systems more flexible and to meet higher specifications, actuators were added and also control engineers got involved. However, mechanical design principles and control theory have evolved separately and we believe that significant progress in the development of motion systems is possible if these two fields are integrated right from the start of the design process.

## 2 Status of control in mechatronic systems in industry

Most motion system designs use (decentralized) SISO control loops, since proper mechanical design aims at decoupling the dynamics for the desired degrees of freedom. In general, based on specifications of the process, a bandwidth is selected. The mechanical design is optimized, providing all resonances above this bandwidth. In general, this means stiff coupling between actuator and sensor positions, which will result in a relatively heavy construction. In order to design stiff constructions, high E-module materials are used, which are intrinsically badly damped. However, up to the proposed bandwidth such system behaves as a rigid body, and proper control design is relatively straightforward (SISO-loopshaping). Such controller can be simple and of low order, which is desirable for the purpose of implementation. Due to variability in production of motion systems, non-modelled nonlinearities (i.e. position dependencies) and time-varying behavior (i.e. friction), a controller has to be robust against variations in the plant model. In general, one controller design is used for all produced plants, which also induces some conservatism, which limits performance. Due to this, sophisticated model-based controllers are undesired in practice. Overviewing these facts, one can conclude that there is little integration of mechanical design and control design in industry nowadays.

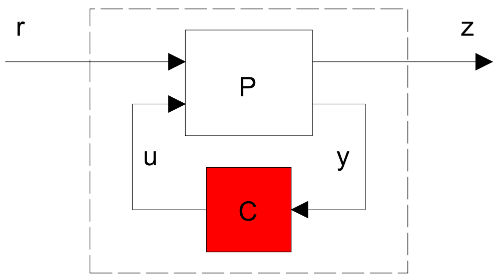



Figure 1: A motion system as a standard plant. For a control engineer the only freedom is the design of the controller $C$ to obtain the desired closed-loop performance ($r \rightarrow z$).

Some problems can be solved by proper constructive design (i.e. friction,backlash, hysteresis), others will always cause problems. For example resonances will be always present in a motion system, and will often be the main reason for limited bandwidth. For stability reasons, it is always easier to design the controller based on a *collocated* plant. In theory the closed-loop can always be stable [11] and in case an intrinsically passive controller is used, robust stability can be guaranteed. In general the controlled position in collocated control does not coincide with the position of the end-effector. The performance output of the system $z$ is different from the controlled measurement $y$ (Fig. 1). A very stiff controller will not automatically give the best performance due the resonance modes of the plant.

On the other hand, in *non-collocated* control the actuator and sensor (mostly placed as close as possible to the end-effector) placement are not the same. From the viewpoint of stability the control of a non-collocated plant is always less robust (i.e. against plant variations). But the controlled position coincides with the performance output of the system. Since performance of both collocated and non-collocated SISO-control topologies is always limited by resonances, the next chapter will give an overview of strategies to overcome resonance problems: vibration control.

# 3 Vibration control

Since vibrations are limiting the performance of mechanical systems, this chapter gives an overview of existing vibration control strategies. Vibration problems can be driven by two mechanisms: *non-resonant vibrations* induced by an external system and excitation of *structural resonances* in the mechanics. Our research is mainly focussed on the latter mechanism.

## 3.1 Passive solutions

In general vibration control can be split up in two parts; active and passive vibration control. In *passive vibration control* one tries to solve the vibration problem by modifying the plant's properties, like its stiffness, mass and damping. The available passive methods can be divided into four categories [6] as listed below:

- **Vibration isolation** tries to minimize transmission of vibrations between parts of the mechanical structure. This can be accomplished by using interconnecting elements which are sufficiently soft. One of the biggest problem in designing such *vibration isolators* or *anti-vibration mounts* is to combine good isolation properties with enough static stiffness and strength.

- **Structural redesign** can be used if vibration problems already are visible in the design stage. Along the wide range of available techniques the most important methods deal with vibration problems dominated by a single resonance mode (*detuning* and *nodalising*).

- **Improve damping** is a strategy that can be used best if the vibration problem is driven by structural resonances. The most commonly used method is to include viscoelastic materials along the mechanical construction as distributed damping system.

- **Localized additions** can be applied at discrete points in the construction. Such an extra system (a single mass or spring) can lead to the desired performance improvement of the combined mechanics. A combination of a mass and a spring can act as a vibration absorber [7].

