DAMPING POSITION-DEPENDENT PARASITIC VIBRATIONS

Parasitic resonances are often limiting the performance of precision machinery. The associated resonance frequency and mode shape may change over the workspace of the machine, making robust suppression or damping of these parasitic resonances a non-trivial task. This article outlines an active damping method for the suppression of position-dependent parasitic resonances through small integrated piezoelectric sensors and actuators, and shows its application to a flexure mechanism. Applying a resonant control technique, the limited actuation budget is efficiently used. A scheduling approach is followed to deal with the position-dependent behaviour. This leads to a significant improvement of the disturbance sensitivity of the flexure-based manipulator.

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Introduction

Parasitic or unwanted resonance frequencies are often a limiting factor for the performance of precision mechatronic systems [1]. Good tracking is only achieved if a machine operates within its servo bandwidth, which typically cannot exceed the first parasitic resonance frequency. Furthermore, disturbance sources such as floor vibrations can be amplified by parasitic resonances, resulting in unwanted vibration of the end-effector. Lightweight and stiff design is conventionally used to ensure that any parasitic resonances have high associated resonance frequencies.

However, with a push for higher cycle times and better disturbance suppression, damping the parasitic resonances is a crucial tool for achieving even more performance. Parasitic resonance frequencies may depend on the position of the manipulator in its workspace. Depending on the machine configuration, this is due to shifting end-effector mass, changing support stiffness or a combination of both.

As a simple example, consider a gantry robot as depicted in Figure 1. As the carriage moves along the gantry, the support stiffness in the vertical direction changes significantly. Consequently, the parasitic resonance frequency associated with the depicted z-motion depends greatly on the y-position of the gantry. In the presence of floor vibrations or a direct disturbance force d, this low support stiffness and resonance frequency can lead to large vibrations in the z-direction. This position-dependent behaviour makes robust damping of parasitic resonances over the entire workspace a non-trivial task, especially when the use of passive materials is restricted by design constraints, such as vacuum or cryogenic compatibility.

Similar to the gantry example, position-dependent support stiffness can be observed in flexure mechanisms. Such mechanisms achieve motion through the use of elastic elements, referred to as flexures. As the mechanism is deflected, the induced deformation of the flexure elements decreases the stiffness they provide in support directions [2]. Consequently, flexure mechanisms perform well in the centre of their workspace, but exhibit low parasitic resonance frequencies for larger deflections.

This article showcases an active damping approach for position-dependent parasitic vibrations. A flexure-based manipulator, shown in Figure 2, is used as a demonstration case. As the mechanism performs its nominal motion, floor vibrations and direct disturbance forces lead to excitation of a parasitic resonance. The associated resonance frequency

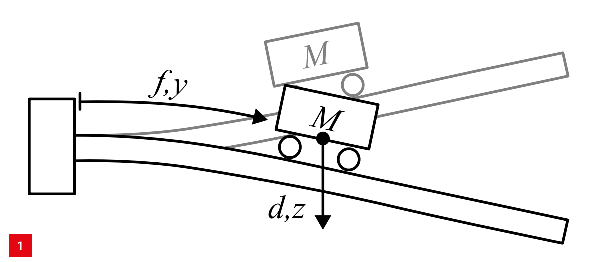

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Illustration of a position-dependent resonance in a deformable gantry. As the gantry with mass M moves along y, the support stiffness in the z-direction decreases. Therefore, the system becomes more sensitive to a disturbance force d.

