Chapter 22: Principles of Passive Vibration Control: Basics

INTRODUCTION

The generally accepted methods for vibration control of industrial equipment include; Force Reduction, Mass Addition, Tuning, Isolation, and Damping. Vibrations can be isolated from equipment using active or passive technology. With active methods, equal but opposite forces are created electronically using sensors and actuators to cancel out the unwanted vibrations. With passive methods, isolation is achieved by limiting the ability of vibrations to be coupled to the item to be isolated. This is done using a mechanical connection which dissipates or redirects the energy of vibration before it gets to the item to be isolated. The added structural modification can be thought of as a passive control. Added stiffness chosen to improve the vibrational response of the system, then it can be thought of as a passive control procedure. Passive control is distinguished from active control by the use of added power or energy in the form of an actuator, required in active control. The material on isolators and absorbers represents two possible methods of passive control. Indeed, the most common passive control device is the vibration absorber. Much of the other work in passive control consists of added layers of damping material applied to various structures to increase the damping ratios of troublesome modes. Adding mass and changing stiffness values are also methods of passive control used to adjust a frequency away from resonance. Damping treatments increase the rate of decay of vibrations, so they are often more popular for vibration suppression.

Passive methods sometimes involve electromechanical controls for adjusting the system, but the isolation mechanism itself is passive. Passive systems may use elastomers, springs, fluids, or negative-stiffness components. Passive systems generally cost less than active systems and their relative simplicity makes them more reliable and safe. One of the most basic passive isolators is a spring placed between the surface transmitting shock or vibration and the item to be isolated. The spring opposes the impulse on it and absorbs some energy as it deforms. A fluid or elastomeric element is added to the spring element for damping. A simple example is the shock absorber in a car. In this case, mechanical energy from the shock or vibration does work on the fluid and is converted to thermal energy in the fluid,

reducing the amount of energy transmitted to the body of the car. Elastomers are rubber-like materials which absorb mechanical energy by deforming. Examples of elastomeric isolators are shock and vibration mounts for automobile engines, aircraft components, industrial machinery, and building foundations. Because rubber does not have the same characteristics in all directions, isolation may be much better in one axis than the others. The most sophisticated passive isolators use air or negative-stiffness technology.

In advanced technology applications, such as interferometry, microscopy (including SPM, SEM, etc.), nano-fabrication and micro-hardness testing, the best passive vibration isolation devices allow the instruments to perform at their highest possible level. Pneumatic systems support a heavy table or platform on compressed air pistons, which provide the decoupling link between the ground and the table, i.e. the table floats on the air. The most sophisticated air tables do a good job of isolating floor vibrations at small amplitudes, but can be quite expensive. While they isolate in a passive manner, they require an air supply, a leveling system, and associated maintenance and controls. The less sophisticated air systems do not isolate well in all directions or at the low frequencies which are handled so well by negative-stiffness systems. The air supply system for pneumatic isolators can create problematic ambient vibration. The use of electrical methods of leveling the table and controlling air cylinder pressure adds complexity and the potential for failure. Negative-stiffness isolators provide a simple, reliable, and highly effective isolation solution for sensitive instrumentation at the low frequencies and amplitudes of floor and building vibrations. They are relatively small, light weight, stable, and cost-effective.

DESIGN CONSIDERATION OF ABSORBER AND ISOLATOR ALONG-WITH DYNAMIC PROPERTIES OF VIBRATION ABSORBER

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 as shown in Fig. 7.1. Isolation of a vibrating mass refers to designing the connection of a mass (machine part or structure) to ground in such a way as to reduce unwanted effects or disturbances through that connection. Vibration absorption, on the other hand, refers to adding an additional degree of freedom (spring and mass) to

the structure to absorb the unwanted disturbance. The typical model used in vibration isolation design is the simple single-degree-of-freedom system of Figure 7.2 (a) without damping, or Figure 7.2(b) with damping.

