1

Single Degree of Freedom Systems

1.1 Introduction

In this chapter, the vibration of a single degree of freedom system (SDOF) will be analyzed and reviewed. Analysis, measurement, design and control of SDOF systems are discussed. The concepts developed in this chapter constitute a review of introductory vibrations and serve as an introduction for extending these concepts to more complex systems in later chapters. In addition, basic ideas relating to measurement and control of vibrations are introduced that will later be extended to multiple degree of freedom systems and distributed parameter systems. This chapter is intended to be a review of vibration basics and an introduction to a more formal and general analysis for more complicated models in the following chapters.

Vibration technology has grown and taken on a more interdisciplinary nature. This has been caused by more demanding performance criteria and design specifications of all types of machines and structures. Hence, in addition to the standard material usually found in introductory chapters of vibration and structural dynamics texts, several topics from control theory are presented. This material is included not to train the reader in control methods (the interested student should study control and system theory texts), but rather to point out some useful connections between vibration and control as related disciplines. In addition, structural control has become an important discipline requiring the coalescence of vibration and control topics. A brief introduction to nonlinear SDOF systems and numerical simulation is also presented.

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1.2 Spring-Mass System

Simple harmonic motion, or oscillation, is exhibited by structures that have elastic restoring forces. Such systems can be modeled, in some situations, by a spring-mass schematic (Figure 1.1). This constitutes the most basic vibration model of a structure and can be used successfully to describe a surprising number of devices, machines and structures. The methods presented here for solving such a simple mathematical model may seem to be more sophisticated than the problem

Vibration with Control, Second Edition. Daniel John Inman.
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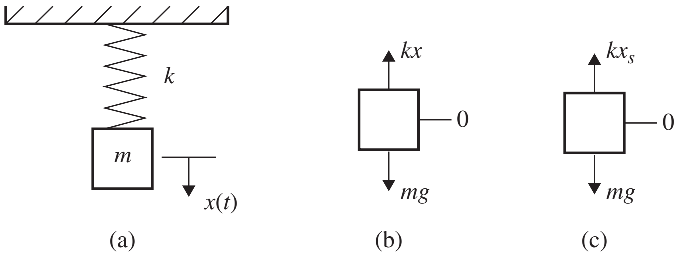

Figure 1.1 (a) A spring-mass schematic, (b) a free body diagram, and (c) a free body diagram of the static spring mass system.

requires. However, the purpose of this analysis is to lay the groundwork for solving more complex systems discussed in the following chapters.

If $x = x(t)$ denotes the displacement (in meters) of the mass m (in kg) from its equilibrium position as a function of time, t (in sec), the equation of motion for this system becomes (upon summing the forces in Figure 1.1b)

$$m\ddot{x} + k(x + x_s) - mg = 0$$

where k is the stiffness of the spring (N/m), x_s is the static deflection (m) of the spring under gravity load, g is the acceleration due to gravity (m/s^2) and the over dots denote differentiation with respect to time. A discussion of dimensions appears in Appendix A and it is assumed here that the reader understands the importance of using consistent units. From summing forces in the free body diagram for the static deflection of the spring (Figure 1.1c), $mg = kx_s$ and the above equation of motion becomes

$$m\ddot{x}(t) + kx(t) = 0 \tag{1.1}$$

This last expression is the equation of motion of an SDOF system and is a linear, second-order, ordinary differential equation with constant coefficients.

Figure 1.2 indicates a simple experiment for determining the spring stiffness by adding known amounts of mass to a spring and measuring the resulting static deflection, x_s. The results of this static experiment can be plotted as force (mass times acceleration) versus x_s, the slope yielding the value of k for the linear portion of the plot. This is illustrated in Figure 1.3.

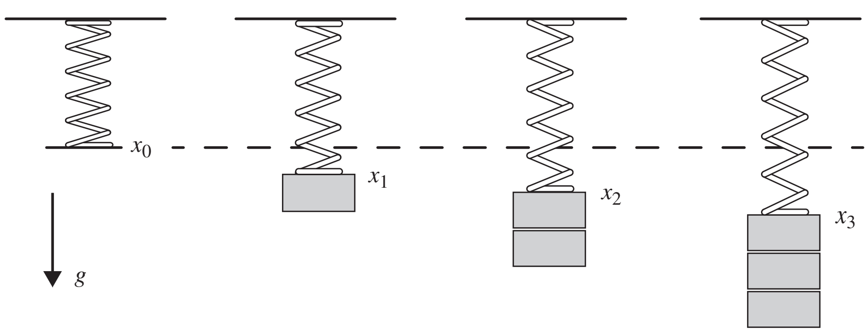

Figure 1.2 Measurement of spring constant using static deflection caused by added mass.

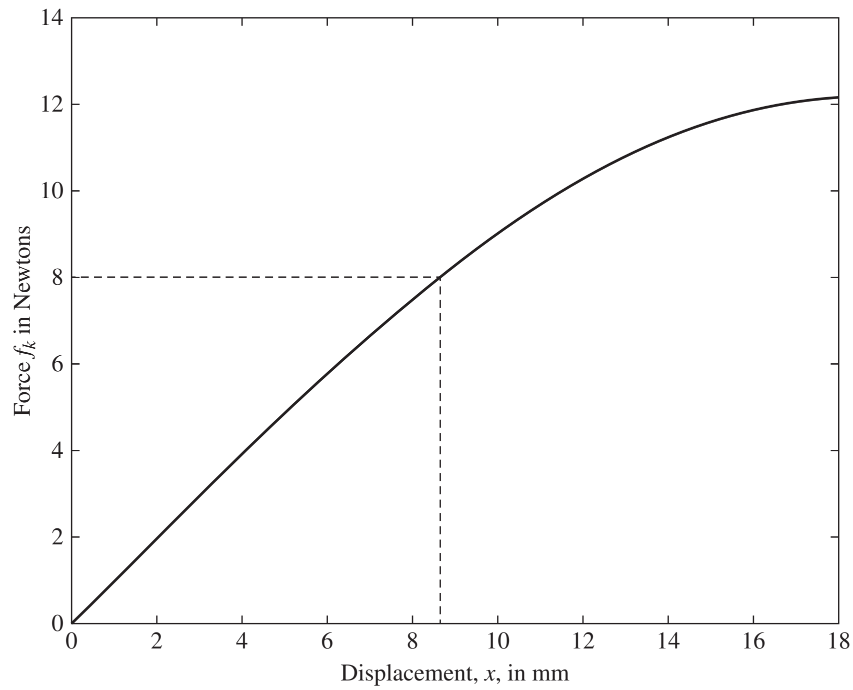

Figure 1.3 Determination of the spring constant. The dashed box indicates the linear range of the spring.

Once m and k are determined from static experiments, Equation (1.1) can be solved to yield the time history of the position of the mass m, given the initial position and velocity of the mass. The form of the solution of Equation (1.1) is found from substitution of an assumed periodic motion (from experience watching vibrating systems) of the form

$$x(t) = A\sin(\omega_n t + \phi) \tag{1.2}$$

where $\omega_n = \sqrt{k/m}$ is called the *natural frequency* in radians per second (rad/s). Here A, the *amplitude*, and ϕ, the *phase shift*, are constants of integration determined by the initial conditions.

The existence of a *unique* solution for Equation (1.1) with two specific initial conditions is well known and is given in Boyce and DiPrima (2012). Hence, if a solution of the form of Equation (1.2) is guessed and it works, then it is *the* solution. Fortunately, in this case, the mathematics, physics and observation all agree.

To proceed, if x_0 is the specified initial displacement from equilibrium of mass m, and v_0 is its specified initial velocity, simple substitution allows the constants of integration A and ϕ to be evaluated. The unique solution is

$$x(t) = \sqrt{\frac{\omega_n^2 x_0^2 + v_0^2}{\omega_n^2}} \sin\left[\omega_n t + \tan^{-1}\left(\frac{\omega_n x_0}{v_0}\right)\right] \tag{1.3}$$

Alternately, $x(t)$ can be written as

$$x(t) = \frac{v_0}{\omega_n} \sin \omega_n t + x_0 \cos \omega_n t \tag{1.4}$$

by using a simple trigonometric identity or by direct substitution of the initial conditions (Example 1.2.1).

A purely mathematical approach to the solution of Equation (1.1) is to assume a solution of the form $x(t) = Ae^{\lambda t}$ and solve for λ, i.e.

$$m\lambda^2 e^{\lambda t} + ke^{\lambda t} = 0$$

This implies that (because $e^{\lambda t} \neq 0$ and $A \neq 0$)

$$\lambda^2 + \left(\frac{k}{m}\right) = 0$$

or that

$$\lambda = \pm j \left(\frac{k}{m}\right)^{1/2} = \pm \omega_n j$$

where $j = (-1)^{1/2}$. Then the general solution becomes

$$x(t) = A_1 e^{-\omega_n j t} + A_2 e^{\omega_n j t} \tag{1.5}$$

where A_1 and A_2 are arbitrary complex conjugate constants of integration to be determined by the initial conditions. Use of Euler's formulas then yields Equations (1.2) and (1.4) (Inman, 2014). For more complicated systems, the exponential approach is often more appropriate than first guessing the form (sinusoid) of the solution from watching the motion.

Another mathematical comment is in order. Equation (1.1) and its solution are valid only as long as the spring is linear. If the spring is stretched too far or too much force is applied to it, the curve in Figure 1.3 will no longer be linear. Then Equation (1.1) will be nonlinear (Section 1.10). For now, it suffices to point out that initial conditions and springs should always be checked to make sure that they fall into the linear region, if linear analysis methods are going to be used.

Example 1.2.1
Assume a solution of Equation (1.1) of the form

$$x(t) = A_1 \sin \omega_n t + A_2 \cos \omega_n t$$

and calculate the values of the constants of integration A_1 and A_2 given arbitrary initial conditions x_0 and v_0, thus verifying Equation (1.4).

Solution: The displacement at time $t = 0$ is

$$x(0) = x_0 = A_1 \sin(0) + A_2 \cos(0)$$

or $A_2 = x_0$. The velocity at time $t = 0$ is

$$\dot{x}(0) = v_0 = \omega_n A_1 \cos(0) - \omega_n x_0 \sin(0)$$

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Solving this last expression for A_1 yields $A_1 = v_0/x_0$, so that Equation (1.4) results in

$$x(t) = \frac{v_0}{x_0} \sin \omega_n t + x_0 \cos \omega_n t$$

Example 1.2.2
Compute and plot the time response of a linear spring-mass system to initial conditions of $x_0 = 0.5$ mm and $v_0 = 2\sqrt{2}$ mm/s, if the mass is 100 kg and the stiffness is 400 N/m.

Solution: The frequency is

$$\omega_n = \sqrt{k/m} = \sqrt{400/100} = 2 \text{ rad/s}$$

Next compute the amplitude from Equation (1.3):

$$A = \sqrt{\frac{\omega_n^2 x_0^2 + v_0^2}{\omega_n^2}} = \sqrt{\frac{2^2(0.5)^2 + (2\sqrt{2})^2}{2^2}} = 1.5 \text{ mm}$$

From Equation (1.3) the phase is

$$\phi = \tan^{-1}\left(\frac{\omega_n x_0}{v_0}\right) = \tan^{-1}\left(\frac{2(0.5)}{2\sqrt{2}}\right) \approx 10 \text{ rad}$$

Thus the response has the form

$$x(t) = 1.5 \sin(2t + 10)$$

and this is plotted in Figure 1.4.

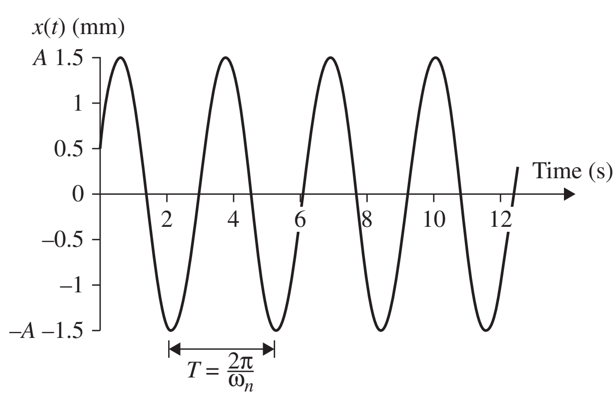

Figure 1.4 The response of a simple spring-mass system to an initial displacement of $x_0 =$ 0.5 mm and an initial velocity of $v_0 = 2\sqrt{2}$ mm/s. The period, defined as the time it takes to complete one cycle off oscillation, $T = 2\pi/\omega_n$, becomes $T = 2\pi/2 = \pi$s.

1.3 Spring-Mass-Damper System

Most systems will not oscillate indefinitely when disturbed, as indicated by the solution in Equation (1.3). Typically, the periodic motion dies down after some time. The easiest way to treat this mathematically is to introduce a velocity term, $c\dot{x}$, into Equation (1.1) and examine the equation

$$m\ddot{x} + c\dot{x} + kx = 0 \tag{1.6}$$

This also happens physically with the addition of a *dashpot* or *damper* to dissipate energy, as illustrated in Figure 1.5.

Equation (1.6) agrees with summing forces in Figure 1.5 if the dashpot exerts a dissipative force proportional to velocity on the mass m. Unfortunately, the constant of proportionality, c, cannot be measured by static methods as m and k are. In addition, many structures dissipate energy in forms not proportional to velocity. The constant of proportionality c is given in Newton-second per meter (Ns/m) or kilograms per second (kg/s) in terms of fundamental units.

Again, the unique solution of Equation (1.6) can be found for specified initial conditions by assuming that $x(t)$ is of the form

$$x(t) = Ae^{\lambda t}$$

and substituting this into Equation (1.6) to yield

$$A\left(\lambda^2 + \frac{c}{m}\lambda + \frac{k}{m}\right)e^{\lambda t} = 0 \tag{1.7}$$

Since a trivial solution is not desired, $A \neq 0$, and since $e^{\lambda t}$ is never zero, Equation (1.7) yields

$$\lambda^2 + \frac{c}{m}\lambda + \frac{k}{m} = 0 \tag{1.8}$$

Equation (1.8) is called the *characteristic equation* of Equation (1.6). Using simple algebra, the two solutions for λ are

$$\lambda_{1,2} = -\frac{c}{2m} \pm \frac{1}{2}\sqrt{\frac{c^2}{m^2} - 4\frac{k}{m}} \tag{1.9}$$

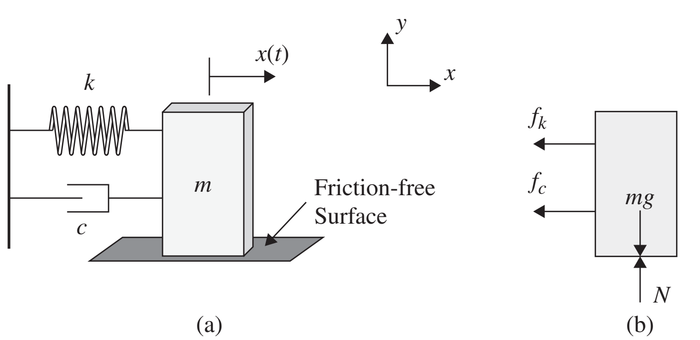

Figure 1.5 (a) Schematic of spring-mass-damper system. (b) A free-body diagram of the system in part (a).

