Chapter 10: Vibration Isolation of the Source

Introduction:

High vibration levels can cause machinery failure, as well as objectionable noise levels. A common source of objectionable noise in buildings is the vibration of machines that are mounted on floors or walls. Obviously, the best place to mount a vibrating machine is on the ground floor. Unfortunately, this is not always possible. A typical problem is a rotating machine (such as a pump, AC compressor, blower, engine, etc) mounted on a roof, or on a floor above the ground floor. The problem is usually most apparent in the immediate vicinity of the vibration source. However, mechanical vibrations can transmit for long distances, and by very circuitous routes through the structure of a building, sometimes resurfacing hundreds of feet from the source.

Vibration isolation is the process of isolating an object, such as a piece of equipment, from the source of vibration. Passive vibration isolation' refers to vibration isolation or mitigation of vibrations by passive techniques such as rubber pads or mechanical springs, as opposed to 'active vibration isolation' or 'electronic force cancellation' employing electric power, sensors, actuators, and control systems. vibrations have undesirable effects on both human quality of life, and on our material goods. Vibration-generating machinery and processes contribute, to a large extent, to the total noise and vibration exposure. A very useful strategy to reduce noise and vibrations is to interrupt the propagation path between the source and the receiver. *Elastic mounting* is a simple method to hinder the spread of structural vibrations. In practice, an elastic mounting system is realized by incorporating so-called vibration isolators along the propagation path. Strongly vibrating machines in factories, dwellings, and office buildings can be placed on elastic elements. The passenger compartments in vehicles are isolated from vibrations generated at the wheel-roadway contact by incorporating springs between the wheel axles and the chassis. With properly design and implementation, elastic mounting of machines is both an effective and an inexpensive approach to noise and vibration mitigation. The objective of this chapter is both to provide the essential knowledge required to properly design vibration isolation systems, and to impart a physical understanding of the principles used in vibration isolation.

A vibration problem can also be nicely described by the same source – path – receiver model we previously used to characterize the noise control problem. [J. S. Lamancusa Penn State USA]

Mechanism of vibrations transmission:

Source: a mechanical or fluid disturbance, generated internally by the machine, such as unbalance, torque pulsations, gear tooth meshing, fan blade passing, etc. These typical occur at frequencies which are integer multiples of the rotating frequency of the machine.

Path: the structural or airborne path by which the disturbance is transmitted to the receiver

Receiver: the responding system, generally having many natural frequencies which can potentially be excited by vibration frequencies generated by the source. (Murphy says the natural frequency of the system will always coincide with an excitation frequency.)

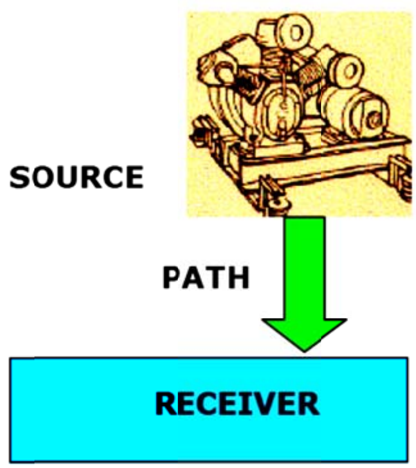

Fig.3.14 Flow diagram of Vibration transmission

Vibration Control solution:

The best solution to a vibration problem is to avoid it in the first place. Intelligent design is far more cost effective than building a bad design and having to repair it later. Methods for vibration control of industrial equipment include; Force Reduction, Mass Addition, Tuning, Isolation, and Damping.

The intelligent solution to any vibration problem involves the following steps:

1) Characterize the system parameters (mass, stiffness, damping) by experimental methods, manufacturers data, or a combination of both.

2) Model the system dynamics using a simple lumped parameter model

a) Identify natural frequencies; look for coincidence with excitation frequencies

b) If excitation forces and frequencies are known, system response can be calculated

3) Use the model to assess the effect of changes in system parameters as;

- Force Reduction of excitation inputs as, unbalance or misalignment will decrease the corresponding vibration response of the system.
- Mass Addition will reduce the effect (system response) of a constant excitation force for higher order frequency response.
- Tuning (changing) the natural frequency of a system or component will reduce or eliminate amplification due to resonance.
- Isolation rearranges the excitation forces to achieve some reduction or cancellation.
- Damping is the conversion of mechanical energy (vibrations) into heat mainly at resonance frequency.