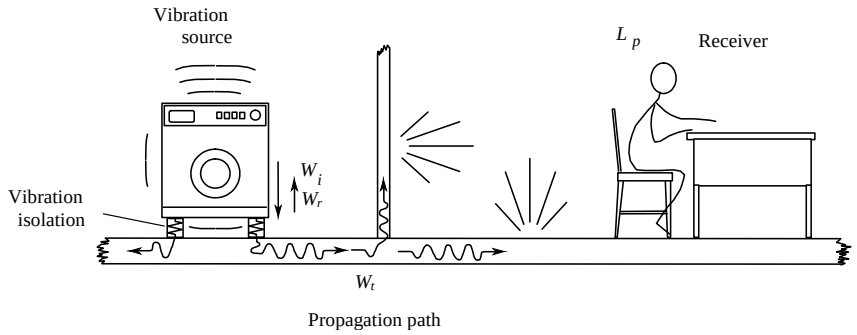


Figure 7.1 Example of a situation in which the vibrations emanating from a machine are reduced by isolation. A power W_i impinges on the isolators, a power W_r is reflected back towards the source, and a power W_t is transmitted to the floor. [1]

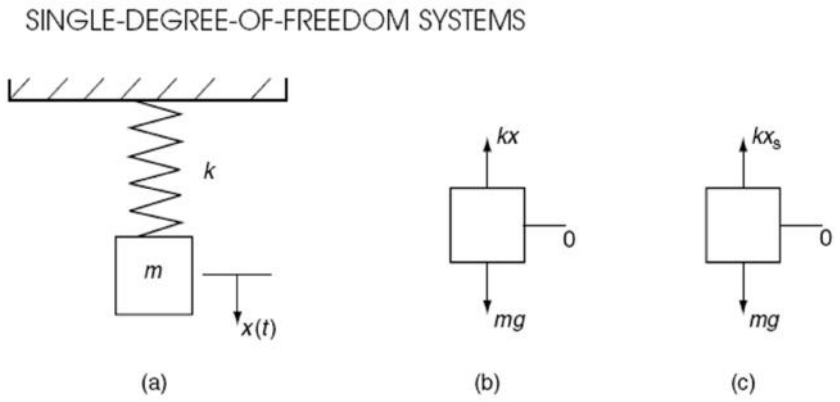


Figure 7.2 (a) Spring–mass schematic (b) free body diagram and (c) free body diagram of the static spring–mass system

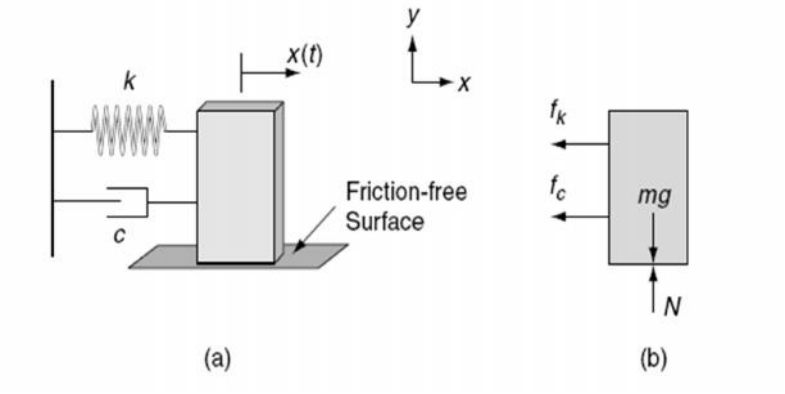


Figure 7.3 (b) Schematic of the spring–mass–damper system and (b) free body diagram of the system in part (a).

The idea here is twofold. First, if a harmonic force is applied to the mass through movement of the ground (i.e., as the result of a nearby rotating machine, for instance), the values of c and k should be chosen to minimize the resulting response of the mass. The design isolates the mass from the effects of ground motion. The springs on an automobile serve this purpose. A second use of the concept of isolation is that in which the mass represents the mass of a machine, causing an unwanted harmonic disturbance. In this case the values of m, c, and k are chosen so that the disturbance force passing through the spring and dashpot to ground is minimized. This isolates the ground from the effects of the machine. The motor mounts in an automobile are examples of this type of isolation.