The quantity under the radical is called the *discriminant* and together with the sign of m, c and k determines whether or not the roots are complex or real. Physically, m, c and k are all positive in this case, so the value of the discriminant determines the nature of the roots of Equation (1.8).

It is convenient to define the dimensionless *damping ratio*, ζ, as

$$\zeta = \frac{c}{2\sqrt{km}}$$

In addition, let the *damped natural frequency*, ω_d, be defined by (for $0 < \zeta < 1$)

$$\omega_d = \omega_n\sqrt{1-\zeta^2} \tag{1.10}$$

Then Equation (1.6) becomes

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x = 0 \tag{1.11}$$

and Equation (1.9) becomes

$$\lambda_{1,2} = -\zeta\omega_n \pm \omega_n\sqrt{\zeta^2-1} = -\zeta\omega_n \pm \omega_d j, \quad 0 < \zeta < 1 \tag{1.12}$$

Clearly the value of the damping ratio, ζ, determines the nature of the solution of Equation (1.6). There are three cases of interest. The derivation of each case is left as an exercise and can be found in almost any introductory text on vibrations (Inman, 2014; Meirovitch, 1986).

Underdamping occurs if the system's parameters are such that

$$0 < \zeta < 1$$

so that the discriminant in Equation (1.12) is negative and the roots form a complex conjugate pair of values. The solution of Equation (1.11) then becomes

$$x(t) = e^{-\zeta\omega_n t}(A\cos\omega_d t + B\sin\omega_d t) \tag{1.13}$$

or

$$x(t) = Ce^{-\zeta\omega_n t}\sin(\omega_d t + \phi)$$

where A, B, C and ϕ are constants determined by the specified initial velocity, v_0 and position, x_0

$$\begin{aligned} A &= x_0 \qquad & C &= \frac{\sqrt{(v_0+\zeta\omega_n x_0)^2 + (x_0\omega_d)^2}}{\omega_d} \\ B &= \frac{(v_0+\zeta\omega_n x_0)}{\omega_d} \qquad & \phi &= \tan^{-1}\left[\frac{x_0\omega_d}{(v_0+\zeta\omega_n x_0)}\right] \end{aligned} \tag{1.14}$$

The underdamped response has the form given in Figure 1.6 and consists of a decaying oscillation of frequency ω_d.

Overdamping occurs if the system's parameters are such that

$$\zeta > 1$$

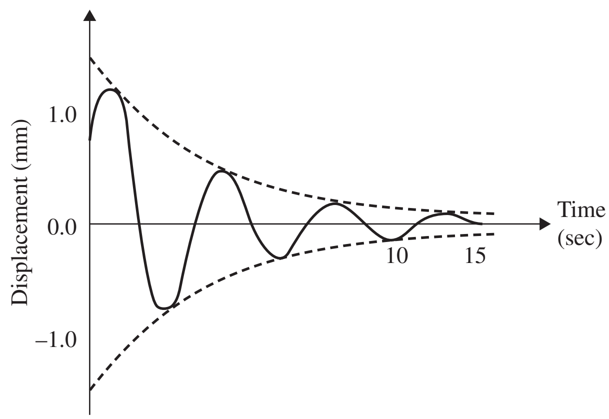

Figure 1.6 Response of an underdamped system illustrating oscillation with exponential decay.

so that the discriminant in Equation (1.12) is positive and the roots are a pair of negative real numbers. The solution of Equation (1.11) then becomes

$$x(t) = Ae^{\left(-\zeta+\sqrt{\zeta^2-1}\right)\omega_n t} + Be^{\left(-\zeta-\sqrt{\zeta^2-1}\right)\omega_n t} \tag{1.15}$$

where A and B are again constants determined by v_0 and x_0. They are

$$A = \frac{v_0 + \left(\zeta + \sqrt{\zeta^2 - 1}\right)\omega_n x_0}{2\omega_n\sqrt{\zeta^2 - 1}} \text{ and } B = -\frac{v_0 + \left(\zeta - \sqrt{\zeta^2 - 1}\right)\omega_n x_0}{2\omega_n\sqrt{\zeta^2 - 1}} \tag{1.16}$$

The overdamped response has the form given in Figure 1.7. An overdamped system does not oscillate, but rather returns to its rest position exponentially.

Critical Damping occurs if the system's parameters are such that $\zeta = 1$, so that the discriminant in Equation (1.12) is zero and the roots are a pair of negative real repeated numbers. The solution of Equation (1.11) then becomes

$$x(t) = e^{-\omega_n t}[(v_0 + \omega_n x_0)t + x_0] \tag{1.17}$$

The critically damped response is plotted in Figure 1.8 for values of the initial velocity v_0 of different signs and $x_0 = 0.25$ mm.

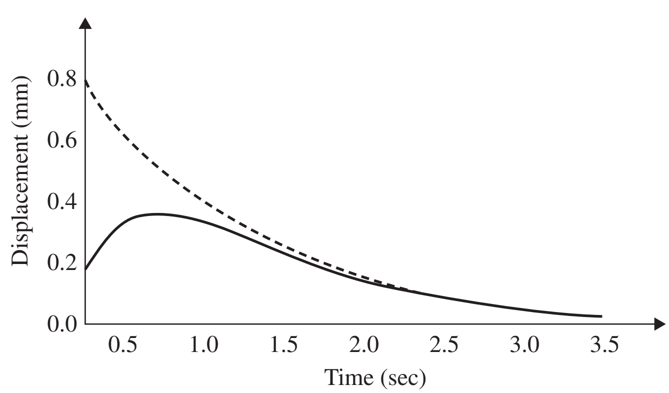

Figure 1.7 Response of an overdamped system illustrating exponential decay without oscillation.

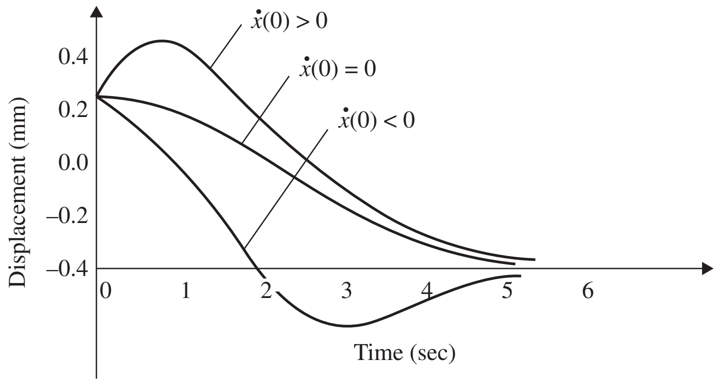

Figure 1.8 Response of critically damped system to an initial displacement and three different initial velocities indicating no oscillation.

It should be noted that critically damped systems can be thought of in several ways. First, they represent systems with the minimum value of damping rate that yields a non-oscillating system (Exercise 1.5). Critical damping can also be thought of as the case that separates non-oscillation from oscillation.

Example 1.3.1

Derive the constants A and B of integration for the overdamped case of Equation (1.15).

Solution: Substitution of $x(0) = x_0$ into Equation (1.15) yields

$$x(0) = Ae^0 + Be^0 \text{ or } x_0 = A + B \tag{1.18}$$

Differentiating Equation (1.15) and setting $t = 0$ in the result yields

$$\dot{x}(0) = A\lambda_1 e^0 + B\lambda_2 e^0 \text{ or } v_0 = \lambda_1 A + \lambda_1 B \tag{1.19}$$

where λ_1 and λ_2 are defined in Equation (1.12). These two initial conditions result in two independent equations in two unknowns, A and B, which can be solved in many ways. Writing Equations (1.17) and (1.18) as a single matrix equation yields

$$\begin{bmatrix} x_0 \\ v_0 \end{bmatrix} = \begin{bmatrix} 1 & 1 \\ \lambda_1 & \lambda_2 \end{bmatrix} \begin{bmatrix} A \\ B \end{bmatrix} \text{ or } \begin{bmatrix} A \\ B \end{bmatrix} = \begin{bmatrix} 1 & 1 \\ \lambda_1 & \lambda_2 \end{bmatrix}^{-1} \begin{bmatrix} x_0 \\ v_0 \end{bmatrix}$$

Solving by computing matrix inverse (see Appendix B for details on computing a matrix inverse) yields

$$\begin{bmatrix} A \\ B \end{bmatrix} = \frac{1}{\lambda_2 - \lambda_1} \begin{bmatrix} \lambda_2 & -1 \\ -\lambda_1 & 1 \end{bmatrix} \begin{bmatrix} x_0 \\ v_0 \end{bmatrix}$$

Expanding, substituting in the values for λ_1 and λ_2, recalling that they are real numbers (i.e. $\zeta^2 > 1$) and writing as two separate equations results in

$$A = \frac{-v_0 + (-\zeta - \sqrt{\zeta^2 - 1})\omega_n}{-2\omega_n\sqrt{\zeta^2 - 1}} \text{ and } B = \frac{v_0 + (\zeta - \sqrt{\zeta^2 - 1})\omega_n}{-2\omega_n\sqrt{\zeta^2 - 1}}$$

Factoring out the minus sign in the denominator results in Equations (1.16).

1.4 Forced Response

The preceding analysis considers the vibration of a device or structure due to some initial disturbance (nonzero v_0 and x_0). In this section, the vibration of a spring-mass-damper system subjected to an external force is considered. In particular, the response to harmonic excitations, impulses and step forcing functions is examined.

In many environments, rotating machinery, motors, etc., cause periodic motions of structures to induce vibrations into other mechanical devices and structures nearby. It is common to approximate the driving forces, $F(t)$, as periodic of the form

$$F(t) = F_0 \sin \omega t \tag{1.20}$$

where F_0 represents the amplitude of the applied force and ω denotes the frequency of the applied force, or the driving frequency, in rad/s. On summing forces, the equation for the forced vibration of the system in Figure 1.9 becomes

$$m\ddot{x} + c\dot{x} + kx = F_0 \sin \omega t \tag{1.21}$$

Recall from the discipline of differential equations (Boyce and DiPrima, 2012), that the solution of Equation (1.21) consists of the sum of the homogeneous solution Equation (1.5) and a particular solution. These are usually referred to as the *transient response* and the *steady-state response*, respectively. Physically, there is motivation to assume that the steady state response will follow the forcing function. Hence, it is tempting to assume that the particular solution has the form

$$x_p(t) = X \sin(\omega t - \theta) \tag{1.22}$$

where X is the steady-state amplitude and θ is the phase shift at steady state. Mathematically, the method is referred to as the method of undetermined coefficients. Substitution of Equation (1.22) into Equation (1.21) yields

$$X = \frac{F_0/k}{\sqrt{(1 - m\omega^2/k)^2 + (c\omega/k)^2}}$$

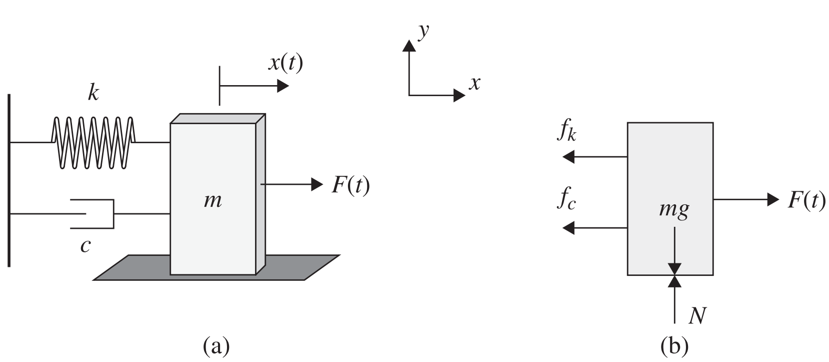

Figure 1.9 **(a)** The schematic of the forced spring-mass-damper system, assuming no friction on the surface. **(b)** The free-body diagram of the system of part (a).

or

$$\frac{Xk}{F_0} = \frac{1}{\sqrt{\left[1-(\omega/\omega_n)^2\right]^2 + [2\zeta(\omega/\omega_n)]^2}} \tag{1.23}$$

and

$$\tan\theta = \frac{(c\omega/k)}{1-m\omega^2/k} = \frac{2\zeta(\omega/\omega_n)}{1-(\omega/\omega_n)^2} \tag{1.24}$$

where $\omega_n = \sqrt{k/m}$ as before. Since the system is linear, the sum of two solutions is a solution, and the total time response for the system in Figure 1.9 for the case $0 < \zeta < 1$ becomes

$$x(t) = e^{-\zeta\omega_n t}(A\sin\omega_d t + B\cos\omega_d t) + X\sin(\omega t - \theta) \tag{1.25}$$

Here A and B are constants of integration determined by the initial conditions and the forcing function (and in general will be different than the values of A and B determined for the free response). See Examples 1.4.2 and 1.5.1 for the case where the driving force is a cosine function.

Examining Equation (1.25), two features are important and immediately obvious. First, as t gets larger, the transient response (the first term) becomes very small – hence the term steady-state response is assigned to the particular solution (the second term). The second observation is that the coefficient of the steady state response, or particular solution, becomes large when the excitation frequency is close to the undamped natural frequency, i.e. $\omega \approx \omega_n$. This phenomenon is known as *resonance* and is extremely important in design, vibration analysis and testing.

