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Cost-Effective Design of Practically Vibration-Free High-Technology Facilities

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Abstract and Introduction

Advanced electro-optical equipment, such as that used in the production of integrated circuits, requires clean environments with extremely limited vibrations. Ground motions, personnel activities, and mechanical equipment tend to produce unacceptably severe vibrations in high-technology facilities, unless suitable precautions are taken.

The first of the following sections deals with recommended vibration criteria and the second presents an overview of the important vibration sources. Subsequent sections provide general guidance and suggests means for control of vibrations produced by external sources, interior activities, and mechanical equipment.

Vibration Criteria

A completely vibration-free environment is as unachievable as are such other idealized abstractions as immovable objects, irresistible forces or perfect vacua. Fortunately, it is generally sufficient in practice to provide a vibration environment that is adequately vibration-free - that is, an environment that does not exceed suitably selected vibration limits.

For high-technology facilities - for example, such as are used for fabrication of integrated circuits - limits on the permissible vibrations logically are set by the most severe vibration environments under which critical items of vibration-sensitive equipment can operate satisfactorily. If all equipment items to be placed in a facility under design are fully identified, and if the acceptable vibration limits of all items are known, one may readily derive a vibration criterion for the facility, e.g., by requiring the environmental vibration in each frequency band not to exceed the vibrations that are acceptable for the equipment with the most stringent limitation in that band. (Note that different equipment items may determine the criterion values in different frequency bands.)

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However, establishment of realistic vibration criteria for a facility usually involves some complications. Quite often, the equipment to be installed in a facility is not fully known at the time the facility is being designed - and, more importantly, the installed equipment may be changed in the future, as new processes and equipment become available. Furthermore, even for some items of sensitive equipment that are presently in wide use, the acceptable vibration limits are not known adequately. Although most equipment manufacturers provide some vibration specifications, many such specifications are based only on estimates and tend to be overconservative, many do not indicate the range of frequencies within which a specified limitation applies, and virtually none identify the frequency bandwidths in which measurements are to be made.

In order to develop meaningful facility environmental vibration criteria in light of the aforementioned difficulties, we have reviewed a considerable number of equipment criteria supplied by equipment manufacturers and also have made exploratory measurements on several different items of equipment. This work has led us to develop the criteria shown in Fig. 1, which we have used extensively for design and evaluation purposes.

The frequency range associated with the criteria of Fig. 1 takes account of two important considerations. The lower end of the frequency range is based on the fact that the operation of optical or mechanical devices is affected by internal (relative) deflections, and not overall (absolute) deflections - and that base motions at frequencies that are well below a device's fundamental natural frequency produce only very small relative deflections. The higher end of the frequency range is set by the fact that the vibration isolation systems, as well as the structural components of the sensitive equipment items themselves, transmit less vibration at higher frequencies.

The vibration criteria indicated in Fig. 1 are expressed in terms of vibrational velocity, even though vibrations generally are measured most directly in terms of acceleration, and even though vibrational displacements can be visualized more readily. Of course the amplitudes of displacement x, velocity v, and acceleration a, at any frequency f, are interrelated simply by

$$a = 2\pi f v = (2\pi f)^2 x,$$

so that one readily can convert from one representation to another. Our use of velocity is not based on a fundamental principle, but on our observations that the many relative mimima of the typically wildly fluctuating curves which represent the frequency-variations of the measured maximum acceptable vibration amplitudes for items of sensitive equipment generally lie approximately on a curve of constant velocity. Such a curve constitutes a simple conservative vibration criterion which has the advantage that it can be characterized essentially by a single number (without a frequency dependence) - namely, the maximum acceptable velocity - against which measured or predicted vibrations can be judged.