Vibration Isolators

Consider a vibrating machine, bolted to a rigid floor (Figure 3.15 a). The force transmitted to the floor is equal to the force generated in the machine. The transmitted force can be decreased by adding a suspension and damping elements (often called vibration isolators) Figure 3.15 b, or by adding what is called an inertia block, a large mass (usually a block of cast concrete), directly attached to the machine (Figure 3.15 c). Another option is to add an additional level of mass (sometimes called a seismic mass, again a block of cast concrete) and suspension (Figure 3.15 d).

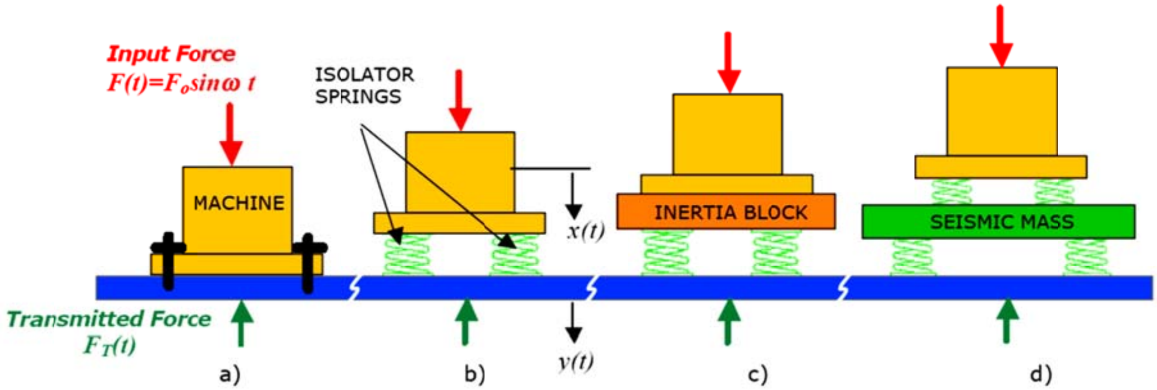

Figure 3.15 Vibration isolation systems: a) Machine bolted to a rigid foundation b) Supported on isolation springs, rigid foundation c) machine attached to an inertial block d) Supported on isolation springs, non-rigid foundation (such as a floor); or machine on isolation springs, seismic mass and second level of isolator springs [J. S. Lamancusa Penn State USA]

SOURCE AND SHIELD ISOLATION

Vibration isolation seeks to reduce the vibration level in one or several selected areas. The idea is to hinder the spread of vibrations along the path from the source to the receiver; see figure 3.16.

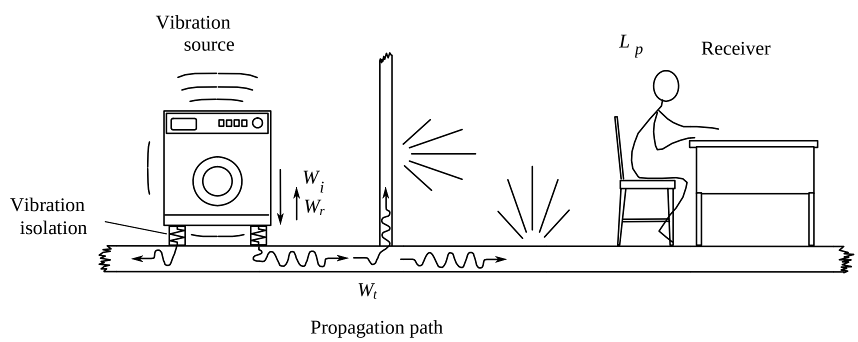

Figure 3.16 Example of a situation in which the vibrations emanating from a machine are reduced by isolation

A power W_i impinges on the isolators, a power W_r is reflected back towards the source, and a power W_t is transmitted to the floor. [1]

Of course, there are various options inherent in the actual realization of a vibration isolation system. Firstly, there are options as to where along the path to deploy the isolation; secondly, the isolators themselves can be designed in many different ways. It is essential to locate and design the isolation in the best possible way for the specific situation.

Regarding the location of the isolation, one can distinguish two extreme cases: placement near the source; and, placement near the receiver. In the first case, in which the source is isolated from the surroundings, one speaks of *source isolation*. In the second case, in which the receiver is isolated, one instead speaks of *shielding isolation*. Both cases are illustrated in figure 3.17. Note that one can, of course, combine shielding and source isolation. If there are very demanding requirements for a low vibration level, it is natural to isolate both the source and the receiver.