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. 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.

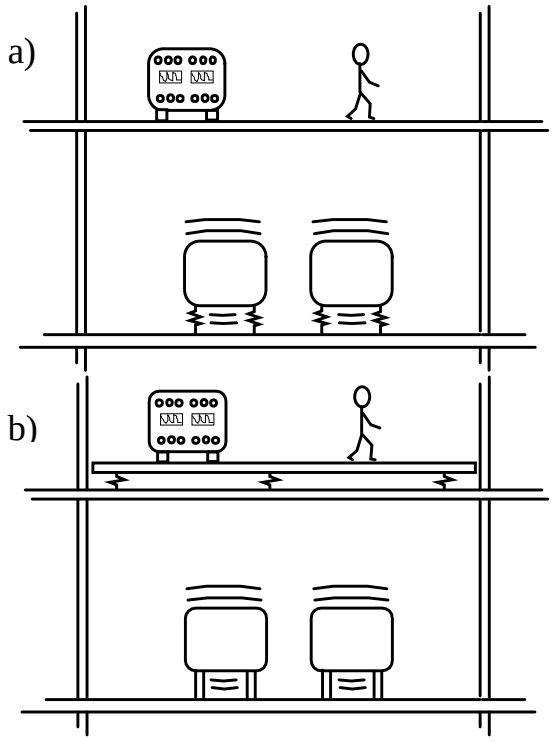
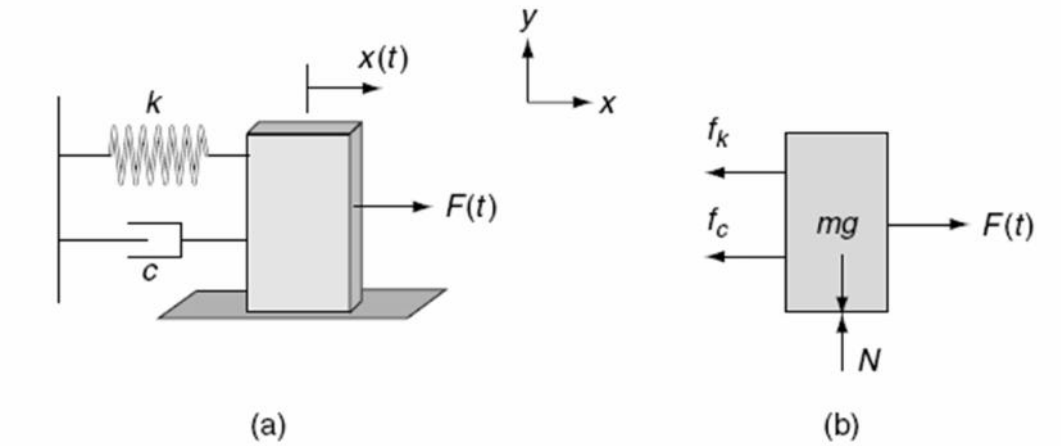
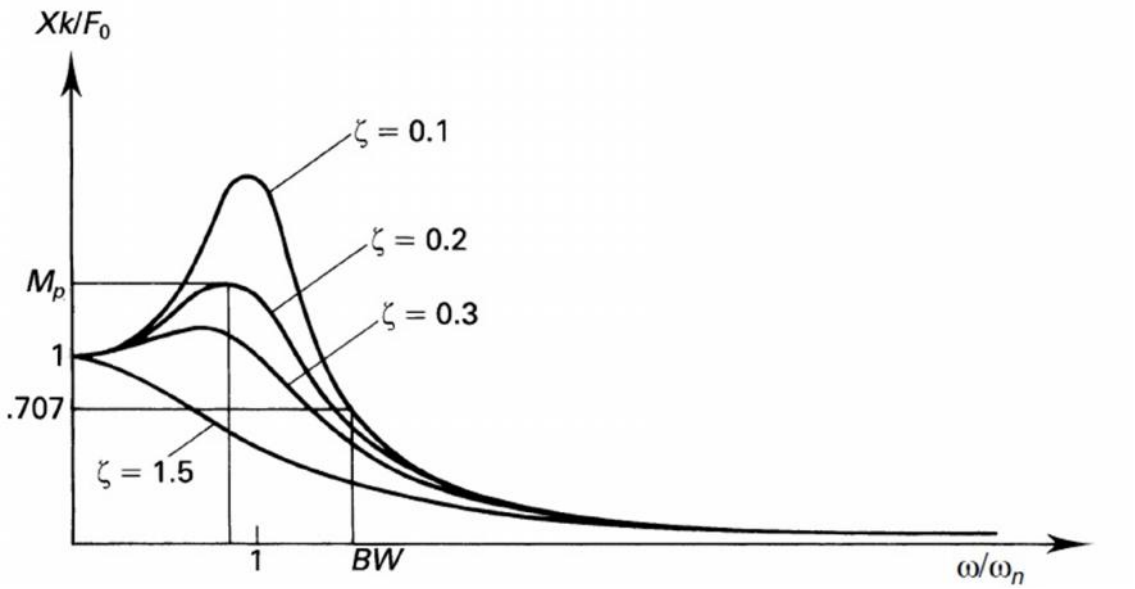