Example 1.4.1

Compute the response of the following system (assuming consistent units)

$$\ddot{x}(t) + 0.4\dot{x}(t) + 4x(t) = \frac{1}{\sqrt{2}}\sin 3t, \quad x(0) = \frac{-3}{\sqrt{2}}, \quad \dot{x}(0) = 0$$

Solution: First solve for the particular solution by using the more convenient form of

$$x_p(t) = X_1 \sin 3t + X_2 \cos 3t$$

rather than the magnitude and phase form, where X_1 and X_2 are the constants to be determined. Differentiating x_p yields

$$\dot{x}_p(t) = 3X_1\cos 3t - 3X_2 \sin 3t$$
$$\ddot{x}_p(t) = -9X_1 \sin 3t - 9X_2\cos 3t$$

Substitution of x_p and its derivatives into the equation of motion and collecting like terms yields

$$\left(-9X_1 - 1.2X_2 + 4X_1 - \frac{1}{\sqrt{2}}\right)\sin 3t + (-9X_2 + 1.2X_1 + 4X_2)\cos 3t = 0$$

Since the sine and cosine are independent, the two coefficients in parenthesis must vanish, resulting in two equations in the two unknowns, X_1 and X_2. This solution yields

$$x_p(t) = -0.134 \sin 3t - 0.032 \cos 3t$$

Next consider adding the free response to this. From the problem statement

$$\omega_n = 2 \text{ rad/s}, \quad \zeta = \frac{0.4}{2\omega_n} = 0.1 < 1, \quad \omega_d = \omega_n\sqrt{1-\zeta^2} = 1.99 \text{ rad/s}$$

Thus, the system is underdamped, and the total solution is of the form

$$x(t) = e^{-\zeta\omega_n t}(A \sin \omega_d t + B \cos \omega_d t) + X_1 \sin \omega t + X_2 \cos \omega t$$

Applying the initial conditions requires the derivative

$$\begin{aligned}\dot{x}(t) &= e^{-\zeta\omega_n t}(\omega_d A \cos \omega_d t - \omega_d B \sin \omega_d t) + \omega X_1 \cos \omega t \\ &\quad - \omega X_2 \sin \omega t - \zeta\omega_n e^{-\zeta\omega_n t}(A \sin \omega_d t + B \cos \omega_d t)\end{aligned}$$

The initial conditions yield the constants A and B

$$\begin{aligned}x(0) &= B + X_2 = \frac{-3}{\sqrt{2}} \Rightarrow B = -X_2 - \frac{3}{\sqrt{2}} = -2.089 \\ \dot{x}(0) &= \omega_d A + \omega X_1 - \zeta\omega_n B = 0 \Rightarrow A = \frac{1}{\omega_d}(\zeta\omega_n B - \omega X_1) = -0.008\end{aligned}$$

Thus the total solution is

$$x(t) = -e^{-0.2t}(0.008 \sin 1.99t + 2.089 \cos 1.99t) - 0.134 \sin 3t - 0.032 \cos 3t$$

Example 1.4.2
Calculate the form of the forced response if, instead of a sinusoidal driving force, the applied force is given by

$$F(t) = F_0 \cos \omega t.$$

Solution: In this case, assume that the response is also a cosine function out of phase or

$$x_p(t) = X \cos(\omega t - \theta)$$

To make the computations easy to follow, this is written in the equivalent form using a basic trig identity

$$x_p(t) = A_s \cos \omega t + B_s \sin \omega t$$

where the constants $A_s = X \cos \theta$ and $B_s = X \sin \theta$ satisfying

$$X = \sqrt{A_s^2 + B_s^2} \quad \text{and} \quad \theta = \tan^{-1} \frac{B_s}{A_s}$$

are undetermined constant coefficients. Taking derivatives of the assumed form of the solution and substitution of these into the equation of motion yields

$$\left(-\omega^2 A_s + 2\zeta\omega_n\omega B_s + \omega_n^2 A_s - f_0\right)\cos\omega t + \left(-\omega^2 B_s - 2\zeta\omega_n\omega A_s + \omega_n^2 B_s\right)\sin\omega t = 0$$

This equation must hold for all time, in particular for $t = \pi/2\omega$, so that the coefficient of sin ωt must vanish. Similarly, for $t = 0$, the coefficient of cos ωt must vanish. This yields the two equations

$$\left(\omega_n^2 - \omega^2\right) A_s + \left(2\zeta\omega_n\omega\right) B_s = f_0$$

and

$$(-2\zeta\omega_n\omega)A_s + \left(\omega_n^2 - \omega^2\right) + B_s = 0$$

in the two undetermined coefficients A_s and B_s. Solving yields

$$A_s = \frac{(\omega_n^2 - \omega^2)f_0}{\left(\omega_n^2 - \omega^2\right)^2 + (2\zeta\omega_n\omega)^2}$$

$$B_s = \frac{2\zeta\omega_n\omega f_0}{\left(\omega_n^2 - \omega^2\right)^2 + (2\zeta\omega_n\omega)^2}$$

Substitution of these expressions into the equations for X and θ yields the particular solution

$$x_p(t) = \overbrace{\frac{f_0}{\sqrt{\left(\omega_n^2 - \omega^2\right)^2 + (2\zeta\omega_n\omega)^2}}}^{X} \cos\left(\omega t - \overbrace{\tan^{-1}\frac{2\zeta\omega_n\omega}{\omega_n^2 - \omega^2}}^{\theta}\right)$$

Resonance is generally to be avoided in designing structures, since it means large amplitude vibrations, which can cause fatigue failure, discomfort, loud noises, etc. Occasionally, the effects of resonance are catastrophic. However, the concept of resonance is also very useful in testing structures and in certain applications such as energy harvesting (Section 7.10). In fact, the process of modal testing (Chapter 12) is based on resonance. Figure 1.10 illustrates how ω_n and ζ affect the amplitude at resonance. The dimensionless quantity Xk/F_0 is called the *magnification factor* and Figure 1.10 is called a *magnification curve* or *magnitude plot*. The maximum value at resonance, called the *peak resonance*, and denoted by M_p, can be shown (Inman, 2014) to be related to the damping ratio by

$$M_p = \frac{1}{2\zeta\sqrt{1-\zeta^2}} \tag{1.26}$$

Also, Figure 1.10 can be used to define the *bandwidth* of the structure, denoted by BW, as the value of the driving frequency at which the magnitude drops below 70.7% of its zero frequency value (also said to be the 3-dB down point from the zero

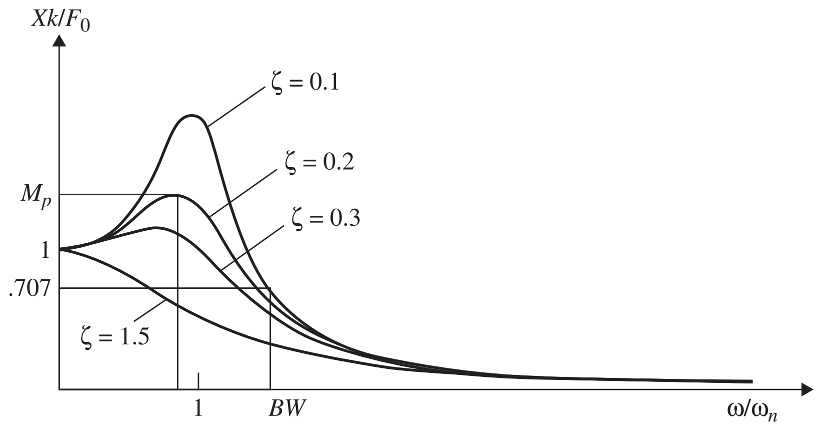

Figure 1.10 Magnification curves (dimensionless) for an SDOF system showing the normalized amplitude of vibration versus the ratio of driving frequency to natural frequency ($r = \omega/\omega_n$).

frequency point). The bandwidth can be calculated (Kuo and Golnaraghi, 2009: p. 359) in terms of the damping ratio by

$$BW = \omega_n \sqrt{(1 - 2\zeta^2) + \sqrt{4\zeta^4 - 4\zeta^2 + 2}} \tag{1.27}$$

Two other quantities are used in discussing the vibration of underdamped structures. They are the *loss factor* defined at resonance (only) to be

$$\eta = 2\zeta \tag{1.28}$$

and the *Q value*, or *resonance sharpness* factor, given by

$$Q = \frac{1}{2\zeta} = \frac{1}{\eta} \tag{1.29}$$

Another common situation focuses on the transient nature of the response, namely, the response of Equation (1.6) to an impulse, to a step function, or to initial conditions. Many mechanical systems are excited by loads, which act for a very brief time. Such situations are usually modeled by introducing a fictitious function called the *unit impulse function*, or the *Dirac delta function*. This delta function, denoted δ, is defined by the two properties

$$\begin{aligned} &\delta(t - a) = 0 \qquad t \neq a \\ &\int_{-\infty}^{\infty} \delta(t - a)\, dt = 1 \end{aligned} \tag{1.30}$$

where a is the instant of time at which the impulse is applied. Strictly speaking, the quantity $\delta(t)$ is not a function; however, it is very useful in quantifying important physical phenomena of an impulse.

The response of the system of Figure 1.9 for the underdamped case (with $a = x_0 = v_0 = 0$) can be given by

$$x(t) = \begin{Bmatrix} 0 & t < a \\ \dfrac{1}{m\omega_d} e^{-\zeta\omega_n t} \sin \omega_d t & t \geq a \end{Bmatrix} \tag{1.31}$$

Note from Equation (1.13) that this corresponds to the transient response of the system to the initial conditions $x_0 = 0$ and $v_0 = 1/m$. Hence, the impulse response is equivalent to giving a system at rest an initial velocity of $(1/m)$. This makes the impulse response, $x(t)$, important in discussing the transient response of more complicated systems. The impulse is also very useful in making vibration measurements, as described in Chapter 12.

A physical impact applied to a structure can be modeled by using the Dirac delta function with a magnitude representing the size of the impact. In this case, the impulse applied to the structure is modeled as having a magnitude F applied over a short time period Δt so that the effective change in momentum is $mv_0 - 0 = F\,\Delta t$, assuming the structure is initially at rest. This is equivalent to imparting an initial velocity of $v_0 = F\,\Delta t/m$. Thus, for an impulse of magnitude F applied over time Δt, the response becomes

$$x(t) = \begin{Bmatrix} 0 & t < a \\ \dfrac{F\Delta t}{m\omega_d} e^{-\zeta\omega_n t} \sin \omega_d t & t \geq a \end{Bmatrix} \tag{1.32}$$

Often design problems are stated in terms of certain specifications based on the response of the system to step function excitation. The response of the system in Figure 1.9 to a step function (of magnitude $m\omega_n^2$ for convenience), with initial conditions both set to zero, is calculated for underdamped systems from

$$m\ddot{x} + c\dot{x} + kx = m\omega_n^2 \mu(t), \qquad \mu(t) = \begin{cases} 0 & t < 0 \\ 1 & t \geq 0 \end{cases} \tag{1.33}$$

to be

$$x(t) = 1 - \frac{e^{-\zeta\omega_n t} \sin(\omega_d t + \phi)}{\sqrt{1-\zeta^2}} \tag{1.34}$$

where

$$\phi = \arctan\left[\frac{\sqrt{1-\zeta^2}}{\zeta}\right] \tag{1.35}$$

A sketch of the response is given in Figure 1.11, along with the labeling of several significant specifications for the case $m = 1$, $\omega_n = 2$ and $\zeta = 0.2$.

In some situations, the steady-state response of a structure may be at an acceptable level, but the transient response may exceed acceptable limits. Hence, one important measure is the *overshoot*, labeled O.S. in Figure 1.11 and defined to be

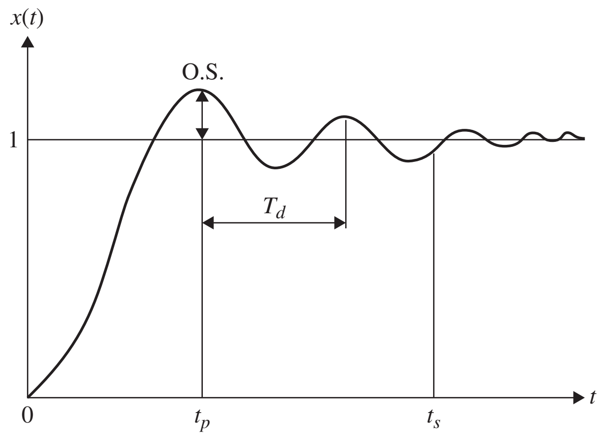

Figure 1.11 Step response of an SDOF system.

the maximum value of the response minus the steady-state value of the response. From Equation (1.34) it can be shown that

$$\text{overshoot} = \text{O.S.} = x_{\max}(t) - 1 = e^{-\zeta\pi/\sqrt{1-\zeta^2}} \tag{1.36}$$

This occurs at the *peak time*, t_p, which can be shown to be

$$t_p = \frac{\pi}{\omega_n\sqrt{1-\zeta^2}} \tag{1.37}$$

In addition, the period of oscillation, T_d, is given by

$$T_d = \frac{2\pi}{\omega_n\sqrt{1-\zeta^2}} = 2t_p \tag{1.38}$$

Another useful quantity, which indicates the behavior of the transient response, is the *settling time*, t_s. This is the time it takes the response to get within ±5% of the steady-state response and remain within ±5%. One approximation of t_s is given by Kuo and Golnaraghi (2009: p. 263)

$$t_s = \frac{3.2}{\omega_n\zeta} \tag{1.39}$$

The preceding definitions allow designers and vibration analysts to specify and classify precisely the nature of the transient response of an underdamped system. These definitions also give some indication of how to adjust the physical parameters of the system so that the response has a desired shape.

The response of a system to an impulse may be used to determine the response of an underdamped system to any input $F(t)$ by defining the *impulse response function* by

$$h(t) = \frac{1}{m\omega_d} e^{-\zeta\omega_n t} \sin\omega_d t \tag{1.40}$$

Then the solution of

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = F(t)$$

can be shown to be

$$x(t) = \int_0^t F(\tau)h(t-\tau)d\tau = \frac{1}{m\omega_d}e^{-\zeta\omega_n t}\int_0^t F(\tau)e^{\zeta\omega_n\tau}\sin\omega_d(t-\tau)d\tau \tag{1.41}$$

for the case of zero initial conditions. This last expression gives an analytical representation for the response to any driving force that has an integral.

Example 1.4.3
Consider a spring-mass-damper system with $m = 1$ kg, $c = 2$ kg/s and $k =$ 2000 N/m, with an impulsive force applied to it of 10,000 N for 0.01 s. Compute the resulting response.

Solution: A 10,000 N force acting over 0.01 s provides (area under the curve) a value of $F\Delta t = 10000 \times 0.01 = 100$ N · s Using the values given, the equation of motion is

$$\ddot{x}(t) + 2\dot{x}(t) + 2000x(t) = 100\delta(t)$$

Thus the natural frequency, damping ratio and damped natural frequency are

$$\omega_n = \sqrt{\frac{2000}{1}} = 44.721 \text{ rad/s}, \zeta = \frac{2}{2\sqrt{1\times 2000}} = 0.022,$$

$$\omega_d = 44.721\sqrt{1-0.022^2} = 44.71 \text{ rad/s}$$

Using Equation (1.32), the response becomes

$$x(t) = \frac{\hat{F}e^{-\zeta\omega_n t}}{m\omega_d}\sin\omega_d t = 2.237e^{-0.1t}\sin(44.71t)$$

1.5 Transfer Functions and Frequency Methods

The preceding analysis of the response was carried out in the time domain. Current vibration measurement methodology (Ewins, 2000), as well as much control analysis (Kuo and Golnaraghi, 2009), often takes place in the frequency domain. Hence, it is worth the effort to reexamine these calculations using frequency domain methods (a phrase usually associated with linear control theory). The frequency domain approach arises naturally from mathematics (ordinary differential equations) via an alternative method of solving differential equations, such as Equations (1.21) and (1.33), using the Laplace transform (Boyce and DiPrima, 2012; Chapter 6).

