Tutorial at Cleanrooms '96 West

Vibration Control in Microelectronics Cleanrooms

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(1) Introduction:

Vibration is one of several contaminants that can affect the "yield" of the microelectronics fabrication process. The reasons are clear when one considers the two following statements:

- The per-unit-area power consumption of a fabrication cleanroom exceeds that of a "normal" building (such as an office) by a factor of 100 or more.
- The vibration sensitivity of the most critical metrology and photolithography tools commonly used in fabrication can exceed human sensitivity to vibration by a factor of 100 or more.

In this tutorial we shall try to unravel some of the mysteries of cleanroom vibration: sources; methods of propagation; measurements; criteria and specifications; control and isolation.

The plan is to provide the attendees with some of the tools necessary to understand the issues involved in vibration control.

(2) Vibration Sources

Sources of vibration can ideally be quantified by the dynamic (time-varying) forces that they apply to the structure that supports them. Depending upon the source, these forces may occur at different frequencies - in many cases involving a range of frequencies.

The simplest example of a single (discrete) frequency source comes from the dynamic unbalance of a rotating shaft. Here the unbalance force is close to sinusoidal, reaching a maximum value once every shaft revolution. An electric transformer also generates discrete frequencies, due to magnetrostrictive deformation of the core, at integral multiples of twice line frequency (120, 240, 360 Hz, etc.).

Most sources, however, are much more complex. A pump or a fan, for instance, generates forces due to fluid turbulence in and around the machine, in addition to shaft unbalance forces. With few exceptions, turbulence-generated forces are random rather than discrete. Their frequency spectra are characterized by broadband rather than pure tone energy. Turbulence-generated forces encompass a wide range of frequencies, depending upon the nature of the turbulence; its intensity, scale and velocity.

Machines are not the only sources of vibration to be considered in designing a microelectronics cleanroom. Fluid flow in ducts and pipes generates broadband vibration - even at locations far from the machines to which they are connected. Flow in ducts and pipes is, in a practical sense, never laminar; it is always turbulent. From a simple model of fluid flow it can be shown that the net force (F) per unit length generated by a duct or pipe has the parametric dependence:

$$\tilde{F} \sim \rho \eta V^2 A \qquad \text{- - - - (1)}$$

where ρ is the fluid density, η the turbulence intensity, V the flow velocity and A the cross-sectional area.

Other sources of vibration include people and material movements within the building, vehicular movements around the building and environmental sources such as highways and railroads in the general vicinity of the plant.

With few exceptions the sources described above cannot be quantified in terms of the force spectra they generate. Rather, we must rely on measurements and experience from other similar situations. The proximity of a source to the process floor is a major consideration in determining its importance. Thus, large sources in remote locations, such as a central utility building that is separated by some distance from the process floor, may be substantially less critical than small sources that are close. Sub-fab systems are especially critical in this regard.

An important factor to realize is that of the cumulative effects of sources. A single pump, or section of duct, may in itself have an almost negligible vibration effect. When the literally hundreds of individual sources that form the completed facility are combined together, the need for careful vibration control becomes evident.

In Table 1 we list the types and characteristics of many of the sources that can influence the completed facility.

(3) Vibration Propagation - In the Ground

Vibrational energy can propagate efficiently through the ground, depending upon the type and characteristics of the soil. The type of ground wave that commonly dominates the process is called the Rayleigh wave, described by the equation:

$$V_2 = V_1 \sqrt{\frac{R_1}{R_2}} \cdot e^{-\alpha(R_2 - R_1)} \qquad \text{- - - - (2)}$$

where V_1 and V_2 are the wave amplitudes at distances R_1 and R_2 from the vibration source, and α is the material damping coefficient of the soil. At microvibration amplitudes, values of α lie in the range 0.001 to 0.005 per foot.

The Rayleigh wave is a surface wave that spreads out from a point source in the form of a symmetrical annular wave. Most of its energy is contained within a depth from the surface of one to one-and-a-half wavelengths. Since typical Rayleigh wave speeds lie in the range 400 to 1000 ft/sec, the depth of the wave varies, typically, from 60 to 150 feet at 10 Hz to 12 to 30 feet at 50 Hz.

A number of methods are suggested in the literature for attenuating the ground-borne path of vibration propagation. Three of these are illustrated in Fig. 1.

Methods (a) and (b) have the disadvantage that, to be effective, the value of h (the depth of the trench and length of the foundations, respectively) has to equal or exceed the values given above. Since much of the environmental vibration on a site, from near and distant sources, lies close to 10 Hz, these techniques are quite impractical.

Method (c), that of supporting the building on a stiff rock layer close to the surface, is not so impractical. Here, of course, the trick is not to get out of the Rayleigh wave, since Rayleigh waves exist in the stiff layer also, but to support the building on a stiff layer - for which the wave amplitudes are much lower. The use of this method, of course, is highly site specific.

In Figure 2 we illustrate the situation of an on-grade slab with and without an isolation break separating the source from the point of measurement. We have confirmed repeatedly by measurement that the presence of an isolation break in the concrete slab does not have any significant effect on the energy transmitted from source to receiver. The reason, of course, lies in the fact that the vibration travels mainly through the ground; it is not concentrated in the concrete.

In conclusion, it is virtually impossible to isolate buildings one from the other by using separate footings, isolation breaks and the like. The ground is too efficient as a path of vibration propagation to allow this to happen. The most effective method of site attenuation is distance.

(4) Vibration Propagation - Structures

The vibration response of building structures to vibratory inputs, be they applied directly to the building or indirectly through the ground on which the building rests, is complicated by the phenomenon of resonance. In a simple mass/spring system, resonance occurs when the mass and stiffness impedances are equal. The system will respond energetically to the smallest input at the resonance frequency, where equality occurs.

Building structures can be thought of as being made up of infinitely many masses (the distributed structural mass) connected by infinitely many springs (the finite stiffnesses of the structural elements). Therefore, there are infinitely many resonance frequencies or modes in which they can respond. In addition to the properties of mass and stiffness, structures also have damping. Damping determines the rate at which vibratory energy is absorbed and converted to heat.

As a result of the modal complexity of a typical building, the only practical way of studying a structure in detail - especially at the lower frequencies where the "modal density" is low - is by a technique called finite element modeling (FEM). Here, the structure is represented in a computer by a series of basic elements (beams, plates and bricks, to mention a few). Finite element analysis (FEA) is used to identify modes and to model the response to input forces.

Two examples of FEA models are show in Figure 3, each representing a wafer fabrication facility of recent design. Models of this sort can be used to study the modes of energy propagation within the building and the response at various distances from vibration sources.

The utility of FEA methods is limited, at this time, by the difficulty of interfacing the model with the ground on which the foundations rest. This interface is important since it determines ground-borne inputs to the building, ground transmission from one part of the building to another, and the amount of damping provided by the ground.

(5) The Parameters of Vibration

Vibration measurements are more complicated than sound measurements (for instance) because of the variety of parameters that can be used in expressing the amplitude and time variability of the phenomenon. To add further complexity, whereas sound pressure (the normal measure of sound) is scalar, vibration is vectorial and its direction, be it vertical or horizontal, must be taken into account.

(a) Amplitude Vibration amplitude may be measured and expressed, in terms of displacement (D), velocity (V) or acceleration (A). These measures are related simply as follows:

Velocity = Rate of Change of Displacement

$$V = 2\pi f D \qquad \text{- - - - (3)}$$

Acceleration = Rate of Change of Velocity

$$A = 2\pi f V = 4\pi^2 f^2 D \qquad \text{- - - - (4)}$$

In these equations f denotes frequency in Hz (cps)

By using equations 3 & 4 one can convert simply between the different measures of amplitude - using, of course, consistent units.

Example:

A displacement of 1 micrometer (micron) at 5 Hz equals a velocity of 31.4 microns/sec (1240 microinches/sec*).

An acceleration of 0.5 gal (1 gal = 1 Galileo = 1 cm/sec^2) at 50 Hz equals a velocity of 16 microns/sec (630 microinches/sec) and a displacement of 0.05 micron (2 microinches).

(b) Time Variability Back in the days of pen recorders and oscilloscopes, dynamic signals were measured in terms of peak-to-peak and zero-to-peak amplitudes. These measures, for a simple sinusoid, are illustrated in Figure 4.

For more complex waveforms, typical of "real" vibration environments, peak-to-peak and zero-to-peak amplitudes tend to be very variable, especially if the vibration is "random" rather than "ordered". Thus, when using an oscilloscope, it would be difficult to measure peak-to-peak and zero-to-peak excursions of a waveform with any accuracy.

The more modern practice in measuring vibration, or for that matter sound or any other dynamic quantity, is to measure the root-mean-square (rms) value of the signal. The rms value is really a measure of the energy contained in the signal. It is obtained by squaring the signal, so that the negative half cycles in Figure 4 become positive, calculating the mean (or average) value over some time period and taking the square root of the result.