Figure 7.4 Two different strategies for vibration isolation: a) source isolation of machines; and, b) shielding isolation of sensitive equipment. [1]

In either case, the details of the governing equations for the isolation problem consist of analyzing the steady state forced harmonic response of equations. For instance, if it is desired to isolate the mass of Figure 7.5 (a, b) from the effects of a disturbance $F_o = \sin(\omega t)$ then the magnification curves of Figure 7.5 (c) indicate how to choose the damping ζ and the isolator frequency ω_n so that the amplitude of the resulting vibration is as small as possible.





(c)

Figure 7.5 Magnification curves for a single-degree-of-freedom system

Curves similar to the magnification curves, called transmissibility curves, are usually used in isolation problems. The ratio of the amplitude of the force transmitted through the connection between the ground and the mass to the amplitude of the driving force is called the transmissibility. For the system of Figure 7.3, the force transmitted to ground is transmitted through the spring, k, and the damper, c. these forces at steady state are

$$F_k = KX_{ss}(t) = KX\sin(\omega t - \phi) \qquad (7.1)$$

And

$$F_c = Cx_{ss}(t) = C\omega\dot{X}\cos(\omega t - \phi) \quad (7.2)$$

Here, F_k and F_c denote the force in the spring and the force in the damper respectively, and X is the magnitude of the steady state response. The magnitude of the transmitted force is the magnitude of the vector sum of these two forces, denoted by F_T, and is given by

$$F_T^2 = / Kx_{ss} + C\dot{x}_{ss} /^2 = [(KX)^2 + (C\omega X)^2] \quad (7.3)$$

Thus, the magnitude of transmitted force becomes

$$F_T = KX\left[1 + \left(\frac{C\omega}{K}\right)^2\right]^{1/2} \quad (7.4)$$

The amplitude of the applied force is just F_0, so that the transmissibility ratio, denoted by TR, becomes;

$$TR = \frac{F_T}{F_0} = \frac{\sqrt{\left(1+\left(2\zeta\omega/\omega_n\right)^2\right)}}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2 + [2\zeta\omega/\omega_n]^2}} \quad (7.5)$$

Plots of above expression versus the frequency ratio ω/ω_n for various values of ζ are called transmissibility curves. One such curve is illustrated in Figure 7.6. This curve indicates that, for values of $\omega/\omega_n > \sqrt{2}$ (that is, TR <1), vibration isolation occurs, whereas for values of $\omega/\omega_n < \sqrt{2}$ (TR >1) an amplification of vibration occurs. Of course, the largest increase in amplitude occurs at resonance. If the physical parameters of a system are constrained such that isolation is not feasible, a vibration absorber may be included in the design. A vibration absorber consists of an attached second mass, spring, and damper, forming a two-degree-of-freedom system. The second spring–mass system is then 'tuned' to resonate and hence absorb all the vibrational energy of the system.

