euspen's 24th International Conference & Exhibition, Dublin, IE, June 2024
www.euspen.eu

Optimal active damping of a wafer gripper in presence of multiple disturbances

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Abstract

The vibrations of an end effector of a wafer handling system, moving wafers between the atmospheric and vacuum environments, can be excited during systems operation. Conventionally, to cope with this, the operations of a system are slowed down and dwell intervals are introduced so any undesired vibrations can settle. This requires significant time, since the damping levels in the system are low, because of the materials used. Vibration attenuation in thin structures, like the considered wafer gripper, can be improved with active means. Collocated piezoelectric patch sensors and actuators are attached to the gripper and the damping is increased using appropriate controllers. In the literature, such controllers are tuned with the aim of minimizing the transfer of vibrations from a selected disturbance source to a single measurement point or a modal response. This approach is not effective if multiple disturbance sources are present in the system. Especially, the influence of electronic noise is amplified as the gains of a controller are increased. In this paper, the effect of multiple disturbances on beam-like active vibration control systems is investigated using the dynamic error budgeting approach. The system dynamics are studied with a focus on disturbance propagation paths in open and closed loops. The performance of the system is represented by the cumulative power spectrum (CPS) of acceleration at the point of interest. The well-established Positive Position Feedback (PPF) is used as the controller, and a tuning method, based on the optimisation of the predicted CPS, is presented. The overall performance of the proposed and conventional tuning methods is compared, which highlights the trade-off between resonance peak reduction and noise amplification. The improvement over the conventional method is clear, with almost 75% smaller noise amplification and 13% decrease in the total CPS in the considered case.

Keywords: Active Vibration Control, Dynamic Error Budgeting, Positive Position Feedback, Piezoelectric Patch Transducers

1. Introduction

The demand for smaller and more potent chips is ever-growing, leading to higher requirements for all the systems involved in the semiconductor manufacturing process, which must be met while maintaining high productivity. The same applies to the wafer handling systems, moving the wafers in and out of the machines where the production steps happen. The main component of the wafer handling system is a robot arm with a gripper for manipulating the wafers, illustrated in Figure 1. It is a thin structure with beam-like prongs with wafer attachment points at the tips.

During the operation of the wafer handler, the vibration modes of the gripper may be excited. The vibrations of the base of the gripper are amplified at the resonance frequencies, leading to large accelerations at the tips of the prongs, which may result in mispositioning or damaging wafers. Stiffening the end-effector by using a ceramic material was not sufficient to alleviate the problem. To cope with the vibrations, the operations of a system are slowed down, and dwell intervals are introduced so any undesired vibrations can settle. However, this requires significant time, since the damping levels in the system are low, because of the materials used. In the previous investigation it has been shown that the damping of the gripper cannot be sufficiently increased by passive means like viscoelastic materials[1], tuned mass dampers[2] or shunted piezoelectric transducers[3].

Vibration attenuation in thin structures, like the considered wafer gripper, can be improved actively using piezoelectric patch transducers. When attached to a structure, the patch sensor output is related to the average beam curvature at its location, and the actuator produces a pair of moments with amplitudes proportional to the applied voltage[4]. While ample configurations are available in the literature, collocating sensors and actuators assures predictable dynamics[4] and facilitates robust stability of the system[5]. When the loop is closed, vibration attenuation can be improved with an appropriate controller. Low-order fixed-structure controllers are preferred rather than elaborate optimization-based schemes, which are sensitive to model inaccuracies. Especially well-established is Positive Position Feedback (PPF) control, in which signal