## 3.2 Active solutions

In contrast to passive vibration control, *active vibration control* makes use of external power to drive actuators. In general these actuators are specialized for compensating vibrations and are driven by a controller (both feedback and feedforward is possible). If the vibration source is an actuator in the system itself (as in a motion system), the vibration control problem can be regarded as a motion control problem. Motion control is not regarded as active vibration control, although i.e. a notch filter (in feedback) or input shaping (feedforward) can be regarded as strategies to solve a vibration problem. Categorizing active solutions can be done is several ways and it is hard to get a complete overview.

The field of science which is interesting for our application (vibration control within a motion system) is called **structural vibration control** (i.e. [3] and [8]). Much of this work considers the use of distributed sensors and actuators (mostly piezo materials), and robust control techniques. Design is planned as modelling of the structure, modal analysis and robust controller design (mostly model based). **Modal control** plays an important role in structural vibration control. By decoupling the dynamics it is possible to control certain modes. For reconstruction of modal coordinates of a system, an observer can be used [2]. The disadvantage of using this method is that the unmodelled dynamics causes observer and control spillover. This problem can be solved by using modal filters [9]. This approach uses measured displacements and uses algebraic equations to reconstruct modal coordinates. Most distributed sensor research focuses on model-based modal sensing. Exceptions in this field use distributed sensors as spatial filters ([5] and [15]). The latter methods are less model based and are specifically interesting for our design approach.

**Sensor and actuator placement** is a topic related to structural vibration control, which is already widely explored (see [10] for lumped $I/O$ selection or [14] for a general overview). By input-output design certain modes can be made controllable/observable, while others are uncontrollable/unobservable (spillover). In general quantitative methods make use of a cost function, which is usually some combination of controllability gramians and observability gramians. These methods are actually solving an optimization problem.

Almost all of the above literature on these topics present active vibration control as a technique to increase performance by adapting the existing plant. Strategies for integrated design of motion systems are hard to find (i.e [1]) although there is industrial relevance [4].

## 4 New design approach

In order to design more efficient systems for more accurate specifications in future, we explore the potential of an integrated design approach. In such approach plant (mechanics) together with controller dynamics are designed simultaneously. Looking to both plant and controller it can be trade-off what closed-loop problems can be best solved in the mechanical design and what problem can be best solved by the controller. As long as the final closed-loop dynamical behavior (transfer from $r$ to $z$) fulfills the performance specifications, dynamic properties can be interchanged between controller and plant Fig. 2. In such new approach we keep in mind that the complete design must be robust against uncertainties, as the traditional design is. One of the advantages of such an approach is the freedom

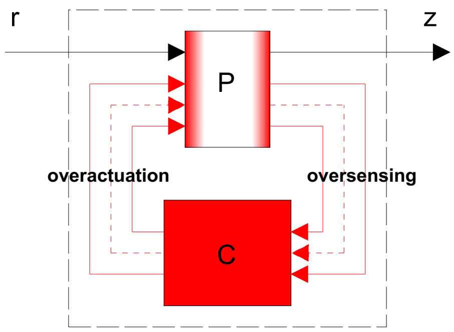



Figure 2: Created freedom in a true mechatronic design approach

in interconnecting plant and controller. This means the placement of sensors and actuators, but also their numbers are free to choose. The idea is that the use of extra actuators (and possibly extra sensors) with a properly designed controller can be beneficial for the (closed-loop) system in several ways.

Since active vibration control implies adding extra actuators this is a way of over-actuation. Why should we use over-actuation in the first place? If we look to over-actuation as a way of implementing active vibration control there are several advantages compared to passive vibration control:

- Further improving the mechanical design can become expensive (i.e. high-quality parts), and will give rise to the product price.
- Compared to passive methods, active vibration control can change the closed-loop dynamics in a variety of ways. However, active methods are in general more expensive and are only considered when passive means have been exhausted.

Traditionally, increased performance is reached by mechanical redesign and or passive add-ons. Controller design is kept as simple as possible. To make real mechatronic improvement, combined redesign of both mechanics and controller must be considered. Having the freedom of designing plant and controller at the same time it is to be expected that we can fully benefit from active vibration control. To make use of the benefits of mechanics and control and to interchange them in the closed-loop system, we have to use of extra actuators (and sensors). However, at this moment we are not able to come up with such a design approach. Therefore small steps are taken to get more insight in the use of extra actuators on mechanical systems. Before a promising design procedure can be optimized, several general topological decisions have to be made before we can pose a certain design procedure.

