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ACTIVE DAMPING BASED ON DECOUPLED COLLOCATED CONTROL

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INTRODUCTION

High-precision machines typically suffer from small but persistent vibrations. As it is difficult to damp these vibrations by passive means, research at the Drebbel Institute at the University of Twente is aimed at the development of an *active structural element* that can be used for vibration control. The active structural element, popularly referred to as 'Smart Disc', is based on a piezoelectric position actuator and a piezoelectric force sensor.

One of the main problems in active control is to ensure stability. In this respect it is often advantageous to consider the use of so-called *collocated* actuator-sensor-pairs, as this enables to actively implement a passive control law, which is robustly stable, irrespective of structural modeling errors. Within the context of vibration control for lightly damped structures, collocated actuator-sensor-pairs are known to be well suited to obtain robust active damping [1]. The Smart Disc concept, based on a position actuator and a collocated force sensor, as such may be used to provide robust active damping within high-precision machines [2].

Control based on collocated actuator-sensor-pairs is inherently in terms of 'local' coordinates. Vibration problems, however, are usually analysed in terms of 'modal' coordinates, corresponding to a limited number of vibration modes, as captured in a simplified model of the mechanical structure. It can be shown that decoupling of collocated actuator-sensor-pairs, i.e., the transformation of the original control problem into modal coordinates, yields control loops that again enable the implementation of a passive control law. Stability of 'decoupled collocated control' thus does not depend on the accuracy of the model that has been used for decoupling.

In the present paper we will illustrate 'decoupled collocated control' based on the Smart Disc concept, by means of experiments performed on an industrial high-precision machine. We will show that the main benefit of decoupling for active damping is not that much in improving the performance, but in gaining *insight* in the control problem. Once this insight has been established, the damping of the targeted vibration modes can be tuned in a relatively simple manner.

WAFER STEPPER LENS VIBRATIONS

The industrial high-precision machine we will consider in this paper, is the advanced microlithography system referred to as *wafer stepper*, that is at the heart of Integrated Circuit (IC) manufacturing. Microlithography is used by IC manufacturers to transfer a circuit pattern from a photomask to a thin slice of silicon referred to as the *wafer*, from which the ICs are cut out in the end. The circuit pattern is projected onto the wafer through a carefully constructed lens, which is in fact a complex system of lenses (Fig. 1). The most important variable to control in the lithography process is the line width of the circuitry on the wafer, as this width has direct impact on the IC speed and performance. The current IC line width is about 0.1 μm.

One of the bottlenecks in decreasing the line width, and thus in the miniaturization of ICs, may in future be caused by badly damped micro-vibrations of the lens of the wafer stepper. Up till now, micro-vibration problems within high-precision machines could often be relieved by means of adequate *isolation* of equipment from the floor, through which most of the disturbing vibrations enter. However, once the equipment is sufficiently isolated from floor vibrations, an other disturbance source becomes dominant: acoustics. It is practically impossible to come up with isolation means for acoustic vibrations, for instance because a well-conditioned airflow is required for 'thermal management' within the machine. *Damping* of the lens vibrations by passive treatments has also turned out to be practically impossible. The wafer stepper thus constitutes a challenging test-bed for evaluation of the active damping potential of the Smart Disc concept.

In order to have a close look at the troublesome lens vibrations in the wafer stepper, Fig. 2 schematically depicts the parts of the wafer stepper that are important to us. Besides the lens, this figure shows the main-plate, which serves as a positional reference for all other parts of the machine. The main-plate is resiliently isolated from the floor, both passively and actively, by means of so-called airmounts. The lens is held in a flange, which is connected to the main-plate by means of three symmetrically located (passive) lens support blocks, only two of which are in sight in Fig. 2.

The lens support blocks are 'simple' steel blocks, equipped with flexure hinges, designed as much as possible according to kinematic design principles, in order to prevent the position of the lens being overconstrained. As a consequence, the overall stiffness of the lens suspension, and the related resonance frequencies of the machine, can not be increased infinitely.