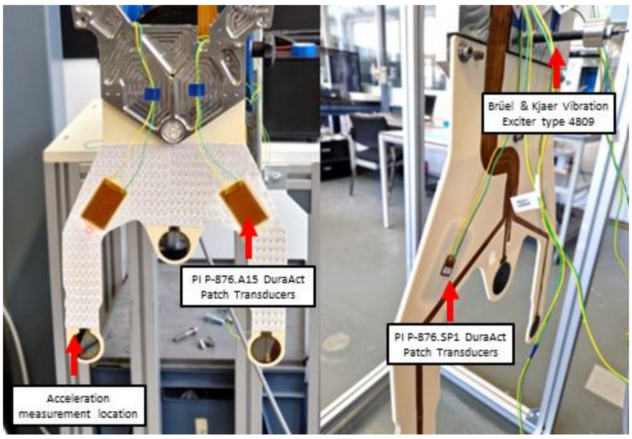

Figure 1. Experimental setup. The wafer gripper with piezoelectric transducers attached is suspended on elastic cords. The base of the gripper is connected to a shaker by a thin strut. Accelerations at the tips of grippers prongs are measured for performance validation.

measured by the patches, related to the generalized position, is fed back to the actuators via a second-order low-pass filter[6]. This provides a stronger vibration attenuation at the target frequency thanks to the presence of the resonance and smaller amplification of high-frequency noise thanks to the roll-off.

In the literature, PPF controllers are tuned with the aim of minimizing the transfer of vibrations from a selected disturbance source to a single measurement point or a modal response[7], [8]. This approach is not effective if multiple disturbance sources are present in the system. The use of more aggressive controllers, for example with a higher gain, leads not only to stronger attenuation of resonance peaks but also to amplification of noise entering the system via electronic components used for implementation. Consequently, the system does not work as intended.

In this paper, the effect of multiple disturbances on beam-like active vibration control systems is investigated using the dynamic error budgeting approach[9]. The performance of the system is represented by the cumulative power spectrum (CPS) of acceleration at the point of interest. The disturbances acting on the system, as well as the transfer functions relating them to the performance measurement, are studied. This information is used to design an optimal PPF controller in the frequency domain, using experimental data. The obtained performance is then compared with the conventionally tuned controllers.

The details of the studied problem are presented in section 2, section 3 presents the obtained results, and the paper is concluded in section 4.

2. System analysis

In this section, we clarify the studied problem. In 2.1 we introduce the plant, with the focus on all the considered input and output signals. In 2.2 we define the performance of the system using the dynamic error budgeting approach. Sections 2.3 and 2.4 introduce the disturbances and their propagation within the system. The controller design is studied in section 2.5.

2.1. Plant description

Figure 1 presents the experimental setup and the signals acting on it are shown in figure 2. The wafer gripper is suspended on flexible cords and attached to a shaker, applying the disturbances z_{in}, that represent the excitation of the system during operation. Two pairs of collocated piezoelectric patch sensors and actuators are attached to the gripper, so the low-frequency vibration modes can be influenced. The location selection for the piezo transducers was a subject of a previous study[10]. With appropriate amplifiers included, the measured and applied signals are denoted $V_{in,i}$, $V_{out,i}$ respectively. To check the performance, the accelerations $z_{out,i}$, are measured at the two tips of the fingers, where the wafer is attached. The complete system is represented by a transfer matrix with three input and four output signals

$$\begin{bmatrix} z_{out,1} \\ z_{out,2} \\ V_{out,1} \\ V_{out,2} \end{bmatrix} = \begin{bmatrix} P_{11} & P_{12} & P_{13} \\ P_{21} & P_{22} & P_{23} \\ P_{31} & P_{32} & P_{33} \\ P_{41} & P_{42} & P_{43} \end{bmatrix} \begin{bmatrix} z_{in} \\ V_{in,1} \\ V_{in,2} \end{bmatrix}.$$