The relationships between the peak-to-peak, zero-to-peak and rms values for the sinusoid shown in Figure 4 are as follows:

$$rms = \frac{zero-to-peak}{\sqrt{2}} = \frac{peak-to-peak}{2\sqrt{2}} \qquad ---- (5)$$

All modern instruments that handle time-fluctuating or oscillating signals measure the rms values of these signals. Unfortunately, we are often faced with vendor specifications for vibration-sensitive tools that use the old peak-based amplitudes. All one can do here is to assume the relationships given in Eq. (5) above.

Example: A vendor specification of 5 microns peak-to-peak can be interpreted 1.77 microns, rms.

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* In the United States we tend to be in defiance of the rest of the world by continuing to use the old fps (foot-pounds-seconds) units. This causes no end of confusion to our more enlightened (and modern) neighbors.

(c) Frequency Spectrum and Bandwidth Most vendor specifications state or imply that the vibration environment on the floor that supports the tool satisfy different requirements at different frequencies. Thus, the vendor sets limits on the details of the frequency spectrum of the vibration to which the tool is exposed. The frequency spectrum describes the vibration amplitude as a function of frequency.

A spectrum (or frequency) analyzer uses some form of frequency filtering to split the total frequency range of the signal into its individual components. There is a wide variety of ways in which this can be done:

(1) Using Fast Fourier Transform (FFT) techniques the range can be divided into equal parts each having the same bandwidth. Depending upon the total frequency range and the number of "lines" into which the range is divided, the effective filter bandwidth can vary widely. The wider the bandwidth the greater the energy that each filter will pass and the higher the values that will be displayed.

Figure 5 shows an example of a random (broadband) vibration spectrum in the range 0 to 100 Hz measured using different bandwidths (corresponding to FFT spectra with 100 through 1600 lines). The effect of bandwidth is clearly seen.

(2) Using Proportional Band Filters, the frequency range is divided into filtered segments in which the bandwidths increase in direct proportion to the frequency. The most common examples of proportional bands are the octave band and the one-third octave band. In these, the bandwidths equal 70.7 percent and 23 percent of the band center frequency, respectively. The one-third octave band filter is commonly used for vibration evaluations as we shall show later. The one-third octave band version of the spectra of Fig 5 is shown in Fig. 6.

Tool specifications rarely provide any information about the bandwidth that should be used in testing for compliance. In cases where the vibration is entirely dominated by pure tones, bandwidth is of secondary important. In cases where the environment is dominated by broadband (random) energy, bandwidth is of critical importance. This is well illustrated in Figures 5 & 6.

Most real life environments, including cleanroom environments, are dominated by broadband energy and the issue of measurement bandwidth is extremely important, therefore.

The rules for converting from one bandwidth to another are simple. Let's say the tool specification sets a vibration limit on measurements made using a fixed bandwidth of 1 Hz, and that the floor measurements used a bandwidth of 0.375

Hz (typical for a 400 line FFT over the frequency range 0 to 100 Hz). The appropriate correction (C) to the floor data is

$$C = \sqrt{\frac{1}{0.375}} = 1.63$$

The measured vibration amplitudes (in microns, microinches/sec, etc.) should be multiplied by C for comparison with the tool specification.

The correction from a measurement bandwidth of 0.375 Hz (say) to a desired bandwidth of a one-third octave is given by

$$C = \sqrt{\frac{0.23f}{0.375}}$$

where f is the band center frequency. At 5 Hz the correction is 1.75, at 20 Hz it is 3.5, and so on. Clearly of course since one-third octave bands are so wide, especially at the higher frequencies, it is far preferable to measure the one-third octave band amplitudes directly or to reconstruct them from the FFT spectrum. This is simply done.

Narrowband spectra should typically be displayed using a linear frequency scale. One-third octave band spectra are normally displayed to a logarithmic scale.

(d) Vibration Level, The Decibel Scale In practice, it is found that vibration amplitudes vary over a very wide range of values as frequency varies. On a typical velocity spectrum it is not unusual to find values ranging from 1 microinch/sec to 1000 microinches/sec. With such a range it is necessary to use a logarithmic amplitude scale since, with a linear scale, much information would be lost. One common logarithmic scale is the decibel scale.

By definition, the decibel (abbreviated dB) is a unit of level which denotes the ratio between two quantities that are proportional to power. The dB gain of an amplifier, for instance, is defined as

$$\text{gain (dB)} = 10\log_{10}\left(\frac{\textit{Power Out}}{\textit{Power In}}\right) \qquad \text{- - - - (6)}$$

If the amplifier is a voltage amplifier the corresponding relationship is

$$\textit{gain}\ \text{(dB)} = 10\log_{10}\frac{(\textit{Voltage Out})^2}{(\textit{Voltage In})^2}$$

$$= 20\log_{10}\left(\frac{Voltage\ Out}{Voltage\ In}\right) \quad ---- (7)$$

since, of course, the power in a circuit is proportional to voltage squared.

We use the same rules when applying the decibel scale to vibration, and when we do so we use the term "level" to denote that the vibration is expressed in decibels. Since the decibel applies to a ratio of two powers we must define the reference level that forms the denominator of the ratio. In vibration measurements it is common practice to use the unit value of measurement (1 micro-g**, 1 micro-inch/sec, etc.) as the reference. Thus we have

$$\text{Velocity Level (in dB re 1 micro-inches/sec)} = 10 \log (V^2) = 20 \log (V) \quad ---- (8)$$

where V is in micro-inches/sec.

$$\text{Acceleration Level (in dB re 1 micro-g)} = 10 \log (A^2) = 20 \log (A) \quad ---- (9)$$

where A is in micro-g

With some practice the process of conversion between linear and dB units becomes simple:

$$\text{Velocity Level (VL)} = 20 \log (V)$$

$$\text{Velocity (V)} = 10^{(VL/20)} \quad ---- (10)$$

The decibel equivalent of 400 microinches/sec is 52 dB (re 1 microinch/sec) and the linear equivalent of 86 dB (re 1 microinch/sec) is 20,000 microinch/sec.

The benefits of using a logarithmic or decibel amplitude scale rather than a linear scale are illustrated in Figure 7. Here, we show the same spectrum in both forms.

(6) Vibration Measurements

The vibration amplitudes that can be allowed in an operating cleanroom are very low. As stated earlier they lie far below the threshold of "feelability". For this reason, these vibrations may be termed "microvibrations" and their measurement requires rather special instrumentation. The instrumentation we most commonly use is illustrated in Figure 8.

** 1 g = 1 gravitational unit = 9.8 m/sec^2

The measurement procedure can be summarized as follows:

(1) Data are acquired and stored in the SA-77 with the integrator of the Model 2635 amplifier set to measure velocity. The frequency range of the SA-77 is normally set to 0 to 100 Hz using 400 lines of analysis.

(2) Measurement locations are selected depending upon the situation. For evaluation of a large floor area, locations are selected randomly, not favoring any type of location relative to the column supports, and in a sufficient quantity to allow statistical analysis of the spatial results. Data are acquired vertically at all locations and horizontally, in two axes, at most locations. On a typical floor, spatial variations in horizontal amplitudes are less than they are vertically.

(3) Again depending upon the situation, data are normally collected on the structural floor, below the raised access floor. It is upon the structural floor that most vibration-sensitive tools are supported - on stiff isolated pedestals or some other means.

(4) The SA-77 data are periodically downloaded to a lap-top computer to free up memory space in the signal analyzer.

(5) The field data are finally processed and formatted using Microsoft Excel. Data are normally displayed in both narrowband and one-third octave band formats, the latter being consistent with the generic vibration criterion curves, discussed later. The vertical and horizontal data are statistically processed separately, and sometimes, depending upon the situation (for example, two floors that display markedly different characteristics) further divisions are made.

(6) It is our custom to display floor evaluation data at the "average-plus-one-standard-deviation" level. For a normally distributed population this level represents the value below which about 84 percent of the population lies at each frequency. This statistic provides a conservative evaluation of the properties of a floor.

Examples of final evaluation data acquired using the instrumentation and methods described above are given in Figures 9 & 10.

(7) Vibration Criteria

It became clear several years ago that tool specifications in themselves do not provide an adequate guide as to what constitutes a "good" or "bad" floor, insofar as vibration is concerned. For one thing, tool specifications are often incomplete; they omit critical information such as the bandwidth that should be used when testing for compliance with the specification. Often, we suspect, the specifications are not based on measurements or analytical studies; they are notional, "plucked from the air" as it were, to satisfy a customer's need. Finally, even if all tool specifications were real, based solidly on measurements and analyses, there is so much variation between them that one would have difficulty in deciding what, indeed, constitutes a good quality floor.

For this reason, early in our involvement with the microelectronics industry, we started creating the "generic vibration criterion curves". The derivation of the curves is described in SPIE Proceedings Volume 1610, Nov. 1991, "Generic Criteria for Vibration-Sensitive Equipment", by Colin G. Gordon.