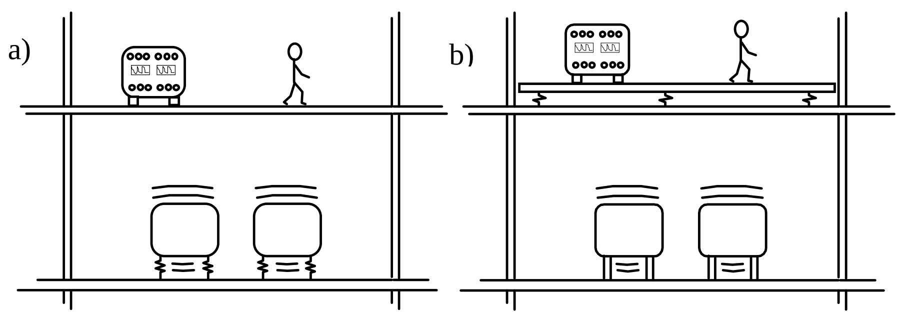

Figure 3.17 Two different strategies for vibration isolation: a) source isolation of machines; and, b) shielding isolation of sensitive equipment. [1]

Vibration Isolation in General

A vibration isolation problem is often schematically described by division into substructures: a *source structure* which is coupled to a *receiver structure*. The vibration isolation is yet another substructure incorporated between the two structures. The objective of *vibration isolation* is to reduce the vibrations in some specific portion of the receiver structure. It is apparent that vibration isolation can be realized in many different ways. It therefore falls upon the designer to arrive at an isolation system design well-suited to the specific situation.

All practical vibration isolation builds on a single physical principle. When a wave propagating in an elastic medium falls upon an abrupt change (discontinuity) in the properties of the medium, only a portion of the wave passes through that is in discontinuity. The remaining portion of the wave is reflected back towards the direction from which the incident wave arrives. The magnitude of the reflected portion of the wave depends on the magnitude of the change in properties. In the case of vibration isolation, one seeks to hinder the propagation of the wave by bringing about such discontinuities in properties along the propagation path.

The most common way to accomplish a discontinuity in the properties of the medium is to incorporate an element that is considerably more compliant, i.e., has a lower stiffness than the surrounding medium (see figure 2-2a). That type of element is usually called a *vibration isolator*. Steel coil springs and rubber isolators in a variety of forms are examples of vibration isolators readily available on the market; see section 2.6 for more details. Note that the stiffness can be changed by incorporating elements that are stiffer than the surrounding parts.

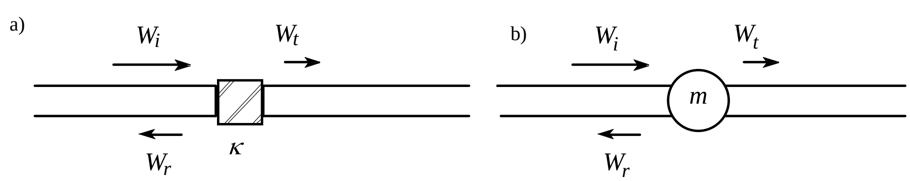

Figure 3.18 Two different vibration isolation methods. a) Reflection against a soft element. b) Reflection against a mass. W_i = incident power, W_r = reflected power and W_t = transmitted power. [1]

On further consideration, it is also apparent that one can bring about a reflection by incorporating an element with a differing inertia from that of the medium. Since elements of that type are often idealized as rigid masses, see figure 3.18 b, they are referred to as *blocking masses*. Practical realizations of the concept of blocking masses are, for example, *seismic blocks* and *added masses* at compliant points; see figure 3.18.

Considering, however, that the most common construction materials are relatively stiff, such as steel and concrete, it is often simpler to accomplish significant discontinuities in the medium properties by the compliant-element approach. For that reason, it is much more common to use compliant than stiff elements. Nevertheless, for structural reasons, there are some cases in which it is necessary to use stiff elements.

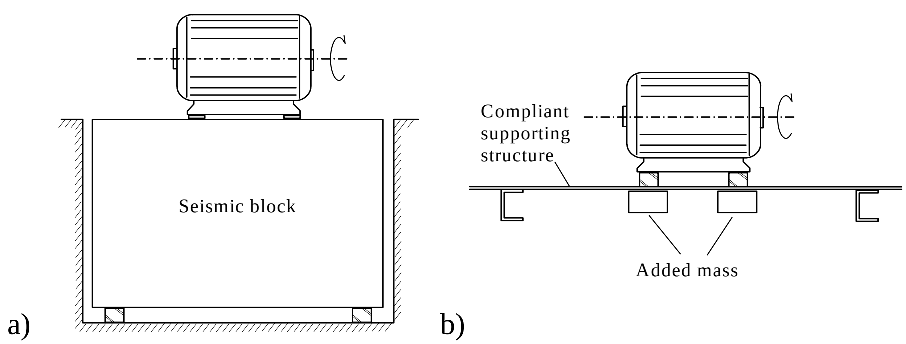

Figure 3.19 Blocking masses, preferably in combination with elastic elements, give very good vibration isolation and are frequently used in practice. a) Seismic block. b) Added mass at a compliant point. [1]