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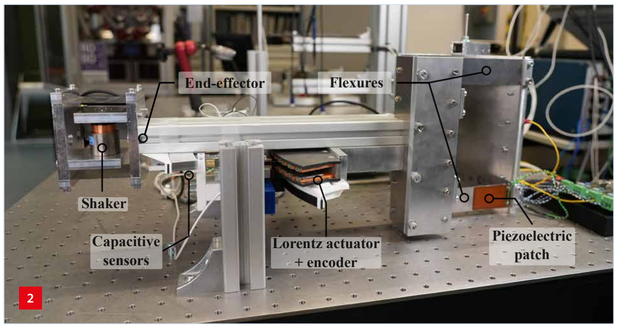
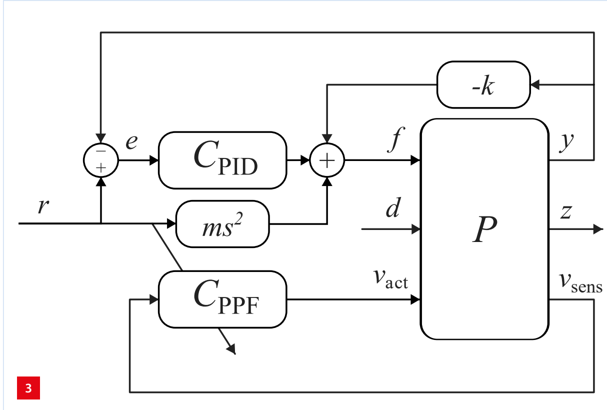


Image of the flexure-based manipulator. The nominal motion is actuated by the Lorentz actuator and measured with an encoder. The parasitic motion is excited using a disturbance force, which is generated with the shaker. The capacitive sensors are used to measure the parasitic movement of the end-effector with respect to the base.



The proposed control approach. The nominal motion y, actuated with f, is controlled with a PID controller C_{PID} in combination with stiffness compensation –k and mass feedforward ms^2. The piezoelectric actuator voltage v_{act} and sensor voltage v_{sens} are used by the Positive Position Feedback controller C_{PPF} to suppress the parasitic resonance. C_{PPF} is scheduled with the the reference of the nominal motion r.

depends on the nominal deflection of the mechanism. A small piezoelectric sensor and actuator are integrated in one of the flexures. Combining these with a scheduled resonant controller, the parasitic resonance peak in the transfer from a direct disturbance to parasitic displacement is actively suppressed over a large portion of the workspace. In turn, this results in significantly lower vibrations at the end-effector.
The remainder of this article starts with outlining the vibration-suppression approach. Thereafter, the flexure hinge case is discussed. Finally, experimental results are presented.

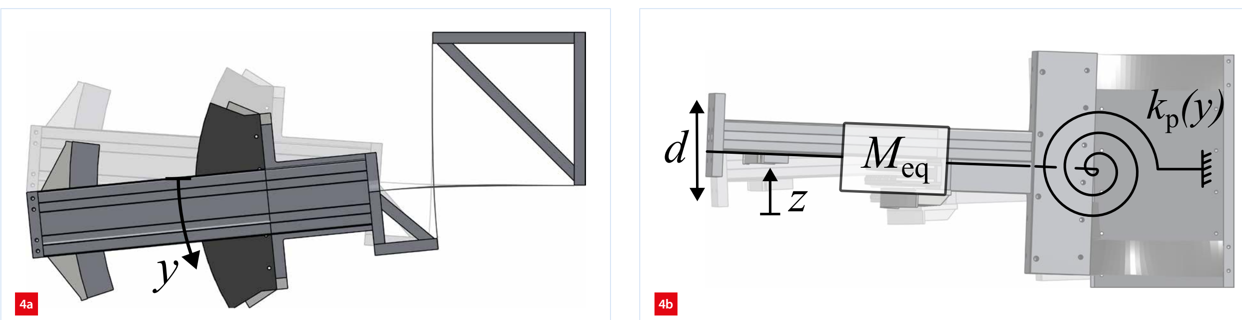
Vibration-suppression approach

Figure 3 shows the control diagram of the active vibration-suppression approach. The system *P* has nominal motion *y*, actuated by force *f*, and parasitic motion *z*, excited by disturbance *d*. In the vibration-suppression approach, it is proposed to integrate small piezoelectric actuators and sensors that couple with the problematic resonance. The actuation of the nominal motion is conventionally controlled by a PID controller C_{PID} combined with mass feedforward ms^2. For flexure mechanisms, it is sensible to also include a stiffness compensation term –*k* that cancels the stiffness of the flexures in the actuation direction.

