ACENTECH

A Vibration Primer for Architects

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The current trend toward the use of stronger materials has led to lighter structures, which tend to vibrate more readily. This, coupled with the increased prevalence of vibration-sensitive occupancies, implies that architects need to be conversant with vibrations in buildings. The present discussion is intended to provide some of the relevant background.

WHY VIBRATIONS ARE OF CONCERN

Vibrations in buildings need to be considered in relation to:

- Personnel comfort and the perception of quality
- Accommodation of vibration-sensitive equipment and activities (e.g., microscopes, surgery)
- Damage to equipment and structural elements

Sources of potentially significant vibrations abound within and outside of the building. They include:

Internal Sources

- The building's equipment and machinery (e.g., HVAC systems, elevators)
- Internal traffic (e.g., service vehicles, carts, personnel movement)
- Users' equipment (e.g., MRIs, production machines)

External Sources

- Micro-seismic background
- Street and rail traffic
- Aircraft, including helicopters
- Mechanical equipment (e.g., emergency generators, ventilation fans)
- Power plants and heavy industrial facilities
- Construction activities

ESTABLISHING AND MEETING VIBRATION GOALS

It generally is advisable to recognize the dominant vibration concerns early in the design process. Ideally, vibration goals and quantitative criteria should be established once the building's requirements have been identified, and these criteria should be taken into account as the design progresses.

Establishment of vibration goals needs to be based on the owner's requirements and on the criteria relevant to any sensitive equipment or activities planned to be housed in the building. Satisfaction of these typically involves providing a favorable building layout, suitable structures, and appropriate selection, location, and vibration isolation of mechanical equipment. Suitable site selection and planning, aided by on-site vibration monitoring, may be important in areas where significant disturbances are likely – for example, at heavily built-up industrial sites, and near airports, busy highways, or rail corridors.

Many specialized tools are available for meeting any vibration requirements. A project's structural and mechanical engineers typically are conversant with many of these, but an architect generally is well served by employing the services of a consultant to ensure that all vibration-related aspects are addressed appropriately.

HOW VIBRATIONS ARE DESCRIBED

Unlike temperature, a vibration cannot be characterized by a single number without considerable added discussion. Even at just a single fixed point, the motions associated with a vibration may occur in any direction – and the motions of adjacent points may occur in different directions. In some situations, however, vibrational motions in one direction tend to predominate. For example, footfall-induced vibrations of floors occur primarily in the vertical direction, whereas wind-induced vibrations of buildings are associated primarily with horizontal motions.

Although everyone has some intuitive understanding of what we mean by vibration, let us just state that by vibration we basically mean a back-and-forth motion of some kind. The simplest such motion may be visualized by a pendulum that is deflected a certain amount and then released. This motion has a character like that illustrated in Fig. 1, below.

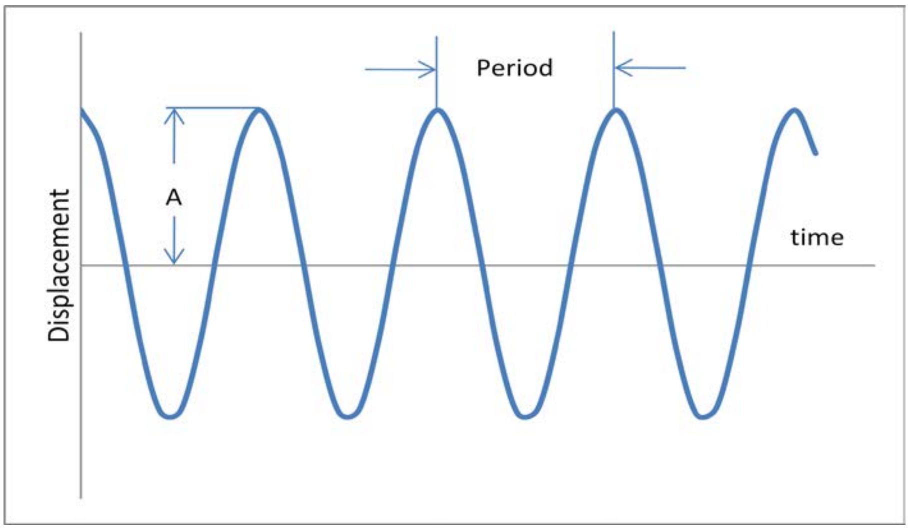


Figure 1

The greatest excursion from the undisturbed (zero) position in any direction is called the *amplitude*, indicated by "A" in the figure. The time taken for the excursion to go from one maximum to the next – that is, through one full *cycle* – is called the *period* of the vibration. The number of cycles that the excursion goes through in one second is called the *frequency*. The frequency is usually given in *Hertz (Hz)*, which is the name given to "cycles per second".