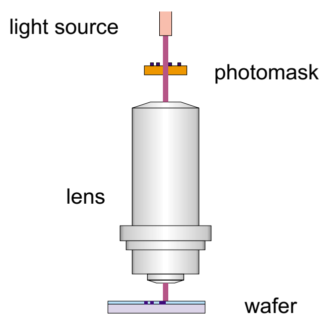

Figure 1 Wafer stepper principle

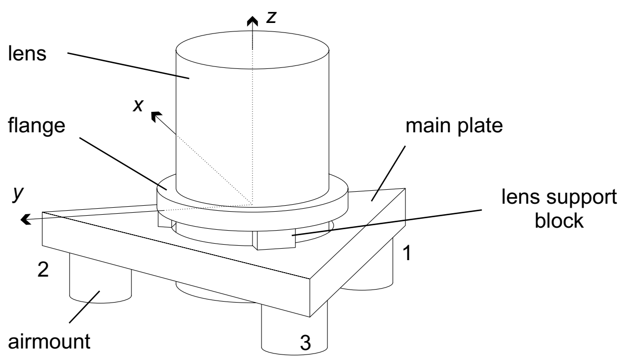

Figure 2 Schematic view on wafer stepper

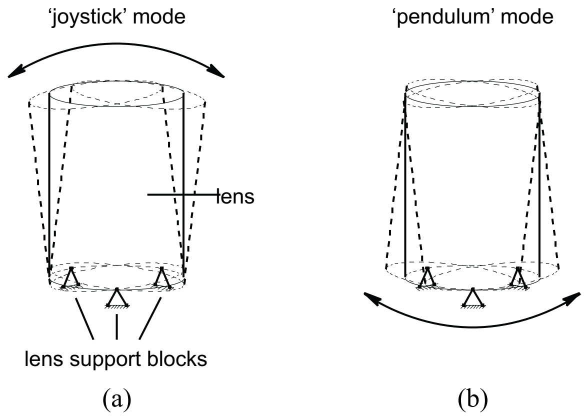

Figure 3 Dominant vibration modes of the lens relative to the main-plate

The two lowest suspension modes of the lens are referred to as the *joystick modes* (at about 100 Hz). They correspond to tilt of the lens, relative to the main -plate, i.e., relative rotation of the lens around two perpendicular axes in the plane of mounting (Fig. 3a). The joystick modes are mainly due to the limited *vertical* stiffness of the lens support blocks. In the sequel we will refer to these modes as φ_x and φ_y (rotation around respectively the *x*- and the *y*-axis).

Two suspension modes of the lens that are also of interest, are referred to as the *pendulum modes* (at about 300 Hz). In these modes the lens moves approximately horizontally with respect to the main-plate in the plane of mounting (Fig. 3b). The pendulum modes are mainly due to the limited *tangential* stiffness of the lens support blocks. In the sequel we will refer to these modes as T_x and T_y (translation, more or less, along respectively the *x*- and the *y*-axis).

The final two suspension modes of the lens (not shown in Fig. 3) are *rotation* around the *z*-axis and the *translation* mode along the *z*-axis (both also above 300 Hz), in the sequel respectively referred to as φ_z and T_z .

Piezo Active Lens Mount. In order to be able to damp the six suspension modes mentioned above, the lens support blocks have been equipped with Smart Disc functionality in two perpendicular directions [3]. A picture of the resulting 'Piezo Active Lens Mount' (PALM) is shown in Fig. 4. The PALM consists of a monolithic steel 'flexure block', in which two piezoelectric stacks have been glued.

The flexure block is symmetric with respect to the *z*-axis. It has been equipped with:

- two so-called 'accordion springs', designed so as to be slightly in tension after gluing of the piezoelectric stacks, in order to provide a compressive elastic preload force for the piezoelectric actuator;
- four flexure hinges, providing the required elastic degrees of freedom between the upper and the lower part of the flexure block, so as to relief the piezoelectric stacks from shear and tilt forces.

The piezoelectric stacks both consist of a position actuator and a force sensor. The black arrows in Fig. 4 indicate the direction along which the actuator may expand, and along which

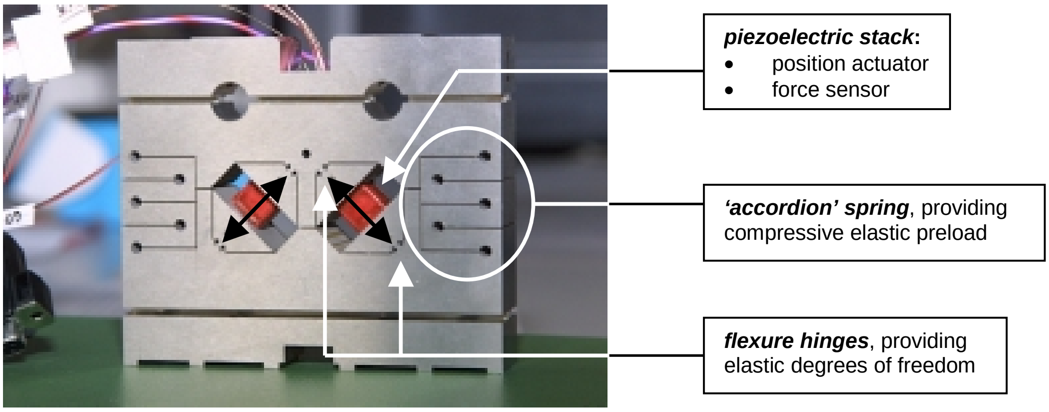

Figure 4 Piezo Active Lens Mount

the force is measured. The orientation of the stacks is such that the gravitational load, due to the mass of the lens, is equally distributed over the individual stacks. From Fig. 4 it may be deduced that:

- if the actuators are operated *in-phase*, the upper part of the flexure block moves *vertically* with respect to the lower part;
- if the actuators are operated *out-of-phase*, the upper part of the flexure block moves *horizontally* (or tangentially) with respect to the lower part.

Similarly:

- the *sum* of the measured forces on the individual sensors is a measure for the *vertical* mechanical load;
- the *difference* of the measured forces is a measure for the *shear* (or tangential) load.

The piezoelectric actuators have been designed for a stroke of 50 nm under normal operation, and a maximum stroke of 0.50 μm. Due to the fact that the piezoelectric actuators operate along perpendicular axes, the vertical stroke and the horizontal stroke of the PALM are equal.

ACTIVE DAMPING WITHIN THE SMART DISC CONCEPT

In order to describe the concept of Smart Disc-based active damping, consider a single piezoelectric stack within the entire mechanical structure (schematically depicted in the upper part of Fig. 5). The piezoelectric stack is modelled as a stiffness element k_s in series with a position actuator, embedded in a mechanical structure $P(s)$. The elastic force that is present in the piezoelectric stack is measured (F_{sens}), and fed to the Smart Disc controller $C(s)$, which in turn should generate a desired position for the actuator (x_{act}), so as to damp the measured vibrations.