The criteria take the form of a set of one-third octave band velocity spectra labeled vibration criterion (VC) curves VC-A through VC-E. These are shown in Figure 11, together with the International Standards Organization (ISO) guidelines for the effects of vibration on people in buildings. The criteria apply to vibration as measured in the vertical and two horizontal directions. The application of these criteria as they apply to people and vibration-sensitive equipment are described in Table 2.

The main elements of the criteria are as follows:

(1) The vibration is expressed in terms of its root-mean-square (rms) velocity (as opposed to displacement or acceleration). It has been found in various studies that while different items of equipment (and people) may exhibit maximum sensitivity at different frequencies (corresponding to internal resonances), often these points of maximum sensitivity lie on a curve of constant velocity.

(2) The use of a proportional bandwidth (the bandwidth of the one-third octave is twenty-three percent of the band center frequency) as opposed to a fixed bandwidth is justified on the basis of a conservative view of the internal damping of typical equipment components. Experience shows that in most environments the vibration is dominated by broadband (random) energy rather than tonal (periodic) energy.

(3) The fact that the criterion curves allow for greater vibration velocity for frequencies below 8 Hz reflects experience that this frequency range, in most instances, lies below the lowest resonance frequency. Relative motions between the components are, therefore, harder to excite and the sensitivity to vibration is reduced.

(4) For a site to comply with a particular equipment category the measured one-third octave band velocity spectrum must lie below the appropriate criterion curve of Figure 11.

These equipment criterion curves have been developed on the basis of data on individual items of equipment and from data obtained from measurements made in facilities before and after vibration-related problems were solved. The curves are generic in the sense that they are intended to apply to broadly defined classes of equipment and processes. They are intended to apply to the more sensitive equipment within each category that is defined.

The criteria assume that bench-mounted equipment will be supported on benches that are rigidly constructed and damped so that amplification due to resonances is limited to a small value.

The criteria take into account the fact that certain types of equipment (such as SEMs) are supplied by the manufacturer with built-in vibration isolation.

It is important to note that these criteria are for guidance only. The "detail sizes" given in Table 2 appear to represent experience at the time of writing. They reflect the fact that the quality of design and of built-in isolation in most equipment tends to improve as dimensional requirements become more stringent. In some instances the criteria may be overly conservative because of the high quality of built-in isolation.

In most instances it is recommended that the advice of equipment manufacturers or of a vibration consultant be sought in selecting a design standard.

(8) Vibration Control

In this, the last, section we shall briefly review the principal methods of vibration control.

(a) Selection of Design Criteria - At the onset of the project the vibration goal for the completed facility must be selected. This requires knowledge of the likely toolsets that will be used, and the line-widths.

(b) Site Vibration Survey - In all instances it is advisable to evaluate the site, be it "greenfield" or an existing building. Vibration conditions on the site must lie below the selected design goal by some substantial margin. The site review must take into account the effects of future developments and construction.

(c) Layout - The layout of the facility must try to separate major sources of vibration from the vibration-sensitive tools. Vibration isolation is far from perfect and is also subject to deterioration. We have already commented on the limitations of building separations and isolation breaks at ground level. Distance separation is good - especially horizontally. A separation of 50 feet or greater is good for many situations; 10 feet or less can spell trouble.

(d) Structural Design - This is of critical importance, especially when the process floor is column-supported - lying above one or two levels of sub-fab. The most important characteristic is stiffness; high stiffness generally means high resonance frequencies and low response to input forces.

From a combination of studies using finite element modeling and statistical analyses of data from many operating facilities we have developed equations which adequately describe the performance of facilities which have been well-designed in terms of layout, selection and isolation of mechanical systems:

$$V_V = \frac{C_1}{K_V} \qquad ----(11)$$

$$V_H = \frac{C_2}{\sqrt{K_H}} \qquad ----(12)$$

where V_V and V_H are the maximum one-third octave band velocity amplitudes, respectively in the vertical and horizontal directions, K_V is the vertical stiffness of the floor at mid-bay between columns, and K_H is the "global" horizontal stiffness. C_1 and C_2 are constants.

To achieve adequate vertical stiffness for the most stringent floor performance, the columns must be close together and the floor adequately deep. Cast-in-place concrete waffle is the most common design for contemporary facilities. Precast concrete floors and even steel floors are possible, however.

To achieve adequate horizontal stiffness, especially for facilities with two levels of sub-fab, it is generally necessary to provide shear walls (or stiffening walls) to augment the column stiffness. Other methods are possible, even the use of steel or concrete diagonal braces at sub-fab level.

(e) Mechanical Equipment - Mechanical equipment and systems, including ductwork and pipework, form the primary source of vibration in an operating facility - assuming, that is, that the site has been sensibly chosen.

Certain types of equipment will generate more vibration than others, and it is well to avoid these wherever possible. Thus, for instance, reciprocating compressors should be avoided in favor of screw compressors. Similarly, direct-drive fans (with variable frequency drives, if necessary) should be favored over belt-driven fans.

When specifying mechanical equipment, it is important that the performance specification encompass dynamic balance (both factory and field, if necessary) and vibration isolation. The vibration isolation specification should be carefully prepared so that the possibility of errors or misunderstandings is minimized. Since there can be substantial variation in the quality of vibration isolators from one manufacturer to another, it will be advisable to list those manufacturers whose products are acceptable.

Fluid flow in pipework and ductwork can generate significant vibration. As shown in Eq. (1) the intensity of this vibration is proportional to the square of the flow velocity, and linearly proportional to the turbulence intensity. It is necessary in many cases to place limits on flow velocities and to strive for system configurations that help minimize flow turbulence. In many cases it is well to vibration-isolate pipework and ductwork where they lie close to the process floor. Direct contact of these systems with the process floor, even via spring hangers, should be avoided as much as possible.

Table 1 - Vibration Sources

Type	Source	Importance
Environmental	Wind	Generally not Important
	Seismicity	Generally not Important
	Railroads	Depends on Distance and Condition
	Highways/Roads	Depends on Distance
	Industry	Depends on Type & Distance
	Construction	Depends on Methods & Distance
Plant Exterior	Vehicles (Trucks)	Generally Not Important
	Nitrogen Plant	Depends on Distance and Type
	Emergency Generator	Depends On Distance and Isolation
	Sub-Station Transformers	Generally Not Important
	Central Utility Bldg.	Potentially a Major Source
Plant Interior	Make-up Air Handlers	Careful Design Required
	Recirculation Air Handlers	Careful Design Required
	Scrubbed Exhaust Fans	Direct Drive + Isolation
	General Exhaust Fans	Careful Design Required
	General Air Handlers	Depends on Distance
	Compressors	Screw Rather than Recip.
	Process and House Vac. Pumps	Location and Isolation
	Process Cooling Water Pumps	Potential Major Source
	DI Water and UPW Pumps	Potential Major Source
	UPS Systems	Depends on Type
	Duct and Pipe Turbulence	Requires Careful Design

Table 2: Application and interpretation of the generic vibration criterion (VC) curves (as shown in Figure 11)

Criterion Curve (see Figure B.1)	Max Level (1) micro-in/sec (dB)	Detail Size (2) microns	Description of Use
Workshop (ISO)	32000 (90)	N/A	Distinctly feelable vibration. Appropriate to workshops and nonsensitive areas.
Office (ISO)	16000 (84)	N/A	Feelable vibration. Appropriate to offices and nonsensitive areas.
Residential Day (ISO)	8000 (78)	75	Barely feelable vibration. Appropriate to sleep areas in most instances. Probably adequate for computer equipment, probe test equipment and low-power (to 20X) microscopes.
Op. Theatre (ISO)	4000 (72)	25	Vibration not feelable. Suitable for sensitive sleep areas. Suitable in most instances for microscopes to 100X and for other equipment of low sensitivity.
VC-A	2000 (66)	8	Adequate in most instances for optical microscopes to 400X, microbalances, optical balances, proximity and projection aligners, etc.
VC-B	1000 (60)	3	An appropriate standard for optical microscopes to 1000X, inspection and lithography equipment (including steppers) to 3 micron line widths.
VC-C	500 (54)	1	A good standard for most lithography and inspection equipment to 1 micron detail size.
VC-D	250 (48)	0.3	Suitable in most instances for the most demanding equipment including electron microscopes (TEMs and SEMs) and E-Beam systems, operating to the limits of their capability.
VC-E	125 (42)	0.1	A difficult criterion to achieve in most instances. Assumed to be adequate for the most demanding of sensitive systems including long path, laser-based, small target systems and other systems

Notes:

(1) As measured in one-third octave bands of frequency over the frequency range 8 to 100 Hz. The dB scale is referenced to 1 micro-inch/sec.

(2) The detail size refers to the line widths for microelectronics fabrication, the particle (cell) size for medical and pharmaceutical research, etc. The values given take into account the observation that the vibration requirements of many items depend upon the detail size of the process.

The information given in this table is for guidance only. In most instances, it is recommended that the advice of someone knowledgeable about applications and vibration requirements of the equipment and process be sought.