In the foregoing example we have neglected the fact that the amplitude of vibrations of a pendulum, or of anything else, that is briefly disturbed and then let go will decrease as time progresses and as energy is lost from the vibrating system. As the result, the excursion will behave somewhat as shown in Fig. 2

Since decaying vibrations are of minor importance for the present discussion, we shall not deal further with them here.

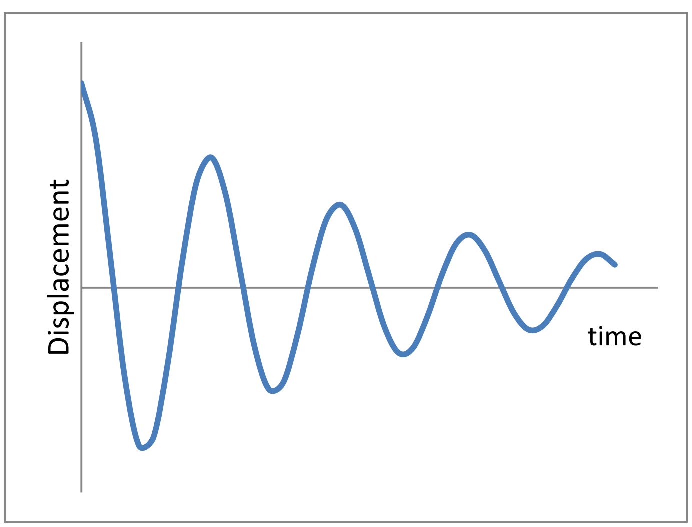


Figure 2

What is more important to note is that most vibrations of interest are not made up of just one single frequency component, but generally of many such components. Figure 3 shows a three-component vibration (the solid purple curve) and the three components of which it is composed.

Note that the three components have different frequencies and amplitudes.

The character of a multi-component vibration typically is shown in terms of a *spectrum* – that is, a plot showing what amplitudes are present at each frequency. The following plot (Figure 4) is a spectrum that corresponds to the three-component vibration of