In order to achieve robust active damping, the only model knowledge that is needed, is the fact that the position actuator and the force sensor are collocated. By 'collocation' we mean that the associated signals for the actuator and the sensor are 'power-conjugated': the product of the actuated *velocity* and the measured *force* represents the *power* that is extracted from the mechanical structure. This implies that, if we impose a linear relation between the measured force F_{sens} and the actuated velocity v_{act} (in Fig. 5: $H(s) = K_p$), we are effectively

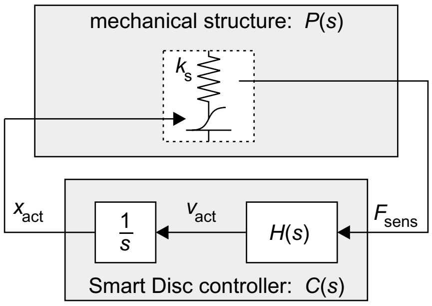

Figure 5 Illustration of power dissipation based on Integral Force Feedback within the Smart Disc concept

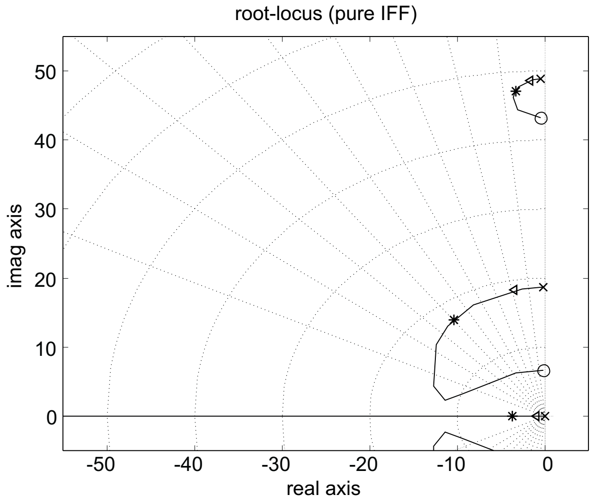

Figure 6 Example of a root-locus upon application of Integral Force Feedback

implementing a mechanical damper (valued K_{p}^{-1}). It is easily seen that this behavior is achieved by incorporating an integrator in the feedback loop $C(s)$:

$$C(s) = \frac{x_{\mathrm{act}}(s)}{F_{\mathrm{sens}}(s)} = \frac{K_{\mathrm{p}}}{s} \quad \rightarrow \quad x_{\mathrm{act}}(s) = \frac{K_{\mathrm{p}}}{s} F_{\mathrm{sens}}(s) \quad \rightarrow \quad v_{\mathrm{act}}(s) = K_{\mathrm{p}} F_{\mathrm{sens}}(s) \tag{1}$$

This active damping strategy is referred to as *Integral Force Feedback* (IFF) [4,1].

In order to take, in addition to this energy-based approach, a brief look from the perspective of controller design, consider the open-loop Smart Disc frequency response:

$$P(\mathrm{j}\omega) = \frac{F_{\mathrm{sens}}(\mathrm{j}\omega)}{x_{\mathrm{act}}(\mathrm{j}\omega)} \tag{2}$$

In the absence of structural damping, the pole-zero-map of this response is characterized by poles and zeros on the imaginary axis. For a collocated actuator-sensor-pair, it can be shown that the pole-zero-map exhibits an *alternating* pole-zero-pattern [1]. This is shown in Fig. 6 for a mechanical structure with two vibration modes (i.e., two pairs of poles and zeros). By adding an extra pole in the origin (the integrator in the feedback loop), all braches of the root-locus are drawn into the left half of the *s*-plane, which implies that all resonances are damped (robustly, as hardly any model knowledge has been used).

In Fig. 6 the location of the closed-loop poles is shown for two values of the feedback gain: $K_{\mathrm{p}} = \beta$, indicated by the triangles, and $K_{\mathrm{p}} = 3\beta$, indicated by the stars. It can be seen that initially, up to a certain level, a higher feedback gain yields higher damping. Beyond this level the closed-loop poles tend to move towards the open-loop zeros on the imaginary axis, and damping decreases again.

Maximizing damping for a single vibration mode. If the Smart Disc frequency response $P(\mathrm{j}\omega)$ is dominated by a single vibration mode, the open-loop transfer function of the mechanical structure in series with the IFF-controller can be denoted by:

$$C(s)P(s) = K_{\mathrm{ol}} \frac{s^2 + \omega_{\mathrm{a}}^2}{s(s^2 + \omega_{\mathrm{e}}^2)} \tag{3}$$

with:

- ω_{e} : the resonance frequency of the dominant vibration mode;
- ω_{a} : the dominant anti-resonance frequency ($\omega_{\mathrm{a}} < \omega_{\mathrm{e}}$), which can be shown to correspond to the dominant resonance frequency of the mechanical structure if the Smart Disc were removed from the structure [1];
- K_{ol} : the overall open-loop gain, i.e., the product of the feedback gain K_{p} and the high-frequency level of the Smart Disc response $P(\mathrm{j}\omega)$.