2.2. Performance definition

The goal of the AVC system is to minimize the movement of tips of the prongs, in the presence of disturbances. Formally, this can be expressed using the Dynamic Error Budgeting (DEB)[9] approach. Due to systems symmetry, a single acceleration measurement $z_{out,1}$ is used for the controller design and the objective is then to minimize the variance of this signal. For all the calculations the signals are assumed to be stochastic and zero-mean. In such a case, the variance of a signal $x(t)$ is equal to its power

$$\sigma_x^2 = \bar{x}^2 = \int_{-\infty}^{\infty} x(t)^2 dt.$$

The power disruption of a signal over frequencies can be modelled using one-sided Power Spectral Density (PSD), denoted $S_x(f)$. The Cumulative Power Spectrum (CPS) shows how different frequencies contribute to the total power of the signal and is defined by

$$C_x(f_0) = \int_0^{f_0} S_x(f) df,$$

with $\lim_{f_0 \to \infty} C_x(f_0) = \sigma_x^2$. The CPS is useful for visualising the biggest contributions to the error, that should get the designers attention. The influence of different disturbance sources on the total PSD can be calculated as

$$S_{z_{out}}(f) = \sum_{j=1}^{n} S_j(f) T_j(f)^2,$$

where S_j represents the PSD of the jth disturbance signal and T_j denotes the transfer function form that source to the performance signal z_{out}, presented in equation (TFS). If the disturbances due to sensors and amplifiers are negligible, the simplification $S_{z_{out}}(f) \approx S_{z_{in}}(f) T_{z_{in}}(f)^2$ can be used. Then, for a given $S_{z_{in}}(f)$ it is sufficient to minimize $T_{z_{in}}(f)$. While this assumption supports the use of H_2 or H_∞ tuning methods for AVC controllers, it is often not satisfied in practice.

2.3. Disturbance signals

The disturbance z_{in} acting on the base of the gripper is created by the shaker. In this paper, a signal consisting of ten 20ms impulses with 3s pauses between them is considered. The disturbances due to the piezo actuator amplifier $n_{u,1/2}$ were neglected, since their contribution was expected to be small. The noise sources acting on the measurements of the piezoelectric patch sensors $n_{v,1/2}$ were assumed to be equal. This is explained by the fact that both measurements were filtered by charge amplifiers implemented using the same integrated circuit and the same power supply. Assuming uncorrelated noise sources in this case would lead to significant overestimation of the total CPS. PSD of z_{in} and n_v are presented in Figure 3.

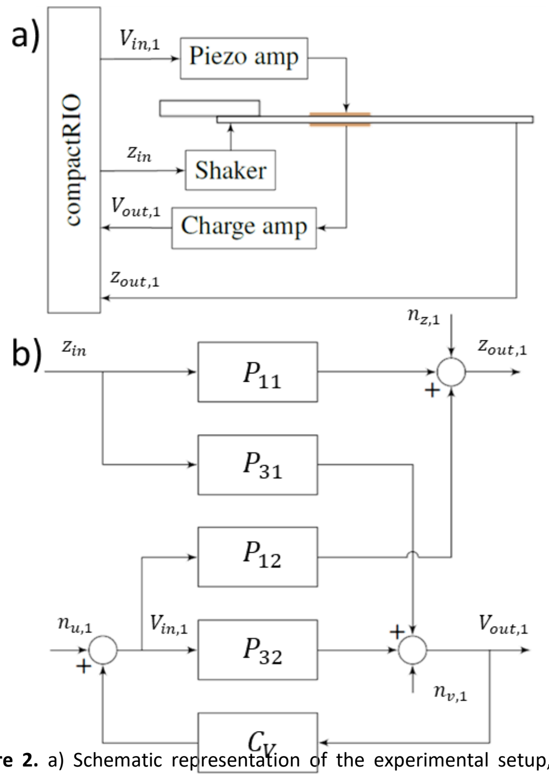

Figure 2. a) Schematic representation of the experimental setup, b) Overview of signals in a single prong of the gripper in closed loop.