Figure 1: Concepts for Attenuation of Ground Vibration

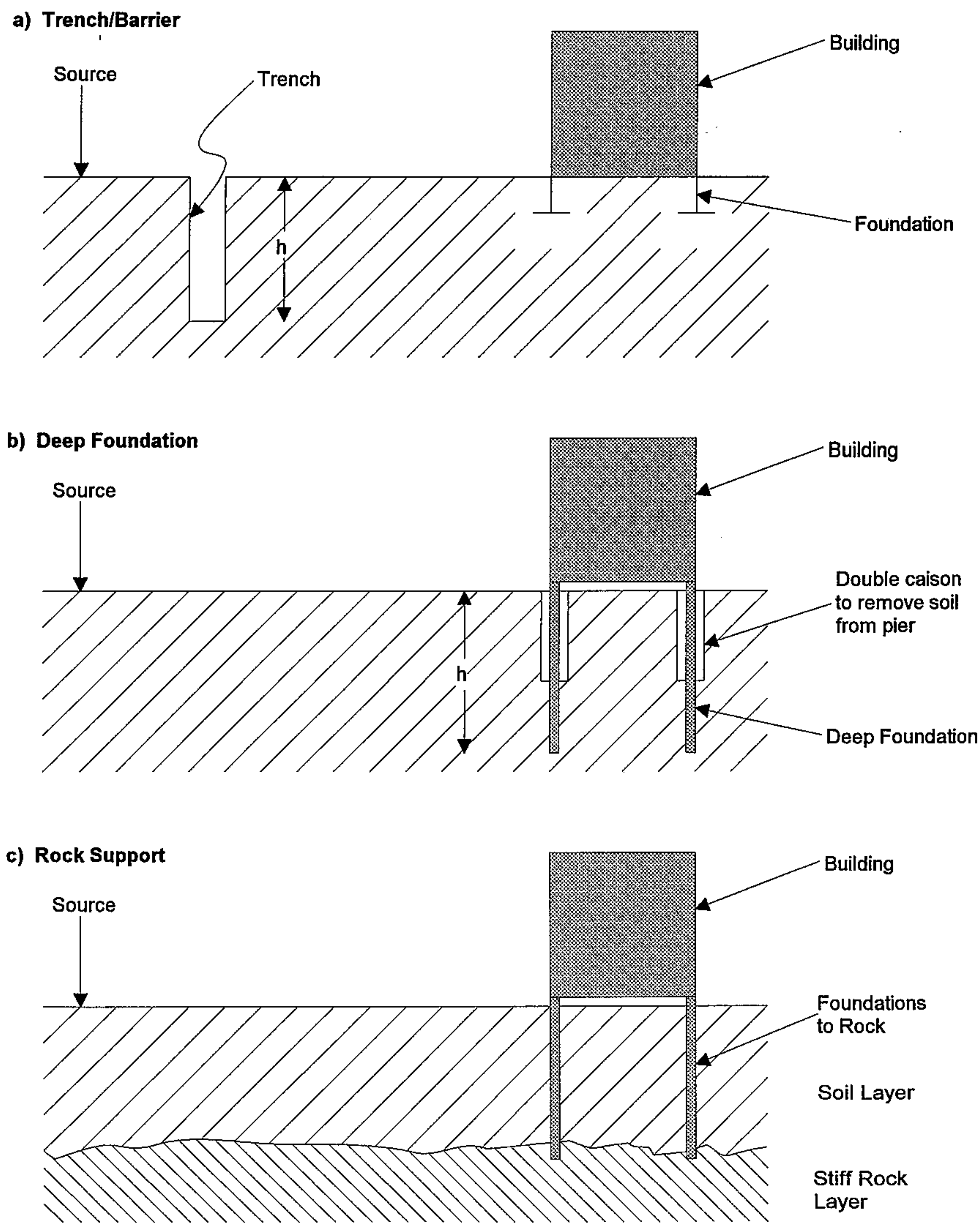


Figure 2: Effect of Isolation Break in Slab

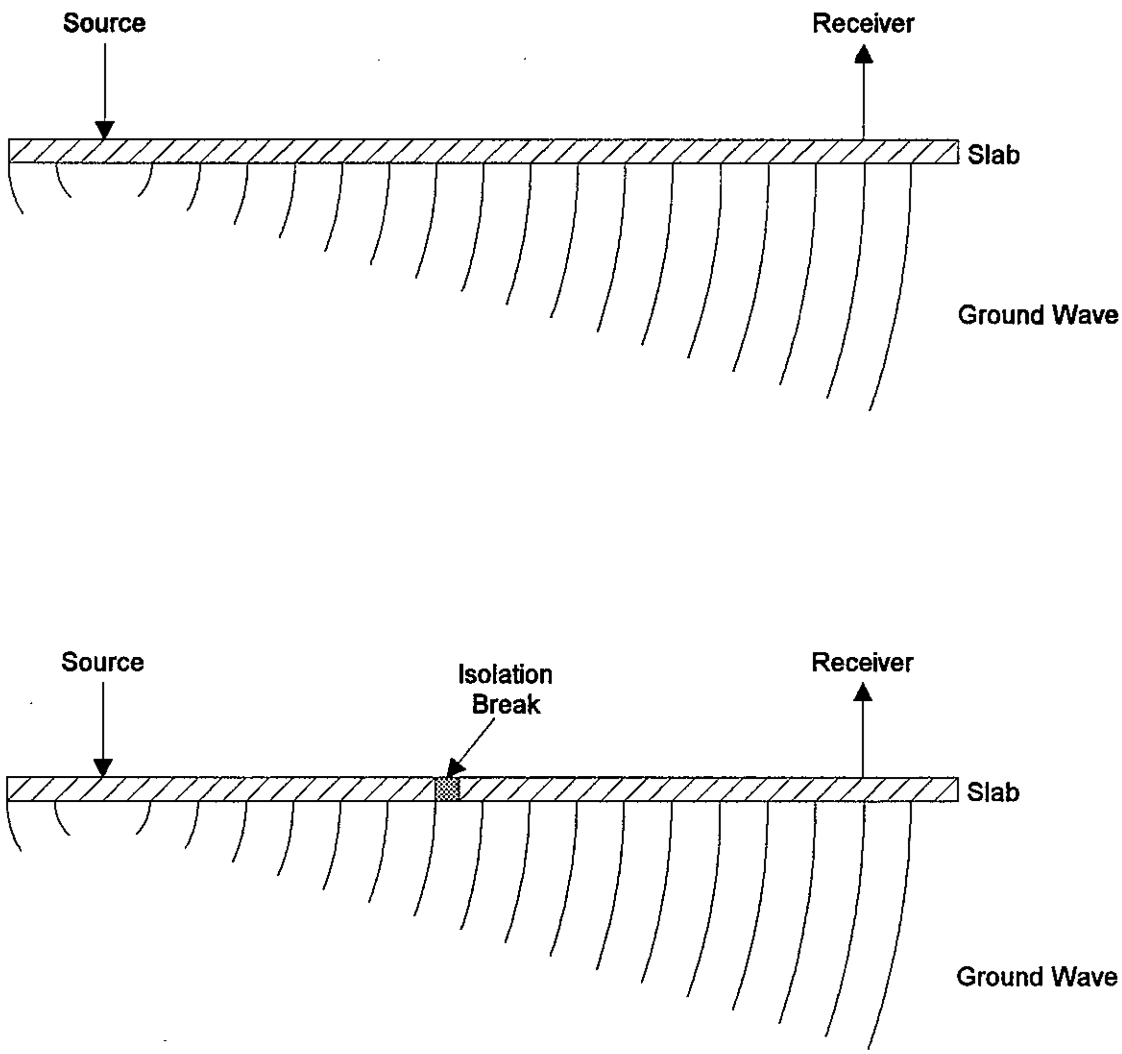


Figure 3: Examples of FEA Models

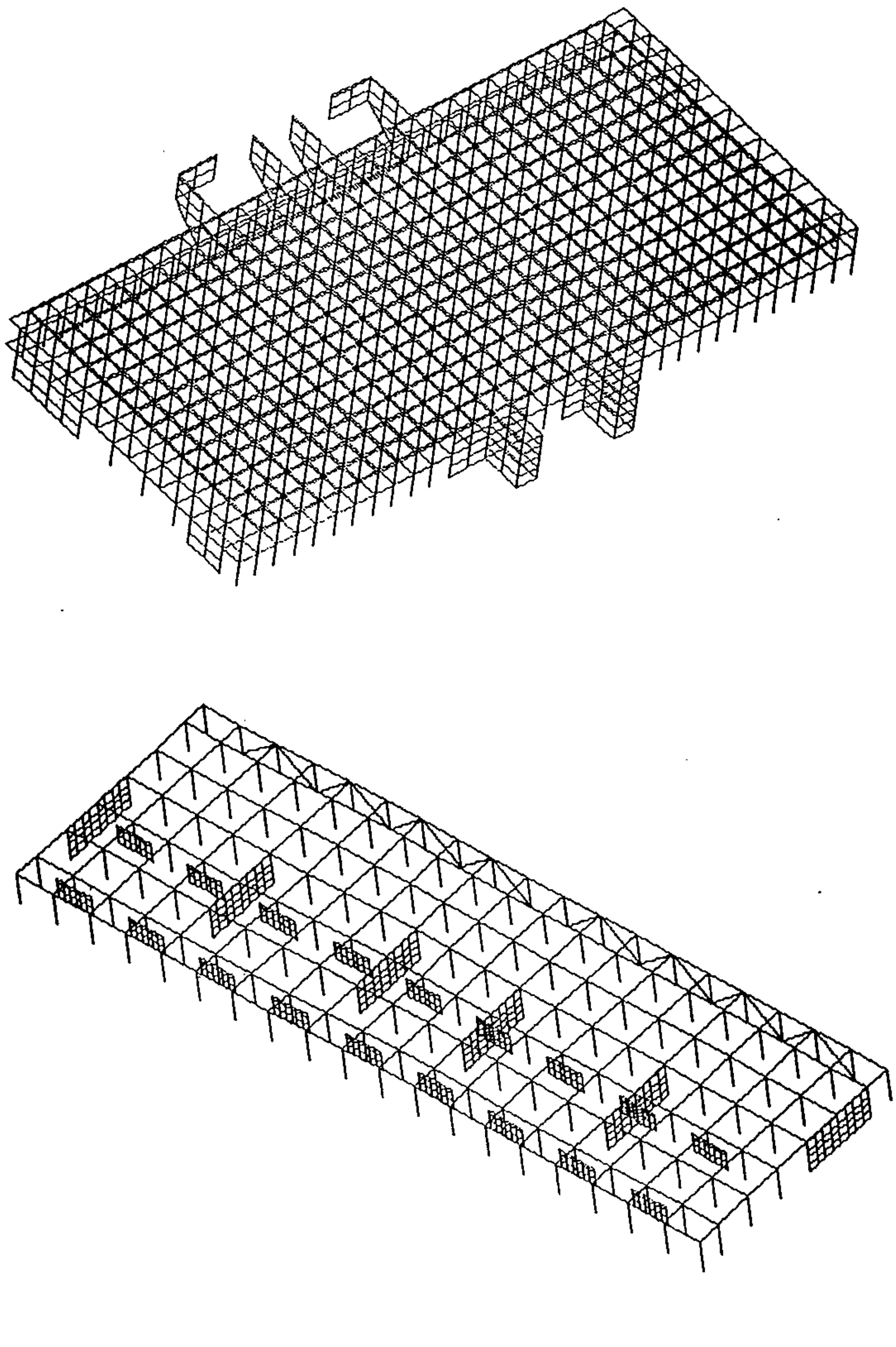