It can be shown [5] that the maximum achievable modal damping in this case is given by (for $\omega_{\mathrm{a}} > \frac{1}{3}\omega_{\mathrm{e}}$):

$$\zeta_{\max} = \frac{\omega_{\mathrm{e}} - \omega_{\mathrm{a}}}{2\omega_{\mathrm{a}}} \tag{4}$$

which is obtained for:

$$K_{\mathrm{ol}}^{\max\zeta} = \omega_{\mathrm{e}} \sqrt{\frac{\omega_{\mathrm{e}}}{\omega_{\mathrm{a}}}} \tag{5}$$

It should be noted here that, for increasing feedback gain, whereas initially the damping increases, the effective *stiffness* of the mechanical structure *decreases*. Tuning of an IFF-controller thus involves balancing of a ‘damping-versus-stiffness’ trade-off. Optimal balancing of this trade-off in general does not correspond to maximization of the damping, but rather to tuning the open-loop gain to a level between 20% and 80% of $K_{\mathrm{ol}}^{\max\zeta}$, depending on the particular vibration problem at hand [6].

Local active damping applied to a piezoelectric stack. In order to examine to what extent the above controller tuning strategy can be applied to the PALMs in the wafer stepper lens support, Fig. 7 shows a typical measured collocated response for a single actuator-sensor-stack. (Due to the symmetry in the set-up, all collocated responses are similar.) It can be seen that this response is *not* dominated by a single vibration mode. It rather consists of the contribution of multiple modes, the most important of which are:

- a joystick mode (at about 70 Hz; the other joystick mode is uncontrollable from this stack)
- a pendulum mode (at about 170 Hz; the other pendulum mode again is uncontrollable)
- the vertical translation mode (at about 170 Hz, close to the pendulum modes)
- the rotation mode around the *z*-axis (at about 270 Hz)

In comparison to the passive lens support, the natural frequencies of the suspension modes have decreased significantly, due to the fact that the effective stiffness of the (passive) PALM is lower than the stiffness of a conventional lens support block.

From Fig. 7 the collocation can clearly be observed, by the alternating pattern of resonance and anti-resonance frequencies. It can also be seen from the phase plot, which is bounded

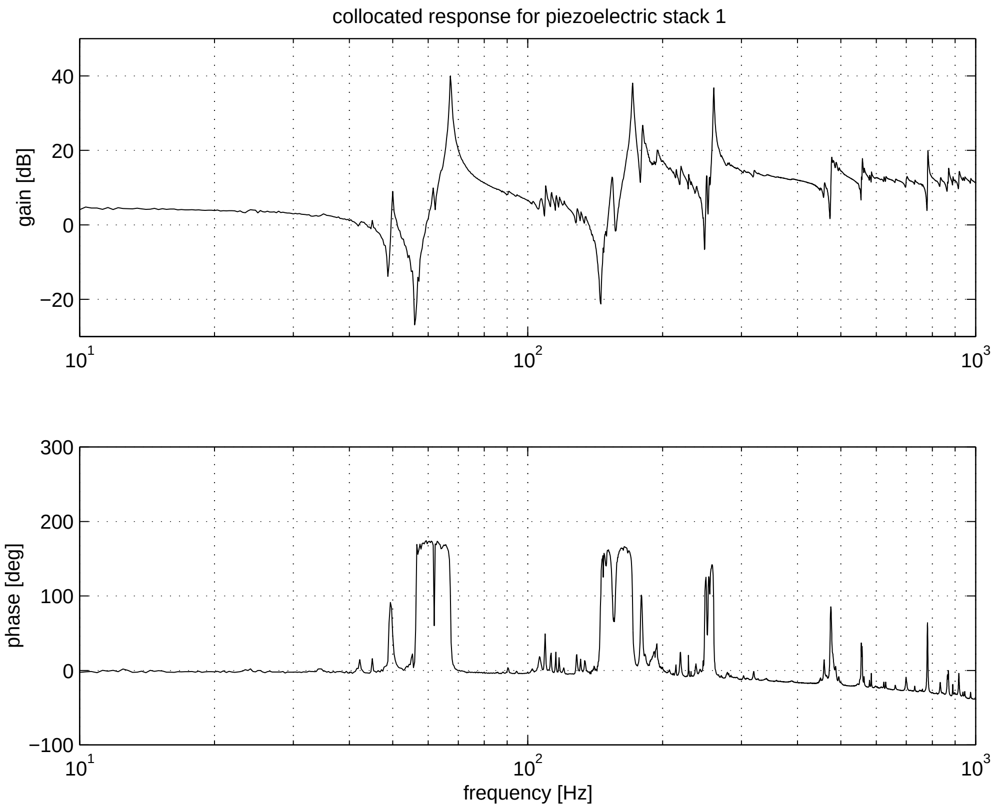

Figure 7 Typical collocated reponse for a single actuator-sensor-stack within a PALM

between 0° and 180°. From the phase plot in Fig. 7 it can however also be seen that stability (guaranteed in theory) may in practice be endangered by additional phase lag for high frequencies. The phase lag is mainly due to the actuator amplifier electronics and the anti-aliasing filter (necessitated by the digital implementation of the Smart Disc controller; sampling frequency at 10 kHz). This implies that, upon closing a collocated control loop, closed-loop stability should in practice always be checked from open-loop response.

In order to examine the difference between a collocated response and non-collocated responses, consider Fig. 8, which shows the response for all six local force sensors in the lens suspension, due to actuation by one and the same stack (actuator 1). In these plots it can clearly be seen that the beneficial alternating pole-zero pattern is *not* present in the response between an actuator and a sensor within different piezoelectric stacks.

