

Chapter 3

Multi-Degree-of-Freedom Systems

For an accurate description of the displacement configuration of a structure subjected to a dynamic loading, often displacements along more than one coordinate are necessary. Such a system is known as a *multi-degree-of-freedom system*.

We begin by discussing the formulation of the equations of motion for multi-degree-of-freedom (MDOF) systems.

3.1 Formulation of the Equation of Motion for an MDOF System

We begin with an example of a MDOF system with three degrees-of-freedom, as shown in Figure 3.1. Note that the displacements are time-varying, but are simply designated as x rather than $x(t)$ for succinctness.

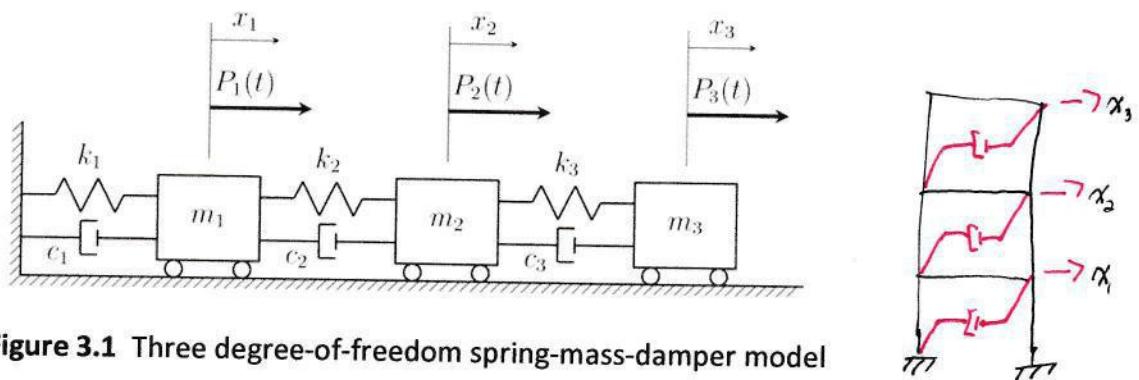
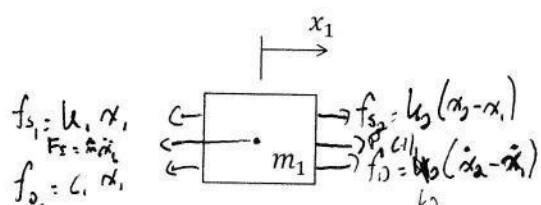


Figure 3.1 Three degree-of-freedom spring-mass-damper model

Example 3.1 Derive the equations of motion for the 3DOF system shown in Figure 3.1.

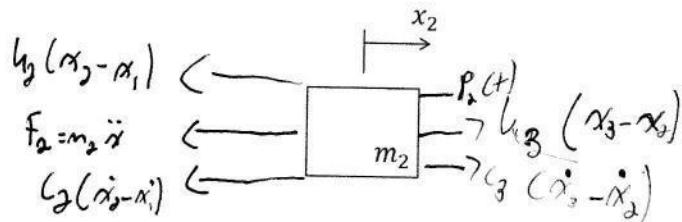
Solution: First, consider the free body diagram of m_1 .



The equation of motion is

$$m_1 \ddot{x}_1 + k_1 x_1 + c_1 \dot{x}_1 + k_2 (x_2 - x_1) - c_2 (\dot{x}_2 - \dot{x}_1) = P_1(t) \quad (3.1)$$

Next, consider the free body diagram of m_2 .

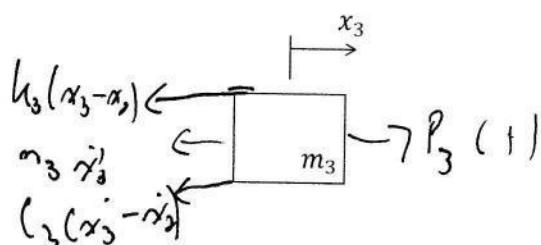


The equation of motion is

$$m_2 \ddot{x}_2 + k_2 (x_2 - x_1) + c_2 (\dot{x}_2 - \dot{x}_1) - k_3 (x_3 - x_2) - c_3 (x_3 - x_2) = P_2 \quad (3.2)$$

$$m_2 \ddot{x}_2 + [c_2 + c_3] \dot{x}_2 + [k_2 + k_3] x_2 - c_2 \dot{x}_1 - k_2 x_1 - k_3 x_3 - c_3 x_3 = P_2$$

Finally, consider the free body diagram of m_3 .



The equation of motion is

$$m_3 \ddot{x}_3 + c_3 x_3 - c_3 x_2 + k_3 x_3 - k_2 x_2 = P_3 \quad (3.3)$$

Equations 3.1 through 3.3 can be written in matrix form as

$$\begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_3 \end{bmatrix} + \begin{bmatrix} c_1 + k_2 & -k_2 & 0 \\ -k_2 & k_2 + k_3 & -k_3 \\ 0 & -k_3 & k_3 \end{bmatrix} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} P_1(t) \\ P_2(t) \\ P_3(t) \end{bmatrix} \quad (3.4)$$

where M is the *mass matrix*, C is the *damping matrix*, and K is the *stiffness matrix*.

$$M \ddot{x} + C \dot{x} + Kx = P(t)$$

We can also formulate MDOF equations of motion using an analytical mechanics approach. Lagrange's Equation is a powerful method but will not be covered in this course.

3.2 Solution for a 2-DOF System

We will now consider the solution for a 2-DOF system. We begin by finding the equation of motion for a 2-DOF system with no damping.

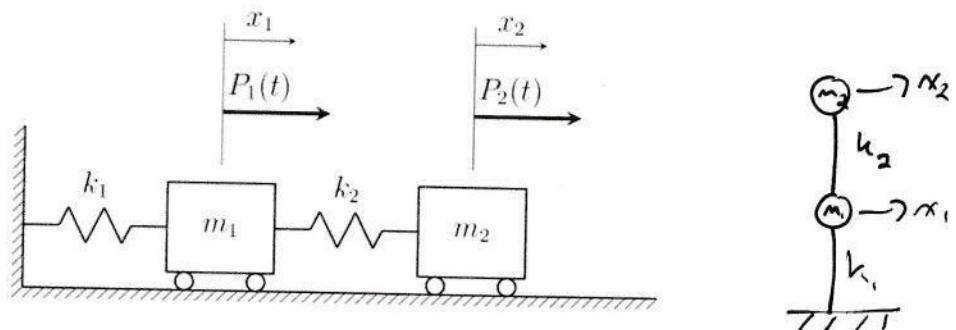
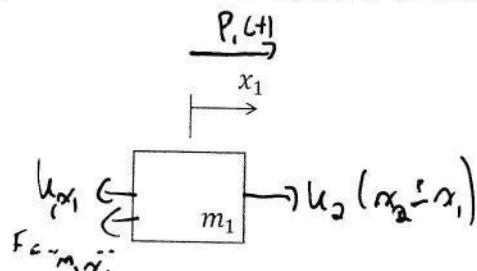


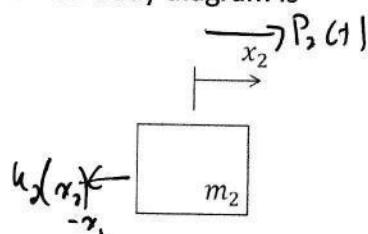
Figure 3.2 Two degree-of-freedom spring-mass model

Using Newtonian Mechanics, we can derive the equation of motion. For mass m_1 , the free body diagram is:

using Blueplan



Similarly, for the mass m_2 , the free body diagram is



The equations of motion are

$$P_1(t) = k_1 x_1 - k_2(x_2 - x_1) + m_1 \ddot{x}_1, \quad m_1 \ddot{x}_1 + (k_1 + k_2)x_1 - k_2 x_2 = P_1 \quad (3.6a)$$

$$P_2(t) = k_2 x_2 + m_2 \ddot{x}_2 \quad (3.6b)$$

which can be written in matrix form as

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} P_1(t) \\ P_2(t) \end{bmatrix} \quad (3.7)$$

$$M \ddot{x} + Kx = P$$

Several observations can be made from this result.

- Equation 3.7 is uncoupled in the mass terms, but coupled in the stiffness terms.
- Coupling is not an inherent property of the system; rather, it is a consequence of the coordinate system used.
- Both the mass and stiffness matrices are symmetric.

We will now turn to solving this system. Assuming the solution is of the form

$$\begin{bmatrix} X_1 \\ X_2 \end{bmatrix} = \begin{bmatrix} X_1 \\ X_2 \end{bmatrix} \sin(\omega_n t + \alpha) , \quad (3.8)$$

where X_1 and X_2 are the amplitudes of x_1 and x_2 , respectively. As before, ω_n is the circular natural frequency, and α is the phase angle. Substituting Equation 3.8 into Equations 3.6 gives,

$$-m_1 \omega_n^2 X_1 \sin(\omega_n t + \alpha) + (k_1 + k_2) X_1 \sin(\omega_n t + \alpha) - k_2 X_2 \sin(\omega_n t + \alpha) = P_1(t) \quad (3.9a)$$

$$-m_2 \omega_n^2 X_2 \sin(\omega_n t + \alpha) - k_2 X_1 \sin(\omega_n t + \alpha) + k_2 X_2 \sin(\omega_n t + \alpha) = P_2(t) \quad (3.9b)$$

3.2.1 Free Vibration Response

For the free vibration response ($P_1(t) = P_2(t) = 0$), the solution becomes

$$\text{Homogeneous system} \rightarrow [-m_1 \omega_n^2 X_1 + (k_1 + k_2) X_1 - k_2 X_2] \sin(\omega_n t + \alpha) = 0 \quad (3.10a)$$

$$[-m_2 \omega_n^2 X_2 - k_2 X_1 + k_2 X_2] \sin(\omega_n t + \alpha) = 0 \quad (3.10b)$$

For a nontrivial solution, X_1 and $X_2 \neq 0$ at some time

$$\therefore \text{Non invertible, determinate} = 0 . \begin{bmatrix} (k_1 + k_2) - m_1 \omega_n^2 & -k_2 \\ -k_2 & k_2 - m_2 \omega_n^2 \end{bmatrix} \begin{bmatrix} X_1 \\ X_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \quad (3.11)$$

Therefore,

$$\boxed{\begin{vmatrix} A & B \\ C & D \end{vmatrix}} \quad AD - BC = 0 \quad (3.12)$$

Expanding the determinant gives

$$\omega_n^4 - \left(\frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2} \right) \omega_n^2 + \frac{k_1 k_2}{m_1 m_2} = 0 \quad (3.13)$$

Equation 3.13 is known as the characteristic equation of the system. The roots are

$$\omega_n^2 = \frac{1}{2} \left[\left(\frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2} \right) \pm \sqrt{\left(\left(\frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2} \right)^2 - 4 \frac{k_1 k_2}{m_1 m_2} \right)} \right] \quad (3.14)$$

The roots $\omega_{n,1}$ and $\omega_{n,2}$ are real and positive. The lower of $\omega_{n,1}$ and $\omega_{n,2}$ is known as the first or fundamental circular natural frequency of the system.

\therefore has 2 undamped natural frequencies, can now exactly determine amplitudes

From Equation 3.14, there are two roots. Corresponding to each value of ω_n^2 , we can determine a ratio of X_2/X_1 in Equation 3.11. However, it is not possible to determine X_1 and X_2 exactly, since Equation 3.11 is homogeneous. That is, if $X = [X_1, X_2]$ is a solution, then $\alpha[X_1, X_2]$ is also a solution, where α is any scalar.

Example 3.2 For $k_1 = k$, $k_2 = 3k/4$, $m_1 = m$, and $m_2 = m/2$, determine the circular natural frequencies and the modes of vibration for the 2-DOF system.