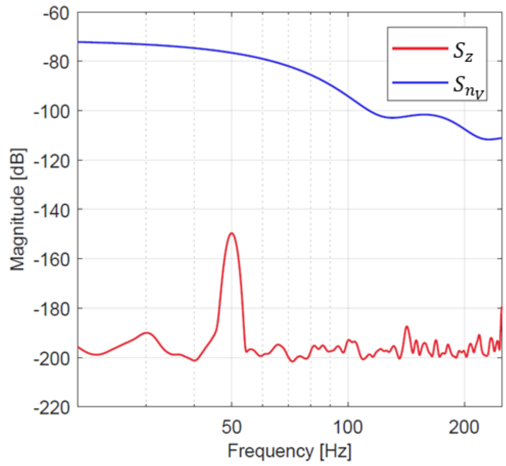

Figure 3. Power spectral densities of the base excitation z_{in} (blue) and the actuator noise n_V (red).

2.4. Disturbance propagation

First, for clarity, the propagation of signals in the wafer gripper will be presented only for a single prong, which is justified by systems symmetry. Figure 2.b presents the closed-loop active vibration control system for a single prong, with the disturbance signals included, where n_u is the noise introduced by the piezo amplifier and n_z, n_v denote the measurement noise of the accelerometer and the piezoelectric transducer respectively. The propagation of the signals through the simplified system is described by transfer functions

$$T_{z,1} = \frac{z_{out,1}}{z_{in}} = P_{11} + P_{31}\frac{C_V}{1 - C_V P_{32}} P_{12},$$
$$T_{n_u,1} = \frac{z_{out,1}}{n_{u,1}} = P_{12}\frac{1}{1 - C_V P_{32}},$$
$$T_{n_V,1} = \frac{z_{out,1}}{n_{V,1}} = P_{12}\frac{C_V}{1 - C_V P_{32}},$$
$$T_{n_z,1} = \frac{z_{out,1}}{n_{z,1}} = 1.$$

In reality, there is a strong coupling between the halves of the gripper which cannot be ignored in final systems analysis. To represent it, the closed loop transfer function is calculated as

$$P_{cl} = C(I + PC)^{-1},$$

with $C_{2,3} = C_{3,4} = C_V$ and all other components of C equal to 0. Note, that the same controller is applied for both piezoelectric sensor-actuator pairs, which is justified by systems symmetry. Considering all the transducers, the transfer function from the base disturbance to acceleration at the tip of the first prong is

$$T_{z,1} = \frac{N_z}{D_z},$$
$$N_z = P_{11} + C_V(P_{11}P_{32} - P_{12}P_{32} + P_{11}P_{43} - P_{13}P_{41}) + C_V^2(P_{11}P_{32}P_{43} - P_{11}P_{33}P_{42} - P_{12}P_{31}P_{43} + P_{12}P_{33}P_{41} + P_{13}P_{31}P_{42} - P_{13}P_{32}P_{41}),$$
$$D_z = C_V^2(P_{32}P_{43} - P_{33}P_{42}) + C_V(P_{32} + P_{43}) + 1.$$

Using the assumption that sensor noise acting on both piezo patch sensors is equal, the noise contribution to the acceleration on the tip of the first prong depends on the transfer function

$$T_{nV,1} = T_{n_V,1} + T_{n_V,2} = C_V P_{cl,12} + C_V P_{cl,13} = \frac{N_{n_V}}{D_{n_V}},$$
$$N_{n_V} = C_V^2(P_{13}P_{42} - P_{12}P_{33} - P_{13}P_{32} - P_{12}P_{43}) + C_V(P_{12} + P_{13}),$$
$$D_{n_V} = C_V^2(P_{32}P_{43} - P_{33}P_{42}) + C_V(P_{32} + P_{43}) + 1.$$

The transfer function from $n_{u,1/2}$ is not shown for the full system as these contributions are neglected. As a result, we have for the total PSD of the AVC system

$$S_z(f) \approx S_z T_{z,1}^2 + S_{n_V} T_{nV,1}^2 + S_{n,z}.$$

Since $S_{n,z}$ is not influenced by control, the optimal controller can be found by minimizing

$$S_C(f) = S_z T_{z,1}^2 + S_{n_V} T_{nV,1}^2 \propto S_z(f).$$

2.5. Controllers

In this section, the controller designs used in the study are presented. Positive position feedback (PPF) controllers