Furthermore, from both Figs. 7 and 8 it can be seen that, in addition to the suspension modes, the dynamic behavior of the wafer stepper is affected by many more vibration modes. Despite the fact that these modes are not dominant, the presence of these modes, in combination with the lack of damping, considerably complicates model-based controller design. If these modes are not accounted for in the model, they may easily give rise to closed-loop instability [1]. As illustrated previously, instability due to unmodelled modes does not occur for active damping based on local IFF-control.

In Fig. 9 we have shown the effect of IFF applied to a single piezoelectric stack, for two values of the feedback gain (left plot: $K_p = 100$, right plot: $K_p = 200$). A higher gain

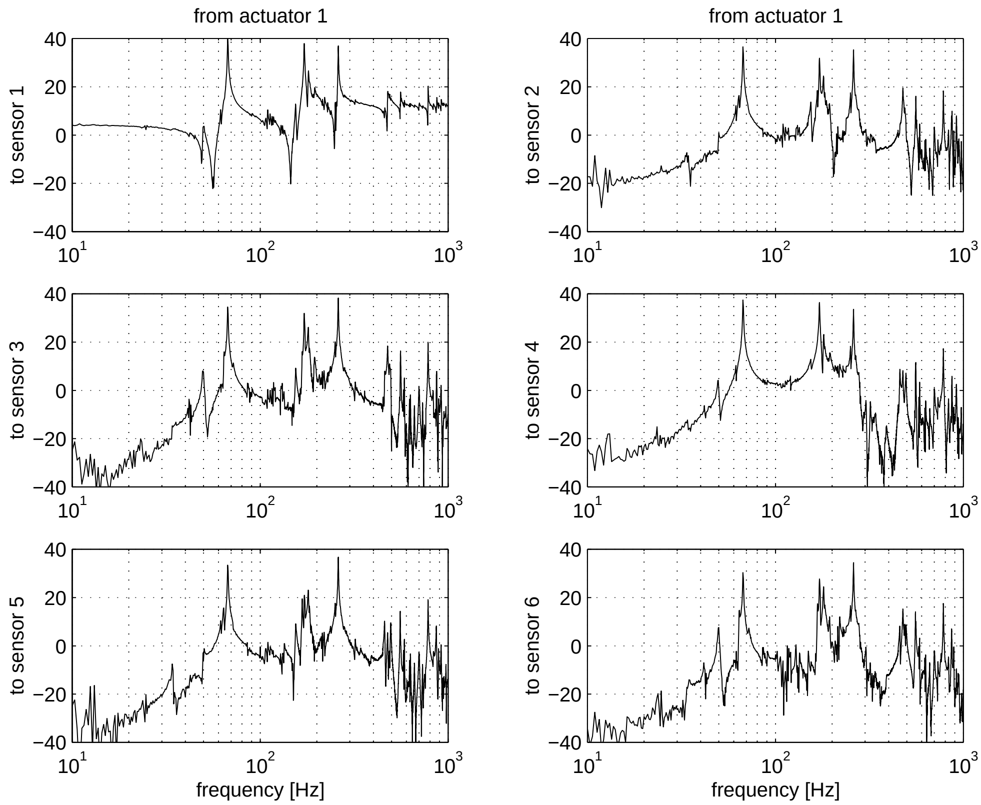

Figure 8 Frequency responses between a single position actuator and all six force sensors

obviously results in a higher open-loop response, as indicated by the dash-dotted curves in the upper plots in Fig. 9. Closing the feedback loop 'flattens' the part of the response that is above 0 dB (the dashed straight line in the upper plots), and yields the closed-loop response as indicated by the solid curves in the upper plots. It can be seen that for a higher gain, the resonance peaks are lifted higher, and thus 'flattened' more upon closing of the loop.

Closed-loop stability can easily be checked from the open-loop response by means of the dashed lines in the Bode plots. At about 800 Hz the phase drops below -180°, which implies that beyond this frequency the magnitude of the open-loop response should be kept below 0 dB. For the two values of the gain considered, it can be seen that the closed-loop is stable indeed.

MODAL ACTIVE DAMPING BASED ON DECOUPLING

The main benefit of local IFF has been stated to be the robust stability associated to the collocation. The main drawback of local control, however, is that it is not straightforward to tune the gain for the individual local IFF controllers, so as to achieve optimal damping for the six suspension modes. From the perspective of vibration control, it would be desirable to have six independent feedback loops, each of them corresponding to a single suspension mode, the damping of which can then be tuned independently. To that end, 'modal decoupling' is needed. Perfect modal decoupling however requires a detailed model of the dynamic behavior of the mechanical structure. As we have seen in Fig. 7, the mechanical structure may exhibit