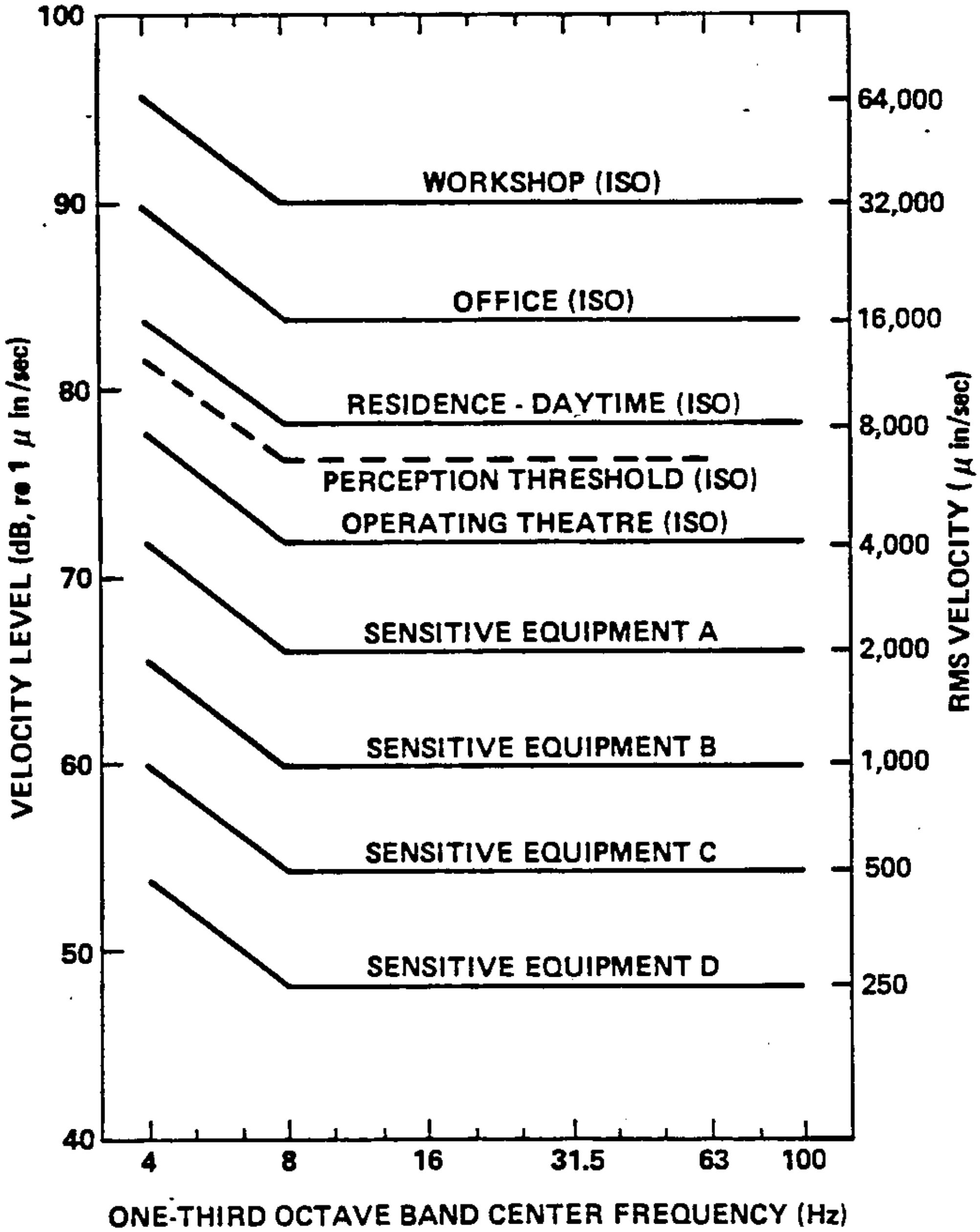

Fig. 1 Vibration Criteria for Sensitive Equipment in Buildings

A: Optical balances, bench microscopes
B: Aligners, steppers, etc. for 5 μm or larger geometries
C: Aligners, steppers, etc. for 1 μm or larger geometries
D: E-beam and other 1 μm or sub-micron equipment; electron microscopes

The "Sensitive Equipment" curves of Fig. 1 represent, in our view, the requirements for the most vibration-sensitive equipment items in each category. (They thus are more restrictive than necessary for other items in the category that are less sensitive to vibration.) Criterion D represents a conservative requirement for electron-beam equipment and for scanning electron microscopes at magnifications up to about 2000x (as normally used in the microelectronics industry); a more stringent requirement may apply to some electron microscopes at their upper limit of magnification. Criterion C represents, in our opinion, a suitable design standard for current state-of-the-art microelectronics manufacturing facilities, and may also be quite adequate for future sub-micron microelectronics technology, given some selectivity in the process equipment to be used.

In addition to the "Sensitive Equipment" criteria curves, Fig. 1 shows, for the sake of comparison, several curves indicating vibration limits in relation to human comfort, as well as the whole-body threshold of perception set forth by the International Standards Organization (Ref. 1). One may note that sensitive equipment requires considerably tighter limits on vibration than even the most sensitive activities of people.

Vibration Sources

The major sources of vibrations of concern in relation to high-technology facility buildings fall into three categories: (1) external sources, (2) internal activities and (3) service machinery.

External sources include ambient vibrations at the site (sometimes called micro-tremors), nearby road and rail traffic (including underground and elevated roads and rail systems), construction activities (including blasting), and machinery operating in the vicinity (either outdoors or in nearby buildings).