For a long time, machinery designers have mainly provided for vibration isolation using “trial and error” methods in combination with rough estimates obtained from very simple calculations. That approach tends to yield good results at lower frequencies, up to about 100 Hz, say. On the other hand, in the lion’s share of the audible frequency range, that approach provides little or no control over the actual isolation results obtained. In order to be able to achieve good vibration isolation by design, throughout the entire relevant frequency range, access to more advanced theoretical, as well as experimental, techniques is a necessity.

Vibration Isolators are usually specified by their static deflection D, or how much they deflect when the weight of the machine is placed on them. This is equivalent to specifying their stiffness and has the additional benefit of making it easy to calculate the system natural frequency. Coil spring isolators are available in up to 3 inch static deflection. If more flexibility is needed, air springs are used.

Important Considerations with Vibration Isolator Selection

1) Machine Location: As far away from sensitive areas as possible

· And on as rigid a foundation as possible (on grade is best)

2) Proper sizing of isolator units

· Correct stiffness (specified by the static deflection, more flexible is generally better)

· Sufficient travel to prevent bottoming out during shock loads, or during system startup

and shutdown.

3) Location of isolators – isolators should be equally loaded, and the machine should be level.

4) Stability – sideways motion should be restrained with snubbers. The diameter of the spring should also be greater than its compressed height. Isolator springs should occupy a wide footprint for stability.

5) Adjustment – springs should have free travel, should not be fully compressed, nor hitting a mechanical stop

6) Eliminate vibration short circuits – any mechanical connection between machine and foundation which bypasses the isolators, such as pipes, conduits, binding springs, poorly adjusted snubbers or mechanical stops.

7) Fail safe operation – should a spring break or become deflated, you must have mechanical supports on which the machine can rest without tipping.

Measures of Transmission Isolation

In order to be able to design in optimal vibration isolation, there is a need for, not only the determination of the vibration levels, but also for some measure of the vibration isolation obtained; that latter would permit comparison of alternative isolation strategies that may be applicable in a given situation. A number of different measures are in use for various specific applications. The most universally applied of these is the so-called *insertion loss* D_{IL}.; it is defined in either of the two following alternative ways:

$$\boxed{D_{IL}^{v} = L_{v}^{before} - L_{v}^{after}} \quad [\mathrm{dB}] \tag{3.44}$$

$$\boxed{D_{IL}^{F} = L_{F}^{before} - L_{F}^{after}} \quad [\mathrm{dB}] \tag{3.45}$$

where the velocity and force levels L_v and L_F . The insertion loss is, therefore, defined as the difference in level at a given point before and after the vibration isolation is provided; see figure 3.20. With these definitions as a model, it is of course possible to devise other such measures of the isolation effectiveness based on weighting different frequency components and bands, e.g., using A-weighting for instance. The choice of the relevant gauge of effectiveness is ultimately determined by the specific application.

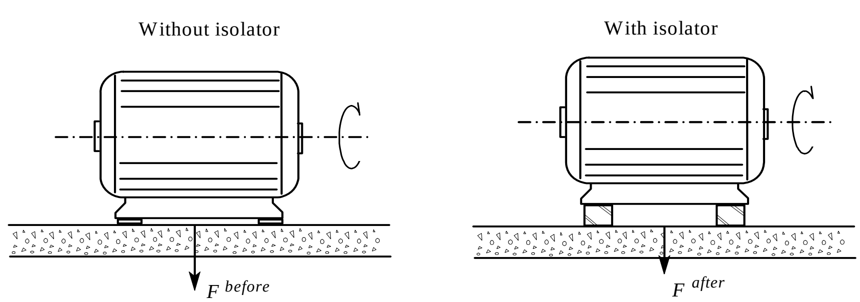

Figure 3.20 The insertion loss can be defined as the difference in the force level acting on the foundation before and after the implementation of isolation. [1]

Calculation of Vibration Isolation

The design of vibration isolation, for frequencies up to 1000 Hz, calls for relatively complicated calculations. In principle, computational models of machines, isolators, and foundations, which correctly describe the relationship between loads and deformations throughout that entire frequency band, are needed. That is, as quickly becomes apparent to the analyst, a very difficult problem. Even very detailed finite element models are so time-consuming, and give such uncertain results, that they are of doubtful value.