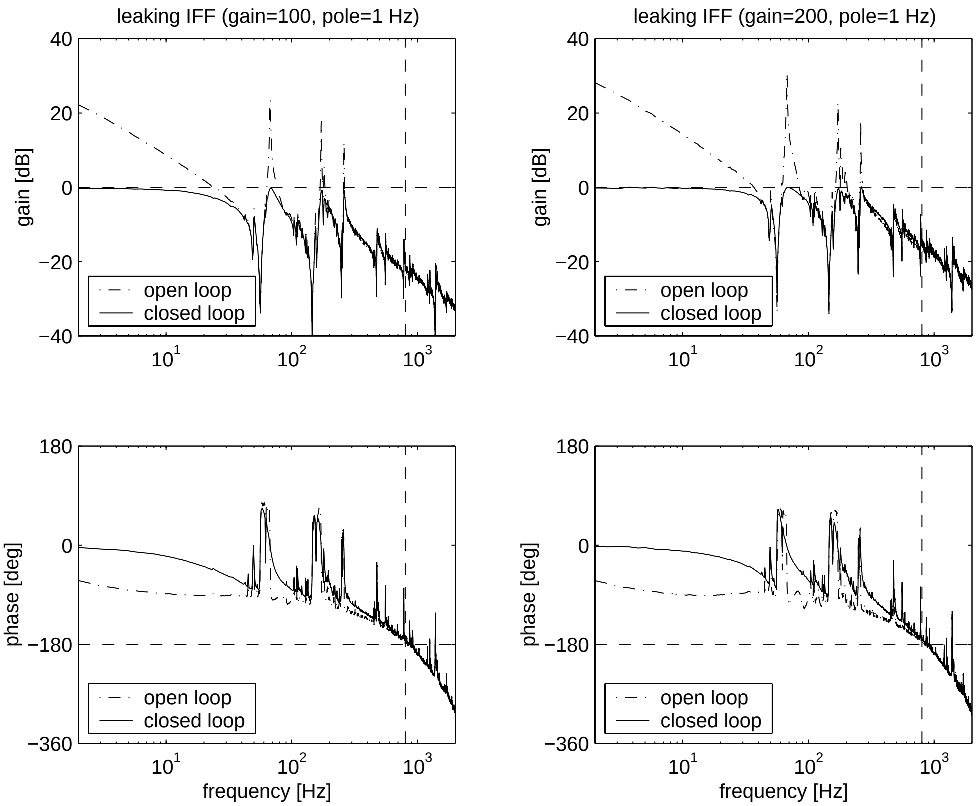

Figure 9 Integral Force Feedback applied to a single piezoelectric stack

far more modes than the vibration modes of interest. This makes it practically impossible to perform perfect decoupling.

The above-mentioned problem can be solved by realizing that decoupling of collocated actuator-sensor-pairs, i.e., the transformation of the original control problem into modal coordinates, yields control loops that again enable the implementation of a passive control law. In other words: *stability of 'decoupled collocated control' does not depend on the accuracy of the model that has been used for decoupling* [6].

We will show the use of this important insight for the PALMs in the wafer stepper lens suspension. We have performed straightforward 'intuitive' decoupling, in the sense that:

- we assumed the behavior (i.e., the gain) of all individual actuators and sensors to be equal;
- we assumed rotational symmetry for the set-up.

Furthermore we have 'tuned' only two parameters in the model, representing the level of the horizontal plane for the axes of rotation for the respectively the joystick modes (φ_x, φ_y) and the pendulum modes (T_x, T_y). Tuning of these two parameters was aimed at minimization of the mutual influence between the diagonal terms of the resulting 6x6-matrix of frequency responses. The two diagonal 3x3-blocks of this matrix are shown in:

- Fig. 10, indicating the mutual influence between the 'rotation' modes ($\varphi_x, \varphi_y, \varphi_z$);
- Fig. 11, indicating the mutual influence between the 'translation' modes (T_x, T_y, T_z).

From the diagonal elements in Figs. 10 and 11, the mutual influence between the joystick modes and the pendulum modes appears to be negligible: the pendulum modes are hardly

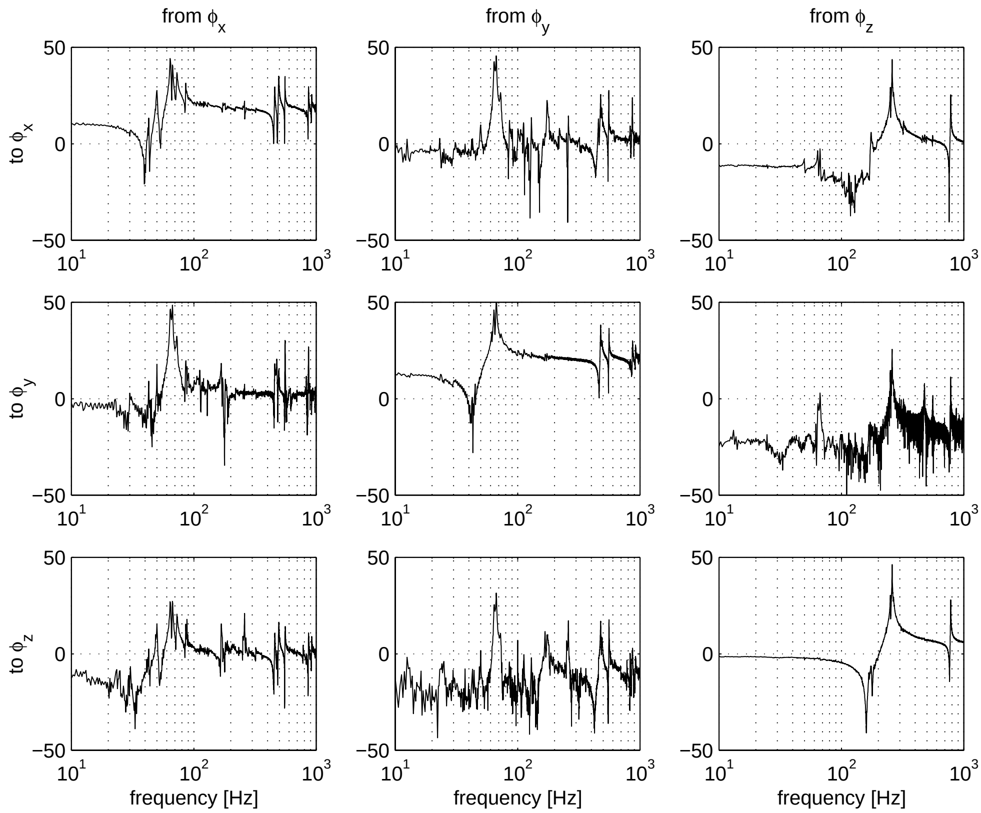

Figure 10 Frequency responses resulting from 'intuitive' decoupling (rotation modes)