Internal activities include personnel walking (footfalls) and service activities (e.g., repair and construction) in-plant vehicles (such as forklifts and carts), and production work (e.g., actuation of production machines or other tools).

Service machinery includes all mechanical and electrical equipment that services the building and supports the production processes. It includes air-conditioning and distribution fans, chillers, cooling towers, furnaces, and all pumps, compressors and vacuum pumps, as well as elevators and mechanically actuated doors and loading platforms.

Dealing with External Sources

It stands to reason that extremely vibration-sensitive facilities should be located in areas where ambient vibrations in the ground are acceptably small, where there exists no significant nearby road or rail traffic, and where one expects no continuing construction activity or other heavy machinery operation in the vicinity. A site vibration survey is essential for evaluating potential facility sites in this regard. In such a survey, careful attention needs to be paid to such considerations as the seasonal variations in vibration-transmitting properties of the ground (which may depend on the moisture content of the soil and the height of the water table, among other things) and the daily variations in traffic.

At a given building site, the vibration-sensitive activities generally should be located as far from external vibration sources as possible, in order to take advantage of the vibration attenuation that usually is associated with increased distance. However, this attenuation may not be realized if the soil at the site is not uniform; a detailed vibration survey can define the vibration distribution at the site and enable placement of the most sensitive operations in the most vibration-free locations.

In some instances, one may be able directly to reduce the vibrations generated by an external source. For example, it is well known that the most severe vibrations associated with road traffic result from heavy vehicles with stiff suspensions moving rapidly along roads with surface irregularities. Thus, one may keep heavy trucks away from sensitive facilities, one may limit the permissible speeds, and one may smoothe the road surface. Certainly, "speed bumps," potholes, misaligned slabs and expansion joints (in bridges) should not be permitted near vibration-sensitive facilities. Similary, if the related costs are acceptable, one may consider replacing jointed rail by continuously welded rail in rail lines passing near such facilities, and/or placing such rail lines on thick ballast beds or on resilient rail support systems (e.g., Ref. 4).

We are aware of no practical means for attenuating vibrations that propagate along the ground. Berms, heavy walls, and other structures above the ground have very little effect on waves at the frequencies of primary concern here. The same is true of trenches, sheet piling and slurry walls and similar underground structures or geotechnical means (e.g., grout injection) of practical size.

However, some benefit can be obtained from appropriate foundation design and from isolating key parts of the facility from soil vibrations. In situations where the ambient vibrations of the bedrock are of relatively small magnitute, compared to those of the surface soil, it is advisable to base the building foundations on the bedrock and to avoid their coupling to the surface soil. On the other hand, where the surface soil vibrates less than the bedrock, mat foundations or spread footings are preferable. Which of these situations exists depends on the soil conditions and on the locations and types of the predominant external vibration sources. By choosing appropriate footing designs, one also may make the soil and footing behave like a suitably tuned spring, acting in conjunction with the mass it supports to attenuate transmitted vibrations above a selected frequency, much as does a classical mechanical spring-mass system. The frequency-distribution of the ground vibration must be taken into account here, however, because such spring-mass systems also amplify vibrations in the vicinity of the systems' resonance frequencies.

Where footing design cannot provide sufficient attentuation, selected parts of the building may be isolated from ground vibrations by supporting them on resilient elements, such as neoprene bride-bearing pads or "air mounts" (pneumatic springs) (Ref. 2). Here again, these resilient elements and the mass they support act like a mass-spring system, which attenuates vibrations above its resonance frequency, but amplifies vibrations near that frequency. Such a specially designed system has the advantage that it can be made very resilient, with a resonance frequency below the range of concern, and with the potential for providing considerable attenuation of intruding groundborne vibrations. It has the disadvantages of considerable complexity and attendant costs, as well as the potential for increasing vibrations resulting from internal activities.

Reduction of intruding vibrations in selected frequency ranges may also be achieved by mounting resiliently supported masses to the foundations, so as to produce in essence in a "tuned absorber" system. Such an arrangement, which has been used successfully in buildings in Japan in a different context, (Ref. 5), tends to require considerable mass and careful design.

Control of Vibrations due to Internal Activities

The problem of vibrations caused by footfalls (walking personnel) needs to be addressed early in the facility design process, because it generally requires a structural or architectural solution. Footfall-induced vibrations tend to be relatively insignificant for slabs on grade, but usually are of major importance for above-grade floors.

