Mitigating Cleanroom Noise Can Improve Tool Performance

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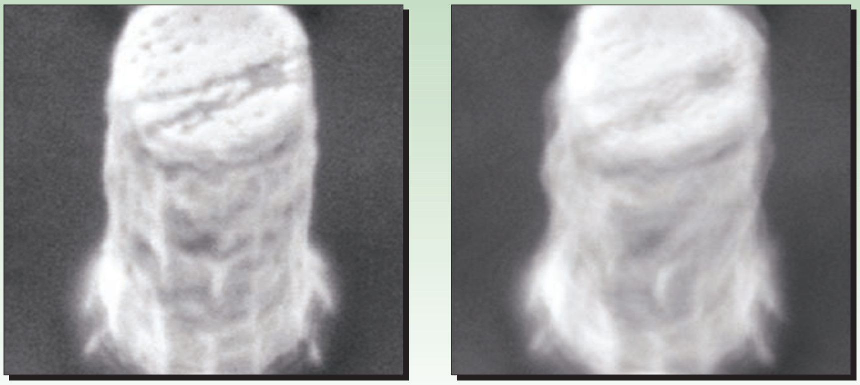
Most high-resolution semiconductor inspection tools are extremely sensitive to the cleanroom environment. Even a minute disturbance can have a significant impact on image quality for a high-magnification tool such as a scanning electron microscope (SEM). These disturbances disrupt the path between the imaging optics and the wafer in a random and/or oscillatory fashion. Common cleanroom sources that can affect the tool performance include floor vibrations, acoustic noise and electro-magnetic interference (EMI). Figure 1 illustrates the effect of these noise sources on image quality.



1. SEM image without vibration (left) and with vibration (right).

Vibrations via floor excitation

Cleanrooms, like other structures, respond to excitations when the frequency of some external source (e.g., wind gusts, trains, construction, road traffic, etc.) or internal source (e.g., fans, hoists, elevators, conveyor systems, internal vehicles, foot traffic, etc.) matches one or more of the natural frequencies of the facility and its support structure. This effect is especially detrimental when it is above the ground level, and is generally stronger in the horizontal direction.

Generic ANSI vibration criteria (VC) curves are generally used during the construction of fabs to minimize noise within the cleanroom. However, the VC curves, which were first published in the 1980s, have become outdated for today's high-resolution applications. As such, site surveys are required to determine if a candidate site is suitable for a particular tool. The site survey data can be used to estimate the degree of image vibrations, as well as to determine whether it is possible to reduce or eliminate the effect of a source.

Floor vibration data can be collected by mounting a high-sensitivity (10 mV/g), low-noise (<0.01 mg/$\sqrt{\text{Hz}}$) seismic accelerometer on the floor (or pedestal) where the tool is to be placed, along the X, Y and Z directions on a typical day. It is recommended that the accelerometer have a frequency range of ~0.1-300 Hz, and be able to provide a broadband resolution of at least 1.0 µG RMS. To reveal the frequency content of the vibration, a spectrum analyzer or a data acquisition instrument is used to perform fast Fourier transform on the accelerometer data. The obtained spectrum should indicate whether there are any strong vibration sources in the bandwidth of interest.

Figure 2 shows the composite floor vibration spectra in both the vertical and

At a Glance

Today's high-resolution wafer metrology and inspection applications demand high sensitivity and throughput. To meet these performance requirements in cleanroom environments, equipment designers must understand the relation between noise sources and tool precision.

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horizontal directions, compiled using vibration measurements from more than 20 cleanroom sites. Each frequency component in the spectra represents 95% Weibull statistic. Large vibration amplitudes are seen at multiples of 30 Hz, which can be traced to frequencies generated by common cleanroom equipment.

Vibrations via acoustic excitations

Acoustic noise is probably the most common contributor to the image vibration problem in the cleanroom. The ambient noise, mostly generated by various equipment such as fans and pumps inside the facility, constantly bombards the tool. This acoustic energy, when interacting with the tool, is transformed into mechanical vibrations that then excite the various components of the tool, even the wafer itself. In most cases, the low-frequency acoustic energy (≤300 Hz) posts the greatest threat to the tool's performance, because the low-frequency energy is omni-directional and can travel through obstacles.