## 5 Overactuation

To see what overactuation can contribute to an integrated design approach, some first explorations are made using two actuators. Although not often encountered in industry, intuitively the effect of multiple actuators in a mechanical structure must be advantageous in handling resonances. Extending the freedom in choosing multiple inputs and outputs for the controllers almost gives an unlimited freedom in the design if no further constraints are included. However, the focus in this research will be on a limited number of actuators to increase performance. For several reasons the number of actuators will be limited in an industrial motion system:

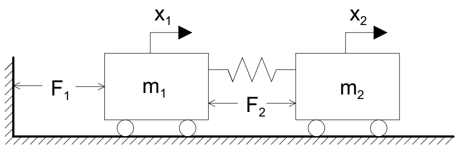
- it keeps design relatively simple, following the trend in industry nowadays
- looking to the costs in production, extra actuators imply extra work but also extra complexity in assembling the system
- constraints for the mechanical design (i.e. thermal and geometrical constraints), will limit the number of actuators

Using MIMO design methods (i.e. $H_2$, $H_\infty$ and $\mu$-synthesis) little insight in the design process is obtained and the resulting compensators are often of high order. In order to get more feeling for the problems resonances cause in closed-loop, more pragmatic methods tested on simple systems are better to start with. For DISO-plants (Dual-input, single output) a design approach exists [13] based on SISO loopshaping, which is also known in industry [12]. This PQ-method splits up the design problem into two SISO design problems. This design method is especially suited for systems with a fine and a coarse actuator, since the individual contribution of both actuators can easily be tuned. However, in what way both actuators change the closed-loop characteristics of resonances is not completely clear.

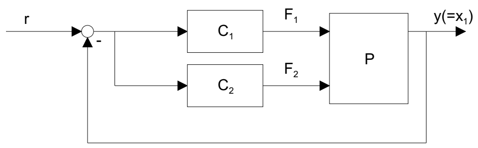
### 5.1 First exploration

To explore the effects of resonances in closed-loop using more actuators, we focus on a ideal lumped parameter model of a 1-

DOF motion structure (Fig. 3) which represents the behavior of a real structure for frequencies just beyond the first resonance.




(a) Schematic representation of the lumped parameter model




(b) General control diagram for a DISO plant

Figure 3: 1-DOF mechanical system with one resonance mode

First objective is to eliminate the effect of the resonance, using the combination of 2 actuators and only 1 sensor. To study this effect the control structure as depicted in Fig. 3.b is used. The open-loop transfer from $r$ to $y$ can be expressed as Eq. 1. Note that in this description damping $b$ is added (parallel to the stiffness $k$) but this value will be small in practice.

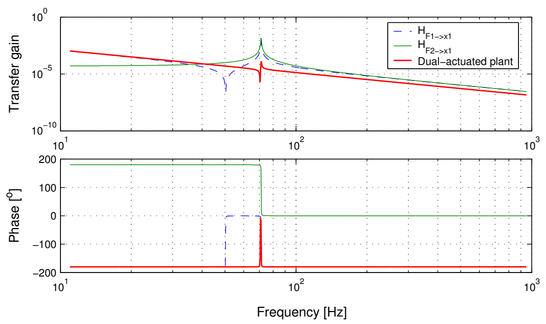



Figure 4: Bode plots for both transfers separate and for the *overactuated* situation (2% mismatch in tuning gain $K$).

$$H_{ol} = \frac{(1-\frac{C_2}{C_1})m_2 s^2 + bs + k}{s^2(m_1 m_2 s^2 + b(m_1+m_2)s + k(m_1+m_2))} C_1 \quad (1)$$

This equation makes clear that exact cancellation of the first resonance in the overall plant transfer (from $u$ to $x_1$) is possible for a certain relation between the two actuator forces (Eq. 2). Note that this static gain difference between the first and second actuator is the only way to exactly cancel the resonance.

$$\frac{C_2}{C_1} = \frac{m_2}{m_1+m_2} = K \approx 1 - \frac{\omega_{ar}^2}{\omega_r^2} \quad (2)$$

$$P^* = \frac{1}{(m_1+m_2)s^2} \frac{(1-K)m_2 s^2 + bs + k}{\frac{m_1 m_2}{m_1+m_2}s^2 + bs + k} \quad (3)$$

For a single resonance system this value is uniquely given by the mass-distribution of $m_1$ and $m_2$. In the lightly damped case, $K$ can be calculated from the resonance and anti-resonance frequencies, which can be determined experimentally.