$$C_V = \frac{g}{s^2/\omega_c^2 + 2\zeta_c s/\omega_c + 1},$$

are considered, with ω_c denoting the resonance frequency, ζ_c the damping ration and g the gain of the controller. As explained earlier, due to systems symmetry, the same values of ω_c and ζ_c are applied for both transducer pairs in each considered case. The stability of a system with a PPF controller is determined from a condition on the steady-state loop gain

$$C_{V,1}(0)P_{32}(0) < 1, \quad C_{V,1}(0)P_{43}(0) < 1.$$

To satisfy this condition with a sufficient margin, the gains $g_1 = kP_{32}^{-1}(0), g_2 = kP_{43}^{-1}(0)$ with $P_{32}(0) \approx P_{43}(0)$ and $k = 0.7$ are selected for each of the considered cases. The remaining parameters of the controllers are selected using three methods:

1. Analytical solution minimizing the H_2 norm of the transfer function $T_{z,1}$ [7]
2. Analytical solution minimizing the H_∞ norm of the transfer function $T_{z,1}$ [7]
3. Minimizing the $S_C(f)$, which corresponds to minimal $S_z(f)$.

3. Results

The transfer functions of controllers designed in the three considered cases are show in figure 4.a. The controller designed for minimizing the $S_z(f)$ is characterized by significantly higher damping and lower resonance peak than the two others. This corresponds to less aggressive attenuation of the resonance peaks of $T_{z,1}$, visible in figure 4.b, where the closed loop transfer functions are shown.

Figure 5 compares the cumulative power spectra obtained in absence of control and in all the considered closed-loop scenarios. In all the cases, the predicted values of the CPS underestimate the measured value but are sufficiently close for designing the controllers. From the open-loop plot it is clear that the second resonance mode at 78 Hz has the strongest influence on the systems performance and is therefore the target for the active damping controllers. A strong reduction of the CPS was achieved in all the considered closed-loop cases. For the analytically derived H_2 and H_∞ controllers' aggressive reduction of the resonance peak corresponds to a large decrease of the contribution from the base vibration. However, the overall performance is deteriorated by the influence of amplified noise, highest at the frequency of 50 Hz. When the controller is designed for $S_z(f)$, improved reduction of total CPS is possible, despite less radical attenuation of the resonance peaks. While the contribution due to the base excitation remains stronger than for H_2/H_∞ optimized controllers, the noise in the system is amplified to a lesser extent. For the proposed controller, the contribution due to the noise is nearly 75% smaller than in the H_2 case, which leads to 13% decrease in the total CPS.

While the obtained results are promising, the issue of optimal active damping of the wafer gripper requires further studies. The measured transfer functions, crucial for the tuning of the controllers, depend strongly on the boundary conditions of the gripper. Additionally, the final optimization results depend on the disturbance present in the systems. For these reasons, experiments in operational conditions of the device are necessary to fully validate the usability of the proposed design. Additionally, the large differences in the noise amplification between the analytical H_2/H_∞ controllers and the optimization-based one cannot be intuitively explained by the differences in the final controllers presented in Figure 4.a, which requires further investigation. To further reduce the noise contribution at 50 Hz additional passive and active filters can be used. The