Taking the Laplace transform of Equation (1.33), assuming both initial conditions to be zero, yields

$$X(s) = \left[\frac{1}{ms^2 + cs + k}\right]\mu(s) \tag{1.42}$$

where $X(s)$ denotes the Laplace transform of $x(t)$, and $\mu(s)$ is the Laplace transform on the right-hand side of Equation (1.33). If the same procedure is applied to Equation (1.21), the result is

$$X(s) = \left[\frac{1}{ms^2 + cs + k}\right] F_0(s) \tag{1.43}$$

where $F_0(s)$ denotes the Laplace transform of $F_0 \sin \omega t$. Note that

$$G(s) = \frac{X(s)}{\mu(s)} = \frac{X(s)}{F_0(s)} = \frac{1}{ms^2 + cs + k} \tag{1.44}$$

Thus, it appears that the quantity $G(s) = [1/(ms^2 + cs + k)]$, the ratio of the Laplace transform of the output (response) to the Laplace transform of the input (applied force) to the system characterizes the system (structure) under consideration. This characterization is independent of the input or driving function. This ratio, $G(s)$, is defined as the *transfer function* of this system in control analysis (or of this structure in vibration analysis). The transfer function can be used to provide analysis of the vibrational properties of the structure, as well as to provide a means of measuring the structure's dynamic response.

In control theory, the transfer function of a system is defined in terms of an output to input ratio, but the use of a transfer function in structural dynamics and vibration testing implies certain physical properties, depending on whether position, velocity or acceleration is considered as the response (output). It is common, for instance, to measure the response of a structure by using an accelerometer. The transfer function resulting is then $s^2X(s)/U(s)$, where $U(s)$ is the Laplace transform of the input and $s^2X(s)$ is the Laplace transform of the acceleration. This transfer function is called the *inertance* and its reciprocal is referred to as the *apparent mass*. Table 1.1 lists the nomenclature of various transfer functions. The physical basis for these names can be seen from their graphical representation.

The transfer function representation of a structure is very useful in control theory as well as in vibration testing. It also forms the basis of impedance methods discussed in the next section. The variable s in the Laplace transform is a complex variable, which can be further denoted by

$$s = \sigma + j\omega_d$$

where the real numbers σ and ω_d denote the real and imaginary parts of s, respectively ($j = \sqrt{-1}$). Thus, the various transfer functions are also complex-valued.

Table 1.1 Various transfer functions.

Response Measurement	Transfer Function	Inverse Transfer Function
Acceleration	Inertance	Apparent mass
Velocity	Mobility	Impedance
Displacement	Compliance	Dynamic stiffness

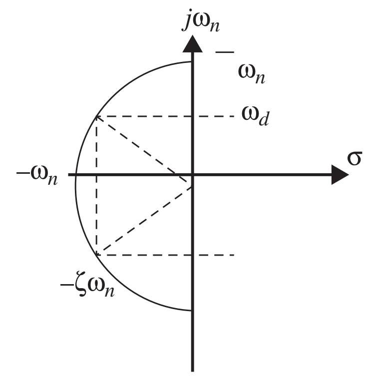

Figure 1.12 Complex *s*-plane of the poles (roots of the characteristics of Equation (1.39).

In control theory, the values of s where the denominator of the transfer function $G(s)$ vanishes are called the *poles* of the transfer function. A plot of the poles of the compliance (also called receptance) transfer function for Equation (1.44) in the complex s-plane is given in Figure 1.12. The points on the semi-circle occur where the denominator of the transfer function is zero. These values of s ($s = -\zeta\omega_n \pm \omega_d j$) are exactly the roots of the characteristic equation for the structure. The values of the physical parameters m, c and k determine the two quantities ζ and ω_n, which in turn determine the position of the poles in Figure 1.12.

Another graphical representation of a transfer function useful in control is the *block diagram* illustrated in Figure 1.13a. This diagram is an icon for the definition of a transfer function. The control terminology for the physical device represented by the transfer function is the *plant*, whereas in vibration analysis the plant is usually referred to as the structure. The block diagram of Figure 1.13b is meant to imply the formula

$$\frac{X(s)}{U(s)} = \frac{1}{(ms^2 + cs + k)} \tag{1.45}$$

exactly.

The response of Equation (1.21) to a sinusoidal input (forcing function) motivates a second description of a structure's transfer function called the *frequency response function* (often denoted by FRF). The FRF is defined as the transfer function evaluated at $s = j\omega$, i.e. $G(j\omega)$. The significance of the FRF follows from Equation (1.22), namely, that the steady-state response of a system driven sinusoidally is a sinusoid of the same frequency with different amplitude and phase. In fact,

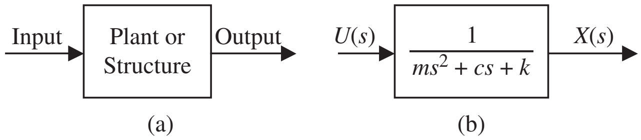

Figure 1.13 Block diagram representation of an SDOF system.

substitution of $j\omega$ into Equation (1.45) yields exactly Equations (1.23) and (1.24) from

$$\frac{X}{F_0} = |G(j\omega)| = \sqrt{x^2(\omega) + y^2(\omega)} \tag{1.46}$$

where $|G(j\omega)|$ indicates the magnitude of the complex FRF

$$\phi = \tan^{-1} G(j\omega) = \tan^{-1}\left[\frac{y(\omega)}{x(\omega)}\right] \tag{1.47}$$

indicates the phase of the FRF, and

$$G(j\omega) = x(\omega) + y(\omega)j \tag{1.48}$$

This mathematically expresses two ways to represent a complex function, as the sum of its real part (Re $G(j\omega) = x(\omega)$) and its imaginary part (Im $(G(j\omega)) = y(\omega)$), or by its magnitude ($|G(j\omega)|$) and phase (ϕ). In more physical terms, the FRF of a structure represents the magnitude and phase shift of its steady-state response under sinusoidal excitation. While Equations (1.23), (1.24), (1.46) and (1.47) verify this for an SDOF viscously damped structure, it can be shown in general for any linear time invariant plant (Melsa and Schultz, 1969: p. 187)).

It should also be noted that the FRF of a linear system can be obtained from the transfer function of the system and vice versa. Hence, the FRF uniquely determines the time response of the structure to any known input.

Graphical representations of the FRF form an extensive part of control analysis and also form the backbone of vibration measurement analysis. Next, three sets of FRF plots that are useful in testing vibrating structures are examined. The first set of plots consists simply of plotting the imaginary part of the FRF versus the driving frequency and the real part of the FRF versus the driving frequency. These are shown for the damped SDOF system in Figure 1.14 (the compliance FRF for $\zeta = 0.01$ and $\omega_n = 20$ rad/s).

The second representation consists of a single plot of the imaginary part of the FRF versus the real part of the FRF. This type of plot is called a *Nyquist plot* (also called an *Argand plane plot*) and is used for measuring the natural frequency and damping in testing methods and for stability analysis in control system design. The

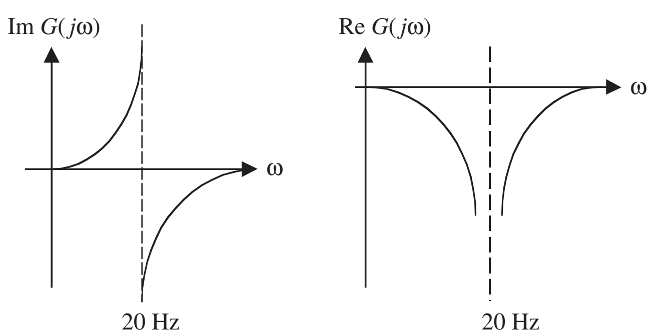

Figure 1.14 Plots of the real part and the imaginary part of the FRF.

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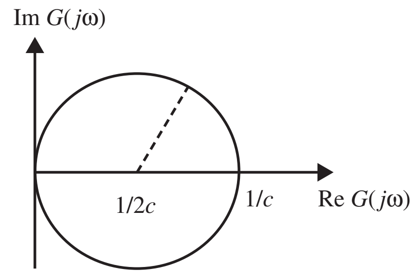
Figure 1.15 Nyquist plot for Equation 1.44.



Nyquist plot of the mobility FRF of a structure modeled by Equation (1.44) is given in Figure 1.15.

The last plots considered for representing the FRF are called *Bode plots* and consist of a plot of the magnitude of the FRF versus the driving frequency and the phase of the FRF versus the driving frequency (a complex number requires *two* real numbers to describe it completely). Bode plots have long been used in control system design and analysis as well as for determining the plant transfer function of a system. More recently, Bode plots have been used in analyzing vibration test results and in determining the physical parameters of the structure.

In order to represent the complete Bode plots in a reasonable space, $\log_{10}$ scales are often used to plot $|G(j\omega)|$. This has given rise to the use of the decibel and decades in discussing the magnitude response in the frequency domain. The magnitude and phase plots (for the compliance transfer function) for the system in Equation (1.21) are shown in Figures 1.16 and 1.17 for different values of ζ. Note the phase change at resonance (90°), as this is important in interpreting measurement data.

Note that Figures 1.10 and 1.17 show the same physical phenomenon and are both plots of the compliance transfer function. However, the magnitude in Figure 1.10 is dimensionless versus dimensionless frequency, while Figure 1.17 is usually the magnitude in decibels versus frequency on a semi-log scale.

Example 1.5.1

Solve the following system using the Laplace Transform method and using a Table of Laplace Transform pairs (from the Internet)

$$m\ddot{x}(t) + kx(t) = F_0 \cos\omega(t),\ x(0) = x_0,\ \dot{x}(0) = v_0$$

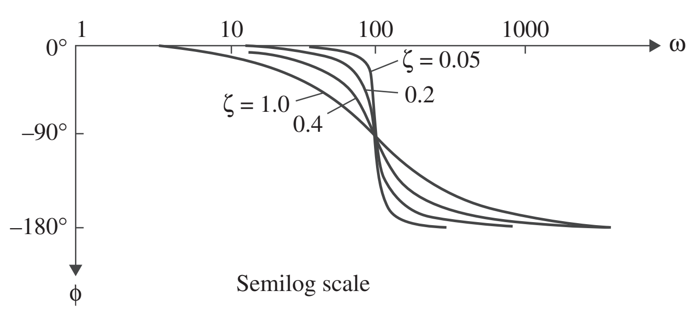


Figure 1.16 Bode phase plot for Equation (1.39) showing resonance at –90°.

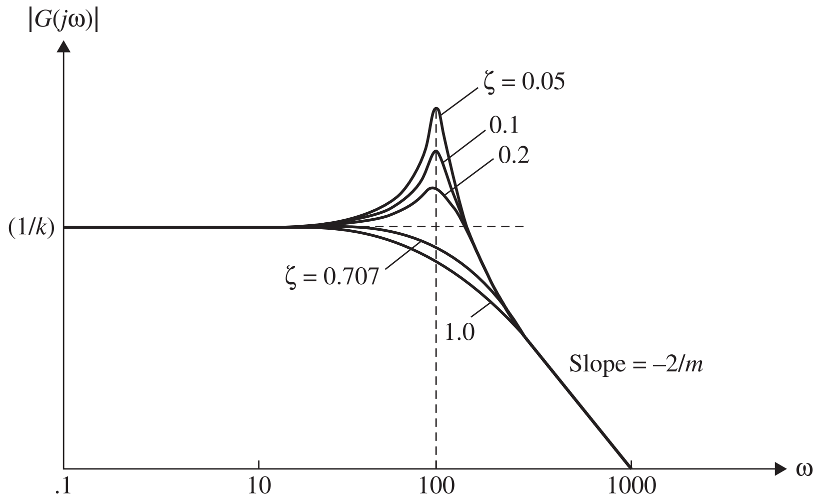


Figure 1.17 Bode magnitude plot for Equation (1.39) showing resonance and values of mass and stiffness.

Solution: First divide through by the mass to get

$$\ddot{x}(t) + \omega_n^2 x(t) = f_0 \cos \omega t,\ x(0) = x_0,\ \dot{x}(0) = v_0$$

Here $f_0 = F_0/m$. Taking the Laplace Transform (see the Table of Laplace Transforms: from the Internet) of the equation of motion considering the initial conditions yields

$$s^2 X(s) - sx_0 - v_0 + \omega_n^2 X(s) = \frac{sf_0}{s^2 + \omega^2}$$

$$\Rightarrow \left(s^2 + \omega_n^2\right) X(s) = sx_0 + v_0 + \frac{sf_0}{s^2 + \omega^2}$$

Solving this for $X(s)$ yields

$$\begin{aligned} X(s) &= \frac{sx_0 + v_0}{s^2 + \omega_n^2} + \frac{sf_0}{\left(s^2 + \omega_n^2\right)\left(s^2 + \omega^2\right)} \\ &= (x_0)\frac{s}{s^2 + \omega_n^2} + \left(\frac{v_0}{\omega_n}\right)\frac{\omega_n}{s^2 + \omega_n^2} + \frac{sf_0}{\left(s^2 + \omega_n^2\right)\left(s^2 + \omega^2\right)} \end{aligned}$$

Taking the Inverse Laplace Transform using an online table of each term yields

$$\begin{aligned} x(t) &= x_0 \cos \omega_n t + \frac{v_0}{\omega_n} \sin \omega_n t + \frac{f_0}{\omega_n^2 - \omega^2}(\cos \omega t - \cos \omega_n t) \\ &= \frac{v_0}{\omega_n} \sin \omega_n t + \left(x_0 - \frac{f_0}{\omega_n^2 - \omega^2}\right) \cos \omega_n t + \frac{f_0}{\omega_n^2 - \omega^2} \cos \omega t \end{aligned}$$

In comparing this with the solution given in Equation (1.25) for zero damping, note that Equation (1.25) is the solution for the case where the driving force is a sine function instead of a cosine as solved here.

1.6 Complex Representation and Impedance

Table 1.1 formally defines impedance as the ratio of a sinusoidal driving force, F, acting on the system to the resulting velocity, v, of the system. Impedance is usually denoted by the symbol Z and is a measure of a structure's resistance to motion. In working with impedance methods it is common to use the complex exponential notation to represent harmonic quantities. Using the exponential notation, the sinusoidal force in Equation (1.21) can be written as

$$F(t) = F_0 e^{j\omega t} \tag{1.49}$$

Here, ω is the driving frequency as before. The impedance approach offers an alternative way to examine systems vibrating harmonically based on using complex functions to represent the response.

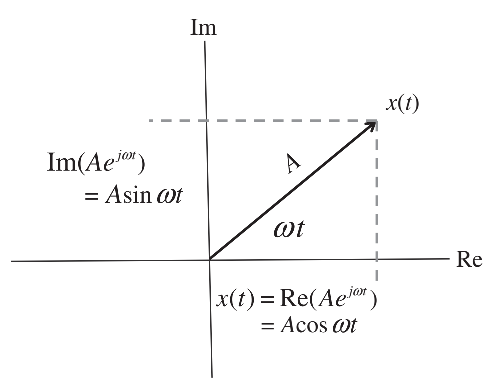
A useful way to visualize harmonic motion is to think of the response $x(t)$ as a vector rotating in the complex plane, as illustrated in Figure 1.18. Here the vector has magnitude A and rotates an angle ωt in the complex plane. From Euler's formula for the complex exponential function

$$x(t) = Ae^{j\omega t} = A\cos\omega t + Aj\sin\omega t \tag{1.50}$$

which agrees with representation in Figure 1.18. Differentiation of the complex exponential yields simply

$$\begin{aligned} \frac{d}{dt}(Ae^{j\omega t}) &= j\omega A e^{j\omega t} = j\omega x(t) \\ \frac{d^2}{dt^2}(Ae^{j\omega t}) &= j^2\omega^2 A e^{j\omega t} = -\omega^2 x(t) \end{aligned} \tag{1.51}$$

Figure 1.18 Graphic illustration of Euler's formula of the complex exponential.