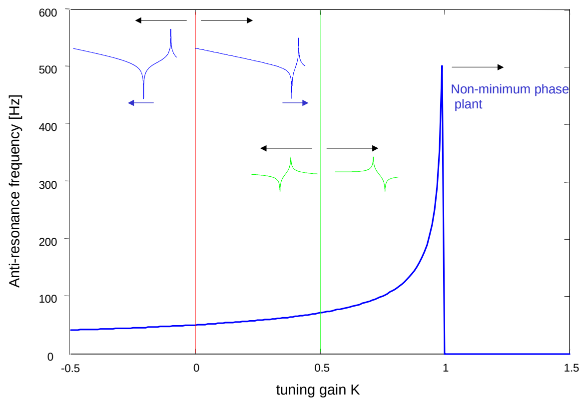



Figure 5: Anti-resonance frequency as a function of tuning gain K. Positive gain shifts the anti-resonance towards the resonance.

From Eq. 3 it is clear that $K$ changes the value of the complex zero, which corresponds to a shift of the anti-resonance in the plant transfer. From Fig. 5 it becomes clear that a positive $K$ move the anti-resonance towards the resonance. Exceeding the value for exact cancellation ($K = 0.5$ in the case of equal masses as in Fig. 5) will result in a resonance/anti-resonance pair in the plant transfer function. This will lead to a $90°$ phase lag, which is undesirable for control design. The obtained effect is the same as tuning a *skew notch* filter (actually a complex double lead filter), however now only one parameter has to be tuned, whereas the double lead filter counts at least 3 independent tunable parameters. Note that the value of the anti-resonance $\omega_{ar}$ is very sensitive to changes in the domain $K \in < 0, 1 >$ which is mandatory for exact cancellation.

Next the adapted plant $P^*$ is evaluated in collocated closed-loop. In general concerning the SISO case (without $C_2$ being actuated) a PD-like controller is implemented in motion control problems. Normally higher performance is gained by loop-shaping. Using such a controller acting on $F_1$ (which is equivalent to fixing the structure at position $x_1$ to the setpoint), all resonances will benefit from the added damping in the closed-loop situation. By using the second actuator as given in Eq. 1 and Eq. 2 ($C_2 = C_1 K$), the benefits of the damping in the closed-loop completely disappears (Fig. 8). However in the sensitivity closed-loop transfers this effect is strongly cancelled by the influence of the zeros of $\omega_{ar}$ which is very close the the pole if $K$ is properly tuned. To show this effect, a root-locus of the closed-loop system is presented in Fig. 6.

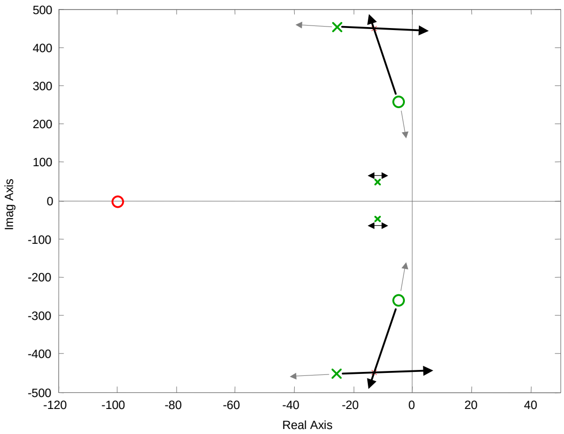



Figure 6: Root-locus of the closed-loop system. Increasing $K$ moves the complex poles towards the imaginary axis (decreasing damping drastically). Zeros increase in frequency and around the point of exact cancellation, closed-loop damping equals the mechanical damping $b$. For negative $K$ (grey arrows) damping further increases. Note that the value of $K$ has minimal influence on the controller induced complex poles.

This diagram shows the loci as function of the tuning gain $K$ of the closed-loop system of Fig. 3.b. For weakly damped systems the poles can even become unstable for overtuned gains. As long as we focus on sensitivity or closed-loop transfers we do not see any problems. However, concerning the transfers of the process sensitivity (Fig. 7), zero-pole cancellation is not the case any more, equal to the situation tuning a normal notch filter. Any disturbances on the actuator inputs or offsets between to two actuators will cause excitation of the badly damped resonance.