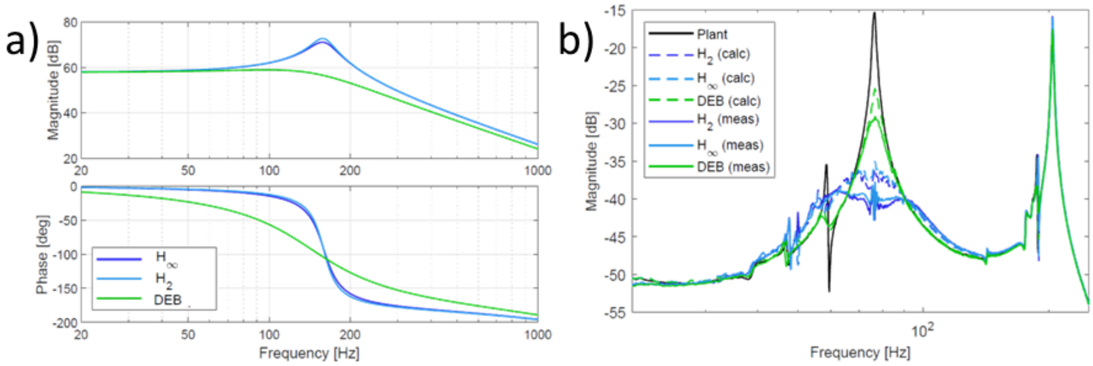

Figure 4. a) Bode plots of controllers designed using the analytical H_2 and H_∞design approaches and with the DEB method. b) Calculated and measured closed-loop transfer functions from the base excitation z_{in} to tip acceleration $z_{out,1}$.

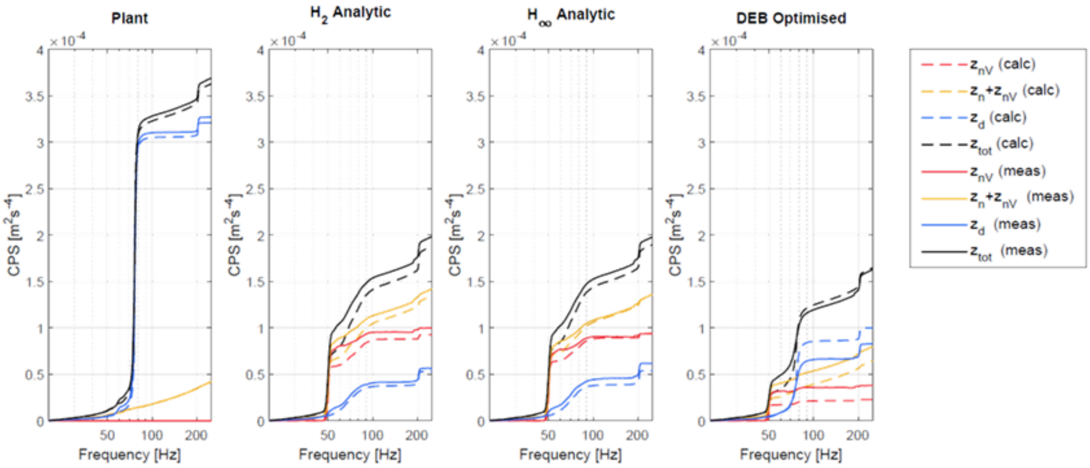

Figure 5. Cumulative power spectra of the tip acceleration $z_{out,1}$ in open and closed loop with different considered controllers.

obtained results suggest that even simple measures like use of notch filters could be sufficient for to achieve strong performance improvements.

4. Conclusion

In this paper, the effect of multiple disturbances on an active vibration control system for a wafer gripper was investigated using the dynamic error budgeting approach. The system dynamics were studied with a focus on disturbance propagation paths. The performance of the system was represented by the cumulative power spectrum (CPS) of acceleration at the point of interest. A tuning method for a PPF controller, based on the optimisation of the predicted CPS, was applied. A clear improvement over conventional tuning method was achieved, with almost 75% smaller noise amplification and 13% decrease in the total CPS in the considered case.

This paper demonstrates how both the structure excitations and electronic noise determine the performance of an AVC system. This means one must not only look at a transfer function from the excitation to performance measurement, but also the propagation paths and frequency domain characteristics of all the expected excitation and disturbances to design an effective AVC system. The best overall performance of the system can be achieved by balancing the resonance peak attenuation and noise amplification. As a continuation of this research, further studies on the disturbance sources and propagation in distributed AVC systems should be conducted. To facilitate the practical use, more intuitive design methods sufficient for an initial design of a well performing controller will be developed. Moreover, the applicability of the proposed method will be validated in the operational conditions of the device.

Funding

This work was supported by the NWO HTSM Applied and Technical Science Program under project MetaMech with number 17976.

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