Solution: From Equation 3.14, the circular natural frequencies are computed from

$$\begin{aligned}\omega_n^2 &= \frac{1}{2} \left[\left(\frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2} \right) \pm \sqrt{\left(\left(\frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2} \right)^2 - 4 \frac{k_1 k_2}{m_1 m_2} \right)} \right] \\ &= \frac{1}{2} \frac{k}{m} \left[3.25 \pm \sqrt{(3.25)^2 - 6} \right] \\ &= \frac{1}{2} \frac{k}{m} (3.25 \pm \sqrt{4.563})\end{aligned}\quad (3.15)$$

Therefore, the first and second circular natural frequencies are

$$\omega_{n,1} = 0.746 \sqrt{k/m} \text{ rad/s} \quad (3.16) \qquad \omega_{n,2} = 1.64 \sqrt{k/m} \text{ rad/s} \quad (3.17)$$

Recall, $[k_1 + k_2 - m_1 \omega^2] X_1 - k_2 X_2 = 0$

$$-k_2 X_1 + (k_2 - m_2 \omega^2) X_2 = 0 \quad (3.18a)$$

$$(3.18b)$$

Substituting $\omega_{n,1} = 0.746 \sqrt{k/m}$ along with the other relevant quantities gives

$$\beta_1 = \frac{X_2}{X_1} = 1.591 \quad (3.19)$$

Similarly, substituting $\omega_{n,2} = 1.64 \sqrt{k/m}$ into Equation 3.18

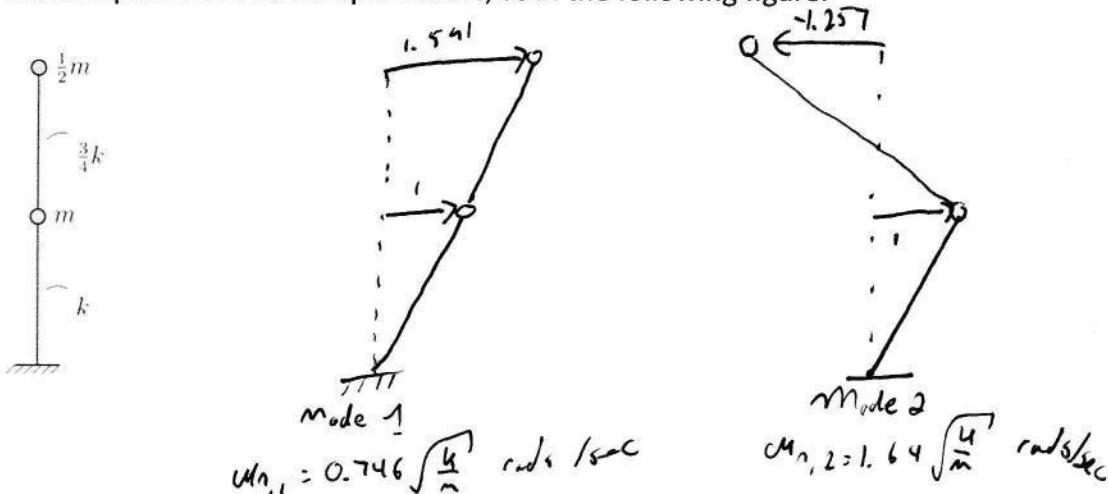
$$\beta_2 = \frac{X_2}{X_1} = -1.257 \quad (3.20)$$

We now have the mode shape vector for the first and second modes,

$$\underbrace{\phi_1 = \begin{bmatrix} X_1 \\ X_2 \end{bmatrix}}_{\text{Corresponds to } \omega_{n,1}} = \begin{bmatrix} 1 \\ 1.591 \end{bmatrix} \quad (3.21a)$$

$$\underbrace{\phi_2 = \begin{bmatrix} X_1 \\ X_2 \end{bmatrix}}_{\text{Corresponds to } \omega_{n,2}} = \begin{bmatrix} 1 \\ -1.257 \end{bmatrix} \quad (3.21b)$$

We can plot the mode shape vectors, as in the following figure.



We can now write the solution for the 2-DOF system as

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = A_1 \phi_1 \sin(\omega_{n,1} t + \alpha_1) + A_2 \phi_2 \sin(\omega_{n,2} t + \alpha_2) \quad (3.22)$$

where A_1 , A_2 , α_1 , and α_2 are constants which can be determined from initial conditions. If $x_1 = 1$, then Equation 3.22 can be rewritten as

$$x_1 = A_1 \sin(\omega_{n,1} t + \alpha_1) + A_2 \sin(\omega_{n,2} t + \alpha_2) \quad (3.23a)$$

$$x_2 = A_2 \beta_1 \sin(\omega_{n,1} t + \alpha_1) + A_2 \beta_2 \sin(\omega_{n,2} t + \alpha_2) \quad (3.23b)$$

or in matrix form

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = A_1 \begin{bmatrix} 1 \\ \beta_1 \end{bmatrix} \sin(\omega_{n,1} t + \alpha_1) + A_2 \begin{bmatrix} 1 \\ \beta_2 \end{bmatrix} \sin(\omega_{n,2} t + \alpha_2) \quad (3.24)$$

We can write Equation 3.24 in a slightly different form.

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = a_1 \begin{Bmatrix} 1 \\ \beta_1 \end{Bmatrix} \cos \omega_{n,1} t + b_1 \begin{Bmatrix} 1 \\ \beta_2 \end{Bmatrix} \sin \omega_{n,2} t + b_1 \begin{Bmatrix} 1 \\ \beta_1 \end{Bmatrix} \sin \omega_{n,1} t + a_2 \begin{Bmatrix} 1 \\ \beta_2 \end{Bmatrix} \cos \omega_{n,2} t \quad (3.25)$$

where $A_1 \sin \alpha_1 = a_1$, $A_1 \cos \alpha_1 = b_1$, $A_2 \sin \alpha_2 = a_2$, and $A_2 \cos \alpha_2 = b_2$ are constants. Equation 3.25 can be rewritten as

$$x = \sum_{i=1}^2 (a_i \phi_i \cos \omega_{n,i} t + b_i \phi_i \sin \omega_{n,i} t) = \sum_{i=1}^2 \phi_i A_i \sin(\omega_{n,i} t + \alpha_i) \quad (3.26)$$

where

$$\phi_i = \begin{Bmatrix} 1 \\ \beta_i \end{Bmatrix} \quad (3.27)$$

and the constants a_i and b_i can be determined from initial conditions.

- The natural modes and frequencies are intrinsic properties of the system and depend only on the mass and stiffness.
- The modes of a system are the simplest possible cases of undamped free vibration response – one exactly described by simple harmonic motion.
- When an undamped system is vibrating in one of its natural modes, all DOFs are oscillating in harmonic motion with the same natural frequency.
- As we can see from the results in Example 3.2, the general response of an MDOF system is given by a combination of the natural modes. This is a very important observation that will become clear later.

Before proceeding further, let's investigate some important properties of the mass matrix, \mathbf{M} , and the stiffness matrix, \mathbf{K} .

Note :

- \mathbf{m} and \mathbf{k} are symmetric: $\therefore \mathbf{M}^T = \mathbf{M}$, $\mathbf{k}^T = \mathbf{k}$

- for system w/o rigid body modes, \mathbf{m} & \mathbf{k} are positive, definite matrices

$$V = \frac{1}{2} \mathbf{x}^T \mathbf{K} \mathbf{x} > 0 \quad (3.28a)$$

$$\text{If system able to store potential energy.}$$

$$T = \frac{1}{2} \dot{\mathbf{x}}^T \mathbf{M} \dot{\mathbf{x}} > 0 \quad (3.28b)$$

$\leftarrow \dot{\mathbf{x}}^T \mathbf{M} \dot{\mathbf{x}} = \text{kinetic energy}$

If system able to store kinetic energy

3.3 Modes of the System

3.3.1 Background on Eigenvalues and Eigenvectors

- Eigenvalues and eigenvectors play a central role in structural dynamics problems. We discuss the standard eigen problem of the form:

$$\mathbf{A}\mathbf{x} = \lambda\mathbf{x}, \quad \mathbf{A} \text{ is vector, } \lambda \text{ is scalar} \quad (3.29)$$

where, \mathbf{x} is a vector of length n and \mathbf{A} is a matrix of size $n \times n$. λ is some scalar.

- Let's consider an example¹. Let:

$$\mathbf{A} = \begin{bmatrix} 3 & -2 \\ 1 & 0 \end{bmatrix} \quad \mathbf{u} = \begin{bmatrix} -1 \\ 1 \end{bmatrix} \quad \mathbf{v} = \begin{bmatrix} 2 \\ 1 \end{bmatrix}$$

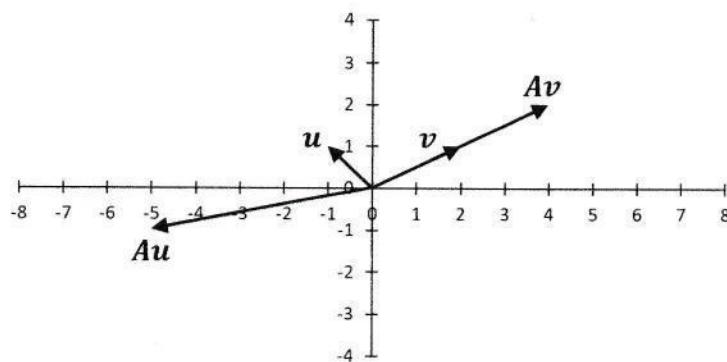
The products:

$$\mathbf{A}\mathbf{u} = \begin{bmatrix} -5 \\ -1 \end{bmatrix}, \quad \mathbf{A}\mathbf{v} = \begin{bmatrix} 4 \\ 2 \end{bmatrix}. \quad \mathbf{v} \text{ is eigenvector, } \mathbf{u} \text{ is not, w/ } \lambda = 2$$

- Clearly, the product $\mathbf{A}\mathbf{v}$ results in a scalar multiple (of 2) of the original vector (see the figure), \mathbf{v} , while the product $\mathbf{A}\mathbf{u}$ does not. In this case \mathbf{v} is an *eigenvector* of \mathbf{A} , and the corresponding *eigenvalue* is 2.

¹Lay, D. (1994) *Linear Algebra and its Applications*, Addison-Wesley, USA.

Def.: An eigenvector of a matrix A is a non-zero vector x such that the product Ax results in a scalar multiple λ of x , where λ is known as the eigenvalue.



- Now, let's examine some important properties of eigenvectors. In doing so, the eigen problem for the transpose of A can be written as:

$$A^T y = \bar{\lambda} y \quad (3.30)$$

Eq. 3.30 corresponds to:

$$[A^T - \bar{\lambda} I] y = 0, \quad I = \begin{bmatrix} 1 & 0 & \dots & 0 \\ 0 & 1 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & 1 \end{bmatrix} \quad (3.31)$$

The characteristic equation to be solved for a non-trivial solution to exist for Eq. 3.31 is:

$$\det(A^T - \bar{\lambda} I) = 0 \quad (3.32)$$

But, we know from the properties of determinants that $|A| = |A^T|$. Hence, Eq. 3.32 is the same as:

$$\det(A^T) = \det(A)$$

$$|A - \bar{\lambda} I| = 0 \quad (3.33)$$

and we can conclude that, $\bar{\lambda} = \lambda$.

Now, we can write Eq. 3.30 as:

$$A^T y = \lambda y \quad (3.34)$$

Taking the transpose on both sides of Eq. 3.34, we get:

$$y^T A = \lambda y^T \quad (3.35)$$

Now, let's write Eq. 3.35 and Eq. 3.29 for the i^{th} and j^{th} eigenvectors as follows:

$$y_i^T A = \lambda_i y_i^T \quad (3.36)$$

$$A_{\alpha_j} = \lambda_j \alpha_j \quad (3.37)$$

Post-multiplying Eq. 3.36 by x_j and pre-multiplying Eq. 3.37 by y_i^T ,

$$y_i^T A x_i = \lambda_i y_i^T x_i \quad (3.38)$$

$$y_i^T A x_i = \lambda_i y_i^T x_i \quad (3.39)$$

Subtracting them, we get:

$$(y_i^T - \lambda_i) y_i^T x_i = 0 \quad (3.40)$$

This leads to the conclusion that:

$$y_i^T x_i = 0, \quad \lambda_i \neq \lambda_j \quad \text{since } y \text{ and } x \text{ are vectors, this is equivalent to } y \cdot x = 0 \quad (3.41)$$

where, y are called the *left eigenvectors* and x are the *right eigenvectors*.

- The results show that the left and right eigenvectors corresponding to two distinct eigenvalues are orthogonal to each other. If A is symmetric, we do not need to differentiate between the left and right eigenvectors; they are the same.
- It can also be shown that the eigenvectors corresponding to distinct eigenvalues of A are linearly independent. $x_1, x_2, x_3, \dots, x_n \neq a_1 x_1 + a_2 x_2 + \dots + a_n x_n$ since distinct. Hence, any arbitrary vector u can be written as a linear combination of N eigenvectors of A . That is:

$$u_{n+1} = c_1 x_1 + c_2 x_2 + \dots + c_n x_n \quad (3.42)$$

To calculate the coefficients, we simply pre-multiply the equation by y_i^T to get:

$$y_i^T u = c_i y_i^T x_i \quad (3.43)$$

This results in:

$$c_i = \frac{y_i^T u}{y_i^T x_i} \quad (3.44)$$

Of course, if A is symmetric, then y and x are the same.

- The solution to the eigen problem (i.e. determining the eigenvalues and eigenvectors) is usually accomplished numerically in practice.