Thus, each differentiation of the complex exponential results in simply multiplying by $j\omega$, similar to multiplying by s in the Laplace domain.

From the Figure 1.18, the physical displacement is interpreted from the complex exponential as just the real part of Equation (1.50). Thus the velocity becomes the real part of the derivative of the complex exponential and the acceleration is the real part of the derivative of that or

$$\begin{aligned} x(t) &= \mathrm{Re}(Ae^{j\omega t}) = A\cos(\omega t) \\ \dot{x}(t) &= \mathrm{Re}(j\omega Ae^{j\omega t}) = -\omega A\sin(\omega t) \\ \ddot{x}(t) &= \mathrm{Re}(j^2\omega^2 Ae^{j\omega t}) = -\omega^2 A\cos(\omega t) \end{aligned} \tag{1.52}$$

If the displacement is thought to be a sine function, then the physical motion variables become the imaginary parts of the complex exponential. Using the complex notation equation for the forced response of an SDOF system becomes

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = F_0 e^{jwt} \tag{1.53}$$

Assuming the resulting displacement is of the form

$$x(t) = A\sin(\omega t - \theta)$$

its complex form is the corresponding velocity as

$$v(t) = Aj\omega e^{j(\omega t+\theta)} \tag{1.54}$$

Here ω and θ are the driving frequency and phase shift between the applied force and the resulting response respectively. Substituting the complex form of $x(t)$ into Equation (1.48) yields

$$[-\omega^2 m + j\omega c + k]Ae^{j\omega - j\theta} = F(t) \tag{1.55}$$

Solving for the complex value A yields

$$A = \frac{F_0 e^{j\theta}}{[-\omega^2 m + j\omega c + k]} \tag{1.56}$$

which has magnitude and phase given by

$$|A| = \frac{F}{\sqrt{(k-\omega^2 m)^2 + (\omega c)^2}} \quad \text{and} \quad \theta = \tan\frac{\omega c}{k - \omega^2 m} \tag{1.57}$$

These values are of course the same as those derived in the previous section in Equations (1.23 and 1.24).

Examination of the force/velocity expressions for each element reveals the impedance of each, and these are given in Table 1.2.

Table 1.2 Impedance values for mass, damping and stiffness.

Mass	$Z = j\omega m$
Damping	$Z = c$
Stiffness	$Z = {}^{-jk}/_{\omega}$

Example 1.6.1
Compute the mechanical impedance of the spring-mass-damper system of Figure 1.9.

Solution: Dividing Equation (1.55) by (1.54) and simplifying yields that directly the mechanical impedance of the spring-mass-damper system becomes

$$\begin{aligned} Z = \frac{F}{v} = \frac{[k - \omega^2 m + j\omega c]Ae^{j\omega t - j\theta}}{Aj\omega e^{j\omega t - j\theta}} &= \frac{1}{j\omega}(k - \omega^2 m + j\omega c) \\ = \omega jm + c - \frac{k}{j\omega} \end{aligned} \tag{1.58}$$

Comparing this expression to the terms in Table 1.2 reveals that the mechanical impedance of the system is just the sum of the impedance expressions for each element. The use of the impedance method is essentially the existence of following rules developed in electrical engineering for combining deferent circuit elements by adding their impedances (e.g. series and parallel combinations) and making the analogy to electrical components of capacitance (reciprocal of stiffness), inductance (mass) and resistance (damping). The units of mechanical impedance are kg/s, the same as the viscous damping coefficient.

1.7 Measurement and Testing

One can also use the quantities defined in the previous sections to measure the physical properties of a structure. As mentioned before, resonance can be used to determine a system's natural frequency. Methods based on resonance are referred to as resonance testing (or modal analysis techniques) (Bishop and Gladwell, 1963) and are briefly introduced here and discussed in more detail in Chapter 8.

As mentioned earlier, the mass and stiffness of a structure can often be determined by making simple static measurements. However, damping rates require a dynamic measurement and hence are more difficult to determine. For underdamped systems one approach is to realize, from Figure 1.6, that the decay envelope is the function $e^{-\zeta\omega_n t}$. The points on the envelope illustrated in Figure 1.19

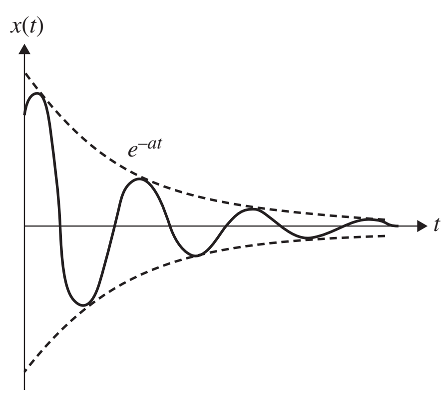


Figure 1.19 Free decay measurement method.

can be used to curve-fit the function e^{-at}, where a is the constant determined by the curve fit. The relation $a = \zeta\omega_n$ can next be used to calculate ζ and hence the damping rate c (assuming that m and k or ω_n are known).

A second approach is to use the concept of logarithmic decrement, denoted by δ (delta) and defined by

$$\delta = \ln\frac{x(t)}{x(t+T_d)} \tag{1.59}$$

where T_d is the period of oscillation. Using Equation (1.13) in the form

$$x(t) = Ae^{-\zeta\omega_n t}\sin(\omega_d t + \phi) \tag{1.60}$$

the value for δ becomes

$$\delta = \ln\left[\frac{e^{-\zeta\omega_n t}\sin(\omega_d t + \phi)}{e^{-\zeta\omega_n(t+T_d)}\sin(\omega_d t + \omega_d T_d + \phi)}\right] = \ln e^{\zeta\omega_n T_d} = \zeta\omega_n T_d \tag{1.61}$$

where the sine functions cancel because $\omega_d T_d$ is a one period shift by definition. Further evaluating δ yields

$$\delta = \zeta\omega_n T_d = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \tag{1.62}$$

Equation (1.62) can be manipulated to yield the damping ratio in terms of the decrement, i.e.

$$\zeta = \frac{\delta}{\sqrt{4\pi + \delta^2}} \tag{1.63}$$

Hence, if the decrement is measured, Equation (1.63) yields the damping ratio.

The various plots of the previous section can also be used to measure ω_n, ζ, m, c and k. For instance, the Bode diagram of Figure 1.17 can be used to determine the natural frequency, stiffness and damping ratio. The stiffness is determined from the intercept of the FRF and the magnitude axis, since the value of the magnitude of the FRF for small ω is $\log(1/k)$. This can be seen by examining the function $\log_{10}|G(j\omega)|$ for small ω. Note that

$$\log|G(j\omega)| = \log\frac{1}{k} - \frac{1}{2}\log\left[\left(1-\frac{\omega^2}{\omega_n^2}\right)^2 + \left(\frac{2\zeta\omega}{\omega_n}\right)^2\right] = \log\left(\frac{1}{k}\right) \tag{1.64}$$

for very small values of ω. Also note that $|G(j\omega)|$ evaluated at ω_n yields

$$k|G(j\omega_n)| = \frac{1}{2\zeta} \tag{1.65}$$

which provides a measure of the damping ratio from the magnitude plot of the FRF.

Note that Equations (1.65) and (1.26) appear to contradict each other, since

$$\frac{1}{2\zeta\sqrt{1-\zeta^2}} = k\max|G(j\omega)| = M_p \neq k|G(j\omega_n)| = \frac{1}{2\zeta}$$

except in the case of very small ζ (i.e. the difference between M_p and $|G(j\omega_n)|$ goes to zero as ζ goes to zero). This indicates a subtle difference between using the damping ratio obtained by using resonance as the value of ω, where $|G(j\omega_n)|$ is a maximum, and using the point, where $\omega = \omega_n$, the undamped natural frequency. This point is also illustrated by noting that the damped natural frequency, Equation (1.8), is $\omega_d = \omega_n\sqrt{1-\zeta^2}$ and ω_p, the frequency at which $|G(j\omega_n)|$ is maximum, is

$$\omega_p = \omega_n\sqrt{1-2\zeta^2} \tag{1.66}$$

Also note that Equation (1.66) is valid only if $0 < \zeta < 0.707$.

Finally, the mass can be related to the slope of the magnitude plot for the inertance transfer function, denoted by $G_I(s)$, by noting that

$$G_I(s) = \frac{s^2}{(ms^2 + cs + k)} \tag{1.67}$$

and for large ω (i.e. $\omega_n \ll \omega$), the value of $|G_I(j\omega)|$ is

$$|G_I(j\omega)| \approx (1/m) \tag{1.68}$$

Plots of these values are referred to as straight-line approximations to the actual magnitude plot (Bode, 1945).

The preceding formulas relating the physical properties of the structure to the magnitude Bode diagrams suggest an experimental way to determine a structure's parameters: namely, if the structure can be driven by a sinusoid of varying frequency and if the magnitude and phase (needed to locate resonance) of the resulting response are measured, then the Bode plots and the preceding formulas can be used to obtain the desired physical parameters. This process is referred to as plant identification in the controls literature and can be extended to systems with more degrees of freedom (see Melsa and Schultz (1969), for a more complete account).

There are several other formulas for measuring the damping ratio and natural frequency from the results of such experiments, sine sweeps. For instance, if the Nyquist plot of the mobility transfer function is used, a circle of diameter $1/c$ results (Figure 1.15). Another approach is to plot the magnitude of the FRF on a linear scale near the region of resonance (Figure 1.20). If the damping is small enough so that the peak at resonance is sharp, the damping ratio can be determined by measuring the frequencies at 0.707 at the maximum value (also called

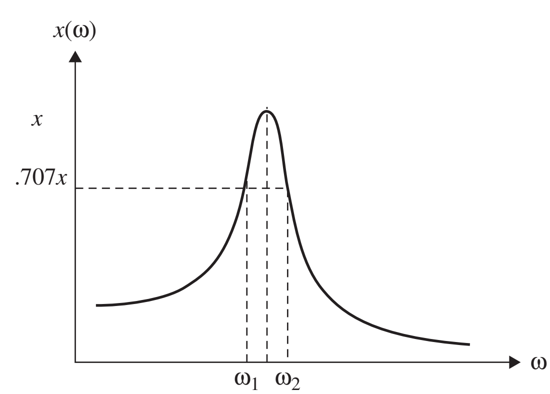


Figure 1.20 Quadrature peak picking method.

the 3-dB down point or half-power points), denoted by ω_1 and ω_2, respectively. Then, using the formula (Ewins, 2000)

$$\zeta = \frac{1}{2}\left[\frac{\omega_2 - \omega_1}{\omega_d}\right] \tag{1.69}$$

to compute the damping ratio. This method is referred to as *quadrature peak picking* and is illustrated in Figure 1.20.

1.8 Stability

In all the preceding analysis, the physical parameters *m, c* and *k* are, of course, positive quantities. There are physical situations, however, in which equations of the form of Equations (1.1) and (1.6) result but have one or more negative coefficients. Such systems are not well behaved and require some additional analysis.

Recalling that the solution to Equation (1.1) is of the form $A\sin(\omega t + \phi)$, where A is a constant, it is easy to see that the response, in this case $x(t)$, is bounded. That is to say that

$$|x(t)| \leq A \tag{1.70}$$

for all t where A is some finite constant and $|x(t)|$ denotes the absolute value of $x(t)$. In this case, the system is well behaved or *stable* (called marginally stable in the control's literature). In addition, note that the roots (also called *characteristic values* or eigenvalues) of

$$\lambda^2 m + k = 0$$

are purely complex numbers $\pm j\omega_n$ as long as m and k are positive (or have the same sign). If k happens to be negative and m is positive, the solution becomes

$$x(t) = A\sinh\omega_n t + B\cosh\omega_n t \tag{1.71}$$

which increases without bound as t does. Such solutions are called *divergent* or *unstable*.

If the solution of the damped system of Equation (1.6) with positive coefficients is examined, it is clear that $x(t)$ approaches zero as t becomes large, because of the exponential term. Such systems are considered to be *asymptotically stable* (called stable in the controls literature). Again, if one or two of the coefficients are negative, the motion grows without bound and becomes unstable as before. In this case, however, the motion may become unstable in one of two ways. Similar to overdamping and underdamping, the motion may grow without bound and not oscillate, or it may grow without bound and oscillate. The first case is referred to as *divergent instability* and the second case as *flutter instability*; together they fall under the topic of self-excited vibrations.

Apparently, the sign of the coefficient determines the stability behavior of the system. This concept is pursued in Chapter 4, where these stability concepts are formally defined. Figures 1.21 to 1.24 illustrate each of these concepts.

These stability definitions can also be stated in terms of the roots of the characteristic Equation (1.8) or in terms of the poles of the transfer function of the

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Figure 1.21 Response of a stable system.

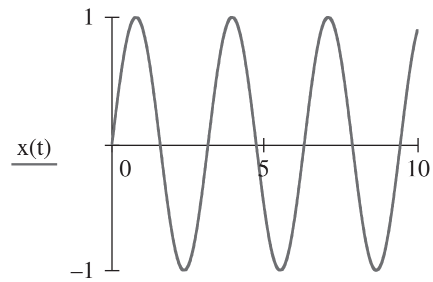


Figure 1.22 Response of an asymptotically stable system.

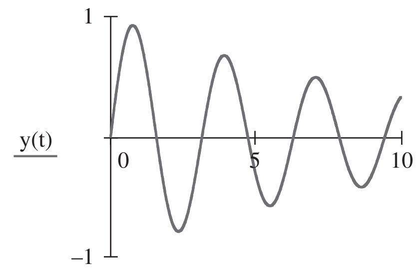


Figure 1.23 Response of a system with a divergent instability.

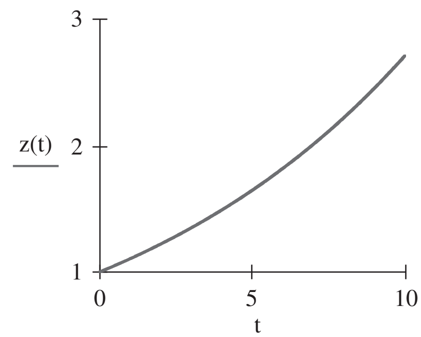
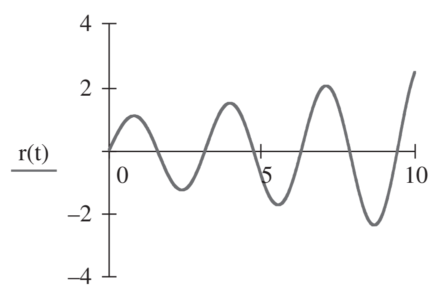


Figure 1.24 Response of a system with flutter instability.



system. In fact, referring to Figure 1.12, the system is stable if the poles of the structure lie along the imaginary axis (called the $j\omega$ axis), unstable if one or more poles are in the right half-plane, and asymptotically stable if all of the poles lie in the left half-plane. Flutter occurs when the poles are in the right half-plane and not on the real axis (complex conjugate pairs of roots with positive real part) and divergence occurs when the poles are in the right-half plane along the real axis. In the simple SDOF case considered here, the pole positions are entirely determined by the signs of m, c and k.

