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CURRENT ISSUE

Whole lot of shakin' goin' on

Minimizing vibration and noise betters image quality in inspection applications.

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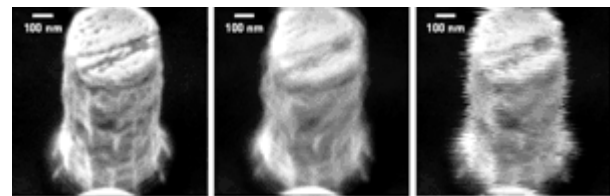
When it comes to high-resolution inspection as in scanning for semiconductor defect review, image stability is paramount. Consider a scanning electron microscope (SEM), which can provide image resolution of 1 nm in theory. In practice, external noise often makes this resolution impractical. Shrinking design rules and nanometer-size defects force equipment designers to focus on the impact that clean-room noise has on the tool.

Common clean-room noise sources include floor vibration, acoustical noise, and EMI. The environmental effect is even more apparent in applications exceeding 25,000X magnification where a small disturbance can severely harm image quality.

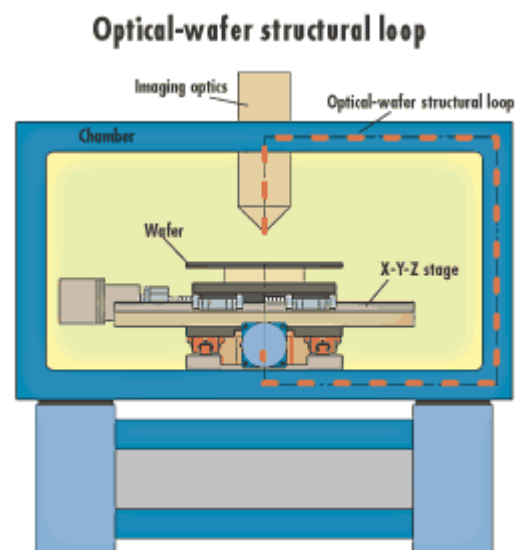
All high-resolution inspection tools incorporate some method of reducing sensitivity to clean-room noise. Isolation systems to attenuate floor vibration are widespread and magnetic shielding, active cancellation, and image processing can eliminate EMI. Using a sealed enclosure around the tool reduces transmission of acoustic noise. Unfortunately, these enclosures are not always used since they require additional clean-room space, complicate tool service, and increase cost.

Because it's difficult to completely eliminate the transmission of clean-room noise, equipment designers should examine its effect on the mechanical structure and positioning system. In addition, the operation of the positioning system can also degrade image quality and should also be carefully reviewed.

Mechanical structures



A scanning-electron microscope image shows how noise can degrade image quality. The image on the left is vibration-free. Center is a time-integrated version under the influence of vibrations. Its structural details are masked by the presence of vibrations, and the image appears to be blurry. On the right is the same vibrating image in slow scan mode. Such images usually help quantify and identify the source of vibration.



The optical-wafer structural loop is the mechanical connection between the imaging optics and the wafer. Resonances in this loop can degrade image quality.

Typically, several resonances in the optics-wafer mechanical structural loop can have a hand in degrading images. The way to keep the image stable is to avoid any relative motion between the loop end points. In practice, there is relative motion when the frequency of a noise source coincides with a mechanical resonance. The amount of relative motion depends on the input noise level and the mechanical gain or amplification at resonance.

The mechanical structure can be modeled as a second-order, mass-damper-spring system. Designers measure the response of the system to input noise by applying a harmonic forcing function to the payload mass over a wide frequency range. This frequency response analysis compares the amplitude ratio or gain of the output and phase shift of the output and input waves.

It's best to determine the lowest resonances in the design phase and compare them with common frequencies known to exist in the tool environment. Usually resonances below 120 Hz can harm image quality because they coincide with most common clean-room noise sources (i.e., 50 to 60 Hz) or their first harmonics. Though an exact value depends on the tool sensitivity and site survey, a realistic target is for all structural resonances to exceed 150 to 200 Hz. To realize this goal, several components in the optics-wafer structural loop need careful scrutiny: the support structure for imaging optics, the support structure for the positioning system, and the positioning system and wafer carrier. The first two items are especially critical for systems using vacuum chambers, which may have resonances caused by thin walls.