In practice, more or less simplified computational models of the different substructures are therefore an inevitable expedient. These should not be relied on for absolute values of the vibration isolation in a narrow frequency band, but they can be used to compare alternative vibration isolation design approaches in octave or third-octave bands. They can also indicated tendencies and suggest how vibration isolation can be improved. By the use of so-called frequency response function methods with input data from measurements, relatively reliable results can be obtained. The measurement of *frequency response functions*, and especially for isolators, is a whole research area in and of itself.

Some Vibration Isolation Computational Models

In order for computational models to serve as practical tools for the comparison of different vibration isolation alternatives, simplified models must inevitably be used. The problem, in its entirety, includes the transmission of vibrations originating in many degrees of freedom, at each *coupling surface* between the elements of the structure. To simplify the problem, one can:

(*i*) Idealize the coupling as occurring via infinitesimally small, i.e., point, contacts.

(*ii*) Assume that only 1 - 2 degrees-of-freedom contribute to the vibration transmission.

(*iii*) Ignore coupling points that make minor contributions to the vibration transmission.

(*iv*) Combine parallel transmission paths into a single equivalent path.

That last simplification is only useful if the contact points both between the isolators and the machine and between the isolators and the foundation move in phase at the same amplitude.

Rigid body – ideal spring – rigid foundation

At much lower excitation frequencies, considerably simplified models of the components are usable. Assume, for example, that we analyze a machine mounted at four points on a system of concrete joists. Assume, moreover, that the machine has an axle that generates sinusoidal bearing forces at the rotational frequency. At very low disturbance frequencies (i.e., low rotational speeds), the deformations of the machine itself are negligible, i.e., the machine acts as a *rigid body*. Physically, one can regard the force acting on the machine as so slowly changing in time that all parts of it have time to react to small changes in the force magnitude before the next such change occurs. Mathematically, the machine's movements can be described by means of equations from rigid body mechanics. The instantaneous state of the machine is then completely described by six degrees-of-freedom, three translational and three rotational. In practice, the number of degrees-of-freedom can normally be further reduced to one or two, eliminating those which are not relevant.

As the rotational speed of the axle increases, we eventually arrive at a situation in which the force changes so rapidly that not all parts of the machine have time to react before the force changes again at the point of its application. At that stage, we can begin to speak of wave propagation in the machine. If the rotational speed continues to increase further, we will arrive at a certain excitation frequency at which the amplitude of the machine deformations has a strong peak. At that frequency, the deformation waves and their reflections interact constructively to bring about the maximum in the response. That phenomenon is the so-called resonance phenomenon. At these frequencies, we can no longer regard the machine as a rigid body. A commonly used rule of thumb is that the rigid body assumption is useful up to frequencies of 1/2 of the first resonance frequency, i.e., for low Helmholtz numbers.

The rigid body assumption for the machine has an analogue that can be used in the description of the foundation. Consider now the example of the machine described above. At very low excitation frequencies, the joists respond with a (quasi-)static bending due to the slowly varying force acting at the machine mounting points. If the excitation frequency is so low that the deformation of the joists is so small as to be negligible in comparison to the deformation of the isolators, then the joists can be regarded, from the vibrations perspective, as a *rigid foundation.* Note that this doesn't imply that the foundation is not excited into vibration; that would apply no transmission whatsoever. Let the excitation frequency now increase, just as it does when considering the machine. At sufficiently high frequencies, the deformation can no longer be ignored. When the frequency has increased sufficiently, an ever more distinct wave propagation becomes apparent in the foundation. If the geometrical limits of the foundation are far away, then we will eventually reach the first resonance frequency of the foundation. The description of the foundation as rigid can, consequently, only be applied at low frequencies, say up to 1/2 of the first resonance frequency, i.e., once again at low Helmholtz numbers.

Assume now that we would like to reduce the vibrations transmitted from the machine into the system of joists by incorporating soft vibration isolators at the mounting positions between the machine and the joists. Under the influence of forces from the machine, the springs are deformed. At low excitation frequencies, all parts of the isolator itself react to the changing of the force. That implies that the cross-sectional load is uniform along the entire isolator. We have, in other words, no considerable wave propagation. Yet another consequence is that the isolator can be considered massless. In contrast to the joists, the isolator is compliant. We can, therefore, not ignore its deformation under load. In these circumstances, the isolator can be regarded as an *ideal massless spring*. As the frequency increases, the motion in the spring takes on the character of wave propagation more and more. Once again, at a certain point, the situation becomes resonant. In exactly the same way as before, we can adopt the rule of thumb that the spring idealization applies up to about 1/2 of the first resonance frequency.