3.3.2 Natural Frequencies and Modes

Now, let's revisit the idea of the natural frequencies and modes of an MDOF system. We will look at the similarities between the standard eigenproblem described earlier, and free vibration problem we encounter in structural dynamics. Consider an arbitrary N -DOF system without damping.

(3.45)

- We assume that the free vibration of the system in one of its natural modes takes the form:

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{k}\mathbf{x} = \mathbf{0} \quad \left| \begin{array}{l} \mathbf{x} = \phi_i y_i(t) \\ \left[\begin{array}{c} x_1 \\ x_2 \\ \vdots \\ x_N \end{array} \right] = \left[\begin{array}{c} \phi_{i1} \\ \phi_{i2} \\ \vdots \\ \phi_{iN} \end{array} \right] A \sin(\omega_n t + \alpha) \cdot \left[\begin{array}{c} y_1 \\ y_2 \\ \vdots \\ y_N \end{array} \right] (B \cos(\omega_n t + C) + \text{constant}) \end{array} \right. \quad (3.46)$$

$\nearrow \nearrow \nearrow \nearrow$
 $N \times N \quad N \times 1 \quad N \times N \quad N \times 1$

where ϕ_n is a vector containing the amplitudes of x , and $y_i(t)$ is nothing more than undamped free vibration response in mode i .

- Substituting Eq. 3.46 into Eq. 3.45 yields

$$[-\omega_n^2 \mathbf{M} \phi_n + \mathbf{K} \phi_n] y_i(t) = \mathbf{0} \quad (3.47)$$

- For a nontrivial solution,

$$\mathbf{K} \phi_n = \omega_n^2 \mathbf{M} \phi_n \quad (3.48)$$

- If we pre-multiply both sides of Eq. 3.45 by \mathbf{M}^{-1} , we get:

$$\mathbf{M}^{-1} \mathbf{K} \phi_n = \omega_n^2 \phi_n \quad (3.49)$$

We can easily see that Eq. 3.46 is the same as the standard eigenproblem in Eq. 3.29 with $\mathbf{M}^{-1} \mathbf{K}$ being analogous to \mathbf{A} and ω_n^2 having the same interpretation as λ .

- The matrix eigenvalue problem in Eq. 3.48 in which ω_n^2 are the eigenvalues and the amplitude vector ϕ_n are the corresponding eigenvectors can also be posed as

$$(-\omega_n^2 \mathbf{M} + \mathbf{K}) \phi_n = \mathbf{0} \quad (3.50)$$

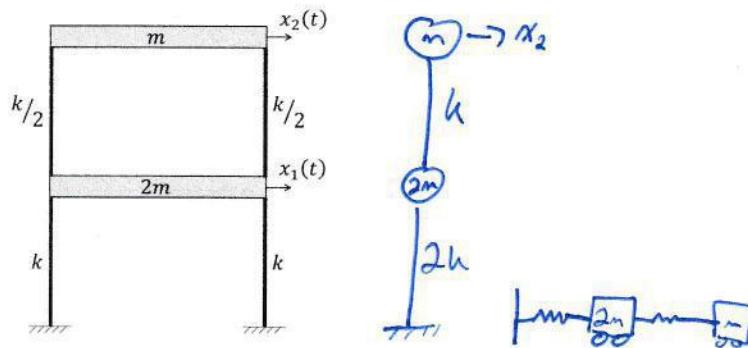
- Which has a nontrivial solution if

$$\det[-\omega_n^2 \mathbf{M} + \mathbf{K}] = 0 \quad (3.51)$$

- If \mathbf{M} and \mathbf{K} are real and positive definite, the eigenvalues ω_n^2 will be real and positive, and the eigenvectors will be real.
- For an N -DOF system, solving Eq. 3.51 will yield N real and positive roots for ω_n^2 .
- For each ω_n , the corresponding eigenvector ϕ_n can be obtained from Eq. 3.50. However, as we saw in Section 3.2, we cannot solve for exact values. It is only possible to know the relative values in ϕ_n .
- The eigenvectors ϕ_n correspond to the response amplitude and are called the *mode shape vectors*. As we saw in Example 3.2, mode shape vectors give us a general sense of what the free vibration response looks like in the natural modes.

- The natural frequencies and associated mode shape vectors are numbered from the lowest frequency to highest from 1 to N . $\omega_{n,1}$ is called the *fundamental circular natural frequency*.

Example 3.3 Determine the natural periods and sketch the mode shapes for the two-storey frame shown below.



Solution: The equation of motion is

$$\begin{bmatrix} 2m & 0 \\ 0 & m \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} 3k & -k \\ -k & k \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \quad M\ddot{x} + Kx = \mathbf{0} \quad \text{Assume } \omega = \sqrt{\lambda} \quad (3.52)$$

To determine the natural frequencies and modes, we solve the eigenvalue problem:

$$(-\omega_n^2 \mathbf{M} + \mathbf{K}) \boldsymbol{\phi}_n = \mathbf{0} \quad \text{(3.53)}$$

Which has nontrivial solutions if

$$\begin{vmatrix} 3k - 2m^2 n & -k \\ -k & k - m^2 n \end{vmatrix} = 0$$

$$(3k - 2m^2 n)(k - m^2 n) - k^2 = 0 \quad (3.54)$$

$$m^4 - \frac{5k}{2m} m^2 + \frac{k^2}{n^2} = 0$$

The 2-DOF system yields two roots,

$$w_{n,1} = \frac{k}{2m}, \quad w_{n,1} = 0.707 \sqrt{\frac{k}{m}} \text{ rads/sec}$$

$$\omega_{n,1}^2 = \frac{2k}{m}, \quad \omega_{n,2} = 1.414 \sqrt{\frac{k}{m}} \text{ rads/sec} \quad (3.55b)$$

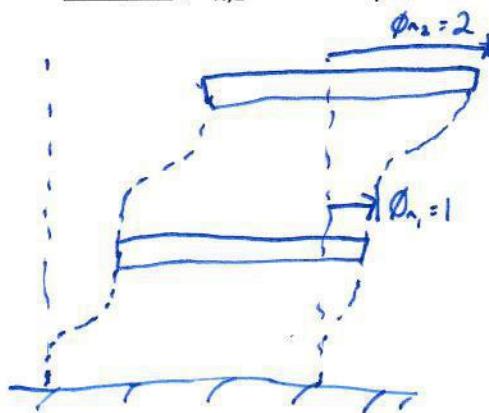
We can now solve for the mode shape vectors. For simplicity take $\phi_1 = 1$.

$$\begin{bmatrix} 3k - 2\omega_n^2 m & -k \\ -k & k - \omega_n^2 m \end{bmatrix} \begin{bmatrix} 1 \\ \phi_2 \end{bmatrix}_n = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \quad (3.56)$$

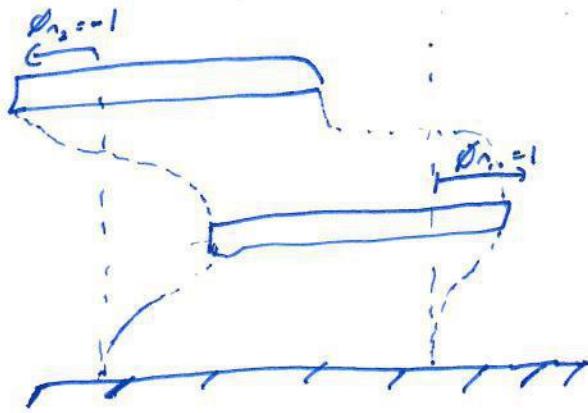
Subbing $m_{n,1} = m$, $3k - 2\omega_n^2 m - \phi_2 k = 0$, $\phi_2 = 2$, Mode shape $1 = \begin{bmatrix} 1 \\ 2 \end{bmatrix}$

$m_{n,2} \rightarrow 3k - 2\omega_n^2 m - \phi_2 k = 0$, $\phi_2 = -1$ $= \begin{bmatrix} 1 \\ -1 \end{bmatrix}$

MODE 1 ($\omega_{n,1} = 0.707\sqrt{k/m}$ rad/s)



MODE 2 ($\omega_{n,2} = 1.414\sqrt{k/m}$ rad/s)



3.3.3 Orthogonality Property

Recall the matrix eigenvalue problem

$$K\phi_n = \omega_n^2 M\phi_n \quad (3.57)$$

Equation 3.57 can be written for each frequency and mode as

$$\text{Mode } i: K\phi_{n,i} = \omega_{n,i}^2 M\phi_{n,i} \quad (3.58a)$$

$$\text{Mode } j: K\phi_{n,j} = \omega_{n,j}^2 M\phi_{n,j} \quad (3.58b)$$

Pre-multiplying Equation 3.58a by $\phi_{n,j}^T$

$$\phi_{n,i}^T K \phi_{n,j} = \omega_{n,i}^2 \phi_{n,i}^T M \phi_{n,j} \quad (3.59)$$

Pre-multiplying Equation 3.58b by $\phi_{n,i}^T$

$$\phi_{n,i}^T K \phi_{n,j} = \omega_{n,j}^2 \phi_{n,i}^T M \phi_{n,j} \quad (3.60)$$

Taking the transpose of both sides in Equation 3.59 gives

$$\begin{aligned} [\mathbf{L}\mathbf{S}]^T &= [\mathbf{R}\mathbf{S}]^T, \quad [\phi_{n,i}^T \mathbf{K} \phi_{n,j}]^T = \omega_{n,i}^2 [\phi_{n,i}^T \mathbf{M} \phi_{n,i}]^T \\ \phi_{n,i}^T \mathbf{K} \phi_{n,j} &= \omega_{n,i}^2 \phi_{n,i}^T \mathbf{M} \phi_{n,i} \end{aligned} \quad (3.61)$$

Subtracting Equation 3.61 from Equation 3.60,

$$(\omega_{n,i}^2 - \omega_{n,j}^2) \phi_{n,i}^T \mathbf{M} \phi_{n,j} = 0 \quad (3.62)$$

Since $\omega_{n,i}^2 \neq \omega_{n,j}^2 \neq 0$ (assuming $i \neq j$), it follows that

$$\phi_{n,i}^T \mathbf{M} \phi_{n,j} = 0 \quad (3.63)$$

which is known as the mass orthogonality property of the modes. Substituting this result back into Equation 3.60, we find

$$\phi_{n,i}^T \mathbf{K} \phi_{n,i} = 0 \quad \text{for } i \neq j \quad (3.64)$$

demonstrating the orthogonality of the stiffness matrix and the modes. Finally, pre-multiplying Equation 3.58a by $\phi_{n,i}^T$ (or Equation 3.58b by $\phi_{n,j}^T$, we get

$$\frac{\phi_{n,i}^T \mathbf{K} \phi_{n,i}}{\phi_{n,i}^T \mathbf{M} \phi_{n,i}} = \omega_{n,i}^2 \quad \underbrace{\mathbf{K} \phi_{n,i} = \omega_{n,i}^2 \mathbf{M} \phi_{n,i}} \quad (3.65)$$

We now introduce the *modal matrix*, Φ .

$$\Phi = \begin{bmatrix} \phi_{11} & \phi_{12} & \cdots & \phi_{1N} \\ \phi_{21} & \phi_{22} & \cdots & \phi_{2N} \\ \vdots & \vdots & \ddots & \vdots \\ \phi_{N1} & \phi_{N2} & \cdots & \phi_{NN} \end{bmatrix} = \begin{bmatrix} \phi_{n,1} & \phi_{n,2} & \cdots & \phi_{n,N} \end{bmatrix} \quad (3.66)$$

The orthogonality of natural modes implies the following:

$$\mathbf{I}^T \mathbf{M} \mathbf{I} = \begin{bmatrix} \phi_{11}^T \mathbf{M} \phi_{11} & 0 \\ 0 & \phi_{22}^T \mathbf{M} \phi_{22} \end{bmatrix} \quad (3.67a)$$

$$\mathbf{I}^T \mathbf{K} \mathbf{I} = \begin{bmatrix} \phi_{11}^T \mathbf{K} \phi_{11} & 0 \\ 0 & \phi_{22}^T \mathbf{K} \phi_{22} \end{bmatrix} \quad (3.67b)$$

It follows that the following matrices are diagonal.

$$\phi_1^T \mathbf{M} \phi_2 = 0 \quad \text{SWITCH} \quad (3.68a)$$

$$\phi_1^T \mathbf{K} \phi_2 = 0 \quad (3.68b)$$

Let's return to the solution of a undamped 2-DOF system under free vibration, given in Equation 3.26.

$$\underline{x}_0 = \begin{bmatrix} x_{1,0} \\ x_{2,0} \end{bmatrix} = \begin{bmatrix} x_{0,1} \\ x_{0,2} \end{bmatrix} \quad x = \sum_{i=1}^2 \phi_i (a_i \cos \omega_{n,i} t + b_i \sin \omega_{n,i} t) \quad (3.67)$$

If $\underline{x}(0) = \underline{x}_0$ and $\dot{\underline{x}}(0) = \underline{0}$ are the displacement and velocity initial conditions, respectively, we can write the initial displacement of the system as

$$\underline{x}(0) = \underline{x}_0 = a_1 \phi_1 + a_2 \phi_2 \quad (3.68)$$

Let us specify a particular configuration, or "shape", for the displacement initial condition. Furthermore, let us specify the initial displacement condition in the form of the first mode of vibration, ϕ_1 . Therefore,

$$\begin{bmatrix} x_{0,1} \\ x_{0,2} \end{bmatrix} = \begin{bmatrix} \phi_1 \\ 0 \end{bmatrix} \quad x_0 = \phi_1 = a_1 \phi_1 + a_2 \phi_2 \quad (3.69)$$

Pre-multiplying each term in Equation 3.69 by $\phi_1^T M$, we get

$$\phi_1^T M \phi_1 = a_1 \phi_1^T M \phi_1 + a_2 \phi_1^T M \phi_2 \quad (\text{if initial condition only mode 1, will only be mode 2}) \quad (3.70)$$

$$= a_1 = 1$$

This implies $a_1 = 1$, which is the only non-zero coefficient. This means the structure continues to vibrate in the first mode.