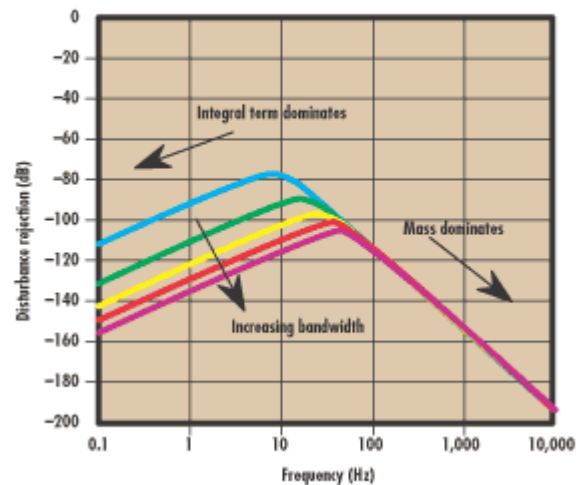
Because models of the support structure can be rather complex, an idea of the resonant frequencies and mode shapes often can only come from dynamic FEA analysis. When a prototype is complete, a modal analysis confirms the previous analysis and helps designers fully understand the limitations of the tool. If modal analysis uncovers a low-frequency resonance that's unexpected, there are several ways to alleviate the sensitivity: increase stiffness, reduce mass, add damping, or a combination of all three.

Positioning systems

The positioning system can degrade image quality in different ways depending on whether the type of motion is continuous scanning or step and repeat. For continuous-scanning systems, the main concern is payload vibration induced by linear bearings or the drive system. With linear rolling bearings, payload vibration comes from imperfections in the guide surface and rolling element. Vibration worsens when the rolling element recirculates through the loading area. This is why more systems now use noncontacting components such as air bearings and linear motors for continuous-scanning systems. Systems operating in a vacuum still use rolling bearings but put up with less performance. While not as widespread, another option is the use of vacuum-scavenged air bearings.

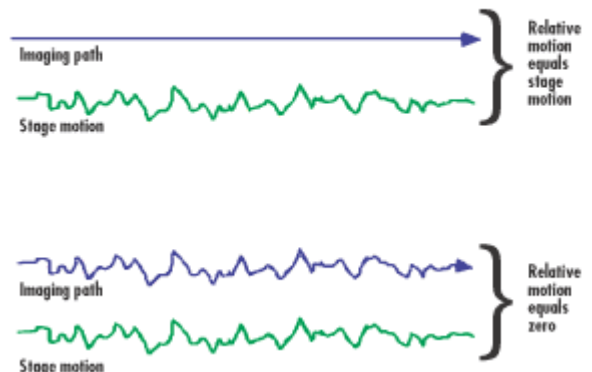
Positioning systems that realize step-and-repeat motion can degrade image quality in other ways. The primary concern for stepmotor-based positioning systems is the lowest resonant frequency of the drive assembly. Positioning systems with servocontrol are primarily concerned with static position

Disturbance rejection as a function of frequency



A linear-motor system with a PID controller rejects disturbances over a wide frequency range depending on servo bandwidth. A higher servo bandwidth will better reject disturbances at low frequencies where the integral term dominates the response.

Effects of active image stabilization



Applications with step-and-repeat motion often use a rotary stepmotor coupled to a precision ground ball-screw assembly.

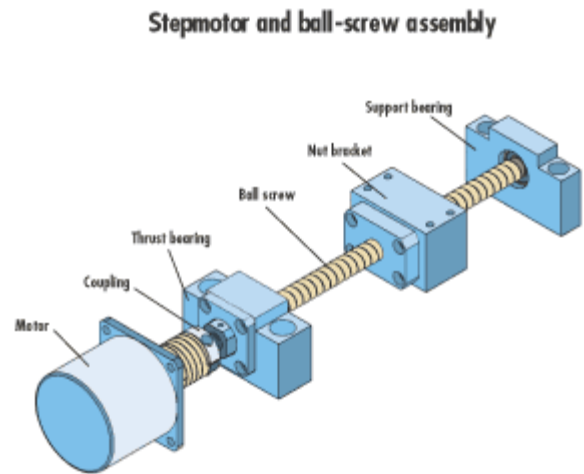
stability, or position jitter, and disturbance rejection.

Steppers move in discrete intervals as excitation of phase windings progressively align the rotor and stator teeth. Their discrete nature gives stepper positioning systems better static position stability than traditional servosystems. Imaging applications with step-and-repeat motion commonly use a rotary stepper coupled to a precision-ground ball-screw assembly. Though such configurations can provide static position stability at the nanometer level, axial resonance in the ball-screw drive assembly can limit imaging performance.

To calculate the axial resonance, we can model the drive assembly as springs in series and assume the assembly behaves like a second-order mass-damper-spring system. These assumptions let the axial resonance be calculated using:

$$K_t = \frac{1}{\frac{1}{K_{bs}} + \frac{1}{K_b} + \frac{1}{K_s} + \frac{1}{K_r} + \frac{1}{K_{nb}}}$$

$$F_{ax} = \frac{1}{2\pi} \sqrt{\frac{K_t}{m}}$$



In active image stabilization, stage accelerometer signals feed to an analog or digital filter that then helps compensate for disturbances to produce zero relative motion.