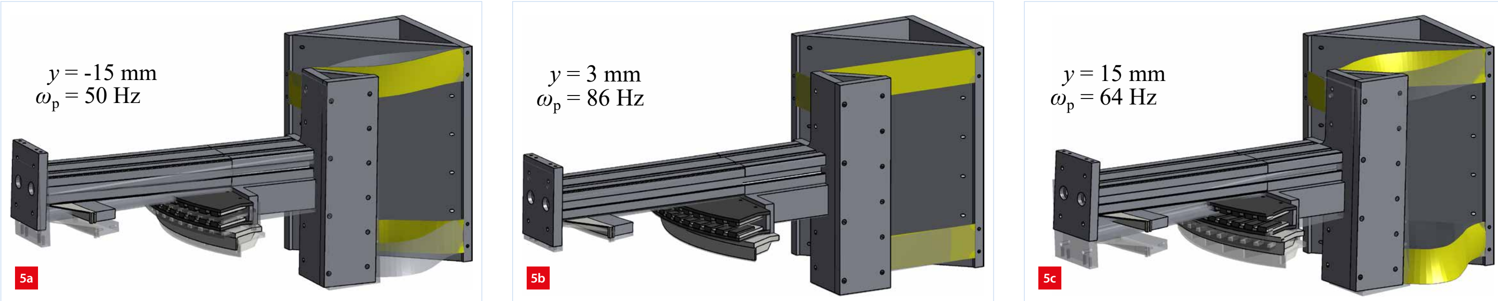
For the suppression of the parasitic resonances, a scheduled Positive Position Feedback controller (PPF) [3] is used. Such a controller mimics the behaviour of a tuned mass damper, damping predominantly at the resonance frequency. The controller structure is given by:

$$C_{\mathrm{PPF}} = K\frac{\omega_c^2}{s^2 + 2\zeta_c\omega_c s + \omega_c^2}$$

Here, *s* is the Laplace variable, *K* is the gain of the controller, ω_c the controller resonance frequency and ζ_c the relative damping of the controller.
In order for the controller to optimally suppress the resonance, one has to choose ω_c slightly higher than the resonance frequency of the to be suppressed parasitic vibration mode ω_p. Since ω_p changes with deflection, the controller can be scheduled with relative ease by taking $\omega_c(r) = \eta\omega_p(r)$, where η is a constant and $\omega_p(r)$ is the parasitic resonance frequency as a function of the reference *r* for the nominal displacement *y*. The gain *K* and relative damping ζ_c can be taken as a constant. As long as the gain *K* is below a given threshold, there is little to no risk of controller instability.
The values for *r*, *K* and ζ_c can be chosen such that the H_2 norm of the transfer from disturbance force *d* to parasitic displacement *z* is minimised [4].



The nominal motion y of the manipulator and the parasitic motion z of the manipulator. The parasitic motion has an equivalent mass of M_{eq} and a support stiffness $k_p(y)$ that depends on the nominal motion.
(a) Top view of the manipulator showing the nominal motion.
(b) Side view of the manipulator showing the parasitic motion.



The parasitic resonance for various nominal deflections y. The shown mode amplifies the parasitic displacement z due to disturbance d. The flexures are coloured yellow for contrast.