- The coefficients a_1 and a_2 represent the coefficients in an expansion of the initial deflection configuration in terms of the natural modes.
- In general, all modes of the system are excited. However, the extent to which each mode participates in the total response depends on the initial conditions.

3.3.4 Rigid Body Modes

Let us investigate what happens when a system is not properly constrained by considering the following example.

Example 3.4 Find the fundamental circular natural frequency and the corresponding natural mode for the system shown in Figure 3.3.

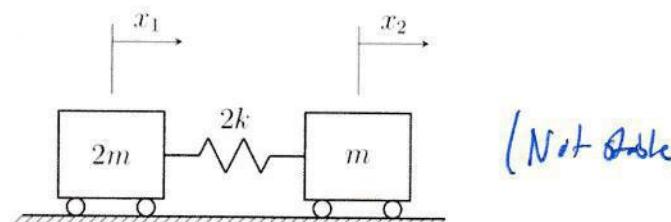
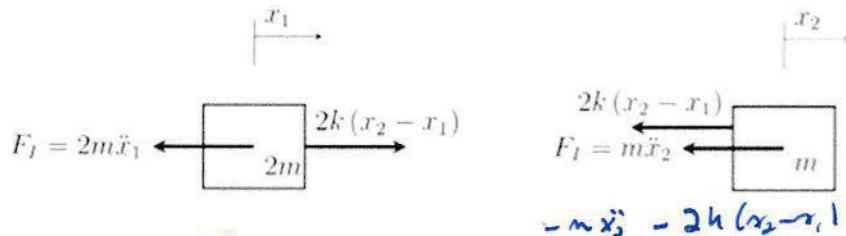


Figure 3.3 System exhibiting a rigid body natural mode

Solution: Using Newtonian Mechanics, we can derive the equations of motion. From the free body diagrams



The equations of motion are

$$\begin{bmatrix} 2m & 0 \\ 0 & m \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} 2k & -2k \\ -2k & 2k \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \quad (3.71a)$$

$$(3.71b)$$

Assuming the solution is of the form

$$\vec{x} = \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} X_1 \\ X_2 \end{bmatrix} \sin(\omega_n t + \phi) \quad (3.72)$$

For the non-trivial solution,

$$\begin{bmatrix} (2k - 2m\omega_n^2) & -2k \\ -2k & (k - m\omega_n^2) \end{bmatrix} \begin{Bmatrix} X_1 \\ X_2 \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \quad (3.73)$$

We can find the roots by setting the following determinant to zero:

$$\begin{vmatrix} 2k - 2m\omega_n^2 & -2k \\ -2k & k - m\omega_n^2 \end{vmatrix} = 0 \quad (3.74)$$

Expanding the determinant to find the characteristic equation and solving for ω_n^2 , we find the following roots:

$$\omega_{n,1}^2 = 0 \quad \text{*2 zero & 1 diff'rent*} \quad \omega_{n,2}^2 = \frac{3k}{m}$$

$$(3.75a)$$

$$(3.75b)$$

The fact that $\omega_{n,1}^2 = 0$ indicates the presence of a rigid body mode. Substituting this result into Equation 3.73 and selecting $X_2 = 1$, we find the corresponding natural mode.

$$\begin{Bmatrix} X_1 \\ X_2 \end{Bmatrix}_1 = \begin{Bmatrix} 1 \\ 1 \end{Bmatrix} \quad (3.76)$$

which is also indicative of a rigid body mode. For the second natural mode,

$$\begin{Bmatrix} X_1 \\ X_2 \end{Bmatrix}_2 = \begin{Bmatrix} 1 \\ -2 \end{Bmatrix} \quad (3.77)$$

3.3.5 Mode Normalization / Scaling

We have already seen that the mode shape vectors (the eigenvectors) can only be determined to a scaling factor. That is, for a 2-DOF system, only the relative values can be obtained for $\{X_1 \ X_2\}^T$. We can set $X_1 = 1$ and find $\beta_1 = (X_2/X_1)_1$ to obtain $\phi_1 = \{1 \ \beta_1\}^T$. Similarly, $\phi_2 = \{1 \ \beta_2\}^T$, where $\beta_2 = (X_2/X_1)_2$. In other words, the elements of a modal matrix are only known in a relative sense; their absolute magnitudes are indeterminate. We will now investigate other ways to scale the natural modes.

3.3.6 Mass Normalization

As discussed above, we can only ever determine relative values for the mode shape vectors. Any vector proportional to ϕ_i yields the same mode shape and satisfies the eigenvalue problem. As a result, it is typical to normalize the mode shape vectors in some way. It may be done such that the largest element in ϕ_i is unity or such that the element corresponding to a certain DOF is unity. It is also common to normalize modes such that

$$M_{ii} = \phi_i^T M \phi_i = 1 \quad (3.78)$$

This is known as *mass normalization*. Note that due to mode orthogonality, $\phi_i^T M \phi_j = 0$ for $i \neq j$. Equation 3.78 can also be written as

$$\underline{\Phi}^T M \underline{\Phi} = I \leftarrow \text{identity matrix} \quad (3.79)$$

where $\Phi = [\phi_1 \ \phi_2 \ \dots \ \phi_N]$. Hence, from Equation 3.65,

$$\underline{\Phi}^T K \underline{\Phi} = \begin{bmatrix} \omega_{1,1}^2 & 0 & \dots & 0 \\ 0 & \omega_{2,2}^2 & \dots & 0 \\ 0 & 0 & \dots & \omega_{N,N}^2 \end{bmatrix} = \Lambda \leftarrow \text{big lambda} \quad (3.80)$$

Let us consider an illustrative example.

Example 3.5 Determine the mass normalized modes, $\Phi = [\phi_1 \ \phi_2]$ for $M = \begin{bmatrix} 2m & 0 \\ 0 & m \end{bmatrix}$, assuming we have already calculated the natural modes $\underline{\Phi} = \begin{bmatrix} 1 & 1 \\ 2 & -1 \end{bmatrix}$.

Solution: Begin by letting $\phi_1 = \alpha \begin{pmatrix} 1 \\ 2 \end{pmatrix}$ and $\phi_2 = \beta \begin{pmatrix} 1 \\ -1 \end{pmatrix}$, where α and β are to be determined. We would like to normalize the modes such that $\phi_1^T M \phi_1 = 1$.

$$\begin{bmatrix} \alpha & \beta \end{bmatrix} \begin{bmatrix} 2m & 0 \\ 0 & m \end{bmatrix} \begin{bmatrix} \alpha \\ \beta \end{bmatrix} = 1$$

$$2\alpha^2 m + 4\alpha\beta m = 1, \quad \alpha = \frac{1}{\sqrt{6m}}, \quad \phi_1 = \frac{1}{\sqrt{6m}} \begin{bmatrix} 1 \\ 2 \end{bmatrix} \quad (3.81)$$

Solving for α , we find that

$$\alpha = \frac{1}{\sqrt{6m}} \quad (3.82)$$

Similarly, for $\phi_2^T M \phi_2 = 1$, we have

$$\begin{bmatrix} 3 & 1 \\ 1 & 3 \end{bmatrix} \begin{bmatrix} 2 & 0 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \alpha \\ \beta \end{bmatrix} = 1 \quad , \quad (3.83)$$

Solving for β , we find that

$$\beta = \frac{1}{\sqrt{3m}} \rightarrow \phi_2 = \frac{1}{\sqrt{3m}} \begin{bmatrix} 1 \\ -1 \end{bmatrix} \quad (3.84)$$

Therefore, the mass normalized modal matrix is

$$\Phi = \begin{bmatrix} \frac{1}{\sqrt{6m}} & \frac{1}{\sqrt{3m}} \\ \frac{1}{\sqrt{6m}} & -\frac{1}{\sqrt{3m}} \end{bmatrix} \quad \text{Verif: } \quad \begin{bmatrix} I^T \\ \Phi^T \end{bmatrix} M \begin{bmatrix} I \\ \Phi \end{bmatrix} = I \quad (3.85)$$

$$I^T K \Phi = 1$$

3.3.7 Mode Superposition

Recall the solution for the undamped MDOF system under free vibration.

$$(3.86)$$

Let

$$\Phi = [\phi_1 \ \phi_2 \ \dots \ \phi_N] = \begin{bmatrix} \phi_{11} & \phi_{12} & \dots & \phi_{1N} \\ \phi_{21} & \phi_{22} & \dots & \phi_{2N} \\ \vdots & \vdots & \ddots & \vdots \\ \phi_{N1} & \phi_{N2} & \dots & \phi_{NN} \end{bmatrix} \quad (3.87)$$

be the modal matrix. Expanding Equation 3.86, we have

$$x_1 = a_1 \phi_{11} \cos \omega_{n,1} t + b_1 \phi_{11} \sin \omega_{n,1} t + a_2 \phi_{12} \cos \omega_{n,2} t + b_2 \phi_{12} \sin \omega_{n,2} t + \dots + a_N \phi_{1N} \cos \omega_{n,N} t + b_N \phi_{1N} \sin \omega_{n,N} t \quad (3.88a)$$

$$x_2 = a_1 \phi_{21} \cos \omega_{n,1} t + b_1 \phi_{21} \sin \omega_{n,1} t + a_2 \phi_{22} \cos \omega_{n,2} t + b_2 \phi_{22} \sin \omega_{n,2} t + \dots + a_N \phi_{2N} \cos \omega_{n,N} t + b_N \phi_{2N} \sin \omega_{n,N} t \quad (3.88b)$$

⋮

3.3.7 Mode Superposition

Recall the solution for the undamped MDOF system under free vibration.

$$x = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_N \end{bmatrix} = \sum_{i=1}^N \phi_i (a_i \cos(\omega_{n,i} t) + b_i \sin(\omega_{n,i} t)) = \sum_{i=1}^N \phi_i A_i \sin(\omega_{n,i} t + \alpha_i) \quad (3.86)$$

Let

$$\Phi = [\phi_1 \ \phi_2 \ \cdots \ \phi_N] = \begin{bmatrix} \phi_{11} & \phi_{12} & \cdots & \phi_{1N} \\ \phi_{21} & \phi_{22} & \cdots & \phi_{2N} \\ \vdots & \vdots & \ddots & \vdots \\ \phi_{N1} & \phi_{N2} & \cdots & \phi_{NN} \end{bmatrix} \quad (3.87)$$

be the modal matrix. Expanding Equation 3.86, we have

$$x_1 = a_1 \phi_{11} \cos \omega_{n,1} t + b_1 \phi_{11} \sin \omega_{n,1} t + a_2 \phi_{12} \cos \omega_{n,2} t + b_2 \phi_{12} \sin \omega_{n,2} t + \cdots + a_N \phi_{1N} \cos \omega_{n,N} t + b_N \phi_{1N} \sin \omega_{n,N} t \quad (3.88a)$$

$$x_2 = a_1 \phi_{21} \cos \omega_{n,1} t + b_1 \phi_{21} \sin \omega_{n,1} t + a_2 \phi_{22} \cos \omega_{n,2} t + b_2 \phi_{22} \sin \omega_{n,2} t + \cdots + a_N \phi_{2N} \cos \omega_{n,N} t + b_N \phi_{2N} \sin \omega_{n,N} t \quad (3.88b)$$

$$x_N = a_1 \phi_{N1} \cos \omega_{n,1} t + b_1 \phi_{N1} \sin \omega_{n,1} t + a_2 \phi_{N2} \cos \omega_{n,2} t + b_2 \phi_{N2} \sin \omega_{n,2} t + \dots + a_N \phi_{NN} \cos \omega_{n,N} t + b_N \phi_{NN} \sin \omega_{n,N} t \quad (3.88c)$$

In matrix form, Equations 3.88 can be written as

$$\mathbf{x} = \begin{bmatrix} \phi_{11} & \phi_{12} & \dots & \phi_{1N} \\ \phi_{21} & \phi_{22} & \dots & \phi_{2N} \\ \vdots & \vdots & \ddots & \vdots \\ \phi_{N1} & \phi_{N2} & \dots & \phi_{NN} \end{bmatrix} \begin{Bmatrix} a_1 \cos \omega_{n,1} t + b_1 \sin \omega_{n,1} t \\ a_2 \cos \omega_{n,2} t + b_2 \sin \omega_{n,2} t \\ \vdots \\ a_N \cos \omega_{n,N} t + b_N \sin \omega_{n,N} t \end{Bmatrix} \quad (3.89)$$

$\mathbf{x} = \Phi \mathbf{y}$ very general

The modal matrix, Φ , we have calculated can be very powerful to solve the equations of motion for MDOF system. Begin with the equations of motion for an undamped MDOF system under free vibration.