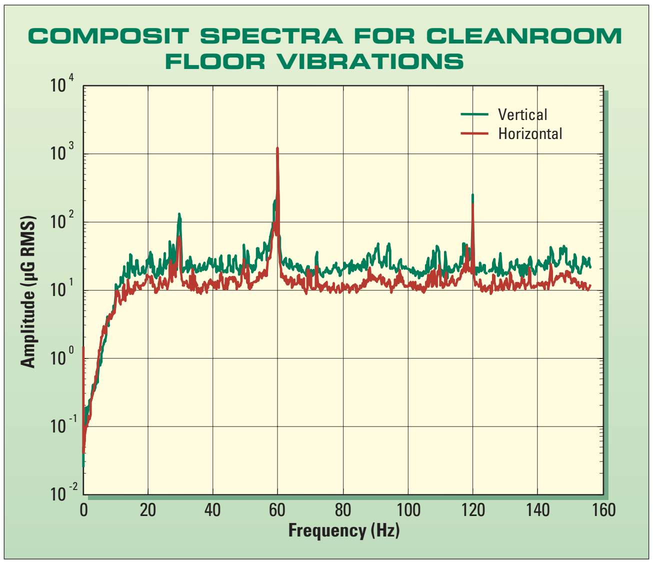
It is recommended to also measure acoustic noise levels for acoustic-sensitive tools during the site survey. Acoustic noise can be measured with a high-sensitivity sound pressure meter or microphone that has a flat response down to ≤20 Hz. C-weighting is preferred when setting the sound meter or microphone, providing a uniform response across the audible range. Like measuring floor vibrations, the same spectrum analyzer can be used to obtain a noise spectrum of the site. The sound pressure level (measured in decibels or Pascal) can be taken in 1/3 octave band or linear frequency manner. When taking acoustic data, it is important to cover the full 360° so that the maximum noise level is captured.

Electromagnetic fields

In addition to mechanically induced vibrations, a SEM can be affected by electromagnetic interference (EMI) and RF interference (RFI) caused by strong electromagnetic fields (EMF) generated by nearby equipment (such as implanters or furnaces) or installations (such as large steel beams, power lines, transformers or conduits).

EMFs are composed of electric and magnetic fields, both found across a wide range of frequencies, from extremely low frequency (ELF) up to gamma rays. Electric equipment, either on or off, generates electric fields (e-fields). The electric fields are measured in volts per meter (V/m) and have relatively constant field strength over time. They are normally easy to shield with common conductive materials, if properly grounded. Magnetic fields (H- or B-fields) are produced by electric current, and their strengths vary over time. They can pass through earth, concrete and most building materials, and are expensive to shield. The magnetic fields are measured in milligauss (mG), and their strengths decrease with distance as a function of the source type. The field spectrum can be obtained by using a magnetometer (Gauss meter) and a spectrum analyzer, similar to the techniques described previously for acquiring vibration and acoustic spectra.

Unlike mechanical vibrations, the EMF affects the electron beam (or other charged particle beams) traveling down the column to the wafer because the EMF attracts or repels the electrons. The effect of the EMF on the SEM image is visually indistinguishable from mechanical vibrations induced by either the floor or the acoustic noise.



2. These composite spectra for cleanroom floor vibrations were compiled using measurements from more than 20 cleanroom sites. Large vibration amplitudes are seen at multiples of 30 Hz, which can be traced to frequencies generated by common cleanroom equipment.

Noise reduction methods

All high-resolution inspection tools incorporate some type of vibration isolation system. The main function of these systems is to reduce the transmission of vibration from the floor to the tool within a certain frequency range. Selecting the correct isolation system is dependent on the application, and can be divided into three categories: vibration-sensitive, settling time-sensitive, or both. While all high-resolution imaging applications are vibration-sensitive, only tools with high throughput are also sensitive to payload settling time. The majority of vibration-sensitive applications can use a passive isolation system, while applications in which payload settling time is also critical may incorporate an active isolation system. An active system might also be incorporated on a vibration-sensitive tool if the site survey indicates a large-amplitude noise near the natural frequency of the isolation system. The distinction is critical because an active system can cost as much as 5-10× more than a passive system.