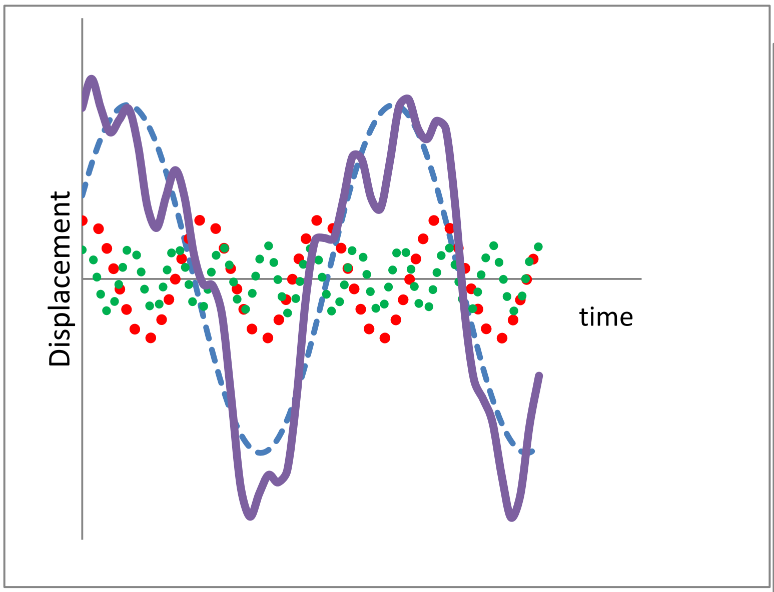


Figure 3

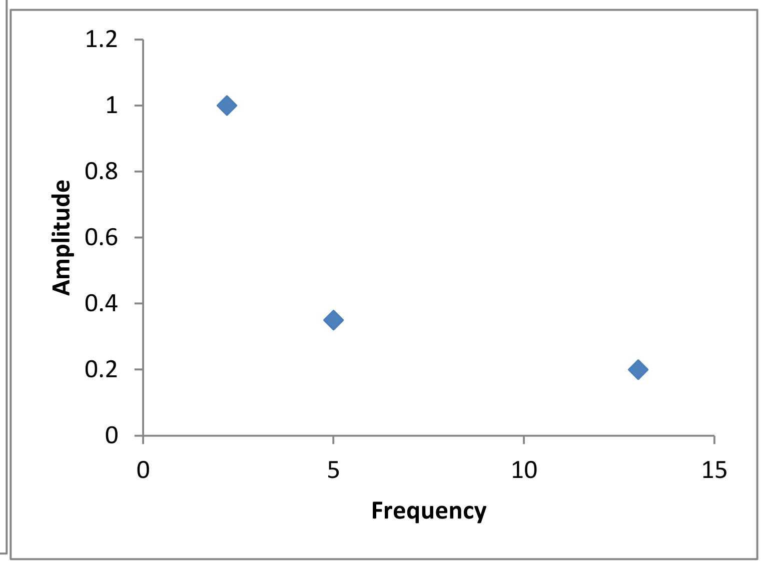


Figure 4

Fig. 3. Many real-world vibrations involve a multitude of components, leading to a spectrum like that of Fig. 5, below.

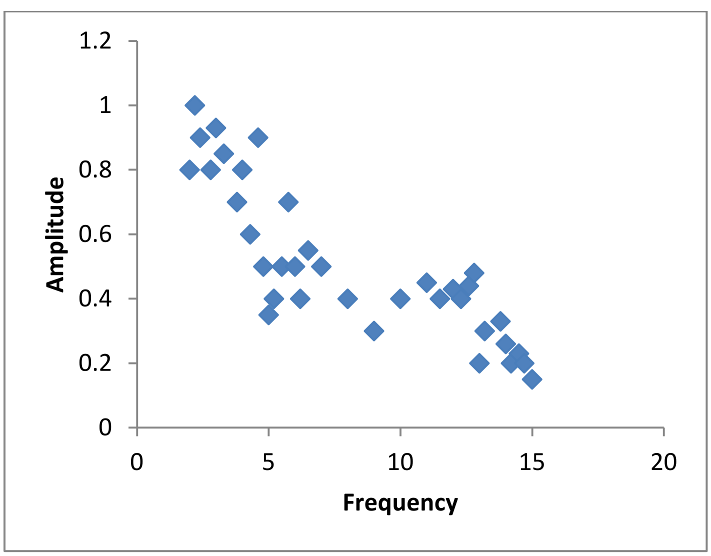


Figure 5

Because this spectrum representation requires many points it usually is convenient to describe a multi-component vibration in terms of representative amplitudes in bands of frequency, rather than indicating the amplitude at each frequency. The representative amplitude in a band may be chosen as the greatest amplitude of any component in the band or as some average or sum of the component amplitudes. In many cases the root-mean square (the square-root of the average of the squared amplitudes) is used. This often is abbreviated as “rms” and sometimes called the “energy average”. A spectrum of this sort is entirely meaningful only of the frequency bandwidth and the type of averaging is indicated.

The figure, Fig. 6, below, shows the previous spectrum, together with the maximum values (red lines) and the rms values (green lines) in 2.5 Hz wide frequency bands. In each case, only six amplitudes characterize the whole multi-component spectrum.