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{0} \quad \begin{array}{l} \text{gone from } \mathbf{x} \text{ plane} \\ \text{of displacement and} \\ \text{acceleration to} \end{array} \quad (3.90)$$

Substituting the coordinate transformation found in Equation 3.89 into Equation 3.90

$$\mathbf{x} = \Phi \mathbf{y} \quad \begin{array}{l} \text{y plane of} \\ \text{harmonic functions} \end{array} \quad (3.91)$$

we get

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{0} \quad \begin{array}{l} \text{Coordinate} \\ \text{Transformation} \end{array} \quad (3.92)$$

Pre-multiplying throughout by Φ^T ,

$$\mathbf{\Phi}^T \mathbf{M} \mathbf{\Phi} \ddot{\mathbf{y}} + \mathbf{\Phi}^T \mathbf{K} \mathbf{\Phi} \mathbf{y} = \mathbf{0} \quad (3.93)$$

Using Equations 3.79 and 3.80, Equation 3.93 can be written as

$$\mathbf{I}\ddot{\mathbf{y}} + \mathbf{\Lambda}\mathbf{y} = \mathbf{0} \quad \mathbf{I} = \begin{bmatrix} 1 & 0 & \dots & 0 \\ 0 & 1 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & 1 \end{bmatrix} \quad \mathbf{\Lambda} = \begin{bmatrix} \omega_{n,1}^2 & & & \\ & \omega_{n,2}^2 & & \\ & & \ddots & \\ & & & \omega_{n,N}^2 \end{bmatrix} \quad (3.94)$$

Since \mathbf{I} and $\mathbf{\Lambda}$ are diagonal, we can write Equation 3.94 as N uncoupled SDOF equations using the modal coordinate y .

No longer vector Quantities
Now \sim SDOF systems

$$\ddot{y}_1 + \omega_{n,1}^2 y_1 = 0 \quad (3.95a)$$

$$\ddot{y}_2 + \omega_{n,2}^2 y_2 = 0 \quad (3.95b)$$

\vdots

$$\ddot{y}_N + \omega_{n,N}^2 y_N = 0 \quad (3.95c)$$

Recall from our study of undamped SDOF systems under free vibration, the solutions are

$$y_i(t) = y_{0,i} \cos \omega_{n,i} t + \frac{\dot{y}_{0,i}}{\omega_{n,i}} \sin \omega_{n,i} t \quad (3.96a)$$

$$y_2 = y_{0,2} \cos \omega_{n,2} t + \frac{\dot{y}_{0,2}}{\omega_{n,2}} \sin \omega_{n,2} t \quad (3.96b)$$

⋮

$$y_N = y_{0,N} \cos \omega_{n,N} t + \frac{\dot{y}_{0,N}}{\omega_{n,N}} \sin \omega_{n,N} t \quad (3.96c)$$

where $\mathbf{y}_0 = \mathbf{y}(0) = [y_{0,1} \ y_{0,2} \ \dots \ y_{0,N}]$ and $\dot{\mathbf{y}}_0 = \dot{\mathbf{y}}(0) = [\dot{y}_{0,1} \ \dot{y}_{0,2} \ \dots \ \dot{y}_{0,N}]$ are the initial displacement and velocity, respectively.

3.3.8 Initial Conditions in Modal Coordinates

We have seen earlier that the equation of motion for an undamped MDOF system under free vibration

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{0} \quad (3.97)$$

can be written as

$$\mathbf{I}\ddot{\mathbf{y}} + \mathbf{A}\mathbf{y} = \mathbf{0} \quad (3.98)$$

The solutions to the equations of motion can be found by determining the solution to each of the uncoupled equations of motion as if they were the response of a SDOF system.

$$y_1 = y_{0,1} \cos \omega_{n,1} t + \frac{\dot{y}_{0,1}}{\omega_{n,1}} \sin \omega_{n,1} t \quad (3.99a)$$

$$y_2 = y_{0,2} \cos \omega_{n,2} t + \frac{\dot{y}_{0,2}}{\omega_{n,2}} \sin \omega_{n,2} t \quad (3.99b)$$

⋮

$$y_N = y_{0,N} \cos \omega_{n,N} t + \frac{\dot{y}_{0,N}}{\omega_{n,N}} \sin \omega_{n,N} t \quad (3.99c)$$

where $\mathbf{y}_0 = [y_{0,1} \ y_{0,2} \ \dots \ y_{0,N}]$ and $\dot{\mathbf{y}}_0 = [\dot{y}_{0,1} \ \dot{y}_{0,2} \ \dots \ \dot{y}_{0,N}]$ are the displacement and velocity initial conditions, respectively, in modal coordinates. The solution to Equations 3.99 can be transformed back into the physical coordinates using the following coordinate transformation:

$$\mathbf{x} = \Phi \mathbf{y} \quad (3.100)$$

However, before we can solve Equations 3.99, we require \mathbf{y}_0 and $\dot{\mathbf{y}}_0$. It is impractical to specify the initial conditions in the modal domain. Therefore, we require a coordinate transformation to transform the physical initial conditions into the modal domain. The relationship between the physical and modal coordinate systems for the initial conditions is the same for the response.

That is,

$$\mathbf{x}_0 = \Phi \mathbf{y}_0 \quad \dot{\mathbf{x}}_0 = \Phi \dot{\mathbf{y}}_0 \quad (3.101)$$

$$\Phi^{-1} \mathbf{x}_0 = \Phi^{-1} \Phi \mathbf{y}_0$$

Pre-multiplying both sides of the Equation 3.101 by $\Phi^T M$ gives

$$\begin{aligned} \text{Must be mass normalized} \\ \Xi^T M x_0 &= \Xi^T M \Phi y_0 \\ &= I_{y_0} \\ &= y_0 \end{aligned} \quad (3.102)$$

Therefore, the displacement initial conditions in the modal coordinates are related to the physical coordinates by the following transformation:

$$y_0 = \Xi^T M x_0 \quad (3.103)$$

A similar relationship exists for the velocity initial condition

$$j_0 = \Xi^T M \dot{x}_0 \quad (3.104)$$

Example 3.6 Find the solution for the 2-DOF system shown in Figure 3.4 for $m_1 = 1 \text{ kg}$, $m_2 = 4 \text{ kg}$, $k_1 = k_3 = 10 \text{ N/m}$, and $k_2 = 2 \text{ N/m}$. The initial conditions are $x_0 = [1 \ 1]^T$ and $\dot{x}_0 = [0 \ 0]^T$.

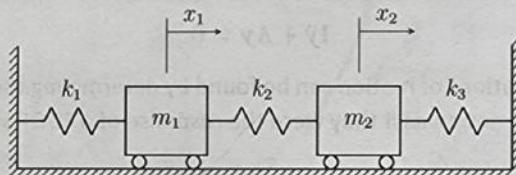
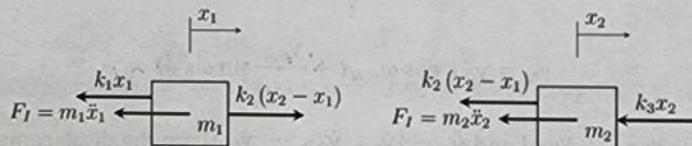


Figure 3.4 Two degree-of-freedom spring-mass model under free vibration

Solution: Begin by the equations of motion for the system. From the free-body diagrams:



the equations of motion are:

$$m_1 \ddot{x}_1 + (k_1 + k_2)x_1 - k_2 x_2 = 0 \quad (3.105a)$$

$$m_2 \ddot{x}_2 - k_2 x_1 + (k_2 + k_3)x_2 = 0 \quad (3.105b)$$

which can also be written as

$$M\ddot{x} + Kx = 0 \quad (3.106a)$$

$$M = \begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \quad (3.106b)$$

$$K = \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 + k_3 \end{bmatrix} \quad (3.106c)$$

Assuming the typical form of the solution for free vibration and expanding the following determinant,

$$\begin{vmatrix} (k_1 + k_2) - m_1 \omega_n^2 & -k_2 \\ -k_2 & (k_2 + k_3) - m_2 \omega_n^2 \end{vmatrix} = 0 \quad (3.107)$$

we get the characteristic equation of the system.

$$\omega_n^4 - \frac{m_1(k_2 + k_3) + m_2(k_1 + k_2)}{m_1 m_2} \omega_n^2 + \frac{(k_1 + k_2)(k_2 + k_3) - k_2^2}{m_1 m_2} = 0 \quad (3.108)$$

Solving for the roots ω_n^2 , we find

$$\omega_{n,1}^2 = 2.89 \text{ (rad/s)}^2 \quad (3.109a)$$

$$\omega_{n,2}^2 = 12.11 \text{ (rad/s)}^2 \quad (3.109b)$$

Selecting $X_1 = 1$ for both modes of vibration, we find the following natural modes:

$$X = \begin{bmatrix} 1 & 1 \\ 4.555 & -0.0549 \end{bmatrix} \quad (3.110)$$

The mass normalized modes are

$$\Phi = \begin{bmatrix} 0.109 & 0.994 \\ 0.497 & -0.055 \end{bmatrix} \quad (3.111)$$

The initial conditions in modal coordinates are

$$\begin{aligned} y_0 &= \Phi^T M x_0 \\ &= \begin{bmatrix} 0.109 & 0.497 \\ 0.994 & -0.055 \end{bmatrix} \begin{bmatrix} 1 & 0 \\ 0 & 4 \end{bmatrix} \begin{bmatrix} 1 \\ 1 \end{bmatrix} \end{aligned} \quad (3.112a)$$

$$= \begin{bmatrix} 2.097 \\ 0.776 \end{bmatrix} \quad y_0 = \Phi^T M x_0 = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \quad (3.112b)$$

From Equation 3.99, the solution is $M\ddot{x} + Kx = 0 \rightarrow I\ddot{y} + A\dot{y} + Ky = 0$

$$y(t) = \begin{bmatrix} y_1(t) \\ y_2(t) \end{bmatrix} = \begin{bmatrix} 2.097 \cos 1.7t \\ 0.776 \cos 3.48t \end{bmatrix} \quad (3.113)$$

Transforming back into physical coordinates, the response is $y_1(t) = y_{10} e^{j\omega_{n1} t} + \frac{y_{1r}}{\omega_{n1}} \sin(\omega_{n1} t)$

$$x = \Phi y \quad x = \begin{bmatrix} 0.109 & 0.994 \\ 0.497 & -0.055 \end{bmatrix} \begin{bmatrix} 2.097 \cos 1.7t \\ 0.776 \cos 3.48t \end{bmatrix} \quad (3.114)$$

$$\begin{bmatrix} x_1(t) \\ x_2(t) \end{bmatrix} = \begin{bmatrix} 0.229 \cos 1.7t + 0.771 \cos 3.48t \\ 1.042 \cos 1.7t - 0.042 \cos 3.48t \end{bmatrix}$$

Mode
Super-
position

3.4 Damped MDOF Systems

We will now consider the damped MDOF system under free vibration.

$$\ddot{\mathbf{x}} + \mathbf{Cx} + \mathbf{Kx} = 0 \quad (3.115)$$

Assume that we can use the same modal transformation used in the undamped case.

$$\mathbf{x} = \Phi \mathbf{y} \quad (3.116)$$

Using this transformation, we get

$$\ddot{\mathbf{M}}\ddot{\mathbf{y}} + \mathbf{C}\dot{\mathbf{y}} + \mathbf{K}\mathbf{y} = 0 \quad (3.117a)$$

$$\ddot{\mathbf{M}}\ddot{\mathbf{y}} + \mathbf{E}^T \mathbf{C} \dot{\mathbf{y}} + \mathbf{E}^T \mathbf{K} \mathbf{y} = 0 \quad (3.117b)$$

$$\mathbf{I}\ddot{\mathbf{y}} + \boxed{\mathbf{E}^T \mathbf{C} \dot{\mathbf{y}}} + \Lambda \mathbf{y} = 0 \quad (3.117c)$$

which is only uncoupled when $\Phi^T \mathbf{C} \Phi$ is diagonal. We will now consider under what conditions $\Phi^T \mathbf{C} \Phi$ is diagonal.