Figure 4: A Simple Sinusoidal Waveform Showing Peak-to-Peak, Zero-to-Peak and RMS Amplitudes

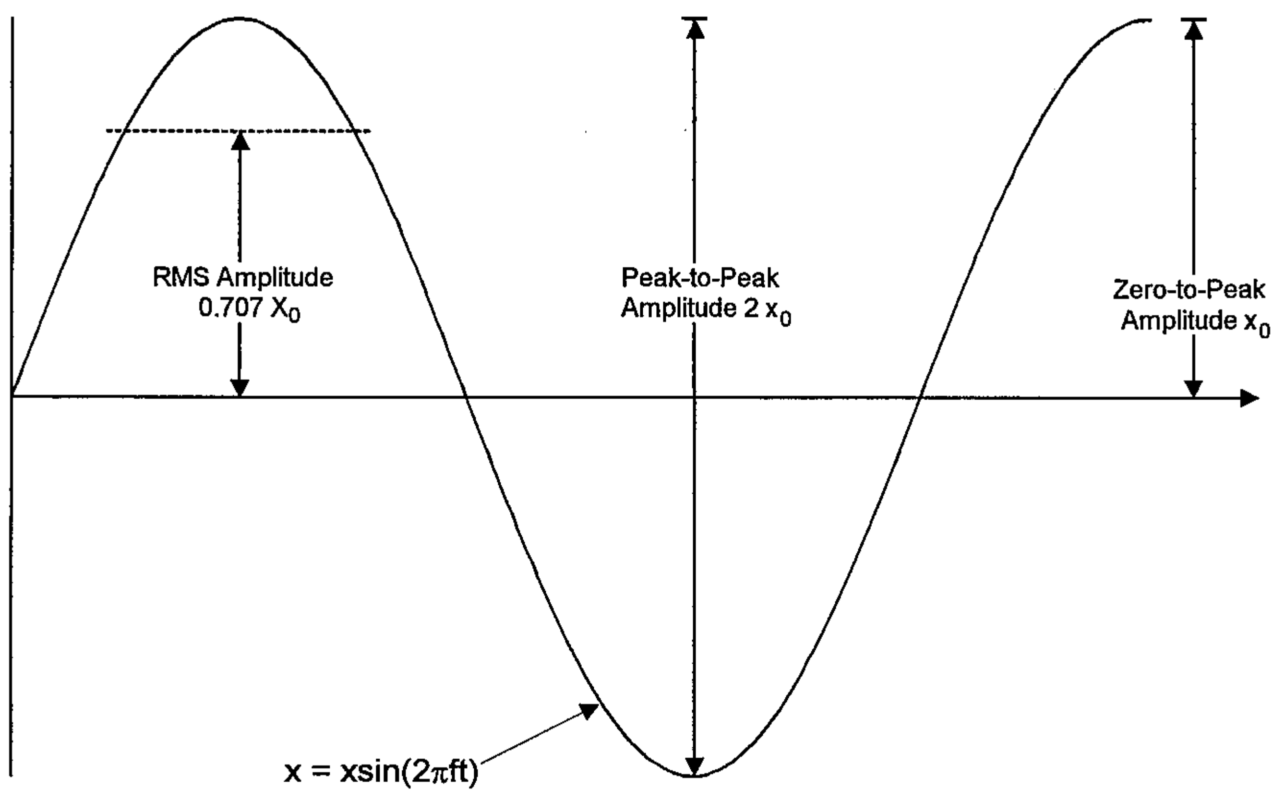


Figure 5: The Effect of Bandwidth on the Measured Amplitude of a Broadband Spectrum