An above-grade floor may be visualized as acting somewhat like a trampoline; footfall impacts on it set it into motion, and thus also set into motion any equipment resting on it. Footfalls near the center of a bay tend to cause the greatest vibrations, and the vibrations always tend to be most severe at mid-bay and least severe near columns. Thus, footfall-induced vibrations and their effects may be reduced by confining heavily travelled areas (e.g., visitors corridors) to regions near column lines, placing sensitive equipment near columns, and keeping as much distance as possible between heavily travelled areas and sensitive equipment.

It is also well-known that rapid walking causes more severe footfall impacts than slower walking and that several people walking in step can cause very strong vibrations. The probability of obtaining such conditions may be reduced by instituting administrative controls or by avoiding long straight corridors that permit rapid walking.

However, the most reliable solutions to the footfall-induced vibration problem usually are structural, consisting of (1) separating the structures on which people walk from those that support the sensitive equipment, and (2) making the floor structures stiff and massive enough so that the footfall-induced vibrations remain within acceptable limits. The structural separations may be horizontal (e.g., in the form of resilient joints between corridors and sensitive fabrication areas) or vertical (e.g., where personnel are confined to a "bridge" that is supported from the columns without making contact with the process floor.)

Analytical methods for predicting the magnitudes and spatial distributions of footfall-induced vibrations have been developed and validated (Ref. 6); they are convenient for evaluating specific structural floor designs and for indicating what changes are necessary to achieve desired vibration criteria. Any type of structure - steel, concrete, or composite - can be designed to perform adequately from this standpoint, but concrete waffle-slabs have been used most widely. Usually the flexural stiffness of the floor structure turns out to be the controlling parameter, and the greatest flexural stiffnesses generally can be obtained by keeping the column spacings as small as possible.

The aforementioned concepts for controlling footfall-induced vibrations also are useful for limiting the vibrations due to in-plant vehicles. However, a vehicle entering on a floor slab or leaving a slab also in effect produces a suddenly applied load; it is desirable to reduce the suddenness of load application - e.g., by using joints with long interlacing fingers or by having joints arranged so that only one wheel of a vehicle crosses the joint at a time. It is also useful to use soft pneumatic tires on all vehicles and keep the roadways smooth and free of surface discontinuities.

The effects of vibration-producing production-related machines can be reduced by keeping these as far from sensitive equipment areas as possible, by locating these machines in areas where the supporting structures are relatively stiff (e.g, near columns), and by supporting these machines on resilient vibration-isolating systems. In general, the same vibration control concepts that are discussed in the following paragraphs in relation to service machinery apply here also.

Control of Vibration Produced by Machinery

In the early stages of design of a facility, there often exist opportunities for selecting mechanical and electrical equipment types that inherently are relatively free of vibration. For example, rotating compressors tend to produce considerably less severe vibrations than reciprocating compressors, because their inertia forces are better balanced; for the same reason, multi-cylinder (particularly opposed-piston) engines and compressors are preferable to single-cylinder machines. Similarly, it is advisable to choose the better balanced of two otherwise similar machine models, and one may do well to opt for the purchase of equipment with the best economically feasible field-balance specifications.

It is advisable to keep as much distance between vibration-sensitive equipment and vibration-producing machinery as possible, to support vibration-producing machinery on stiff structural components, and to provide this machinery with efficient vibration isolation systems.

For the purpose of selecting machinery balance specifications and isolation systems, one may do well to take note of the following easily derived expression for the velocity amplitude V of the point on a structure that supports a piece of vibration-producing machinery, keeping in mind that the vibratory velocity at other locations in the building is proportional to that at the support point:

$$V = (2\pi N)X$$

$$X = \frac{W}{k_o}\frac{x_u}{\delta_{st}} = \delta_o \frac{x_u}{\delta_{st}} .$$

Here x represents the displacement amplitude of the support point and N the rotational speed of the machine (i.e., of the shaft associated with the machine unbalance responsible for the vibration). The unbalance of a machine typically is measured by mounting the machine on a system of very soft springs, running the machine, and measuring the resulting vibratory displacement (or excursion) of its base; this "unbalance displacement" amplitude is designated by x_u. The symbol W represents the total weight of the machine (including the weight of any base on which it may be mounted) and δ_{st} denotes the static deflection of the resilient isolators on which the machine is to be supported in the actual field installation; that is, δ_{st} represents the deflection of the isolators that results from application of the weight W. Finally, k_o represents the stiffness of the supporting structure, and δ_o denotes the supporting structure's static deflection caused by the weight W.

From the foregoing expression, one may observe that the velocity amplitude is proportional to the unbalance amplitude x_u and to the rotational speed N, as well as to the ratio δ_o/δ_{st} of the static deflection of supporting structure to that of the isolation system (which ratio also is proportional to the ratio of the stiffness to the isolator support structure stiffness). One may thus readily deduce the importance not only of using soft resilient isolators, but also of supporting the machines on stiff structures.