3.4.1 Damping Matrix

We first introduce three cases known as proportional damping. When \mathbf{C} is proportional, it is possible to uncouple the equations of motion using the undamped modes. Any other form of the damping matrix is known as non-proportional damping.

Mass proportional damping Let the damping matrix be related to the mass matrix as follows:

$$\mathbf{C} = \alpha \mathbf{M} \quad (3.118)$$

where α is a proportionality factor.

$$\mathbf{E}^T \mathbf{C} \mathbf{E} = \mathbf{E}^T \alpha \mathbf{M} \mathbf{E} = \alpha \mathbf{E}^T \mathbf{M} \mathbf{E} = \alpha \mathbf{I} \quad (3.119)$$

Therefore, $\Phi^T \mathbf{C} \Phi$ is diagonal.

Stiffness proportional damping Let the damping matrix be related to the stiffness matrix as follows:

$$\mathbf{C} = \beta \mathbf{K} \quad (3.120)$$

where β is a proportionality factor.

$$\mathbf{E}^T \mathbf{C} \mathbf{E} = \mathbf{E}^T \beta \mathbf{K} \mathbf{E} = \beta \mathbf{E}^T \mathbf{K} \mathbf{E} = \beta \Lambda \quad (3.121)$$

Therefore, $\Phi^T \mathbf{C} \Phi$ is diagonal.

Rayleigh damping Let the damping matrix be related to the mass and stiffness matrices as follows:

$$C = \alpha M + \beta K \quad (3.122)$$

where α and β are proportionality factors.

$$\Phi^T C \Phi = \Phi^T (\alpha M + \beta K) \Phi = \alpha I + \beta \Lambda \quad (3.123)$$

Therefore, $\Phi^T C \Phi$ is diagonal.

The damping matrix for practical structures cannot be simply calculated from the structural dimensions and member sizes and the corresponding individual damping of each member. Instead, the damping matrix must be estimated from the modal damping ratios, which account for the energy dissipation of the system. Modal damping ratios are typically based on the designer's experience with similar structures and excitations in similar stress ranges.

Recall from our discussion of SDOF systems, the equation of motion in the form

$$m\ddot{x} + c\dot{x} + kx = 0 \quad (3.124)$$

can be also written as

$$\ddot{x} + \frac{c}{m}\dot{x} + \frac{k}{m}x = 0 \quad (3.125a)$$

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2x = 0 \quad (3.125b)$$

From our previous discussion that mass proportional damping, where $C = \alpha M$, stiffness proportional damping, where $C = \beta K$, and Rayleigh damping, where $C = \alpha M + \beta K$, are all forms of proportional damping that result in $\Phi^T C \Phi$ being diagonal. If the system is to take the same form as the SDOF system in Equation 3.125b, the diagonal damping matrix must be of the form,

$$\Phi^T C \Phi = \begin{bmatrix} 2\zeta_1 \omega_{n,1} & & & \\ & 2\zeta_2 \omega_{n,2} & & \\ & & \ddots & \\ 0 & & & 2\zeta_N \omega_{n,N} \end{bmatrix} \quad (3.126)$$

For every modal response, different damping ratio!

where N is the number of degrees of freedom.

In practice, the modal damping ratios are generally specified for the various modes of vibration. With this information, we will now consider the development of the damping matrix, C , by means of a 2-DOF damping system under free vibration. The equations of motion in the modal domain (in terms of $y = \Phi^T M x$) are as follows:

$$\ddot{y}_1 + 2\zeta_1 \omega_{n,1} \dot{y}_1 + \omega_{n,1}^2 y_1 = 0 \quad (3.127a)$$

$$\ddot{y}_2 + 2\zeta_2 \omega_{n,2} \dot{y}_2 + \omega_{n,2}^2 y_2 = 0 \quad (3.127b)$$

We begin by assuming that ζ_1 or ζ_2 are given, or both. The damping matrix C is dependant on

our assumption of either mass proportional, stiffness proportional, or Rayleigh damping. We will investigate each case separately.

Mass proportional Assuming that C is of the form

$$C = \alpha M \quad (3.128)$$

Pre- and post-multiplying by Φ^T and Φ , respectively, we get

$$\begin{aligned} \Phi^T C \Phi &= 2 \Phi^T M \Phi = 2 I \\ \begin{bmatrix} 2\zeta_1 \omega_{n,1} & 0 \\ 0 & 2\zeta_2 \omega_{n,2} \end{bmatrix} &= \begin{bmatrix} 2 & 0 \\ 0 & \alpha \end{bmatrix} \end{aligned} \quad (3.129)$$

Therefore,

$$\alpha = 2\zeta_1 \omega_{n,1} \quad (3.130)$$

or

$$\alpha = 2\zeta_2 \omega_{n,2} \quad \text{or } \text{one alpha.} \quad (3.131)$$

This implies that α can be related to ζ_1 or ζ_2 , but not both. Assuming that α is calculated using Equation 3.130, we can recalculate the damping ratio for the remaining mode as

$$\zeta_2 = \frac{\alpha}{2\omega_{n,2}} \quad (3.132)$$

Therefore, for a N -DOF system, α is calculated by selecting which damping ratio α is related to, and the remaining damping ratios are found in a similar fashion to Equation 3.132. For mass proportional damping, the damping ratio is inversely proportional to the natural frequency.

Stiffness proportional Assuming that C is of the form

$$C = \beta K \quad (3.133)$$

Pre- and post-multiplying by Φ^T and Φ , respectively, we get

$$\begin{aligned} \Phi^T C \Phi &= \beta \Phi^T K \Phi \\ &= \beta \Lambda \\ \begin{bmatrix} 2\zeta_1 \omega_{n,1} & 0 \\ 0 & 2\zeta_2 \omega_{n,2} \end{bmatrix} &= \begin{bmatrix} \beta \omega_{n,1}^2 & 0 \\ 0 & \beta \omega_{n,2}^2 \end{bmatrix} \end{aligned} \quad (3.134)$$

Therefore,

$$\beta = \frac{2\zeta_1}{\omega_{n,1}} \quad (3.135)$$

or

$$\beta = \frac{2\zeta_2}{\omega_{n,2}} \quad (3.136)$$

Similar to mass proportional damping, β can be related to ζ_1 or ζ_2 , but not both. Assuming that β is calculated using Equation 3.135, we can recalculate the damping ratio for the remaining mode as

$$2\zeta_2\omega_{n,2} = \beta\omega_{n,2}^2 \Rightarrow \zeta_2 = \frac{\beta\omega_{n,2}}{2} \quad (3.137)$$

Therefore, for a N -DOF system, β is calculated by selecting which damping ratio β is related to, and the remaining damping ratios are found in a similar fashion to Equation 3.137. For stiffness proportional damping, the damping ratio is directly proportional to the natural frequency.

Rayleigh damping Assuming C is of the form

$$C = \alpha M + \beta K \quad (3.138)$$

Pre- and post-multiplying by Φ^T and Φ , respectively, we get

$$\begin{aligned} \Phi^T C \Phi &= \alpha \Phi^T M \Phi + \beta \Phi^T K \Phi \\ &= \alpha I + \beta \Lambda \\ \begin{bmatrix} 2\zeta_1\omega_{n,1} & 0 \\ 0 & 2\zeta_2\omega_{n,2} \end{bmatrix} &= \begin{bmatrix} \alpha & 0 \\ 0 & \alpha \end{bmatrix} + \begin{bmatrix} \beta\omega_{n,1}^2 & 0 \\ 0 & \beta\omega_{n,2}^2 \end{bmatrix} \end{aligned} \quad (3.139)$$

which can be written as

$$2\zeta_1\omega_{n,1} = \alpha + \beta\omega_{n,1}^2 \quad (3.140a)$$

$$2\zeta_2\omega_{n,2} = \alpha + \beta\omega_{n,2}^2 \quad (3.140b)$$

Subtracting Equation 3.140b from Equation 3.140a, we get

$$\beta = \frac{2(\zeta_1\omega_{n,1} - \zeta_2\omega_{n,2})}{\omega_{n,1}^2 - \omega_{n,2}^2} \quad (3.141)$$

After β is determined, we can solve for α using

$$\alpha = 2\zeta_1\omega_{n,1} - \beta\omega_{n,1}^2 \quad (3.142)$$

or

$$\alpha = 2\zeta_2\omega_{n,2} - \beta\omega_{n,2}^2 \quad (3.143)$$

The relationship between frequency and damping ratio for mass and stiffness proportional damping, and Rayleigh damping is illustrated in Figure 3.5.

Easy to construct since force is to be diagonal in modal coordinates

Which one to use?

Depends

- Rayleigh damping

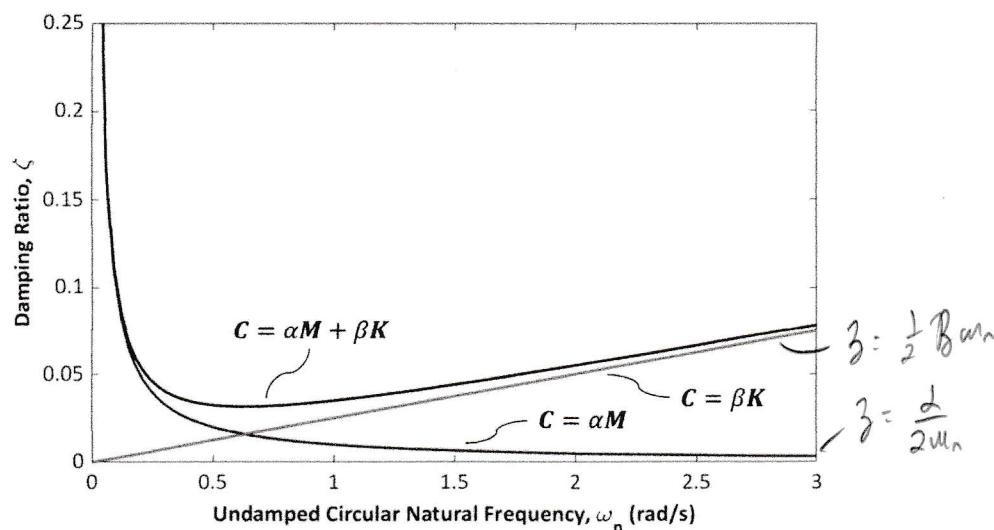


Figure 3.5 Frequency versus damping ratio for mass proportional, stiffness proportional, and Rayleigh damping

Example 3.7 Find the damping matrix for the 3-DOF system with the following natural frequencies:

$$\omega_{n,1} = 11.62 \text{ rad/s} \quad \omega_{n,2} = 27.50 \text{ rad/s} \quad \omega_{n,3} = 45.90 \text{ rad/s}$$

Assume Rayleigh damping and that M and K are known. Use 5% and 3% critical damping in the first two modes, respectively.