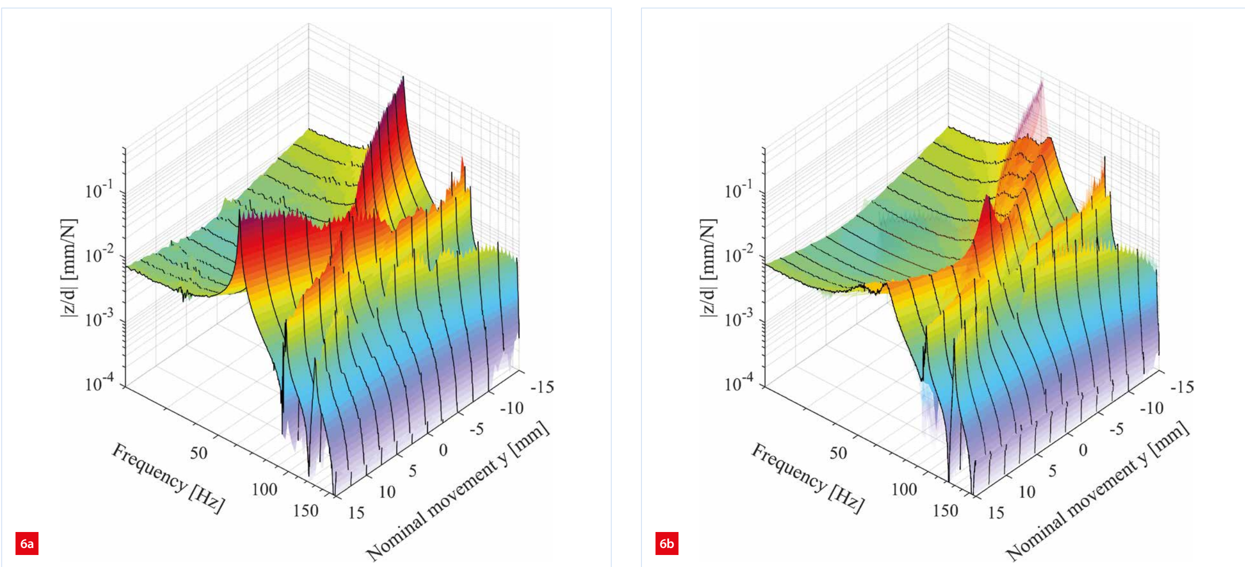
Application to the flexure-based manipulator

Figure 2 shows the considered flexure-based manipulator. It positions a sample at the end-effector along a tangential trajectory by making the nominal motion shown in Figure 4a. As it performs this positioning task, disturbance forces lead to excitation of the parasitic motion shown in Figure 4b. Such disturbance forces can originate from various sources. Here, a direct disturbance force is emulated using the shaker mounted to the end-effector.

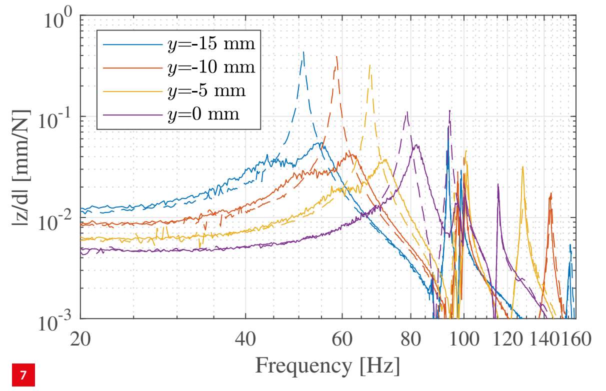
Figure 5 shows a mode shape that is the major contribution to vibrations at the end-effector. If the mechanism is in its undeflected position, all flexures are straight and are loaded in extension. As a result, the support stiffness is high and the mechanism is not very sensitive to disturbance forces. However, as the mechanism is deflected, the nominal motion introduces bending in the flexures. As a result, the flexures now also bend and twist slightly when the parasitic mode shape is excited. This results in a significant decrease of the support stiffness and associated resonance frequency. As mentioned in the previous section, the aim is to integrate a small piezoelectric actuator and sensor that will actively suppress this resonance. Due to gravity, the loading of the flexures is such that the bottom-most flexure exhibits the greatest relative deflection when the parasitic mode shape is excited. By placing piezoelectric material on this bottom flexure, good sensing and actuation of the parasitic resonance can be achieved. Two piezoelectric patches are placed on either side of the flexure, at the location indicated in Figure 2. One of the patches is used for sensing and the other is used for actuation. Since the piezoelectric patches predominantly actuate and sense the bending of the flexure, the best coupling with the parasitic mode shape is expected when the mechanism is deflected.