The preceding definitions and ideas about stability are stated for the free response of the system. These concepts of a well-behaved response can also be applied to the forced motion of a vibrating system. The stability of the forced response of a system can be defined by considering the nature of the applied force or input. The system is said to be *bounded-input, bounded-output stable* (or, simply, BIBO stable) if for *any* bounded input (driving force) the output (response) is bounded for any arbitrary set of initial conditions. Such systems are manageable at resonance.

It can be seen immediately that Equation (1.21) with $c = 0$, the undamped system, is not BIBO stable, since for $f(t) = \sin(\omega_n t)$, the response $x(t)$ goes to infinity (at resonance), whereas $f(t)$ is certainly bounded. However, the response of Equation (1.21) with $c > 0$ is bounded whenever $f(t)$ is. In fact, the maximum value of $x(t)$ at resonance M_p is illustrated in Figure 1.10. Thus, the system of Equation (1.21) with damping is said to be BIBO stable.

The fact that the response of an undamped structure is bounded when $f(t)$ is an impulse or step function suggests another, weaker, definition for the stability of the forced response. A system is said to be *bounded*, or *Lagrange stable*, with respect to a *given* input if the response is bounded for any set of initial conditions. Structures described by Equation (1.1) are Lagrange stable with respect to many inputs. This definition is useful when $f(t)$ is known completely or known to fall in some specified class of functions.

Stability can also be thought of in terms of whether or not the energy of the system is increasing (unstable), constant (stable) or decreasing (asymptotically stable), rather than in terms of the explicit response. Lyapunov stability, defined in Chapter 4, extends this idea. Another important view of stability is based on how sensitive a motion is to small perturbations in the system's parameters (m, c and k) and/or small perturbations in initial conditions. Unfortunately, there does not appear to be a universal definition of stability that fits all situations. The concept of stability becomes further complicated for nonlinear systems. The definitions and concepts mentioned here are extended and clarified in Chapter 4.

Example 1.8.1

Most structures are asymptotically stable (m, c, k are all positive) or at least stable (m, k positive $c = 0$). However, if other forces are present, such as flow through a pipe or over an airfoil, stability can be lost as the effective coefficients in the equation of motion could become negative. In addition, active control systems constructed to improve performance also add forces that can potentially destabilize a structure or machine. Discuss the stability properties of the following equation of motion

$$J\ddot{\theta} + (c - f_d)\dot{\theta} + k\theta = 0$$

Here, θ is the angle of rotation of the flap, J is the moment of inertia of the flap (assumed positive), k is the rotational stiffness of the device plus a control force and assumed positive, c is the internal damping of the device (positive) and f_d is the aerodynamic force applied to the flap (also positive). This is a crude representation of a control surface (flap or tab).

Solution: As long as k and J are positive, stability is controlled by the sign of the equivalent damping term, $c - f_d$. If $c = f_d$, then the system is stable, having the response of the form illustrated in Figure 1.21. If $c - f_d > 0$, then the solution is that of Equation (1.13) and exponential decay and the system is asymptotically stable, as illustrated in Figure 1.22. If, on the other hand, aerodynamic force overcomes the actuation force and internal damping so that $c - f_d < 0$, then the exponent in Equation (1.13) becomes positive and flutter instability occurs, as illustrated in Figure 1.23.

1.9 Design and Control of Vibrations

One can use the quantities defined in the previous sections to design structures and machines to have a desired transient and steady state response to some extent. For instance, it is a simple matter to choose *m, c* and *k* so that the overshoot is a specified value. However, if one needs to specify the overshoot, the settling time and the peak time, then there may not be a choice of *m, c* and *k* that will satisfy all three criteria. Hence, the response cannot always be completely shaped, as the formulas in Section 1.4 may seem to indicate.

Another consideration in designing structures is that each of the physical parameters *m, c* and *k* may already have design constraints that have to be satisfied. For instance, the material the structure is made of may fix the damping rate, *c*. Then, only the parameters *m* and *k* can be adjusted. In addition, the mass may have to be within 10% of a specified value, for instance, which further restricts the range of values of overshoot and settling time. The stiffness is often designed based on the static deflection limitation and strength.

For example, consider the system of Figure 1.11 and assume it is desired to choose values of *m, c* and *k* so that ζ and ω_n specify a response with a settling time $t_s = 3.2$ units and a time to peak, t_p, of 1 unit. Then Equations (1.37) and (1.39) imply that $\omega_n = 1/\zeta$ and $\zeta = 1/\sqrt{1+\pi^2}$. This, unfortunately, also specifies the overshoot, since

$$\text{O.S.} = \exp\left(\frac{-\zeta\pi}{\sqrt{(1+\zeta^2)}}\right)$$

Thus, all three performance criteria cannot be satisfied. This leads the designer to have to make compromises, to reconfigure the structure or to add additional components.

Hence, in order to meet vibration criteria such as avoiding resonance, it may be necessary in many instances to alter the structure by adding vibration absorbers or isolators (Machinante, 1984; Rivin, 2003). Another possibility is to use active vibration control and feedback methods. Both of these approaches are discussed in Chapters 6 and 7.

The choice of the physical parameters m, c and k determines the shape of the response of the system. The choice of these parameters can be considered as the design of the structure. Passive control can also be considered as a redesign process of changing these parameters on an already existing structure to produce a more desirable response. For instance, some mass could be added to a given structure to lower its natural frequency. Although passive control or redesign is generally the most efficient way to control or shape the response of a structure, the constraints on m, c and k are often such that the desired response cannot be obtained. Then the only alternative, short of starting over again, is to try active control.

There are many different types of active control methods, and only a few will be considered to give the reader a feel for the connection between the vibration and control disciplines. Again, the comments made in this text on control should not be considered as a substitute for studying standard control or linear systems texts. Output feedback control is briefly introduced here and discussed in more detail in Chapter 7.

First, a clarification of the difference between active and passive control is in order. Basically, an active control system uses some external adjustable or active (e.g. electronic) device, called an actuator, to provide a means of shaping or controlling the response. Passive control, on the other hand, depends only on a fixed (passive) change in the physical parameters of the structure. Passive control can also involve adding external devices to the structure; however, such devices are not powered. Active control often depends on current measurements of the response of the system and passive control does not. Active control requires an external energy source and passive control typically does not.

Feedback control consists of measuring the output, or response, of the structure and using that measurement to determine the force to apply to the structure to obtain a desired response. The device used to measure the response (sensor), the device used to apply the force (actuator) and any electronics required to transfer the sensor signal into an actuator command (control law) make up the *control hardware*. This is illustrated in Figure 1.25 by using a *block diagram*. Systems with feedback are referred to as closed-loop systems, while control systems without feedback are called open-loop systems, as illustrated in Figures 1.25 and

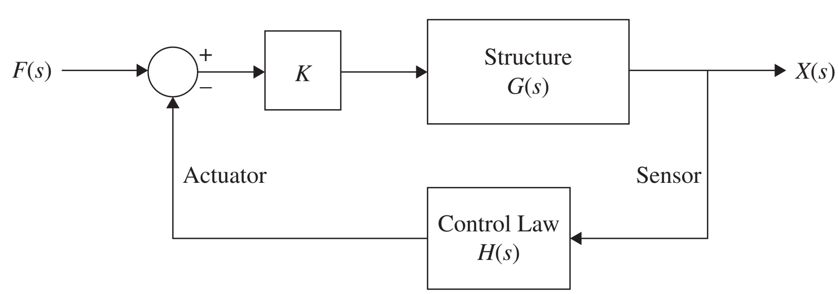
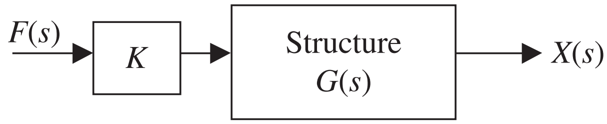


Figure 1.25 Block diagram of closed-loop system.

Figure 1.26 Block diagram of an open-loop system.



1.26, respectively. A major difference between open-loop and closed-loop control is simply that closed-loop control depends on information about the system's response and open-loop control does not.

The rule that defines how the measurement from the sensor is used to command the actuator to effect the system is called the *control law*, denoted $H(s)$ in Figure 1.25. Much of control theory focuses on clever ways to choose the control law to achieve a desired response.

A simple open-loop control law is to multiply (or amplify) the response of the system by a constant. This is referred to as *constant gain control*. The magnitude of the FRF for the system in Figure 1.25 is multiplied by the constant K, called the *gain*. The frequency domain equivalent of Figure 1.25 is

$$\frac{X(s)}{F(s)} = KG(s) = \frac{K}{(ms^2 + cs + k)} \tag{1.72}$$

where the plant is taken to be an SDOF model of structure. In the time domain, this becomes

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = Kf(t) \tag{1.73}$$

The effect of this open-loop control is simply to multiply the steady-state response by K and to increase the value of the peak response, M_p.

On the other hand, the closed-loop control, illustrated in Figure 1.25, has the equivalent frequency domain representation given by

$$\frac{X(s)}{F(s)} = \frac{KG(s)}{(1 + KG(s)H(s))} \tag{1.74}$$

If the feedback control law is taken to be one that measures both the velocity and position, multiplies them by some constant gains g_1 and g_2, respectively, and adds the result, the control law $H(s)$ is given by

$$H(s) = g_1 s + g_2 \tag{1.75}$$

As the velocity and position are the *state variables* for this system, this control law is called *full state feedback*, or PD control (for position and derivative). In this case, Equation (1.74) becomes

$$\frac{X(s)}{F(s)} = \frac{K}{ms^2 + (kg_1 + c)s + (kg_2 + k)} \tag{1.76}$$

The time domain equivalent of this equation is (obtained by using the inverse Laplace Transform) is

$$m\ddot{x}(t) + (c + kg_1)\dot{x}(t) + (k + kg_2)x(t) = Kf(t) \tag{1.77}$$

By comparing Equations (1.73) and (1.77), the versatility of closed-loop control versus open-loop, or passive, control is evident. In many cases the choice of values of K, g_1 and g_2 can be made electronically. By using a closed-loop control, the designer has the choice of three more parameters to adjust than are available in the passive case to meet the desired specifications.

On the negative side, closed-loop control can cause some difficulties. If not carefully designed, a feedback control system can cause an otherwise stable structure to have an unstable response. For instance, suppose the goal of the control law is to reduce the stiffness of the structure so that the natural frequency is lower. From examining Equation (1.77), this would require that g_2 be a negative number. Then suppose that the value of k was over-estimated and g_2 calculated accordingly. This could result in the possibility that the coefficient of $x(t)$ becomes negative, causing instability. That is, the response of Equation (1.77) would be unstable if $(k + Kg_2) < 0$. This would amount to positive feedback and is not likely to arise by design on purpose, but can happen if the original parameters are not well known. On physical grounds, instability is possible because the control system is adding energy to the structure. One of the major concerns in designing high-performance control systems is to maintain stability. This introduces another constraint on the choice of the control gains and is discussed in more detail in Chapter 7. Of course, closed-loop control is also expensive because of the sensor, actuator and electronics required to make a closed-loop system. On the other hand, closed-loop control can always result in better performance provided the appropriate hardware is available.

Feedback control uses the measured response of the system to modify and add back into the input to provide an improved response. Another approach to improving the response consists of producing a second input to the system that effectively cancels the disturbance to the system. This approach, called *feedforward control*, uses knowledge of the response of a system at a point to design a control force that when subtracted from the uncontrolled response yields a new response with desired properties, usually a response of zero. Feedforward control is most commonly used for high frequency applications and in acoustics (for noise cancellation) and is not considered here. An excellent treatment of feedforward controllers can be found in Fuller et al. (1996).

Example 1.9.1

Consider the step response of an underdamped system (Figure 1.11). Calculate the value of the damping ratio ζ in terms of the performance measures t_p and t_s. Show that it is not possible to specify all three performance measures O.S., t_p and t_s in the design of a passive system.

Solution: Rearranging the definition of settling time given in Equation (1.37) yields

$$\omega_n = \frac{3.2}{t_s \zeta}$$

The time to peak is given in Equation (1.39) for underdamped systems as

$$t_p = \frac{\pi}{\omega_n \sqrt{1 - \zeta^2}}$$

Squaring this last expression and solving for ζ yields

$$\zeta = \frac{t_p}{\sqrt{t_p^2 + a^2 t_s^2}}, \quad \text{where} \quad a = \left(\frac{\pi}{3.2}\right)$$

Thus, specifying t_p and t_s completely determines both ζ and ω_n. Since O.S. is only a function of ζ, its value is also determined once t_p and t_s are specified. Recall that

$$\zeta = \frac{c}{2\sqrt{km}}, \quad \text{and} \quad \omega_n = \sqrt{\frac{k}{m}}$$

Thus, no passive adjustment of m, c and/or k can arbitrarily assign all three performance values of O.S., t_p and t_s.

1.10 Nonlinear Vibrations

The force versus displacement plot for a spring of Figure 1.3 curves off after the deflections and/or forces become large enough. Before enough force is applied to permanently deform or break the spring, the force deflection curve becomes nonlinear and curves away from a straight line, as indicated in Figure 1.27. So rather than the linear spring relationship $f_k = kx$, a model such as $f_k = \alpha x - \beta x^3$, called a *softening spring*, might better fit the curve. This nonlinear spring behavior greatly changes the physical nature of the vibratory response and complicates the mathematical description and analysis to the point that numerical integration usually has to be employed to obtain a solution. Stability analysis of nonlinear systems also becomes more complicated.

In Figure 1.27, the force-displacement curves for three springs are shown. Notice that the linear range for the two nonlinear springs is a good approximation until about 1.8 units of displacement or 2000 units of force. If the spring is to be used beyond that range, then the linear vibration analysis of the preceding sections no longer applies.