Solution: From Equation 3.141

$$\begin{aligned} B &= \frac{2[\zeta_1 \omega_{n,1} - \zeta_2 \omega_{n,2}]}{\omega_{n,1}^2 - \omega_{n,2}^2} \\ &= \frac{2[0.05 \cdot 11.62 - 0.03 \cdot 27.50]}{[11.62^2 - 27.5^2]} \\ &= 0.000786 \end{aligned}$$

Using Equation 3.142 or 3.143, we find that

$$\begin{aligned} \alpha &= 2\zeta_1 \omega_{n,1} - B \omega_{n,1}^2 \\ &= 2 \cdot 0.05 \cdot 11.62 - 0.000786 \cdot 11.62^2 \end{aligned}$$

The damping matrix is then computed as

$$\begin{bmatrix} \mathbf{J}^T & \mathbf{C} & \mathbf{I} \\ 2\zeta_1\omega_{n,1} & 0 & 0 \\ 0 & 2\zeta_2\omega_{n,2} & 0 \\ 0 & 0 & 2\zeta_3\omega_{n,3} \end{bmatrix} = \begin{bmatrix} \mathbf{J}^T \mathbf{M} \mathbf{J} \\ \alpha & 0 & 0 \\ 0 & \alpha & 0 \\ 0 & 0 & \alpha \end{bmatrix} + \begin{bmatrix} \beta\omega_{n,1}^2 & 0 & 0 \\ 0 & \beta\omega_{n,2}^2 & 0 \\ 0 & 0 & \beta\omega_{n,3}^2 \end{bmatrix}$$

We can now compute the damping ratio for the third mode.

$$\begin{aligned} 2\zeta_3\omega_{n,3} &= \alpha + \beta\omega_{n,3}^2 \\ \zeta_3 &= \frac{1.056 + (0.000786)(45.9)^2}{2.459} \\ &\approx 0.0295 \end{aligned}$$

Therefore,

$$\alpha = 1.056 \quad \beta = 0.000786 \quad \zeta_3 = 0.0295 \approx 3\%$$

3.4.2 Mode Superposition for Damped Systems

Now that we have established a methodology to calculate the damping matrix, we now turn our attention to solving the equations for motion for a damped MDOF under free vibration. The equations of motion are

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{0} \quad (3.144)$$

Which can be expressed in modal coordinates using the modal transformation $\mathbf{x} = \Phi\mathbf{y}$ as

$$\mathbf{I}\ddot{\mathbf{y}} + \underbrace{\Phi^T \mathbf{C} \Phi \mathbf{y}}_{\text{Diagonal!}} + \Lambda \mathbf{y} = \mathbf{0} \quad (3.145)$$

For the case of proportional damping, the equations of motion in modal coordinates are fully uncoupled and can be written as

$$\ddot{y}_1 + 2\zeta_1\omega_{n,1}\dot{y}_1 + \omega_{n,1}^2 y_1 = 0 \quad (3.146a)$$

$$\ddot{y}_2 + 2\zeta_2\omega_{n,2}\dot{y}_2 + \omega_{n,2}^2 y_2 = 0 \quad (3.146b)$$

there will be \vdots

$$\text{and } \ddot{y}_N + 2\zeta_N\omega_{n,N}\dot{y}_N + \omega_{n,N}^2 y_N = 0 \quad (3.146c)$$

From our study of free vibration of damped SDOF systems, the solution to Equations 3.146 is

$$y_1 = \bar{e}^{\zeta_1\omega_{n,1}t} \left(y_{0,1} \cos \omega_{D,1}t + \frac{\dot{y}_{0,1} + y_{0,1}\zeta_1\omega_{n,1}}{\omega_{n,1}} \sin \omega_{D,1}t \right) \quad (3.147a)$$

$$y_2 = e^{\zeta_2 \omega_{n,2} t} \left(y_{0,2} \cos \omega_{D,2} t + \frac{\dot{y}_{0,2} + y_{0,2} \zeta_2 \omega_{n,2}}{\omega_{n,2}} \sin \omega_{D,2} t \right) \quad (3.147b)$$

⋮

$$y_N = e^{\zeta_N \omega_{n,N} t} \left(y_{0,N} \cos \omega_{D,N} t + \frac{\dot{y}_{0,N} + y_{0,N} \zeta_N \omega_{n,N}}{\omega_{n,N}} \sin \omega_{D,N} t \right) \quad (3.147c)$$

where

$$\omega_{D,i} = \omega_{n,i} \sqrt{1 - \zeta_i^2} \quad i = 1, 2, \dots, N \quad (3.148)$$

Example 3.8 Solve the equations of motion for a damped 2-DOF system with the following system matrices and initial conditions:

$$\mathbf{M} = \begin{bmatrix} 1 & 0 \\ 0 & 3 \end{bmatrix} \text{ kg} \quad \mathbf{K} = \begin{bmatrix} 10 & -4 \\ -4 & 10 \end{bmatrix} \text{ N/m}$$

$$\mathbf{x}_0 = [1 \ 1]^T \text{ m} \quad \dot{\mathbf{x}}_0 = [0 \ 0]^T \text{ m/s}$$



Assume Rayleigh damping with 5% critical damping in both modes.

Solution: The natural frequencies are found by solving the characteristic equation.

$$\omega_{n,1} = 1.616 \text{ rad/s} \quad \omega_{n,2} = 3.274 \text{ rad/s}$$

The mass normalized modes are *(don't have to, but makes it easier)*

$$\Phi = \begin{bmatrix} 0.2983 & 0.9545 \\ 0.5511 & -0.1722 \end{bmatrix} \quad \underline{\Phi}^T \mathbf{M} \underline{\Phi} = \mathbf{I}$$

For Rayleigh damping

$$\underline{\Phi}^T \mathbf{C} \underline{\Phi} = \mathbf{I}$$

$$\alpha = 0.1082 \quad \beta = 0.0204$$

Therefore, the damping matrix is

$$\mathbf{C} = \alpha \mathbf{M} + \beta \mathbf{K}$$

$$= \begin{bmatrix} 0.3127 & -0.0818 \\ -0.0818 & 0.5291 \end{bmatrix} \text{ N} \cdot \text{s/m}$$

$$\underline{\Phi}^T \mathbf{C} \underline{\Phi} = \begin{bmatrix} 2\zeta_1 \omega_{n,1} & 0 \\ 0 & 2\zeta_2 \omega_{n,2} \end{bmatrix}$$

It can be easily verified that the matrix $\underline{\Phi}^T \mathbf{C} \underline{\Phi}$ is indeed diagonal.

The initial conditions in the modal domain are

$$\underline{\mathbf{y}}_0 = \underline{\Phi}^T \mathbf{M} \mathbf{x}_0, \quad \underline{\dot{\mathbf{y}}}_0 = \underline{\Phi}^T \mathbf{M} \dot{\mathbf{x}}_0 = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

$$\begin{bmatrix} \underline{y}_{0,1} \\ \underline{y}_{0,2} \end{bmatrix} = \begin{bmatrix} 1.952 \\ 0.438 \end{bmatrix}$$

The modal response can be calculated as

$$y_1 = e^{-\zeta_1 \omega_{n,1} t} \left(y_{0,1} \cos \omega_{D,1} t + \frac{\dot{y}_{0,1} + y_{0,1} \zeta_1 \omega_{n,1}}{\omega_{n,1}} \sin \omega_{D,1} t \right)$$

$$y_2 = e^{-\zeta_2 \omega_{n,2} t} \left(y_{0,2} \cos \omega_{D,2} t + \frac{\dot{y}_{0,2} + y_{0,2} \zeta_2 \omega_{n,2}}{\omega_{n,2}} \sin \omega_{D,2} t \right)$$

Substituting in the relevant quantities, we get

$$\begin{Bmatrix} y_1 \\ y_2 \end{Bmatrix} = \begin{Bmatrix} e^{-0.081} (1.952 \cos 1.614t + 0.098 \sin 1.614t) \\ e^{-0.164t} (0.483 \cos 3.270t + 0.022 \sin 3.270t) \end{Bmatrix}$$

The displacement response in physical coordinates is found by

$$\begin{aligned} x &= \Phi y \\ &= \begin{bmatrix} 0.2983 & 0.5511 \\ 0.9545 & -0.1722 \end{bmatrix} \begin{bmatrix} y_1 \\ y_2 \end{bmatrix} \end{aligned}$$

$$\begin{bmatrix} x_1(t) \\ x_2(t) \end{bmatrix} = \begin{bmatrix} 0.2983 y_1(t) + 0.9545 y_2(t) \\ 0.5511 y_1(t) - 0.1722 y_2(t) \end{bmatrix}$$

if pre and post multiply by
↓ eigenvectors, just / a while
idea of orthogonality

3.4.3 Caughey Damping

- More than 2 modes, $\mathbf{C} = \alpha_1 \mathbf{M} + \alpha_2 \mathbf{K} + \alpha_3 \mathbf{I}$

We have seen that the expressions for Rayleigh damping allow us to determine the damping matrix by specifying damping in two modes. We now introduce Caughey damping, which allows us to determine the damping matrix by specifying damping in three modes.

Before writing the relevant equations for Caughey damping, we need to derive an additional orthogonality relation. Recall the characteristic equation

$$[\mathbf{K} - \omega_n^2 \mathbf{M}] \begin{Bmatrix} X_1 \\ X_2 \end{Bmatrix} = 0 \quad (3.149)$$

or

$$[\mathbf{K} - \omega_n^2 \mathbf{M}] \phi_i = 0, \quad \mathbf{K} \phi_i = \omega_n^2 \mathbf{M} \phi_i \quad (3.150)$$

Pre-multiplying both sides by $\phi_i^T \mathbf{K} \mathbf{M}^{-1}$, we get [mode shapes must normalize]

$$\begin{aligned} \phi_i^T \mathbf{K} \mathbf{M}^{-1} \mathbf{K} \phi_i &= \omega_n^2 \phi_i^T \mathbf{M} \phi_i \\ &= \omega_n^2 \phi_i^T \mathbf{K} \mathbf{I} \phi_i \\ &= \omega_n^2 \phi_i^T \mathbf{M} \phi_i \end{aligned} \quad (3.151)$$

Therefore,

$$\left. \begin{aligned} \phi_i^T K M^{-1} K \phi_j &= \omega_{n,i}^4 \\ \phi_i^T K M M^{-1} K \phi_j &= 0 \end{aligned} \right\} \quad \checkmark \quad \Phi^T K M^{-1} K \Phi = \Lambda^2 \quad (3.152a)$$

$$\left. \begin{aligned} \phi_i^T K M M^{-1} K \phi_j &= 0 \end{aligned} \right\} \quad \leftarrow 2 \text{ different } \phi_i \text{ yield } (3.152b) \\ \rightarrow \text{from orthogonality property}$$

Assume the damping matrix C takes the following form:

$$C = \alpha_0 M + \alpha_1 K + \alpha_2 K M^{-1} K \quad (3.153)$$

Pre- and post-multiplying each term by Φ^T and Φ , respectively, gives

$$\Phi^T C \Phi = \alpha_0 I + \alpha_1 \Lambda + \alpha_2 \Lambda^2 \quad (3.154)$$

which is diagonal, and therefore a form of proportional damping. Let us revisit Example 3.7 using Caughey damping.

Example 3.9 Find the damping matrix for the 3-DOF system with the following natural frequencies:

$$\omega_{n,1} = 11.62 \text{ rad/s} \quad \omega_{n,2} = 27.50 \text{ rad/s} \quad \omega_{n,3} = 45.90 \text{ rad/s}$$

Assume Caughey damping and that M and K are known. Use 5% critical damping for all three modes.

Solution: Using Equation 3.154, we get

$$2\zeta_1 \omega_{n,1} = \alpha_0 + \alpha_1 \omega_{n,1}^2 + \alpha_2 \omega_{n,1}^4$$

$$2\zeta_2 \omega_{n,2} = \alpha_0 + \alpha_1 \omega_{n,2}^2 + \alpha_2 \omega_{n,2}^4$$

$$2\zeta_3 \omega_{n,3} = \alpha_0 + \alpha_1 \omega_{n,3}^2 + \alpha_2 \omega_{n,3}^4$$

Substituting in the known quantities we get

$$2(0.05)(11.62) = \alpha_0 + \alpha_1(11.62)^2 + \alpha_2(11.62)^4$$

$$2(0.05)(27.50) = \alpha_0 + \alpha_1(27.50)^2 + \alpha_2(27.50)^4$$

$$2(0.05)(45.90) = \alpha_0 + \alpha_1(45.90)^2 + \alpha_2(45.90)^4$$

Solving the three equations simultaneously gives

$$\alpha_0 = 0.755 \quad \alpha_1 = 0.0031 \quad \alpha_2 = -0.606 \times 10^{-6}$$

If M and K are known, the damping matrix can be calculated as

$$C = \alpha_0 M + \alpha_1 K + \alpha_2 K M^{-1} K$$

3.4.4 Direct Method

The final method we will consider to develop the damping matrix is known as the direct method. The result is known as *modal damping*, and permits the damping matrix to be generated considering the damping ratios for all modes. If the damping ratio is specified for all modes, that is, $\zeta_1, \zeta_2, \dots, \zeta_N$ are given, we can calculate the damping matrix as follows. Recall that

$$\Phi^T C \Phi = \begin{bmatrix} 2\zeta_1 \omega_{n,1} & 0 & \cdots & 0 \\ 0 & 2\zeta_2 \omega_{n,2} & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \cdots & 2\zeta_N \omega_{n,N} \end{bmatrix} = \hat{C} \quad (3.155)$$

Pre- and post-multiplying both sides by $(\Phi^T)^{-1}$ and Φ^{-1} , respectively, gives

$$\begin{aligned} (\Phi^T)^{-1} \Phi^T C \Phi \Phi^{-1} &= (\Phi^T)^{-1} \hat{C} \Phi^{-1} \\ ICI &= (\Phi^T)^{-1} \hat{C} \Phi^{-1} \\ C &= (\Phi^T)^{-1} \hat{C} \Phi^{-1} \end{aligned} \quad (3.156)$$