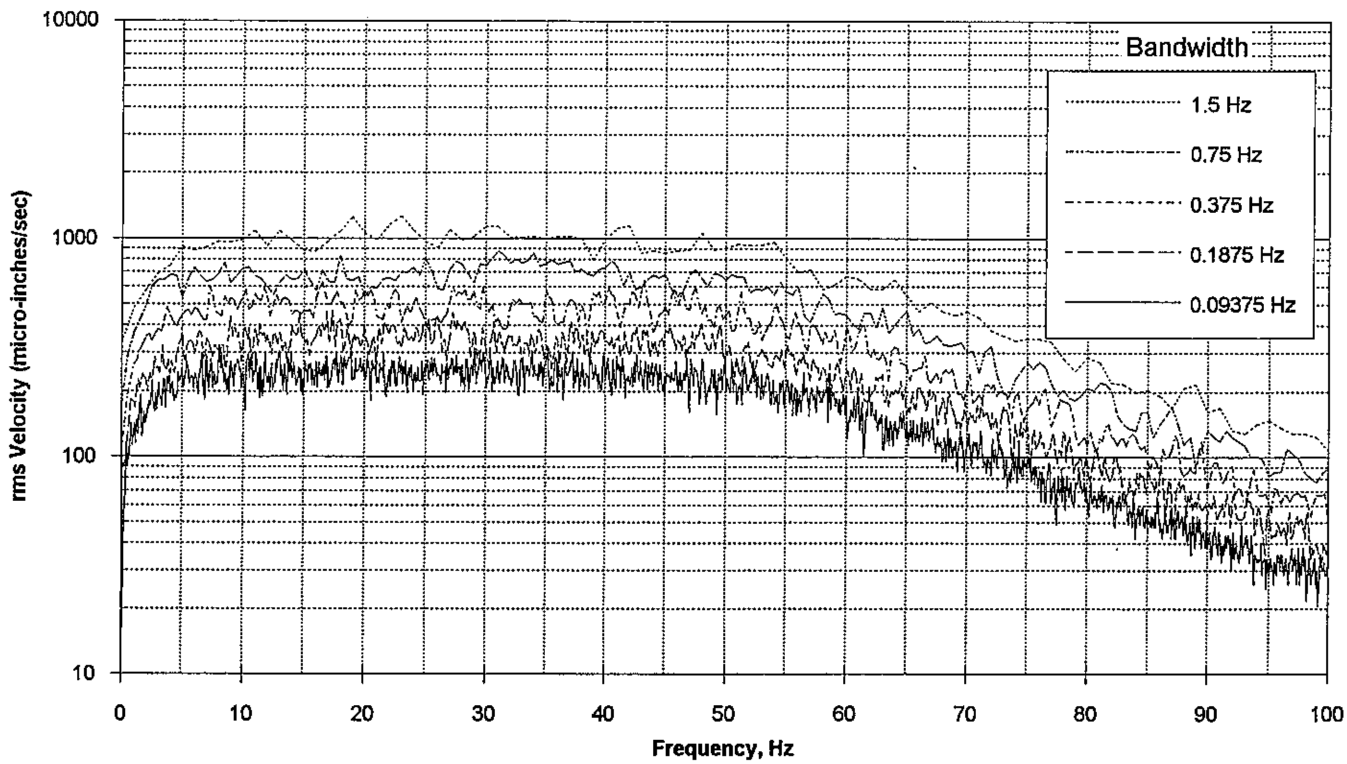


Figure 6: The Signal of Figure 5 Shown as a One-Third Octave Band Spectrum

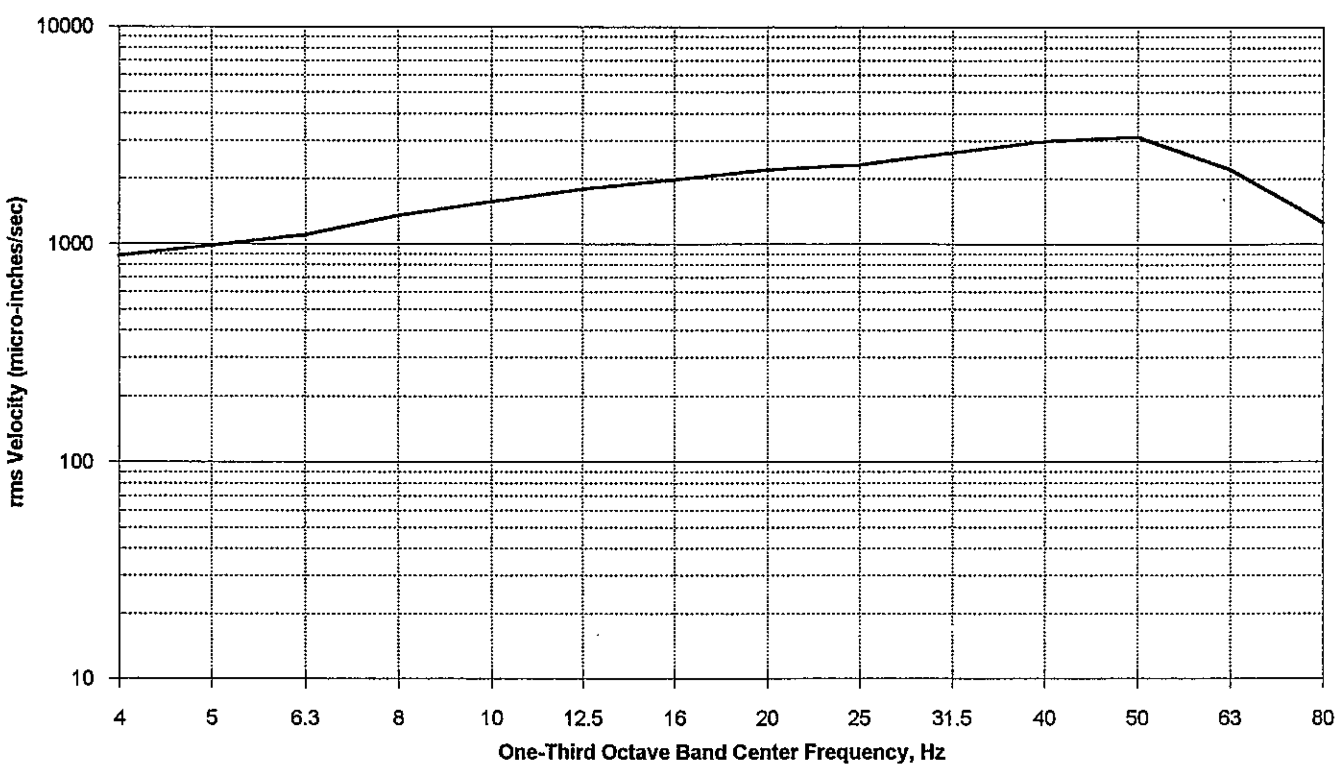
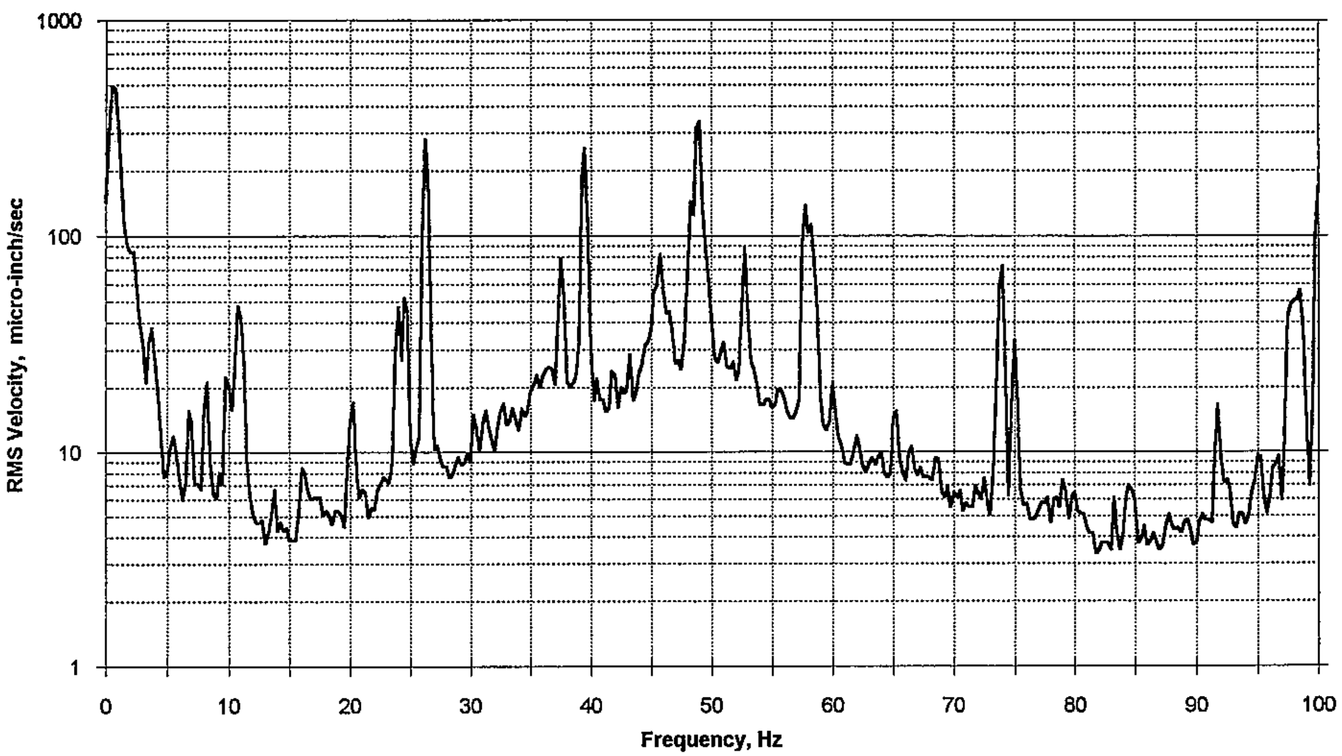


Figure 7: The Comparative Effects of Using Logarithmic and Linear Amplitude Scales