With the integrated piezoelectric sensor and actuator, the flexure mechanism can be characterised. The dependence of the problematic parasitic resonance ω_p on the nominal deflection y is determined. Simultaneously, in line with [4], the tuning of C_{PPF} is set as:

$$\eta = \omega_c / \omega_p = 1.65, \ \zeta_c = 0.07, \ K = 12.9 \ [V/V]$$



The magnitude plots of the frozen frequency response from disturbance force d to the out-of-plane deflection z. The black lines indicate the magnitude plot of the frequency response for a given nominal movement y.
(a) Without active damping.
(b) With active damping, with a transparent overlay of the undamped response on top. It can be seen that the first resonance peak is suppressed over the entire workspace, whereas the rest of the transfer is not significantly altered.



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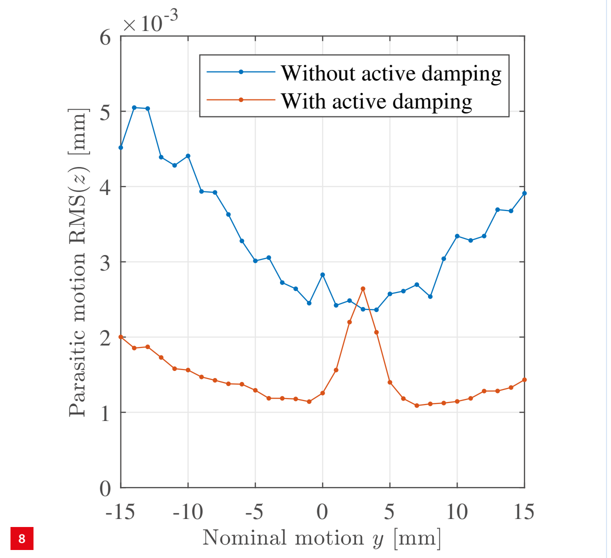
Frozen frequency response for several nominal movements y. The dashed lines indicate the frequency response without active damping and the solid lines the frequency response with active damping.

Experimental results

Figure 6a shows how the magnitude plot of the transfer from disturbance force *d* to parasitic displacement *z* depends on the nominal deflection of the mechanism. The resonance peak associated with the previously discussed parasitic resonance is by far the largest peak over the entire workspace. In the centre of the workspace, this resonance peak is relatively low and the resonance frequency is relatively high. This is due to the more optimal loading of the flexures. However, as the nominal deflection of the mechanism increases, this resonance peak becomes higher and the resonance frequency decreases.

With the active damping strategy, the first resonance can be suppressed and the response plot given in Figure 6b is obtained. It can be seen that the resonance peak is greatly reduced, especially when the mechanism is deflected. This results in a resonance peak reduction of up to a factor 10 for certain deflections. In the centre of the workspace, the resonance peak is difficult to suppress since the coupling of the piezoelectric patches with the parasitic resonance is limited. Figure 7 shows sections of Figure 6b for several nominal deflections.

In order to evaluate the sensitivity of the flexure mechanism to disturbance force *d*, a white-noise disturbance force is applied using the shaker with a magnitude of $6{\cdot}10^{-5}$ N/√Hz in the frequency range 10-300 Hz. The resulting root mean square (RMS) value of the parasitic displacement is shown in Figure 8. It can be seen that in the undamped case, the RMS parasitic displacement becomes larger with the deflection of the mechanism. When the active damping is enabled, the parasitic displacement in the centre of the workspace is similar. However, when the mechanism is deflected, the parasitic displacement is smaller than at the centre, with a reduction factor of up to 2.5 compared to the undamped case.



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Root mean square parasitic deformation RMS(z) due to the white-noise disturbance force d.

Conclusions

This article proposed a vibration-suppression approach for position-dependent parasitic resonances. In the approach, the position-dependent parasitic resonances are actively suppressed over the entire workspace using small integrated sensors and actuators. The low build volume and ability to deal with position-dependent parasitic resonances make the approach a viable alternative to conventional damping methods for a wide class of manipulators.

For the considered flexure hinge, it was shown that a parasitic resonance can be significantly suppressed by integrating a small piezoelectric sensor and actuator in one of the flexures. This results in a much better disturbance sensitivity of the flexure-based manipulator.

This article is based on previous work presented at the ASPE Annual Meeting [5]. Academic papers detailing the tuning of the PPF controller [4] and the suppression of multiple bandwidth-limiting modes [6] are currently under consideration for publication.

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