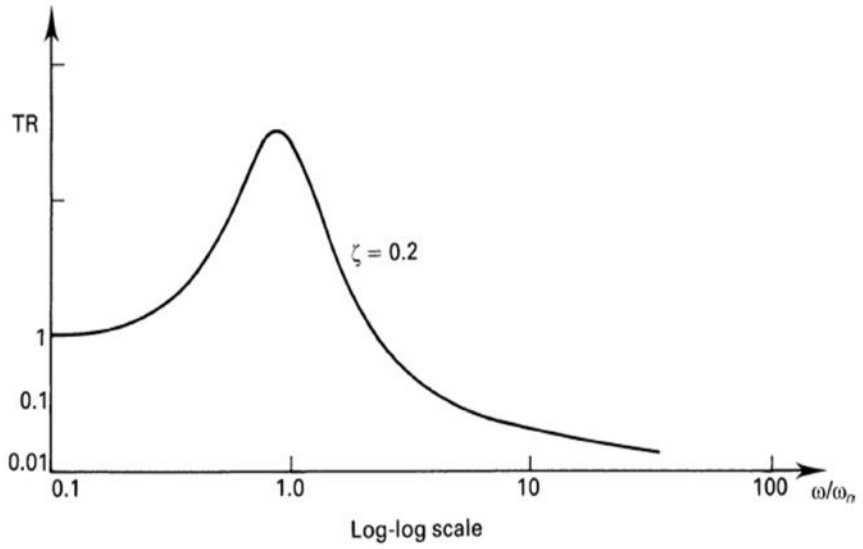


Fig 7.6 Transmissibility ratio on log-log scale

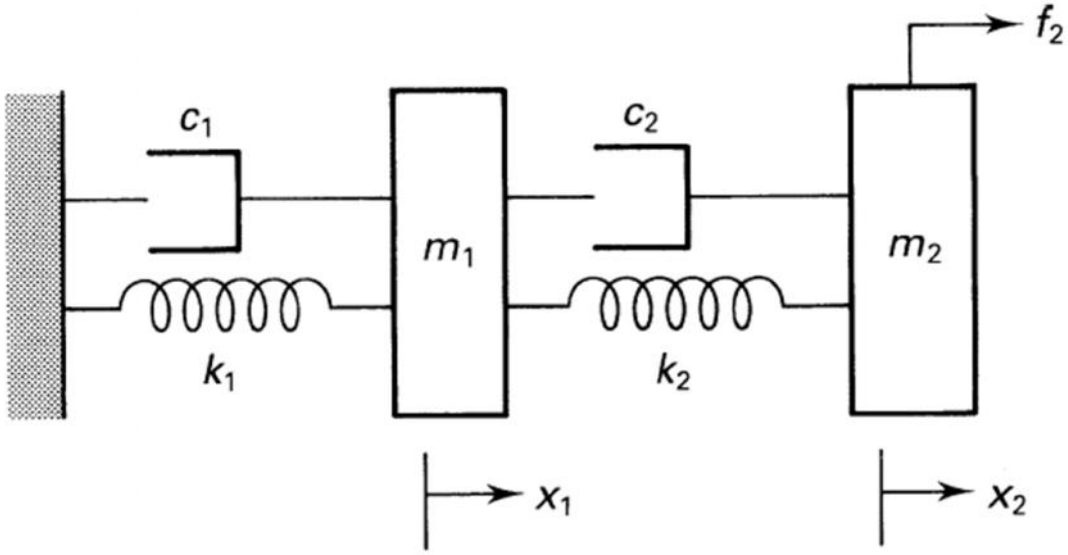


Fig 7.7 Two DOF of system

The basic method of designing a vibration absorber is illustrated here by examining the simple case with no damping. To this end, consider the two-degree-of-freedom system of Figure 7.7 with $c_1 = c_2 = f_2 = 0$, $m_2 = m_a$, the absorber mass, $m_1 = m$, the primary mass, $k_1 = k$, the primary stiffness, and $k_2 = k_a$, the absorber spring constant. In addition, let $x_1 = x$, the displacement of the primary mass, and $x_2 = x_a$, the displacement of the absorber. Also, let the driving force $F_0 = \sin(\omega t)$ be applied to

the primary mass, m. The absorber is designed to the steady state response of this mass by choosing the values of m_a and k_a. Recall that the steady state response of a harmonically excited system is found by assuming a solution that is proportional to a harmonic term of the same frequency as the driving frequency.