(a) Measured Vibration Displayed in Log Format



(b) Measured Vibration Displayed in Linear Format

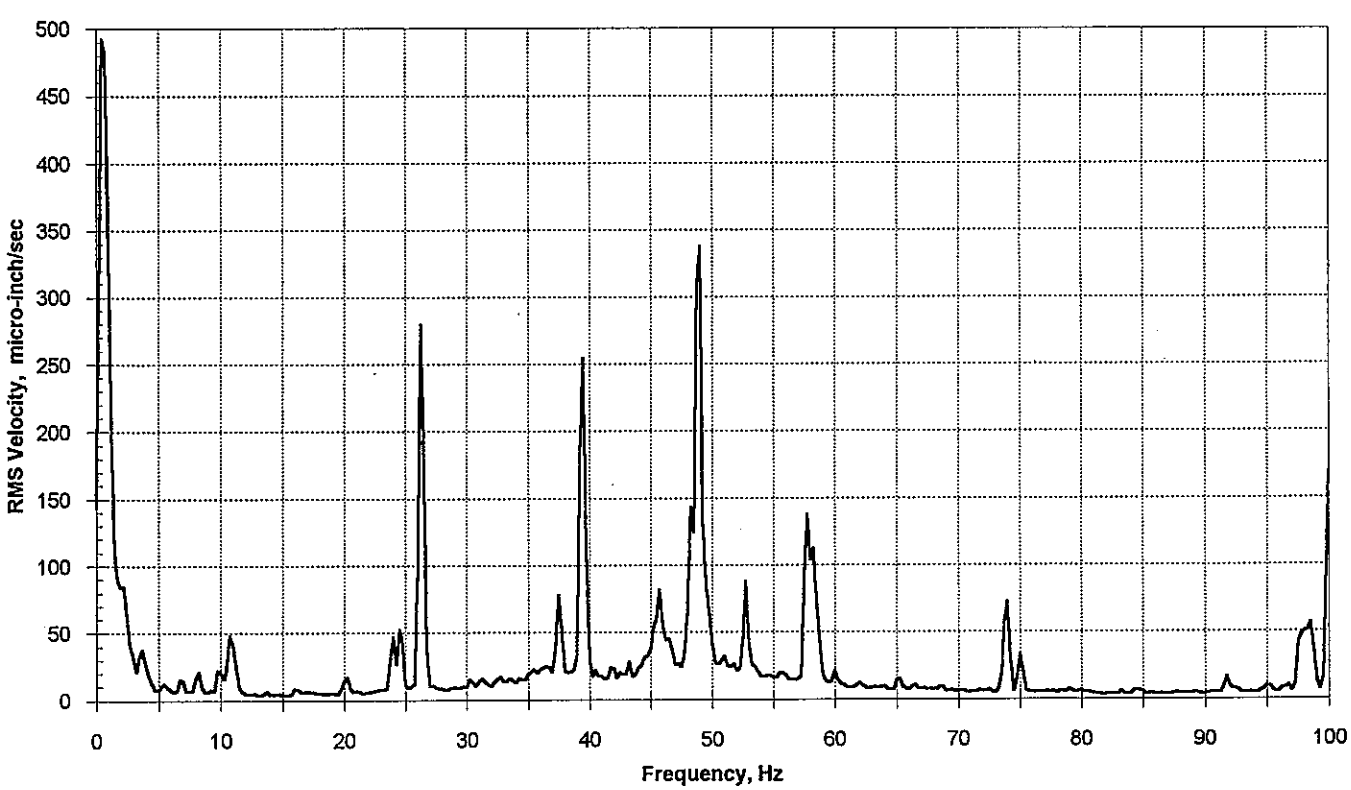
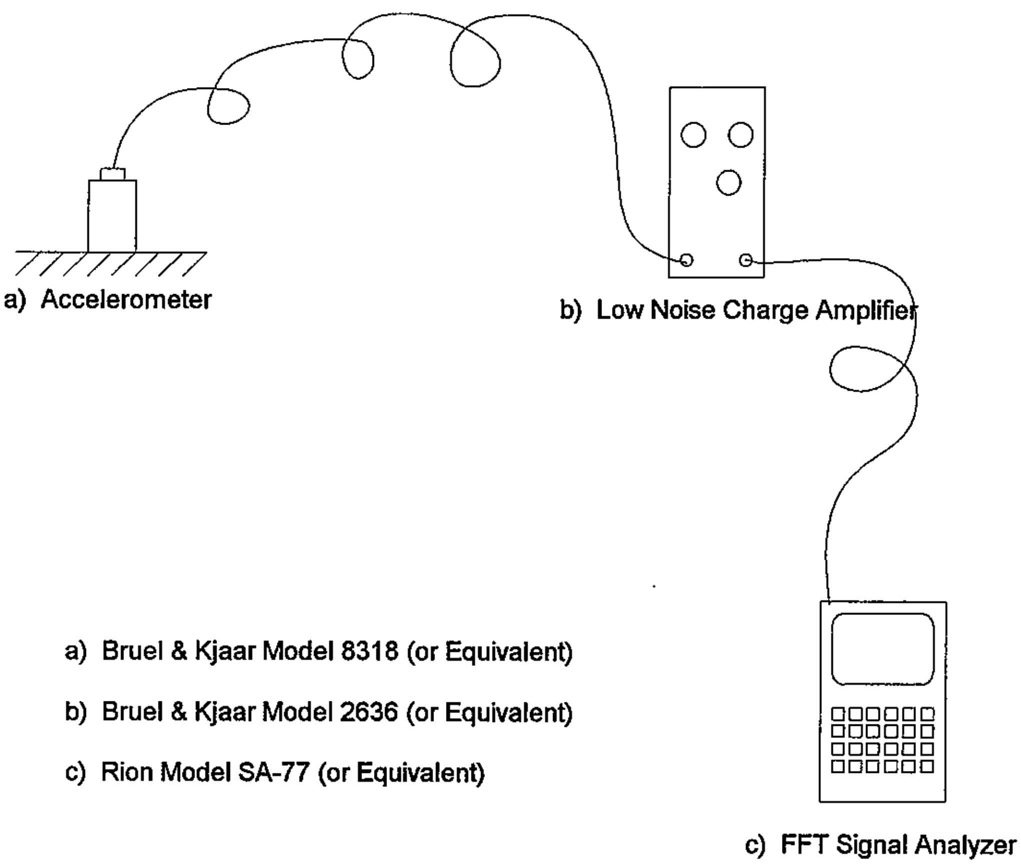


Figure 8: Instrumentation for Micro-vibration Measurements



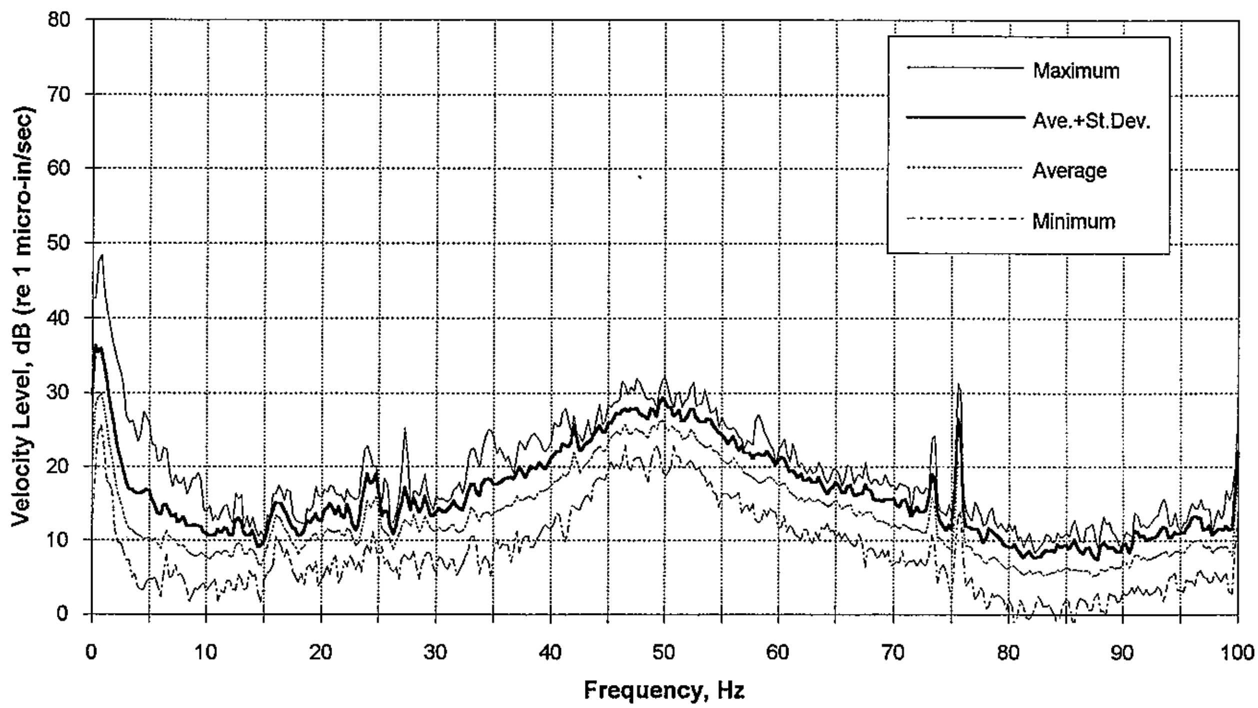
a) Bruel & Kjaar Model 8318 (or Equivalent)

b) Bruel & Kjaar Model 2636 (or Equivalent)

c) Rion Model SA-77 (or Equivalent)