visible in the responses corresponding to the joystick modes, and vice versa. From the off-diagonal elements in Figs. 10 and 11, however, it is clear that mutual influence between the various modes is *not* negligible. The main reason for this is probably the inequality in the six actuators and the six sensors. Intuitive decoupling thus may be simple; it is far from perfect.

Nevertheless, the resulting frequency responses are appropriate to tune modal active damping. The main benefit of applying IFF to the intuitively decoupled responses, is that each response now is dominated by a single resonance and a single anti-resonance, paving the way to perform relatively simple IFF-tuning, based on the single-mode equations (3)-(5). We will illustrate this for two of the decoupled loops for the joystick modes (Fig. 12). Based on these plots, we may draw the following important conclusions:

1. ***Robust stability.*** The left frequency response, which would ideally represent the joystick mode φ_x, can be seen to be affected by the contributions of various other modes (other than the suspension modes, and therefore not present in the simple model that has been used for decoupling). Despite the fact that the model is *not accurate*, stability of the closed-loop can easily be guaranteed on the basis of the phase plot.
2. ***Modal damping performance.*** In the right plot, representing φ_y, we can clearly observe a dominant anti-resonance ($\omega_\mathrm{a} = 42$ Hz) and a dominant resonance ($\omega_\mathrm{e} = 67$ Hz). Consequently, an important result of 'intuitive' decoupling is that, with (4), the maximum achievable modal damping can easily be calculated: $\zeta_{\max} = \dfrac{\omega_\mathrm{e} - \omega_\mathrm{a}}{2\omega_\mathrm{a}} = \dfrac{67-42}{2 \cdot 42} = 0.18$.

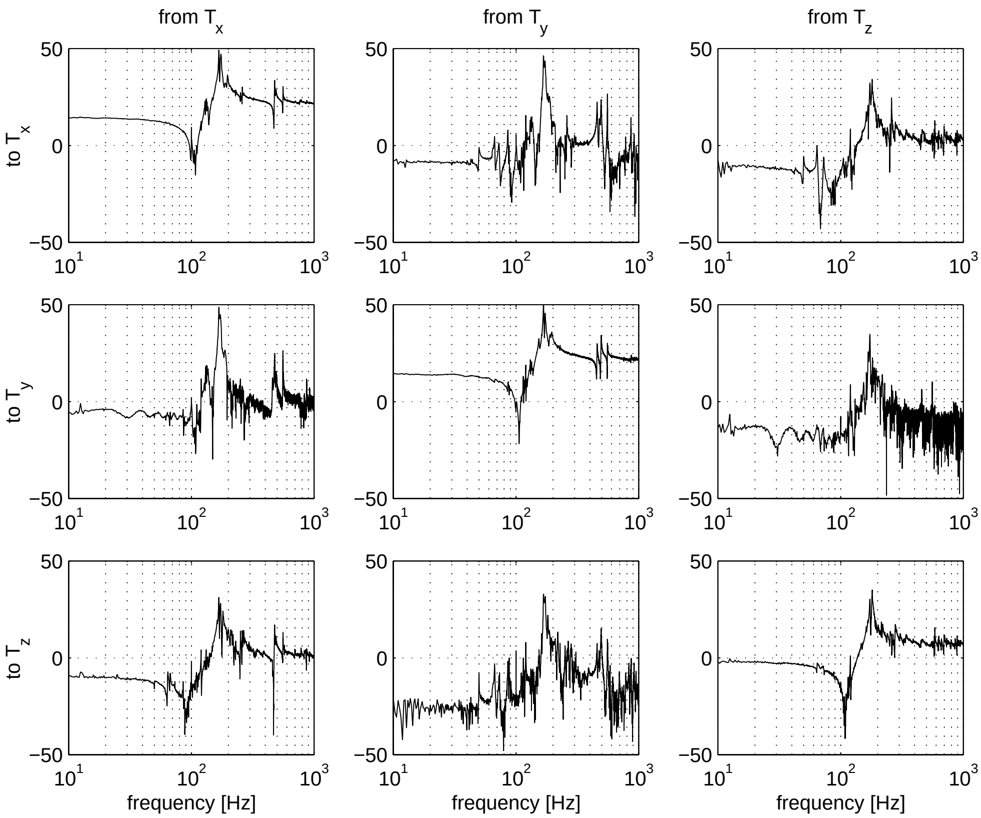

Figure 11 Frequency responses resulting from 'intuitive' decoupling (translation modes)

Furthermore, from (3) and (5) it can be deduced that this maximum is achieved when the upward-part of the open-loop frequency response (between ω_a and ω_e) crosses the 0 dB level halfway ω_a and ω_e (on a logarithmic scale, i.e., for $\omega = \sqrt{\omega_a \omega_e}$) [6]. In the right plot in Fig. 12 (in contrast to the left plot), the 0 dB-crossing can be determined unambiguously. From this plot we may conclude that, in order to achieve maximum damping for the joystick mode φ_y, the feedback gain should be set lower. Moreover, in order to achieve a proper balance for the damping-versus-stiffness trade-off, the feedback gain should be lowered even further.