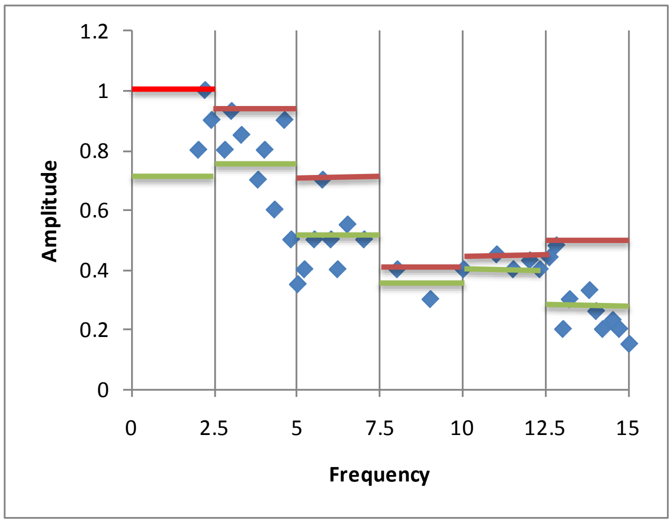


Figure 6

If a vibration includes components that encompass a large range of frequencies, then use of bands of constant width may still involve a great number of points. In that case it generally is convenient to use bands whose width increase in proportion to frequency. *Octave bands* are widely used type of such proportional bands; here each band extends from a beginning frequency to twice that frequency. In *one-third-octave bands* the upper frequency is $\sqrt[3]{2} \approx 1.26$ times the beginning frequency. It is common practice to identify proportional bands by their center frequencies, where the center frequency is the geometric average of the band’s beginning and ending frequencies. Standardized octave and one-third-octave bands are in predominant use.

SOME IMPORTANT NUMBERS; CRITERIA

Previously we discussed vibrations in terms of displacement of a point – that is, how far a point moves from a reference location. Displacements of structural elements generally are important only if one is concerned about deformations or about an element colliding with something else. More often, the velocity of an item – that is, how fast it moves or how fast its displacement changes – is more directly related to perception, structural damage, or equipment malfunction. In many instances an item’s acceleration (how fast its velocity changes) is also of interest in relation to malfunction or damage.

In steady vibrations, the velocity amplitude (in inches per second or millimeters per second) is equal to the displacement amplitude (in inches or millimeters, respectively) times 2π times the frequency (in Hz). Similarly, the acceleration amplitude (in inches per second per second[1] or millimeters per second per

[1] Usually written as in/sec^2

second²) is equal to the velocity amplitude (in the aforementioned units) times 2π times the frequency (in Hz). For example, to a displacement amplitude of 0.03 inches at 70 Hz there corresponds a velocity amplitude of 0.3·2π·70=132 in/sec (approximately) and an acceleration amplitude of 132·2π·70=58,000 in/sec^2.

Many vibration criteria are stated in terms of the greatest acceptable velocity. The frequency range, bandwidth, and the type of averaging to be used often are implied, rather than stated explicitly. Fig. 7 gives an indication of the velocity magnitudes that correspond to some widely used criteria[3].