Figure 9: Statistical Analysis of Vertical Vibration Levels for on Operating Facility at Start-Up

a) Narrowband Data (Bandwidth = 0.375 Hz)



b) One-Third Octave Band Data

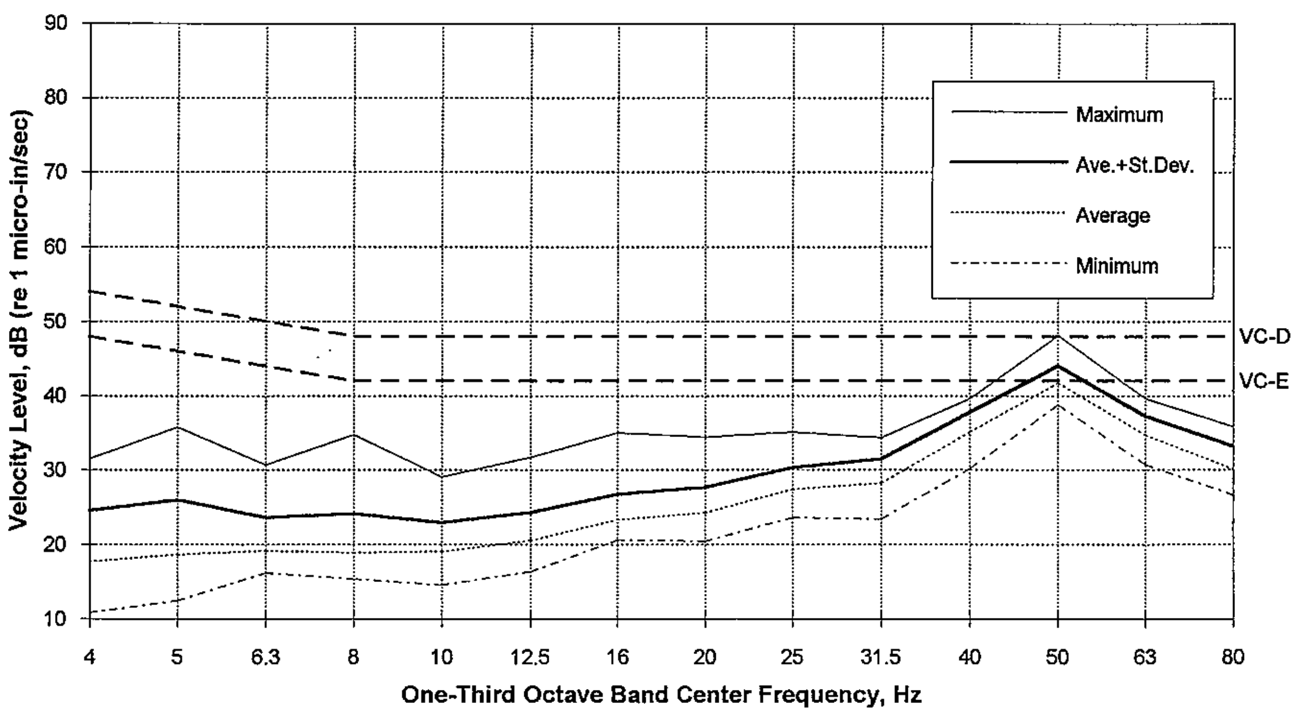
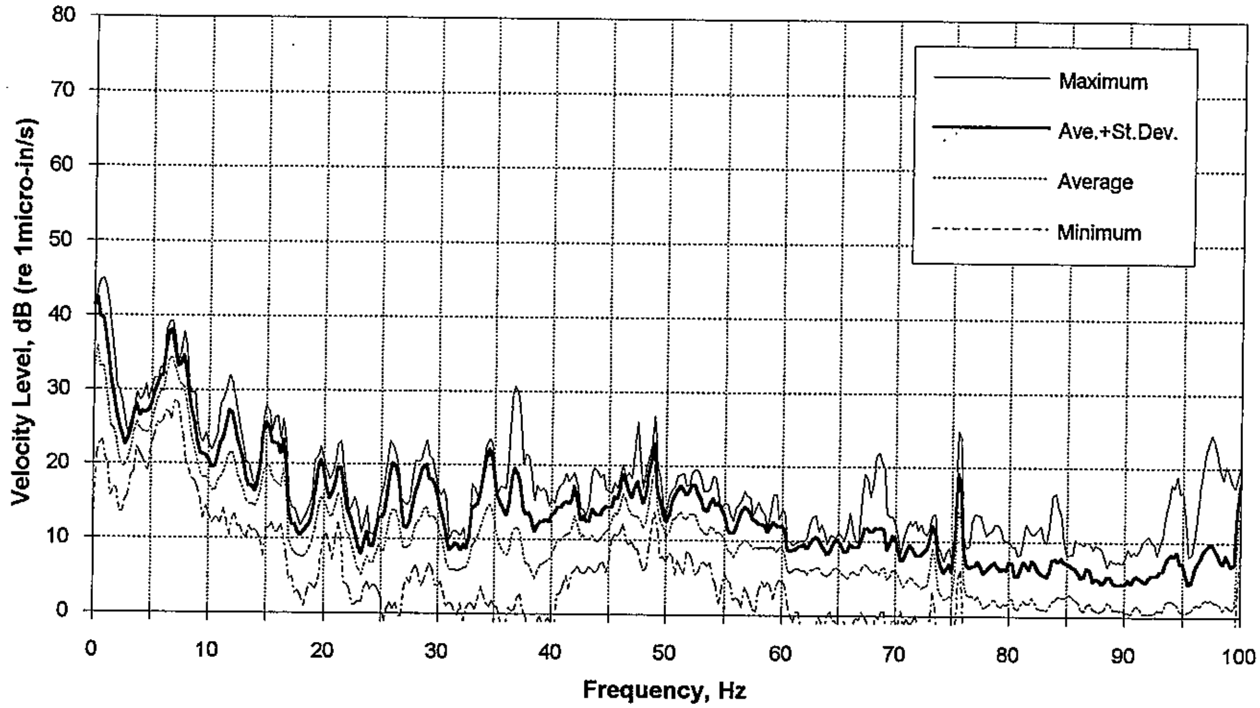


Figure 10: Statistical Analysis of Horizontal Vibration Levels for on Operating Facility at Start-Up

a) Narrowband Data (Bandwidth = 0.375 Hz)



b) One-Third Octave Band

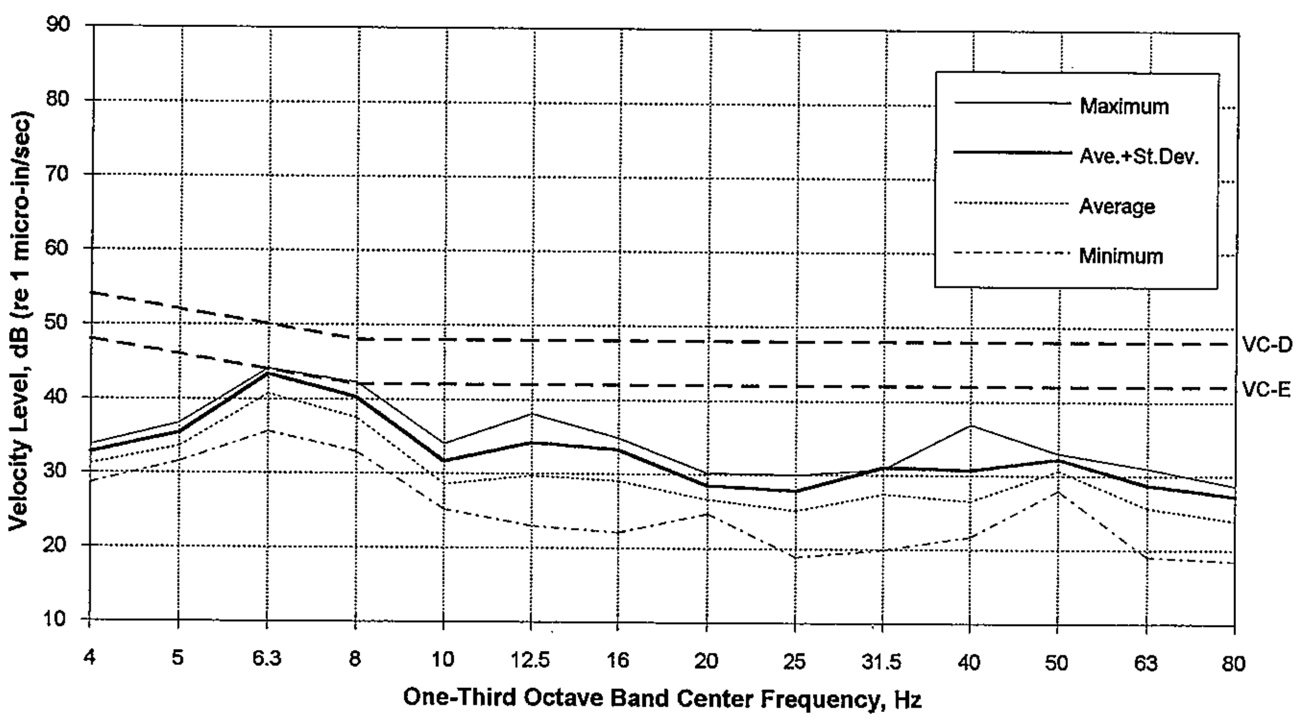


Figure 11: Generic vibration criterion (VC) curves for vibration-sensitive equipment -- Showing also the ISO Guidelines for People in Buildings (See Table 2 for description of equipment and uses.)