The often-heard statement "higher-speed machinery is preferable because it can be isolated better" is not generally true in the present context. (This statement is based on a classical textbook analysis applicable to cases where the magnitude of the force acting on a machine mass does not change with rotational speed, whereas for a given rotating machine the unbalanced inertia force varies as the square of rotational speed.) As is evident from the foregoing expression, the structural vibration velocity V varies as both N and x_u, so that for a given amount of unbalance a higher speed machine may be expected to produce more severe vibrations than a similarly isolated low-speed machine.

It thus is inappropriate to conclude that centrifugal fans installed in a plant constitute more signficant vibration sources than vane-axial fans, because the latter rotate faster and therefore can be isolated better. Although high-quality vane-axial fans typically can be balanced better than well-balanced centrifugal fans, it often turns out that the values of the Nx_u product for competing fan systems are very similar. Because the achievable isolator static deflection is virtually independent of the system being isolated, the selection of fans generally needs to be made on the basis of a comparison of the Nx_u product values for candidate systems.

The addition of an inertia base (i.e., a large mass) to a vibrating machine increases the total weight W while reducing the excursion x_u proportionately, keeping the product Wx_u unchanged. One may conclude from the previously given equations that an inertia base has no effect on the vibration transmitted to the structure, provided that the static deflection of the isolation system is kept unchanged. (This is usually the case, because one typically tries to use the softest practical isolation system.) On the other hand, addition of an inertia base does decrease the vibratory excursion of the isolated machine, thus reducing the vibrations that are induced in attached piping, ducts or conduits.

Ancillary Considerations

In high-technology facilities, as in any complex dynamic system, careful attention needs to be paid to a large number of details, in order to ensure that the desired vibration performance is indeed obtained. All potential vibration transmission paths that may "short circuit" machinery isolation systems or structural breaks need to be considered and eventually treated. This includes piping, ducts, conduits that may bridge the isolation systems or gaps, as well as such auxiliary structures as partitions and pipe racks.

Good design from the start is important, but even the best design is useless unless it is implemented properly. For this reason, we typically advocate not only careful review of relevant shop drawings, but also repeated field inspection in the course of construction. Ideally, vibration measurements should also be performed after the facility is completed, so that meeting of the vibration specifications can be verified and any residual problems can be identified and resolved.

It should also be noted that audible noise tends to be a significant problem in high-technology facilities that involve "clean room" installations. These clean rooms require large amounts of air, and the airhandling systems that supply this air are sources of both noise and vibration. It usually is advangageous to address the noise contol considerations simultaneously with vibration control, so that compact, cost-effective treatments can be devised.

Concluding Remarks

Although the attempt was made in the present paper to summarize the most important factors in the design of practically vibration-free facilities, it obviously does not treat the multitude of significant details that must be considered in practice. Proper consideration of all vibration aspects generally requires the services of specialists who are familiar both with the underlying theory and with the practical hardware. Ideally, such specialist should collaborate with the facility planners, users, architects, structural engineers and mechanical engineers from project site selection through final checkout.

REFERENCES

1. Anon. "Guide for the Evaluation of Human Exposure to Whole-Body Vibration." ISO Standard (Draft) 2631; 1974.

2. Anon. "Sperry IC plant floats on concrete and airbags to prevent vibrations." Electronics, April 19, 1984; pp. 54-55.

3. Gordon, C.G. and E.E. Ungar "Vibration as a Parameter in the Design of Microelectronics Facilities." Transactions, Inter-noise '83, 1983, pp 483-486.

4. Kurzweil, L.G. and E.E. Ungar, "Prediction of Noise and Vibration in Buildings Near the New York City Subway." Transactions, Inter-Noise 82, 1982, pp. 213-216.

5. Konagai, K., K. Shimizu, N. Sano and T. Ikeda. "A Method for Reducing the Ground Tremor and Building Vibration Causes by Exciting Forces on a Tunnel Floor." Transactions, Inter-noise '84, 1984, pp 559-562.

6. Ungar, E.E. and R.W. White "Footfall-Induced Vibrations of Floors Supporting Sensitive Equipment." Sound and Vibration, October 1979, 13, pp 10-13.

7. Ungar, E.E. and C.G. Gordon "Vibration Challenges in Microelectronics Manufacturing." Shock and Vibration Bulletin 52, Part 1, May 1983, pp 51-58.

8. Ungar,E.E. and C.G. Gordon "Vibration Criteria for Microelectronics Manufacturing Equipment." Transactions, Inter-noise '83, 1983, pp 487-490.