It's important to review the stiffness of each drive component carefully during the design phase as the total drive stiffness can be easily compromised by the weakest link in the system. In many cases, developers may have to stiffen up the drive after the design is complete because drive resonance has degraded image quality. But it's often possible to boost resonant frequency without significantly affecting the form factor.

For example, any effort to stiffen the drive should optimize the bearing mount and nut bracket designs using static FEA analysis. Such reviews can offer major improvements especially for brackets assembled from multiple pieces or that have attachment points far from the shaft center. Another example is the thrust bearing stiffness, which comes from an angular contact bearing. A bearing with a 60° contact angle rather than the typical 30° improves axial stiffness.

Another way of raising the drive resonant frequency is to lighten up the payload mass. The best approach is to remove small pockets of material throughout the positioning system carriage. It's best to perform static and dynamic FEA analyses during the review to make sure the carriage retains its structural integrity.

Another approach is use a tuned-mass damper, which reduces the gain at resonance. The damper consists of a second mass-damper-spring added to the stage carriage. If properly tuned, the new system has two resonances. They bracket the original frequency but both have lower gain because of more damping. In practice, the tuned-mass damper is often inexpensive and can be implemented with viscoelastic material serving as the spring and damper. It can also be retrofitted to existing designs. However, one disadvantage of the tuned mass damper is that it only functions over a narrow frequency range.

Servosystems

Unlike stepper positioning systems that use feedback to improve their performance, a servo-system relies on feedback for its operation. Specifically, it compares the reference and feedback signals at a specified update rate to correct for position errors. While superior in most other functions, servosystems do not have the inherent position stability of stepper positioning systems.

This does not mean that servosystems cannot hit nanometer-level stability but rather that they are more sensitive to noise from amplifiers and external sources. In fact, they have become increasingly more sensitive as the semiconductor industry has moved toward zero-friction, noncontact components such as air bearings and linear motors.

These systems require a high-resolution feedback device, linear amplifiers, shielded cables, and a well-tuned servocontroller just to meet position stability demands. In many cases, it may be easier to get stable positioning by selecting drive components such as ball-screw drives or linear piezoelectric motors that have friction. Friction in these

systems helps maintain stability if used in conjunction with a proper deadband setting on the controller.

One measure of position stability is servosystem-disturbance rejection. This is the amount of position change caused by a given external force applied to the carriage. Even a high-bandwidth servosystem of around 50 Hz can't usually provide enough disturbance rejection to meet nanometer level position stability in the presence of typical clean-room noise sources.

One way to boost position stability is to use a brake when the system is stopped. Systems with linear guides might brake by clamping the square rail with a piezoactuator. The brake is much easier to implement for air bearing systems. Here, a pneumatic valve vents air at the specified bearing and the bearing preload holds the stage in place. In both cases, the axis servocontrol is either disabled or the deadband expands to keep the integration term from pushing motor current to peak levels.

But there are several disadvantages, primarily when braking systems with rolling bearings. First, there is a nonrepeatable position shift of 1 to 2 μm when the brake actuates. Also, the resonant frequency of the stage becomes a function of the brake stiffness, which is limited. Finally, braking designs are custom to each stage and stage manufacturer.

Another way to eliminate relative motion is through active image stabilization. Unlike mechanical damping, these methods do not attempt to damp physical vibration. Instead, they constantly change the imaging path to match the motion of the object. Object position gets sensed by high-sensitivity accelerometers, capacitance probes, or a laser interferometer. The net result is little relative motion and stable images though the absolute motion may be quite large.

Software-based image stabilization can also help clean up vibration. One example involves the use of feature recognition on the first image to identify and locate certain features. The software then automatically calculates how the features moved between the first and subsequent images. The resulting offsets let the system adjust the display to line up subsequent images with the reference. The Nyquist sampling theorem shows the software alignment method need only deal with vibrations with frequencies up to half of the frame rate. For example, for a 30-fps imaging system, the software alignment can theoretically address vibrations to 15 Hz.

Nomenclature

F_{ar} = Axial resonance (Hz)

K_b = Thrust-bearing stiffness (N/m)

K_n = Ball-screw-nut stiffness (N/m)

K_s = Screw-shaft stiffness (N/m)

K_t = Total-drive stiffness (N/m)

K_{bb} = Bearing-mount stiffness (N/m)

K_{nb} = Nut-bracket stiffness (N/m)

m = Payload mass (kg)