From the equations of motion of the two-mass absorber systems are

$$\begin{bmatrix} m & 0 \\ 0 & m_a \end{bmatrix}\begin{bmatrix} \ddot{x} \\ \ddot{x}_a \end{bmatrix} + \begin{bmatrix} k + k_a & -k_a \\ -k_a & k_a \end{bmatrix}\begin{bmatrix} x \\ x_a \end{bmatrix} = \begin{bmatrix} F_0 \\ 0 \end{bmatrix} \sin(\omega t) \tag{7.6}$$

Assuming that in the steady state the solution of above Equation will be of the form

$$\begin{bmatrix} x(t) \\ x_a(t) \end{bmatrix} = \begin{bmatrix} X \\ X_a \end{bmatrix} \sin(\omega t) \tag{7.7}$$

and substituting into Equation yields

$$\begin{bmatrix} k + k_a - m\omega^2 & -k_a \\ -k_a & k_a - m_a\omega^2 \end{bmatrix}\begin{bmatrix} X \\ X_a \end{bmatrix} \sin(\omega t) = \begin{bmatrix} F_0 \\ 0 \end{bmatrix} \sin(\omega t) \tag{7.8}$$

Solving for the magnitudes X and Xa yields;

$$\begin{bmatrix} X \\ X_a \end{bmatrix} = \frac{1}{(k+k_a-m\omega^2)(k_a-m_a\omega^2)-(k_a)^2}\begin{bmatrix} k_a - m_a\omega^2 & k_a \\ k_a & k + k_a - m\omega^2 \end{bmatrix}\begin{bmatrix} F_0 \\ 0 \end{bmatrix} \tag{7.9}$$

or

$$X = \frac{(k_a-m_a\omega^2)F_0}{(k+k_a-m\omega^2)(k_a-m_a\omega^2)-(k_a)^2} \tag{7.10}$$

and

$$X_a = \frac{k_a F_0}{(k+k_a-m\omega^2)(k_a-m_a\omega^2)-(k_a)^2} \qquad (7.11)$$

As can be seen by examining Equation, if k_a and k_0 are chosen such that $k_a - m_a\omega^2 = 0$, i.e., such that $\sqrt{k_a/m_a} = \omega$, then the magnitude of the steady state response of the primary ***m*** is zero, i.e., X = 0. Hence, if the added absorber mass, m_a, is 'tuned' to the driving frequency ω, then the amplitude of the steady state vibration of the primary mass, X, is zero and the absorber mass effectively absorbs the energy in the system.

The addition of damping into the absorber-mass system provides two more parameters to be adjusted for improving the response of the mass m. However, with damping, the magnitude X cannot be made exactly zero.

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 is 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 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 7.8). 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. Note that the stiffness can be changed by incorporating elements that are stiffer than the surrounding parts.

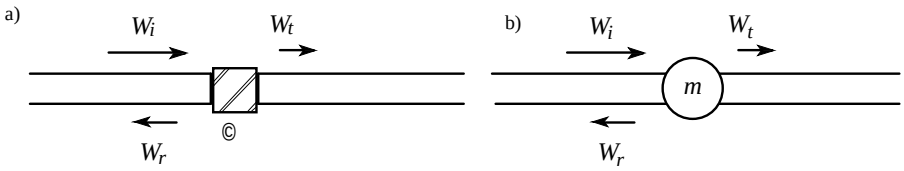


Figure 7.8 Two different vibration isolation methods. a) Reflection against a soft element. b) Reflection against a mass. W_i = incident power, W_r = reflected power and W_t = transmitted power.

"Fundamentals of Sound and Vibrations" by KTH Sweden [1], this book is used under IITR-KTH MOU for course development.