SUMMARY

Collocated actuator-sensor-pairs are ideally suited for adding robustly stable active damping within mechanical structures suffering from badly damped vibrations. The robust stability of local feedback applied to collocated actuator-sensor-pairs, stems from the inherent passivity of the control law. The passivity of control based on collocated actuator-sensor-pairs is preserved upon coordinate transformation. This implies that a relatively simple model may be used to perform 'intuitive' modal decoupling, and that the intuitively decoupled control loops again enable robustly stable active damping. The main benefits of this strategy are:

1. *Robust stability*. Vibration modes that were not present in the model upon which the intuitive decoupling was based, will *not* turn unstable.
2. *Modal damping performance*. The damping for each mode can be tuned independently. The intuitively decoupled open-loop frequency response, together with (3-5), provides a

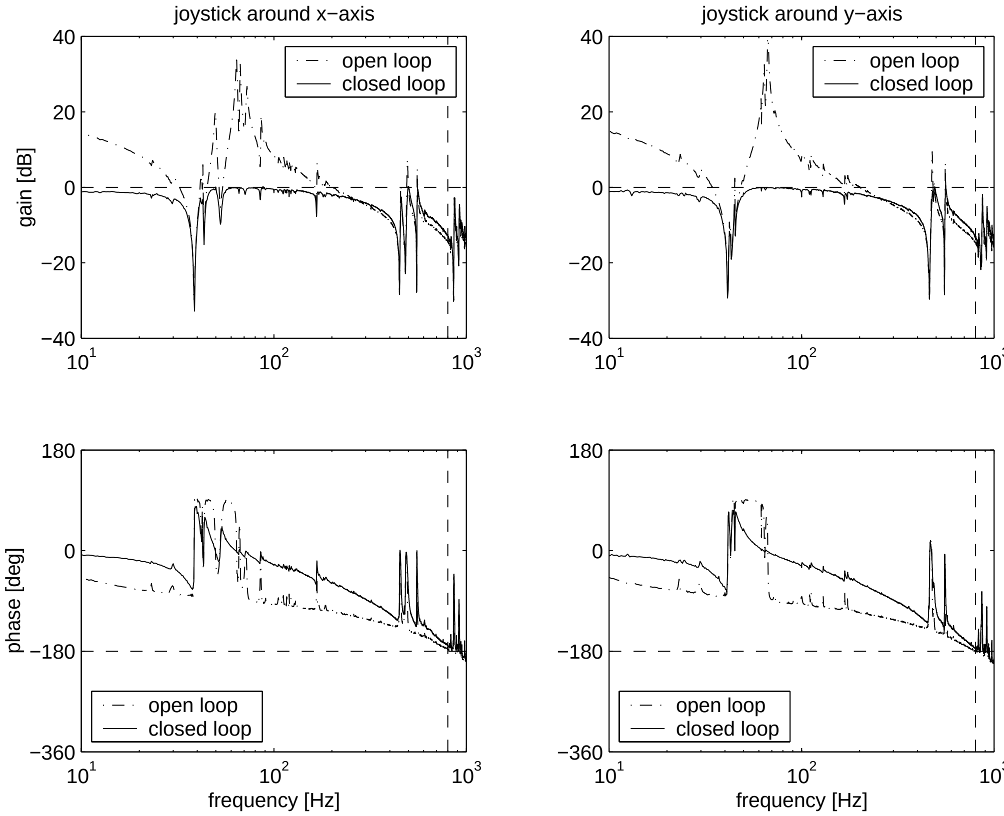

Figure 12 Integral Force Feedback applied to the decoupled 'joystick mode' control loops

simple way to gain insight in the achievable modal damping, and provides relatively simple tools for achieving optimal balancing of the 'damping-versus-stiffness' trade-off.

These benefits have been illustrated experimentally for a practical, industrial application, namely the lens suspension within an advanced microlithography machine (so-called wafer stepper).

REFERENCES

1. A. Preumont, 1997, *Vibration Control of Active Structures, An Introduction*, Kluwer Acedemic Publishers, Dordrecht, The Netherlands.
2. J. Holterman and T.J.A. De Vries, 2002, "Active damping within an advanced microlithography system using piezoelectric Smart Discs". To appear in *Mechatronics*.
3. S.A. Van den Elzen, 2001, *Design of a Smart Lens Support with two active degrees of freedom*, M.Sc. Thesis WA-789, Laboratory of Mechanical Automation, Department of Mechanical Engineering, University of Twente, Enschede, The Netherlands.
4. A. Preumont, J.P. Dufour and C. Malékian, 1992, "Active Damping by a Local Force Feedback with Piezoelectric Actuators", *AIAA Journal of Guidance, Control and Dynamics*, 15 (2), 390-395.
5. A. Preumont and Y. Achkire, 1997, "Active Damping of Structures with Guy Cables", *AIAA Journal of Guidance, Control and Dynamics*, 20 (2), 320-326.
6. J. Holterman, 2002, *Active Structural Elements for High-precision Machine Vibration Control*, Ph.D. Thesis, Control Laboratory, Cornelis J. Drebbel Institute for Mechatronics, University of Twente, The Netherlands. In preparation.