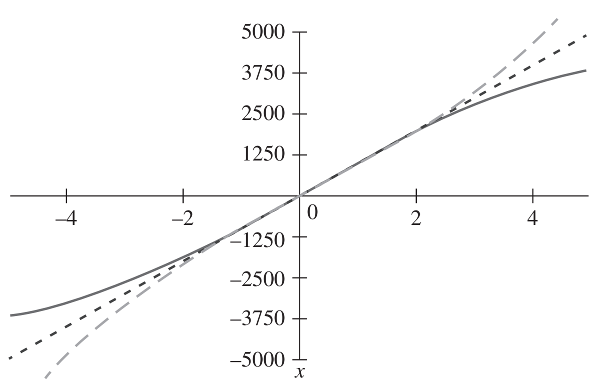


Figure 1.27 Force (vertical axis) deflection (horizontal axis) curves for three different springs in dimensionless terms, indicating their linear range. The curve $g(x) = kx$ is a linear spring (short dashed line), the curve $f(x) = kx - \beta x^3$ is called a softening spring (solid line) and the curve $h(x) = kx + \beta x^3$ is called a hardening spring (long dashed line), for the case $k = 1000$ and $\beta = 10$.

Consider then the equation of motion of a system with a nonlinear spring of the form

$$m\ddot{x}(t) + \alpha x(t) - \beta x^3(t) = 0 \tag{1.78}$$

which is subject to two initial conditions. In the linear system, there was only one equilibrium point to consider, $v(t) = x(t) = 0$. As will be shown in the following, the nonlinear system of Equation (1.78) has more than one equilibrium position. The equilibrium point of a system, or set of governing equations, may be defined best by first placing the equation of motion into *state space* form.

A general SDOF system may be written as

$$\ddot{x}(t) + f(x(t), \dot{x}(t)) = 0 \tag{1.79}$$

where the function f can take on any form, linear or nonlinear. For example, for a linear spring-mass-damper system, the function f is just $f(x, \dot{x}) = 2\zeta\omega_n\dot{x}(t) + \omega_n^2 x(t)$, which is a linear function of the state variables of position and velocity. For a nonlinear system, the function f will be some nonlinear function of the state variables. For instance, for the nonlinear spring of Equation (1.78), the function is

$$f(x, \dot{x}) = \alpha x - \beta x^3$$

The *state space* model of Equation (1.79) is written by defining the two *state variables* the position: $x_1 = x(t)$, and the velocity: $x_2 = \dot{x}(t)$. Then Equation (1.79) can be written as the first-order pair

$$\begin{aligned} \dot{x}_1(t) &= x_2(t) \\ \dot{x}_2(t) &= -f(x_1, x_2) \end{aligned} \tag{1.80}$$

This state space form of the equation of motion is used for numerical integration, in control analysis and for formally defining an equilibrium position. Define the state vector, **x**, and a nonlinear vector function **F**, as

$$\mathbf{x}(t) = \begin{bmatrix} x_1(t) \\ x_2(t) \end{bmatrix}, \quad \text{and} \quad \mathbf{F} = \begin{bmatrix} x_2(t) \\ -f(x_1, x_2) \end{bmatrix} \tag{1.81}$$

Then Equation (1.80) may be written in the simple form of a vector equation

$$\dot{\mathbf{x}} = \mathbf{F}(\mathbf{x}) \tag{1.82}$$

An *equilibrium point* of this system, denoted $\mathbf{x}_e$, is defined to be any value of the vector **x** for which $\mathbf{F}(\mathbf{x})$ is identically zero (called zero phase velocity). Thus the equilibrium point is any vector of constants, $\mathbf{x}_e$, that satisfies the relations

$$\mathbf{F}(\mathbf{x}_e) = \mathbf{0} \tag{1.83}$$

Placing the linear SDOF system into state space form then yields

$$\dot{\mathbf{x}} = \begin{bmatrix} x_2 \\ -2\zeta\omega_n x_2 - \omega_n^2 x_1 \end{bmatrix} \tag{1.84}$$

The equilibrium of a linear system is thus the solution of the vector equality

$$\begin{bmatrix} x_2 \\ -2\zeta\omega_n x_2 - \omega_n^2 x_1 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \tag{1.85}$$

which has the single solution: $x_1 = x_2 = 0$. Thus, for any linear system, the equilibrium point is a single point consisting of the origin. On the other hand, the equilibrium condition of the soft spring system of Equation (1.78) requires that

$$\begin{aligned} x_2 &= 0 \\ -\alpha x_1 + \beta x_1^3 &= 0 \end{aligned} \tag{1.86}$$

Solving for x_1 and x_2, yields the *three* equilibrium points

$$\mathbf{x}_e = \begin{bmatrix} 0 \\ 0 \end{bmatrix}, \quad \begin{bmatrix} \sqrt{\dfrac{\alpha}{\beta}} \\ 0 \end{bmatrix}, \quad \begin{bmatrix} -\sqrt{\dfrac{\alpha}{\beta}} \\ 0 \end{bmatrix} \tag{1.87}$$

In principle, the soft spring system of Equation (1.78) could oscillate around any of these equilibrium points and which one will depend on the initial conditions (and the magnitude of any applied forcing function). Each of these equilibrium points may also have a different stability property.

Note that placing an n^{th} order, ordinary differential equation into n first-order, ordinary differential equations can always be done (Boyce and DiPrima, 2012).

This existence of multiple equilibrium points also complicates the notion of stability introduced in Section 1.7. In particular, solutions near each equilibrium point could potentially have different stability behavior. Since the initial conditions may determine which equilibrium the solution centers around, the behavior of a nonlinear system will depend on the initial conditions. In contrast, for a linear system with fixed parameters, the solution form is the same regardless of the initial conditions. This represents another important difference to consider when working with nonlinear components.

Example 1.10.1
Sliding or Coulomb friction applied to a spring-mass system results in an equation of motion of the form

$$m\ddot{x}(t) + \mu mg\,\text{sgn}(\dot{x}) + kx(t) = 0$$

where m and k are the mass and stiffness values, μ is the coefficient of sliding friction and the function sgn denotes the signum function which is zero when the velocity is zero and is 1 the rest of the time, having the same sign of the velocity. This force reflects the fact that dry friction is always opposite to the direction of motion. Discuss the equilibrium positions of this nonlinear system.

Solution: First put the equation of motion into state space form resulting in

$$\dot{\mathbf{x}} = \begin{bmatrix} x(t) \\ \dot{x}(t) \end{bmatrix} = \frac{d}{dt}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} x_2 \\ -\mu\text{sgn}(x_2) - kx_1 \end{bmatrix}$$

The equilibrium is thus defined as

$$\begin{bmatrix} x_2 \\ -\mu\text{sgn}(x_2) - kx_1 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

This requires $x_2 = 0$ and $-\mu \text{ sgn } (x_2) - kx_1 = 0$. This last expression is a static condition, which is also satisfied for any value of x_1 that satisfies

$$-\frac{\mu_s mg}{k} < x_1 < \frac{\mu_s mg}{k}$$

Thus the equilibrium is not a series of points but rather a region of values. Basically the solution can only move out of this region if the spring force is large enough to overcome static friction. Hence, the response will end up in this equilibrium position depending on the initial condition, as typical of a nonlinear system.

1.11 Computing and Simulation in MATLAB™

Modern computer codes such as MATLAB™ make the visualization and computation of vibration problems available without much programming effort. Such codes can help enhance understanding through plotting responses, can help find solutions to complex problems lacking closed form solutions through numerical integration and can often help with symbolic computations. Plotting certain parametric relations or plotting solutions can often aid in visualizing the nature of relationships or the effect of parameter changes on the response. Most of the plots used in this text are constructed from simple MATLAB commands, as the following examples illustrate. If you are familiar with MATLAB, you may wish to skip this section.

MATLAB is a high-level code, with many built-in commands for numerical integration (simulation), control design, performing matrix computations, symbolic manipulation, etc. MATLAB has two areas to enter information. The first is the command window, which is an active area where the entered command is compiled as it is entered. Using the command window is somewhat like a calculator. The second area is called an m-file, which is a series of commands that are saved then called from the command window to execute. All of the plots in the figures in this chapter can be reproduced using these simple commands.

Example 1.11.1

Plot the free response of the underdamped system to initial conditions $x_0 = 0.01$ m, $v_0 = 0$ for values of $m = 100$ kg, $c = 25$ kg/s and $k = 1000$ N/m, using MATLAB and Equation (1.13).

Solution: To enter numbers in the command window, just type a symbol and use an equal sign after the blinking cursor. The following entries in the command window will produce the plot of Figure 1.28. Note that the prompt symbol "≫" is provided by MATLAB and the information following it is code typed in by the user. The symbol % is used to indicate comments so that anything following this symbol is ignored by the code and is included to help explain the situation. A semicolon typed after a command suppresses the command from displaying the output. MATLAB uses matrices and vectors so that numbers can be entered and computed in arrays. Thus, there are two types of multiplication. The notation `a*b` is a vector operation demanding that the number of rows of `a` be equal to

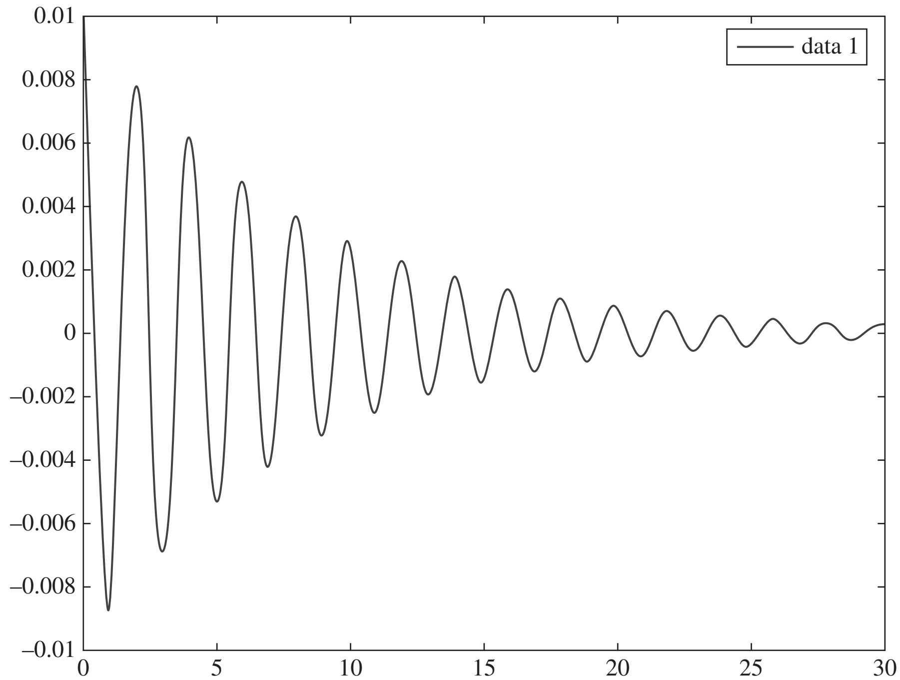


Figure 1.28 The response of an underdamped system ($m = 100$ kg, $c = 25$ kg/s and $k = 1000$ N/m) to the initial conditions $x_0 = 0.01$ m, $v_0 = 0$ plotted using MATLAB.

the number of columns of b. The product a*b, on the other hand, multiplies each element of a times the corresponding element in b.

```
>> clear % used to make sure no previous values are stored
>> %assign the initial conditions, mass, damping and stiffness
>> x0=0.01;v0=0.0;m=100;c=25;k=1000;
>> %compute omega and zeta, display zeta to check if under-
damped
>> wn=sqrt(k/m);z=c/(2*sqrt(k*m))

z =

  0.0395

>> %compute the damped natural frequency
>> wd=wn*sqrt(1-z^2);
>> t=(0:0.01:15*(2*pi/wn));%set the values of time from 0 in
increments of 0.01 up to 15 periods
>> x=exp(-z*wn*t).*(x0*cos(wd*t)+((v0+z*wn*x0)/wd)*sin(wd*t));
% computes x(t)
>> plot(t,x)%generates a plot of x(t) vs t
```

The MATLAB code used in this example is not the most efficient way to plot the response and does not show the detail of labeling the axis, etc. but is given as a quick introduction.

The next example illustrates the use of m-files in a numerical simulation. Instead of plotting the closed-form solution given in Equation (1.13), the equation of motion can be numerically integrated using the ode command in MATLAB. The ode45 command uses a fifth-order Runge-Kutta, automated time step method for numerically integrating the equation of motion (Pratap, 2002).

In order to use numerical integration, the equations of motion must first be placed in first order, or state space form, as in Equation (1.84). This state space form is used in MATLAB to enter the equations of motion.

Vectors are entered in MATLAB by using square brackets, spaces and semicolons. Spaces are used to separate columns and semicolons are used to separate rows. So that a row vector is entered by typing

```
>> u = [1 -1 2]
```

which returns the row

```
u =
    1  -1  2
```

and a column vector is entered by typing

```
>> u = [1; -1; 2]
```

which returns the column

```
u =
    1
    -1
    2
```

To create a list of formulas in an m-file, choose "New" from the file menu and select "m-file". This will display a text editor window, in which you can enter commands. The following example illustrates the creation of an m-file and how to call it from the command window to numerically integrate the equation of motion given in Example 1.11.1.

Example 1.11.2

Numerically integrate and plot the free response of the underdamped system to initial conditions $x_0 = 0.01$ m, $v_0 = 0$ for values of $m = 100$ kg, $c = 25$ kg/s and $k = 1000$ N/m, using MATLAB and Equation (1.13).

Solution: First create an m-file containing the equation of motion to be integrated and save it. This is done by selecting "New" and "M-File" from the File menu in MATLAB, then typing:

```
----------------------
function xdot=f2(t,x)
c=25; k = 1000; m = 100;
% set up a column vector with the state Equations
xdot=[x(2); -(c/m)*x(2)-(k/m)*x(1)];
----------------------
```

This file is now saved with the name `f2.m`. Note that the name of the file must agree with the name following the equal sign in the first line of the file. Now open the command window and enter the following:

```
>> ts=[0 30]; % this enters the initial and final time
>>x0 =[0.01 0]; % this enters the initial conditions
>>[t, x]=ode45('f2',ts,x0);
>>plot(t,x(:,1))
```

The third line of code calls the Runge-Kutta program ode45 and the state equations to be integrated contained in the file named `f2.m`. The last line plots the simulation of the first state variable $x_1(t)$, which is the displacement, denoted `x(:,1)` in MATLAB. The plot is given in Figure 1.29.

Note that the plots of Figures 1.28 and 1.29 look the same. However, Figure 1.28 was obtained by simply plotting the analytical solution, whereas the plot of Figure 1.29 was obtained by numerically integrating the equation of motion. The numerical approach can be used successfully to obtain the solution of a nonlinear state equation, such as Equation (1.78), just as easily.

The forced response can also be computed using numerical simulation and this is often more convenient then working through an analytical solution when the forcing functions are discontinuous or not made up of simple functions. Again the equations of motion (this time with the forcing function) must be placed in state space form. The equation of motion for damped system with general applied force is

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = F(t)$$

In state space form, this expression becomes

$$\begin{bmatrix} \dot{x}_1(t) \\ \dot{x}_2(t) \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -\dfrac{k}{m} & -\dfrac{c}{m} \end{bmatrix} \begin{bmatrix} x_1(t) \\ x_2(t) \end{bmatrix} + \begin{bmatrix} 0 \\ f(t) \end{bmatrix}, \begin{bmatrix} x_1(0) \\ x_2(0) \end{bmatrix} = \begin{bmatrix} x_0 \\ v_0 \end{bmatrix} \tag{1.88}$$

where $f(t) = F(t)/m$ and $F(t)$ is any function that can be integrated. The following example illustrates the procedure in MATLAB.