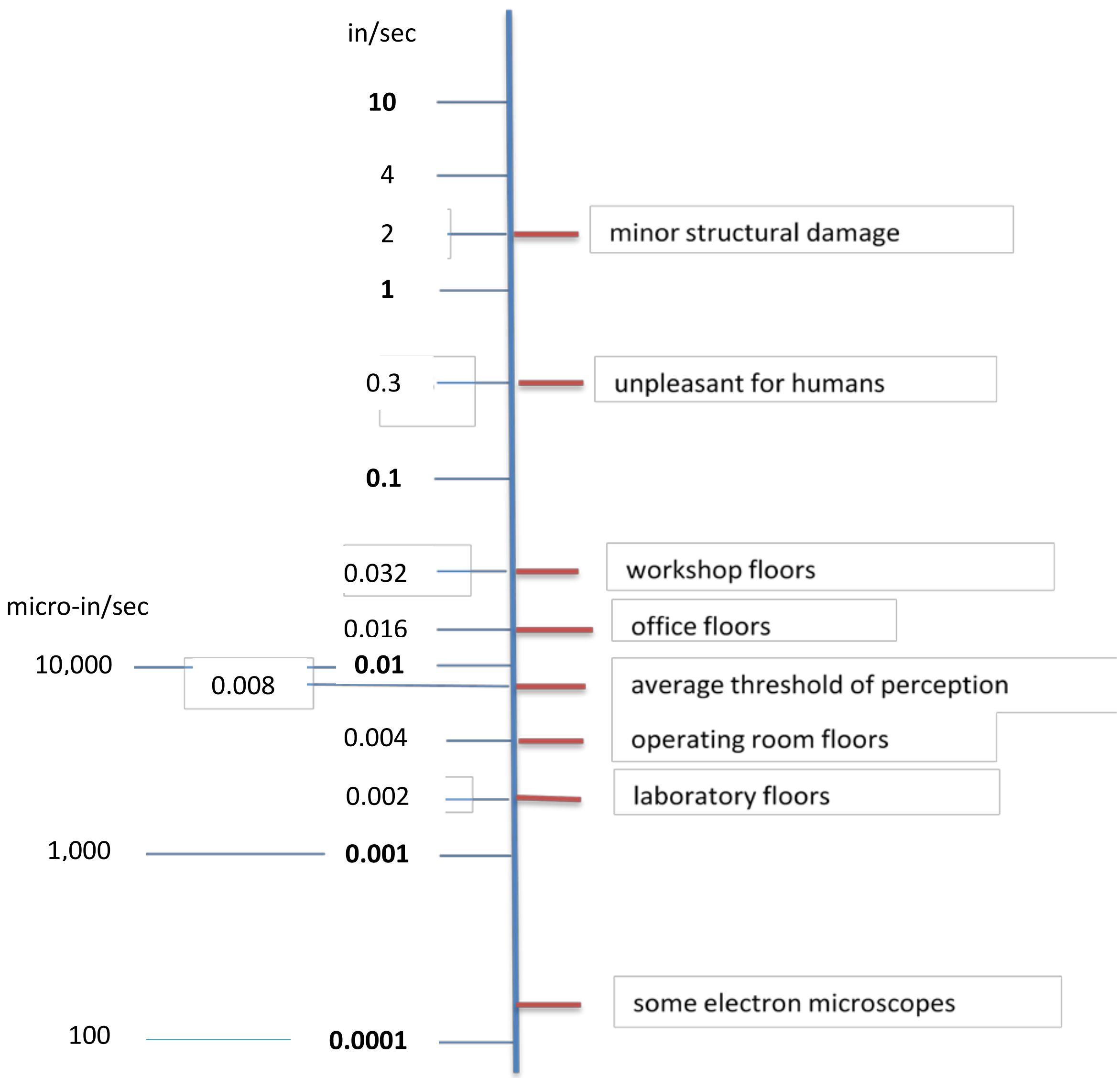


Figure 7

[2] Usually written as mm/s^2

[3] The human perception and floor vibration criteria in essence pertain to rms values in the one-third-octave bands between 4 Hz and 80 Hz. The other limits are not well defined.

HOW THINGS VIBRATE

Consider a weight hanging on a spring, as shown in Fig. 8. If the weight is moved and released, it will bounce at a given frequency (with a motion like that of Fig. 1). This is its so-called *natural frequency* – the frequency at which it "likes" to vibrate. If the spring stiffness is k pounds per inch (that is, if it takes k pounds to deflect the spring one inch), then hanging a weight W onto the unloaded spring – see the left sketch of Fig. 8 – will deflect it statically by an amount $X_{st} = k/W$ inches.

X_{st} is called the *static deflection* of the spring due to the added weight; it is related to the natural frequency of this spring-and-weight system f by

$$f(Hz) = \frac{1}{2\pi}\sqrt{\frac{kg}{W}} \approx \frac{3.13}{\sqrt{X_{st}(in)}}$$

where g denotes the acceleration of gravity =386 in/sec^2. Note that the natural frequency increases as the spring stiffness is increased and/or as the mass is decreased.

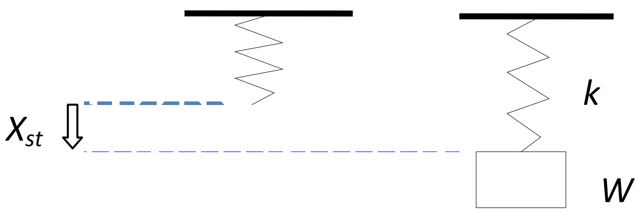


Figure 8

Relations similar to the foregoing one apply also for beams and plates and for more complicated structures, but involving more complicated expressions for the relevant stiffness and weight terms. In fact, beams and other structural elements can vibrate in many different ways if they are deflected and released or if they are impacted – depending on how they are deflected or impacted. Figure 9 illustrates the four simplest deflection shapes for a uniform beam; the top-most corresponds to the lowest, or "fundamental", natural frequency and the others to increasingly higher natural frequencies.

It turns out that the foregoing expression in terms of the static deflection X_{st} applies also for uniform beams, but with the 3.13 replaced by 3.53 and with the static deflection X_{st} replaced by the mid-point deflection of the beam due to its self-weight.