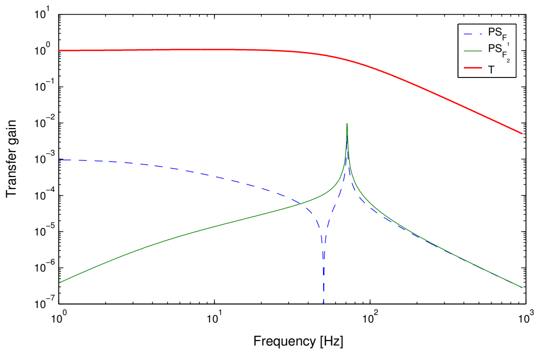
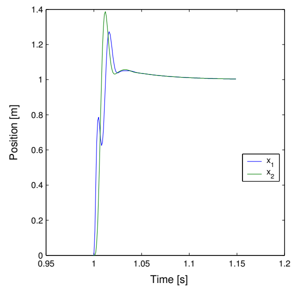


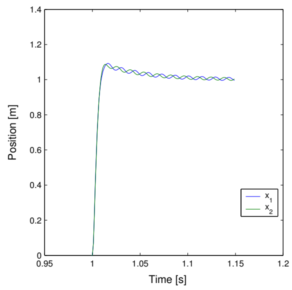

Figure 7: Closed-loop and process-sensitivity transfers for exact cancellation. Although the closed-loop transfer $T$ seems promising, both process-sensitivities suffer from the badly damped resonance.

## 5.2 Conclusions

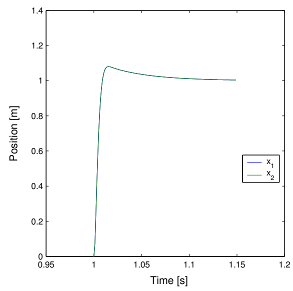
Looking to the overall damping it would even be better to use a negative value of $K$. Although the difference between $F_1$ and $F_2$ is static, damping is increased (Fig. 6). This can be understood concerning resonances in a lumped parameter model. If we want to avoid vibrations, parts between the compliance (stiffness element) must be actuated with the same force (Fig. 9.a). For every real-life system the ratio between the forces strongly depends on the placement of the actuators. Neglecting how to control and what measurement has to be used, the only way to dampen out resonances is to use two forces in opposite direction (Fig. 9.b). The way how this is achieved is completely free since the use of overactuation and oversensing may allow for different solutions.



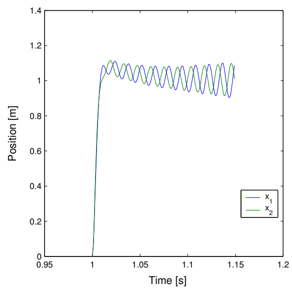

(a) $K = 0$ (1 actuator)




(b) $K = 0.49$ (badly damped, 2% gain mismatch)



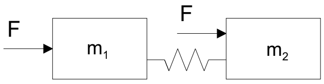

(c) $K = 0.50$ (exact cancellation)



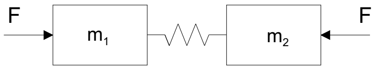

(d) $K = 0.53$ (instable behaviour)

Figure 8: Closed-loop time-domain results: step responses

If exact cancellation is achieved, the resonance cannot be excited by the reference signal. However, the drawback of this effect is the lack of controllability of that resonance mode. Since the resonance can not be excited by the controller, there is also no way to reject the mode. This is equivalent to the spillover problems when using modal control strategies. Overviewing this, *resonance cancellation* is better suited for feedforward control. Known disturbances (so also trajectories in motion systems) will not excite the resonance. For feedback control, we prefer a negative gain $K$, which will result in a better damped closed-loop system without affecting the bandwidth (the damped frequencies of the poles will not change (see Fig. 6). Fortunately, the value of $K$ (for both negative and positive values) does almost have no influence on the disturbance rejection (for low frequencies).




(a) Avoid resonance




(b) Dampen out resonance

Figure 9: Two different ways to influence a compliance

## 6 Discussion and future work

Although current mechatronic systems are examples of state-of-art technology, more integration in the design strategies is still possible. Also for motion systems this integration can be increased. Historically there is a huge gap between mechanical and control design, and a new way of thinking is needed to overcome this. This paper only presents some first ideas, and a lot of work must be carried out to really come to a new design approach. From the exploration it may be clear that the benefits of using extra actuators is strongly depending on design-topology and application field. Further research will be focused on reducing the bandwidth-limiting effects of resonances. Modal filters and modal control strategies will be explored, since these techniques reveal a lot of insight compared the general MIMO-design techniques. Once a good conceptional design is found, a next step is to come up with an optimization of both mechanical and control part of the total system. Objective of the optimization can be the reduction of moving mass in a complete motion system, which is the goal in our current research project.

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