The most common and least expensive passive system uses pneumatic isolators. Pneumatic isolators use a volume of pressurized air acting on a piston area to support the payload. To provide vibration isolation in all degrees of freedom (i.e., vertical, horizontal, tilt and twist), the piston is typically designed to behave like a pendulum that pivots in the plane of the isolator diaphragm. A typical passive isolation system consists of four isolators and three leveling values. The leveling values sense the payload height and change the air pressure acting on the piston as loading requirements vary. To maintain position stability with four isolators (only three points are required to define a plane), two of the isolators are independently controlled and two are operated with a single valve.

For vibration-sensitive applications, there are two parameters critical to isolation system performance: natural frequency (ω_n) and viscous damping factor (ζ). The natural frequency is critical because it determines the regions for which ampli-

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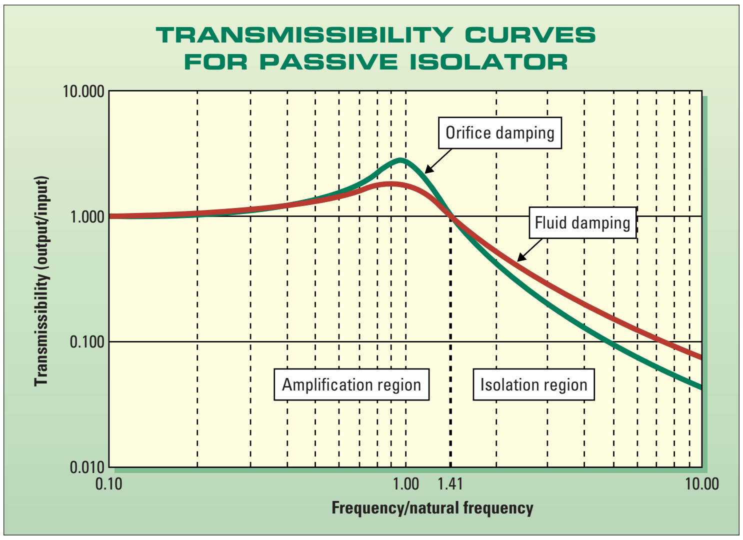
fication and isolation of floor noise will occur. Since the natural frequency of a passive air isolator is typically 1.5-2.0 Hz for both vertical and horizontal motion, isolation will occur only for input frequencies of >2.1-2.8 Hz ($1.41\omega_n$). The viscous damping factor, which is a function of isolator design, determines the peak floor noise amplification, attenuation level in the isolation region, and is also an important factor in determining payload settling time. Transmissibility curves, which plot the ratio of payload motion to ground motion vs. frequency, are important for viewing the effect of these parameters on overall isolation performance. The transmissibility curves for orifice and fluid damping passive air isolators are shown in Figure 3. The fluid damped isolator provides lower peak amplification and slightly less noise attenuation, but improved settling time.

By using servo control, active isolation systems can improve low-frequency isolation and settling time performance. Active systems are available in several configurations, each with different performance specifications and cost. The lowest-cost method uses a pneumatic isolation system as described above, with pneumatic servo control and inertial sensors for feedback. The primary benefit of this system is reduced amplification at resonance in the vertical direction only. The most common active system uses a pneumatic isolation system with voice coil actuators, a digital controller, and a combination of inertial and proximity sensors for feedback. This system will provide improved low-frequency isolation in not only the vertical but also the horizontal direction. Settling time is also significantly improved because payload position changes are corrected using force from the voice coil actuators. Additional improvements in settling time can be obtained by feed-forwarding stage position and acceleration information to the controller.