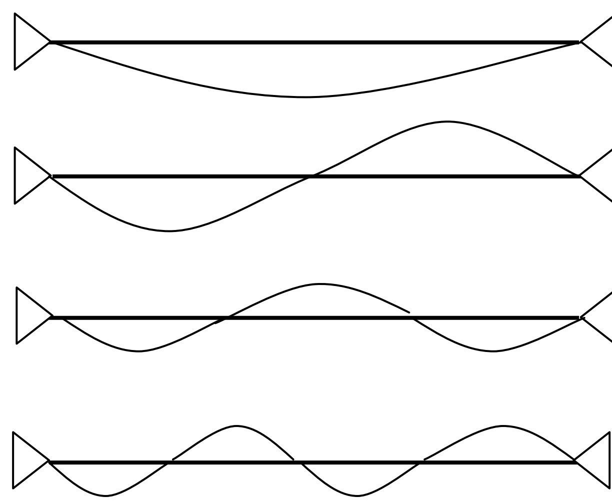


Figure 9

If a component is set into motion and then allowed to vibrate without any further disturbance, it vibrates at its natural frequency or frequencies. However, if the component is subjected to a steady vibratory force or motion, then it will vibrate at the frequency or frequencies at which these disturbances occur. Figure 10 illustrates two basic cases of particular importance. The left diagram represents a spring-supported item (such as a machine with a rotating imbalance) that generates a vibratory force, where forces are transmitted to the support via the springs. The right-hand diagram shows a vibrating support whose oscillatory motion is transmitted to a spring-supported item (perhaps a sensitive instrument). The transmitted forces and motions differ from the applied forces and motions due to the effects of the masses and springs.

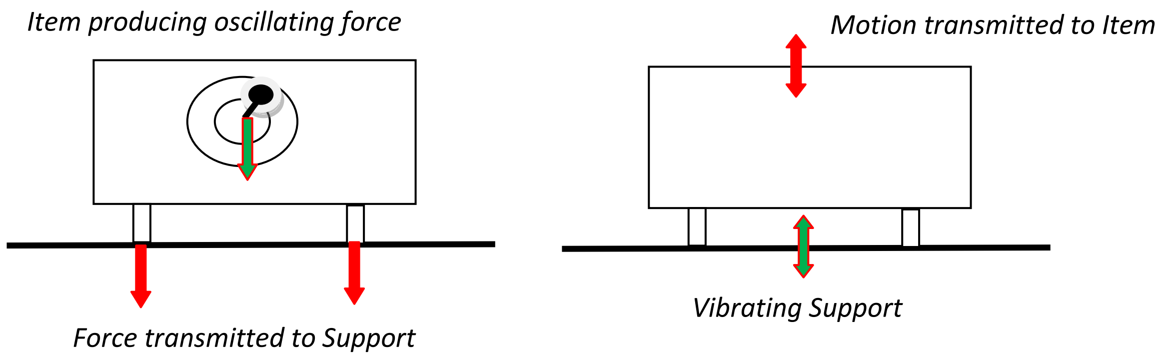
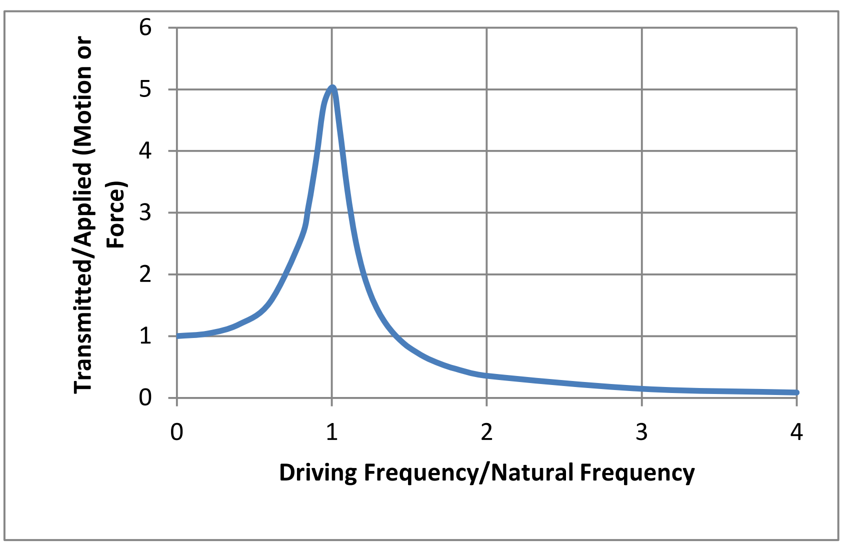