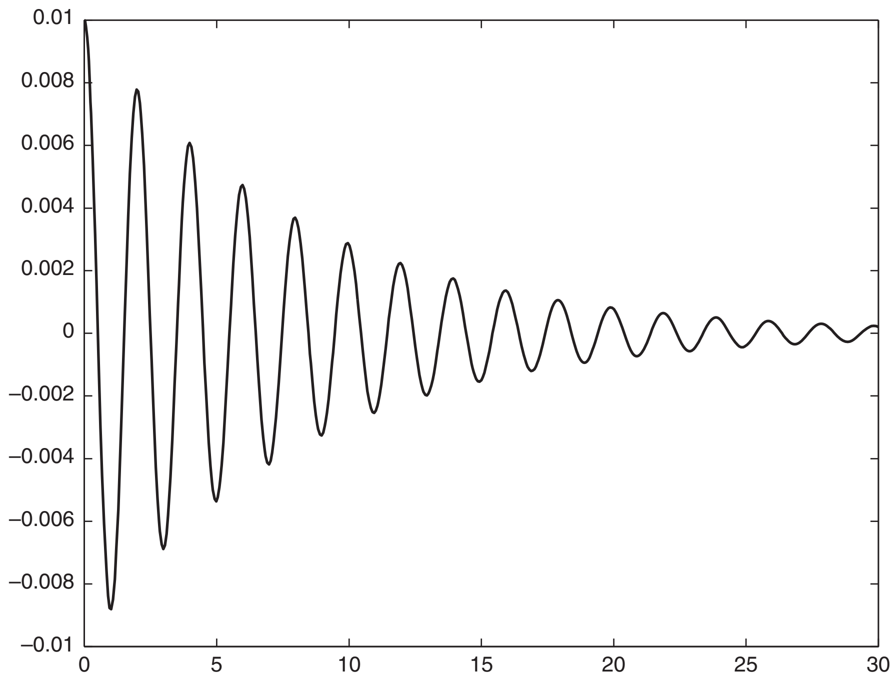


Figure 1.29 A plot of the numerical integration of the underdamped system of Example 1.10.1 resulting from the MATLAB code given in Example 1.10.2.

Example 1.11.3
Use MATLAB to compute and plot the response of the following system

$$100\ddot{x}(t) + 10\dot{x}(t) + 500x(t) = 150\cos 5t,\ x_0 = 0.01, v_0 = 0.5.$$

Solution: The Matlab code for computing these plots is given. First an m-file is created with the equation of motion given in first-order form.

```
--------------------------------------------
function v=f(t,x)
m=100; k=500; c=10; Fo=150; w=5;
v=[x(2); x(1)*-k/m+x(2)*-c/m + Fo/m*cos(w*t)];
--------------------------------------------
```

Then the following is typed in the command window:

```
>>clear all
>>xo=[0.01; 0.5]; %enters the initial conditions
>>ts=[0 40]; %enters the initial and final times
>>[t,x]=ode45('f',ts,xo); %calls the dynamics and integrates
>>plot(t,x(:,1)) %plots the result
```

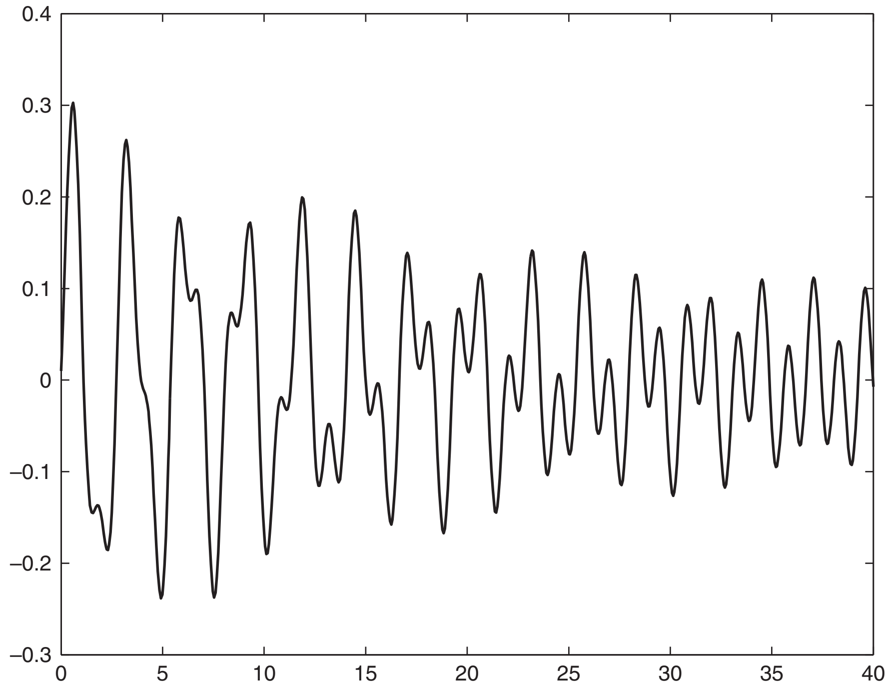


Figure 1.30 A plot of the numerical integration of the damped forced system resulting from the MATLAB code given in Example 11.1.3

This code produces the plot given in Figure 1.30. Note that the influence of the transient dynamics dies off due to the damping after about 20 sec.

Such numerical integration methods can also be used to simulate the nonlinear systems discussed in the previous section. Use of high-level codes in vibration analysis such as MATLAB is now commonplace and has changed the way vibration quantities are computed. More detailed codes for vibration analysis can be found in Inman (2014). In addition, there are many books written on using MATLAB (Pratap, 2002) as well as available online help.

Chapter Notes

This chapter attempts to provide a review of introductory vibrations and to expand the discipline of vibration analysis and design, by intertwining elementary vibration topics with the disciplines of design, control, stability and testing. An early attempt to relate vibrations and control at an introductory level was written by Vernon (1967). More recent attempts are by Meirovitch (1985, 1990) and Inman (1989), which is the predecessor or first edition of this text. Leipholz and Abdel-Rohman (1986) give the civil engineering approach to structural control. The latest attempt to combine vibration and control is by Preumont (2011) and

Benaroya (2004), who also provides excellent treatment of uncertainty in vibrations. Pruemont and Seto (2008) presents control of structures slanted towards civil structures. Moheimani *et al*. (2003) focuses on vibration control of flexible structures. The information contained in Sections 1.2, 1.3 and part of 1.4 can be found in every introductory vibrations text, such as my own (Inman, 2014) and such as the standards by Thomson and Dahleh (1993), Rao (2012) and Meirovitch (1986). A complete summary of most vibration related topics can be found in Braun et al. (2002) and Harris and Piersol (2002).

A good reference for vibration measurement is McConnell (1995). The reader is encouraged to consult a basic text on control, such as the older text of Melsa and Schultz (1969), which contains some topics dropped in modern texts, or Kuo and Golnaraghi (2009), which contains more modern topics integrated with MATLAB. These two texts also give background on specifications and transfer functions given in Sections 1.4 and 1.5, as well as feedback control discussed in Section 1.9. A complete discussion of plant identification, as presented in Section 1.7, can be found in Melsa and Schultz (1969). The book by Neubert (1987) was issued by the Naval Sea Systems Command and provides a treatise on impedance methods (Section 1.6).

There are many introductory controls texts with various slants and focus on sub-topics, as addressed by Davison et al. (2007). The excellent text by Fuller et al. (1996) examines controlling high frequency vibration. Control is introduced here, not as a discipline by itself, but rather as a design technique for vibration engineers. A standard reference on stability is Hahn (1967), which provided the basic ideas of Section 1.8. The topic of flutter and self-excited vibrations is discussed in Den Hartog (1985). Nice introductions to nonlinear vibration can be found in Virgin (2000), Worden and Tomlinson (2001) and the standards by Nayfeh and Mook (1978) and Nayfeh and Balachandra (1995). While there are many excellent texts introducing how to use MATLAB, the website of the MathWorks contains excellent tutorials for using their code.

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Problems

1.1 Derive the solution of $m\ddot{x} + kx = 0$ and sketch your result (for at least 2 periods) for the case.

1.2 Solve $m\ddot{x} - kx = 0$ for $x_0 = 1$, $v_0 = 0$, for $x(t)$ and sketch the solution.

1.3 Derive the solutions given in the text for $\zeta > 1$, $\zeta = 1$ and $0 < \zeta < 1$ with x_0 and v_0 as the initial conditions (i.e. derive Equations 1.14 to 1.16 and corresponding constants).

1.4 Solve $\ddot{x} - \dot{x} + x = 0$ with $x_0 = 1$ and $v_0 = 0$ for $x(t)$, and sketch the solution.

1.5 Prove that $\zeta = 1$ corresponds to the smallest value of c such that no oscillation occurs. (*Hint:* Let $\lambda = -b$, b be a positive real number, and differentiate the characteristic equation.)

1.6 Consider a small spring about 30 mm long, welded to a stationary table (ground) so that it is fixed at the point of contact, with a 12-mm bolt welded to the other end, which is free to move. The mass of this system is about 49.2×10^{-3} kg. The spring stiffness is $k = 857.8$ N/m. Calculate the natural frequency, period and the maximum amplitude of the response if the spring is initially deflected 10 mm.

1.7 A simple model of a vehicle wheel, tire and suspension assembly is just the basic spring-mass equation of motion. If its mass is measured to be about 30 kilograms (kg) and its frequency of oscillation is observed to be 10 Hz, what is the approximate stiffness of the suspension?

1.8 Calculate t_p, OS, T_d, M_p and BW for a system described by

$$2\ddot{x} + 0.8\dot{x} + 8x = f(t)$$

where $f(t)$ is either a unit step function or a sinusoidal as required.

1.9 Derive an expression for the forced response of the undamped system

$$m\ddot{x}(t) + kx(t) = F_0 \sin \omega t,\ x(0) = x_0,\ \dot{x}(0) = v_0$$

to a sinusoidal input and nonzero initial conditions. Compare your result to Equation (1.25) with $\zeta = 0$.

1.10 Compute the total response to the system

$$4\ddot{x}(t) + 16x(t) = 8 \sin 3t,\ x_0 = 1 \text{ mm},\ v_0 = 2 \text{ mm/s}$$

1.11 Calculate the maximum value of the peak response (magnification factor) for the system of Figure 1.10 with $\zeta = 1/\sqrt{2}$.

1.12 Derive Equation (1.26).

1.13 Calculate the impulse response function for a critically damped system.

1.14 Solve for the forced response of an SDOF system to a harmonic excitation with $\zeta = 1.1$ and $\omega_n^2 = 4$. Plot the magnitude of the steady state response versus the driving frequency. For what value of ω_n is the response a maximum (resonance)?

1.15 Consider the forced vibration of a mass m connected to a spring of stiffness 2000 N/m being driven by a 20-N harmonic force at 10 Hz (20π rad/s). The maximum amplitude of vibration is measured to be 0.1 m and the motion is assumed to have started from rest ($x_0 = v_0 = 0$). Calculate the mass of the system.

1.16 Consider a spring-mass-damper system with $m = 100$ kg, $c = 20$ kg/s and $k = 2000$ N/m, with an impulsive force applied to it of 1000 N for 0.01 s. Compute the resulting response.

1.17 Calculate the compliance transfer function for the system described by the differential equation

$$a\ddddot{x} + b\dddot{x} + c\ddot{x} + d\dot{x} + ex = f(t)$$

where $f(t)$ is the input and $x(t)$ is a displacement. Also calculate the FRF for this system.

1.18 Use the frequency response approach to compute the amplitude of the particular solution for the undamped system of the form

$$m\ddot{x}(t) + kx(t) = F_0 \cos \omega t$$

1.19 Derive Equation (1.66).

1.20 Plot (using a computer) the unit step response of an SDOF system with $\omega_n^2 = 4$, $k = 1$ for several values of the damping ratio ($\zeta = 0.01, 0.1, 0.5$ and 1.0).

1.21 Plot ω_p/ω_n versus ζ and ω_d/ω_n versus ζ, and comment on the difference as a function of ζ.

1.22 For the system of Problem 1.8, construct the Bode plots for (a) the inertance transfer function, (b) the mobility transfer function, (c) the compliance transfer function, and (d) the Nyquist diagram for the compliance transfer function.

1.23 The free response of the damped SDOF system with a mass of 2 kg is observed to be underdamped. A static deflection test is performed and the stiffness is determined as 1.5×10^3 N/m. The displacements at two successive maximum amplitudes t_1 and t_2 are measured to be 9 and 1 mm, respectively. Calculate the damping coefficient.

1.24 Discuss the stability of the following system

$$2\ddot{x}(t) - 3\dot{x}(t) + 8x(t) = -3\dot{x}(t) + \sin 2t$$

1.25 An inverted pendulum has equation of motion

$$ml^2\ddot{\theta} + \left(\frac{kl^2}{2}\sin\theta\right)\cos\theta - mgl\sin\theta = 0$$

Linearize the equation and discuss the stability of the result.

1.26 Using the system of Problem 1.8, refer to Equation (1.77) and choose the gains K, g_1 and g_2 so that the resulting closed-loop system has a 5% overshoot and a settling time of less than 10.

1.27 Calculate an allowable range of values for the gains K, g_1 and g_2 for the system of Problem 1.8, such that the closed loop system is stable and the formulas overshoot and peak time of an underdamped system is valid.

1.28 Compute a feedback law with full state feedback (of the form given in Equation 1.77) that stabilizes (makes it asymptotically stable) the following system: $4\ddot{x}(t) + 16x(t) = 0$ and causes the closed loop settling time to be 1 second.

1.29 Compute the equilibrium positions of the pendulum equation

$$ml^2\ddot{\theta}(t) + mgl\sin\theta(t) = 0$$

1.30 Compute the equilibrium points for a system with Coulomb damping given by

$$m\ddot{x}(t) + \mu mg\,\mathrm{sgn}(\dot{x}) + kx(t) = 0$$

where μ is the coefficient of friction and g denotes the acceleration due to gravity. Here sgn denotes the signum function takes on a plus, minus or zero value, depending on whether the argument is plus, minus or zero.

1.31 Compute the equilibrium points for the system defined by

$$\ddot{x} + \beta\dot{x} + x + x^2 = 0$$

1.32 The linearized version of the pendulum equation is given by $\ddot{\theta}(t) + \frac{g}{l}\theta(t) = 0$. Use numerical integration to plot the solution of the nonlinear equation of Problem 1.29 and this linearized version for the case that

$$g/l = 0.01,\ \theta(0) = 0.1 \text{ rad},\ \dot{\theta}(0) = 0.1 \text{ rad/s}$$

Compare your two simulations.