The low-frequency response of the isolation system can be critical because it is not uncommon for cleanrooms with multiple floors to exhibit low-frequency vibration (<5 Hz). This low-frequency vibration is usually attributed to movement of the upper floors, and will be amplified if located near the natural frequency of a passive isolation system. Remedies to this problem include relocating the tool to a more desirable location in the fab, reinforcing the floor/building structure, or using one of the active isolation systems mentioned above or an image processing technique. When comparing the low-frequency performance of various active and passive systems, transmissibility curves should be used. Even with a properly selected

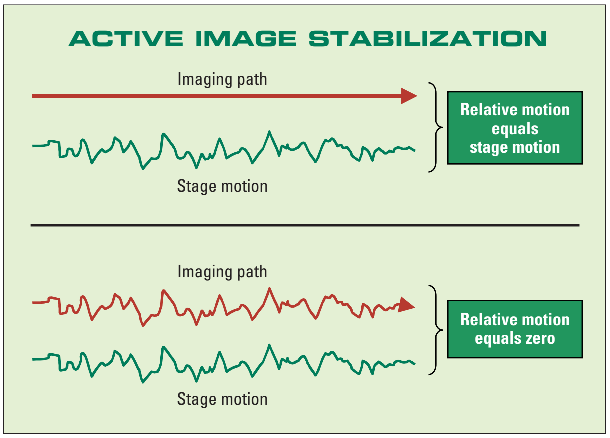


3. This figure shows transmissibility curves for orifice and fluid damping passive air isolators. The fluid damped isolator provides lower peak amplification and slightly less noise attenuation, but improved settling time.

isolation system, low-frequency floor vibration can be transmitted to the tool through cables or flexible conduits.

To reduce the transmission of acoustic noise, sealed enclosures can be used to protect the tool. The performance of these acoustic enclosures varies depending on the construction, but they typically can provide noise isolation of -10 dB at 50 Hz. Unfortunately, there are several disadvantages to acoustic enclosures that limit their use — they require additional cleanroom space, wafer transfer must occur through the enclosure, servicing the tool may be more complicated, and they increase tool cost. However, the effects of acoustic noise can be reduced by another method. Typically, a number of resonances in the optics-wafer structural loop contribute to image degradation. The lowest structural resonances in the tool should be determined in the design phase and compared with common frequencies known to exist in the tool environment. While an exact value depends on the tool sensitivity and site survey, a realistic target is for all structural resonances to be >150-200 Hz. To meet this target, the following components in the optics-wafer structural loop should be examined carefully: (1) support structure for imaging optics, (2) support structure for positioning system, and (3) positioning system and wafer carrier.

To reduce or eliminate the effects of EMI, there are generally three approaches: magnetic shielding, active cancellation or image processing. Magnetic shielding uses specialized materials to protect the column against EMI/RFI. One such material is Mu metal, a nickel-iron alloy with high permeability that shields the magnetic interference by attracting and diverting the field through itself. An active cancellation system, on the other hand, works by measuring the mag-



4. Active image stabilization by beam deflection is one way to maintain a stable image path.

netic field at the sensor and then producing an equal and opposite field to counteract the source field. The cancellation system has the ability to handle multiple fields of varying frequencies and amplitudes automatically. As long as the sensor position is not changed, the system will work without any disruptions. The cancellation system can counteract any fields that are ~25 mG peak-to-peak per axis (X, Y and Z) in amplitude over a frequency range of 1 Hz to 5 kHz. DC magnetic field sensors can be incorporated into the cancellation system to counteract any frequencies below 1 Hz.

As mentioned before, the design objective for high-resolution tools is to maintain a stable image path without any relative motion between the imaging optics and the wafer. If the relative motion cannot be effectively eliminated by the noise reduction techniques mentioned above, alternative methods can be considered. One way to accomplish this is to employ active image stabilization methods. These methods do not attempt to decrease the physical vibration; instead, they constantly change the imaging path so that it matches up with the motion of the object, which can be sensed by high-sensitivity accelerometers, capacitance probes, or laser interferometers. The net result is very little relative motion (and hence perceived stable images) even though the absolute motion may be quite large. For example, stage accelerometer signals can be fed to an analog or digital electronic filter that then incorporates this extra motion to the normal scanning motion, as shown in Figure 4.

In addition to electronic filtering, software-based image stabilization can also be used to "clean up" the vibration. One example of such method involves the use of feature recognition on the first image so that some unique features on the image are identified and located. The software then automatically calculates the offsets of the features between the subsequent images and the first image. These offsets are then applied to adjust the display so that the subsequent images appear to align to the first image. According to the Nyquist sampling theorem, the software alignment method can deal only with vibrations with frequencies up to half of the frame rate, and as such for a 30 fps imaging system, the software alignment can address only vibrations up to 15 Hz, theoretically. •

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