Figure 10

Figure 11 indicates the typical behavior of such arrangements. Each of these systems has a natural frequency that depends on the mass (or weight) and on the spring constant of the supports. If the applied vibration[4] occurs at the natural frequency – that is, if the driving frequency is equal to the natural frequency – then the transmitted vibration is much greater than the applied vibration. This frequency-matching condition is called *resonance*; the amplification of the vibration that occurs in this case depends on the energy dissipation capability (i.e., the "damping") of the system.



If the driving frequency is very low compared to the natural frequency, then the transmitted vibration is very nearly the same as the applied vibration; because the motions are slow (that is, the accelerations are small), the inertia of the mass has little effect. On the other hand, if the applied vibration occurs at a frequency that is considerably above the natural frequency, then the transmitted vibration is much smaller than the applied vibration (and, in fact, decreases rapidly as the driving frequency increases); because the motions are fast (that is, the accelerations are large), the inertia of the mass plays a dominant role in limiting the motion.

[4] For the case corresponding to the left-hand diagram, "vibration" refers to forces; for the case corresponding to the right-hand diagram, "vibration" refers to motions.

The discussion related to the two foregoing figures is the basis for vibration isolation. Since one generally wants the transmitted vibrations to be small, one desires the system to operate in the region where the driving frequency is considerably above the natural frequency. Since the driving frequency generally is determined by the equipment or structure of concern, one typically wants to make the system's natural frequency as low as possible. In practice, this generally means selecting the most flexible practical resilient supporting springs or isolators.

A CHECK-LIST OF DESIGN CONSIDERATIONS

Addressing External Sources

- ✓ Select a quiet site – as distant as possible from current and future sources of disturbances (e.g., rail lines, busy roads, flight paths near airports, power plants, extensive construction activities).
- ✓ Attenuate external sources (e.g., by smoothing roads, restricting speeds and access of heavy vehicles).
- ✓ Shield the building or part of it from intruding vibrations (e.g., provide base isolation of entire building; construct isolated "islands" in the building).
- ✓ Locate sensitive areas in the building as far as possible from major sources.

Dealing with Internal Sources

- ✓ Locate mechanical equipment and other potential sources of vibrations (e.g., users' equipment) as far as possible from sensitive areas.
- ✓ If possible, select alternative mechanical equipment items that inherently produce relatively little vibration (e.g., use rotating rather than reciprocating machines).
- ✓ Provide mechanical equipment and its piping, ducts, and conduits with efficient vibration isolation.
- ✓ Add structural joints to impede the transmission of vibrations to sensitive areas.
- ✓ Design floor structures that are relatively rigid in order to reduce the vibrations produced by personnel activities.
- ✓ Locate corridors outside of the structural bays that house sensitive areas.
- ✓ Keep areas traversed by in-building vehicles smooth, free of joints and bumps. (Also specify carts, etc. with soft pneumatic wheels of large diameter.)

Protecting Sensitive Equipment

- ✓ Generally locate the most vibration-sensitive items of equipment on grade or in a basement, where the vibrations due to internal disturbances typically are relatively small.
- ✓ Locate vibration-sensitive equipment items that are to be housed on supported floors near columns or atop girders or major beams.
- ✓ Provide sensitive equipment with appropriate vibration isolation.

------ End ------

Since 1958 Eric Ungar has been involved in research and consulting in vibration and in the related fields of structure-borne sound, structural and machinery dynamics, stress analysis, and noise and vibration control; he is considered one of the topmost experts in the vibration area. The American Society of Mechanical Engineers awarded him its Gold Medal for Noise Control and Acoustics for "fundamental contributions to noise and vibration control engineering involving structural damping, vibration isolation, vibration of complex structures as applied to aerospace structures, ships, machinery, and buildings." He has received similar recognition from the Acoustical Society of America and the Institute for Noise Control Engineering, of which organizations he also has served as President, and he has been honored with a lifetime achievement award by the Shock and Vibration Information Center. Eric is a registered professional engineer and the author of over 200 publications, including over a dozen handbook and monograph chapters. He holds the Doctor of Engineering Science degree from New York University.