Using the orthogonality property, a convenient expression for Φ^{-1} can be developed. Recall that

$$\underline{\Phi}^T M \underline{\Phi} = \underline{I} \quad (3.157)$$

Pre-multiplying both sides by $(\Phi^T)^{-1}$ gives

$$\begin{aligned} (\underline{I}^T)^{-1} \underline{\Phi}^T M \underline{\Phi} &\sim (\underline{I}^T)^{-1} \underline{I} \\ M \underline{\Phi} &\sim (\underline{I}^T)^{-1} \end{aligned} \quad (3.158)$$

Similarly, post-multiplying both sides by Φ^{-1} , we get:

$$\begin{aligned} \cancel{\underline{\Phi}^T M \underline{\Phi}} \underline{\Phi}^T &\sim \underline{I} \underline{I}^{-1} \\ \underline{I}^T M &\sim \underline{\Phi}^{-1} \end{aligned} \quad (3.159)$$

Substituting the expressions for $(\Phi^T)^{-1}$ and Φ^{-1} , into Equation 3.156 results in

$$C \sim (M \underline{I}) \hat{C} (\underline{I}^T M) \quad (3.160)$$

Note: Φ contains mass normalized modes. If the modes are not mass normalized, then the above expressions will change to:

$$C = (M \Phi M_r^{-1}) \hat{C} (M_r^{-1} \Phi^T M) \quad (3.161)$$

where,

$$M_r = \Phi^T M \Phi \quad (3.162)$$

3.5 Modal Analysis for MDOF Systems

- The treatment of general N -DOF systems is similar to the 2-DOF system we studied earlier.
- All the properties, such as orthogonality apply to N -DOF systems as well.
- The damping matrix, C , requires special attention, as damping must be proportional in order for the equations of motion to be converted into N SDOF systems in the modal domain.
- Having already considered the response to free vibration, we now turn our attention to the study of modal analysis for forced vibration.

3.5.1 Modal Analysis for Forced Vibration

Consider the equation of motion for a MDOF system subjected to forced vibration.

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F}(t) \quad (3.163)$$

where \mathbf{M} is the $N \times N$ mass matrix, \mathbf{C} is the $N \times N$ damping matrix, \mathbf{K} is the $N \times N$ stiffness matrix, $\mathbf{F}(t)$ is the $N \times 1$ force vector, and N is the number of degrees of freedom. Using the following transformation,

$$\mathbf{x} = \Phi \mathbf{y} \quad (3.164)$$

where Φ is the $N \times N$ matrix of undamped modes. Pre-multiplying by Φ^T , we get

$$\Phi^T \mathbf{M} \Phi \ddot{\mathbf{y}} + \Phi^T \mathbf{C} \Phi \dot{\mathbf{y}} + \Phi^T \mathbf{K} \mathbf{y} = \Phi^T \mathbf{F}(t) \quad (3.165a)$$

$$\ddot{\mathbf{y}} + \Phi^T \mathbf{C} \Phi \dot{\mathbf{y}} + \Lambda \mathbf{y} = \Phi^T \mathbf{F}(t) \quad (3.165b)$$

where

$$\Phi^T \mathbf{C} \Phi = \begin{bmatrix} 2\zeta_1 \omega_{n,1} & 0 & \cdots & 0 \\ 0 & 2\zeta_2 \omega_{n,2} & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \cdots & 2\zeta_N \omega_{n,N} \end{bmatrix} \quad (3.166)$$

The N uncoupled equations of motion can be written as

$$\ddot{y}_1 + 2\zeta_1 \omega_{n,1} \dot{y}_1 + \omega_{n,1}^2 y_1 = \phi_1^T \mathbf{F}(t) \\ = P_1(t) \quad (3.167a)$$

$$\ddot{y}_2 + 2\zeta_2 \omega_{n,2} \dot{y}_2 + \omega_{n,2}^2 y_2 = \phi_2^T \mathbf{F}(t) \\ = P_2(t) \quad (3.167b)$$

⋮

$$\ddot{y}_N + 2\zeta_N \omega_{n,N} \dot{y}_N + \omega_{n,N}^2 y_N = \phi_N^T \mathbf{F}(t) \\ = P_N(t) \quad (3.167c)$$

The solution to any of the above SDOF system in Equations 3.167 is

$$y_i(t) = \frac{1}{\omega_{n,i}} \int_0^t p_i(\tau) e^{-\beta \omega_{n,i}(t-\tau)} \sin \omega_{n,i}(t-\tau) d\tau, \quad (3.168)$$

If there are initial conditions present, the response due to the initial conditions should be added to Equation 3.168.

Once the modal responses are calculated using Equation 3.168, the responses in the original physical coordinates can be found by the transformation in Equation 3.164. We will consider the principle of modal superposition to the following forced excitation case.

Example 3.10 Determine the response $x(t)$ to the step force input, $F(t) = [0 \ 0 \ F_0]^T$, assuming zero initial conditions. The mass and stiffness matrix are

$$\mathbf{M} = m \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 2 \end{bmatrix} \quad \mathbf{K} = k \begin{bmatrix} 2 & -1 & 0 \\ -1 & 3 & -2 \\ 0 & -2 & 2 \end{bmatrix}$$

Assume Rayleigh damping where,

$$\alpha = 0.2 \sqrt{\frac{k}{m}} \quad \beta = 0.01 \sqrt{\frac{m}{k}}$$

Solution: Begin by calculating the undamped natural frequencies and mass normalized modes.

$$\omega_{n,1} = 0.373 \sqrt{\frac{k}{m}} \text{ rad/s} \quad \omega_{n,2} = 1.321 \sqrt{\frac{k}{m}} \text{ rad/s} \quad \omega_{n,3} = 2.029 \sqrt{\frac{k}{m}} \text{ rad/s}$$

$$\Phi = [\phi_1 \ \phi_2 \ \phi_3] \\ = \frac{1}{\sqrt{m}} \begin{bmatrix} 0.269 & -0.878 & -0.395 \\ 0.501 & -0.223 & 0.836 \\ 0.582 & 0.299 & -0.269 \end{bmatrix}$$

Using Equation 3.140a, we can compute ζ_1 , ζ_2 , and ζ_3 .

$$2\beta \omega_{n,i} = \alpha + \beta \omega_{n,i}^2, \quad \zeta_i = \frac{0.1 \sqrt{\frac{k}{m}} + 0.01 \sqrt{\frac{m}{k}} (0.373 \sqrt{\frac{k}{m}})^2}{2 (0.373 \sqrt{\frac{k}{m}})} \\ \approx 0.276$$

$$\zeta_1 = 0.0823, \quad \zeta_2 = 0.0594$$

Next, find the damped circular natural frequencies $\omega_{D,1}$, $\omega_{D,2}$, and $\omega_{D,3}$.

$$\begin{aligned}\omega_{D,1} &= \omega_{n,1} \sqrt{1 - \zeta_1^2} & \omega_{D,2} &= \omega_{n,2} \sqrt{1 - \zeta_2^2} & \omega_{D,3} &= \omega_{n,3} \sqrt{1 - \zeta_3^2} \\ &= 0.359 \sqrt{\frac{k}{m}} \text{ rad/s} & &= 1.317 \sqrt{\frac{k}{m}} \text{ rad/s} & &= 2.025 \sqrt{\frac{k}{m}} \text{ rad/s}\end{aligned}$$

Substituting the relevant quantities into Equations 3.167, we get the equations of motion in modal coordinates.

$$\begin{aligned}\ddot{y}_1 + 2(0.270)(0.373) \sqrt{\frac{k}{m}} \dot{y}_1 + 0.139 \frac{k}{m} y_1 &= \frac{1}{\sqrt{m}} \underbrace{[0.269 \quad 0.501 \quad 0.582]}_{\phi_1^T} \begin{Bmatrix} 0 \\ 0 \\ F_0 \end{Bmatrix} \\ &= 0.582 \frac{F_0}{\sqrt{m}} \\ \ddot{y}_2 + 2(0.0823)(1.321) \sqrt{\frac{k}{m}} \dot{y}_2 + 1.746 \frac{k}{m} y_2 &= \frac{1}{\sqrt{m}} \underbrace{[-0.878 \quad -0.223 \quad 0.299]}_{\phi_2^T} \begin{Bmatrix} 0 \\ 0 \\ F_0 \end{Bmatrix} \\ &= 0.299 \frac{F_0}{\sqrt{m}} \\ \ddot{y}_3 + 2(0.0594)(2.029) \sqrt{\frac{k}{m}} \dot{y}_3 + 4.117 \frac{k}{m} y_3 &= \frac{1}{\sqrt{m}} \underbrace{[-0.395 \quad 0.836 \quad -0.269]}_{\phi_3^T} \begin{Bmatrix} 0 \\ 0 \\ F_0 \end{Bmatrix} \\ &= -0.267 \frac{F_0}{\sqrt{m}}\end{aligned}$$

Recall from our discussion of SDOF systems, the solution to the step response

$$m\ddot{x} + c\dot{x} + kx = P_0$$

can be obtained using Equation 2.223

$$\begin{aligned}x(t) &= \frac{1}{m\omega_n} \int_0^t p(\tau) e^{-j\omega_n(t-\tau)} \sin \omega_n(t-\tau) d\tau \\ &\approx \frac{P_0}{m\omega_n} \left[1 - e^{-j\omega_n t} \left(\cos \omega_n t + \frac{j}{\omega_n} \sin \omega_n t \right) \right]\end{aligned}$$

Hence, we can write the solution to the equations of motion in the modal coordinates as

$$y_1(t) = \frac{4.187 F_0 \sqrt{m}}{k} \left[1 - e^{-0.101 \sqrt{\frac{k}{m}} t} \left(\cos 0.359 \sqrt{\frac{k}{m}} t + 0.280 \sin 0.359 \sqrt{\frac{k}{m}} t \right) \right]$$

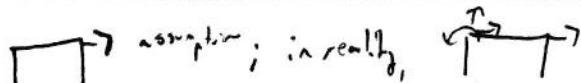
$$y_2(t) = \frac{0.171F_0\sqrt{m}}{k} \left[1 - e^{-0.109\sqrt{\frac{k}{m}}t} \left(\cos 1.317 \sqrt{\frac{k}{m}} t + 0.0826 \sin 1.317 \sqrt{\frac{k}{m}} t \right) \right]$$

$$y_3(t) = \frac{0.065F_0\sqrt{m}}{k} \left[1 - e^{-0.121\sqrt{\frac{k}{m}}t} \left(\cos 2.025 \sqrt{\frac{k}{m}} t + 0.0595 \sin 2.025 \sqrt{\frac{k}{m}} t \right) \right]$$

Having obtained $\mathbf{y} = [y_1 \ y_2 \ y_3]^T$, we can determine the response in physical coordinates using the transformation

$$\mathbf{x} = \Phi \mathbf{y}$$

3.5.2 Truncation of Modes



Often, only a subset of the modal responses, typically the first few, contribute to the majority of the overall response. Say only a subset \bar{N} of the total N modes are retained. That is,

$$\hat{\Phi}_{N \times \bar{N}} = [\phi_1 \ \phi_2 \ \dots \ \phi_{\bar{N}}] \quad (3.169)$$

The overall response, retaining only \bar{N} modes is

$$\mathbf{x}_{N \times 1} = \hat{\Phi}_{N \times \bar{N}} \mathbf{y}_{\bar{N} \times 1}, \text{ only superposing } \bar{N} \text{ responses} \quad (3.170)$$

Truncation becomes important when dealing with very large systems where computational effort is an issue.

*only done if certain solution is adequate
+ one mode is insignificant*

3.6 Vibration Absorber and Vibration Damper

Difference is if considering damping

Since Hermann Frahm introduced the concept over a century ago, vibration absorbers have become ubiquitous for suppressing unwanted harmonic vibrations in machines and sometimes in structures. The idea of a vibration absorber is very simple. It consists of a small (relative to the main structure) mass and a spring attached to the primary structure undergoing unwanted harmonic vibrations. When tuned properly (i.e. for certain specific values of the ratio k_a/m_a), it is possible to eliminate the primary structure response. In the simplest case, both the primary system and secondary system are assumed to be undamped. Only the primary system is assumed to be excited harmonically with a frequency ω .

For this problem, we will not employ modal analysis, as the objective is to determine the absorber parameters (k_a, m_a) and not to compute the response per se. Hence, we will seek expressions for the absorber and primary system responses x and x_a as a function of k_a and m_a , and see if certain values of these parameters will result in a reduction of the primary response

reduction. After all, as a designer we would like to control the primary system response to the extent that we can.

Consider a 2-DOF system with undamped primary and secondary systems as shown

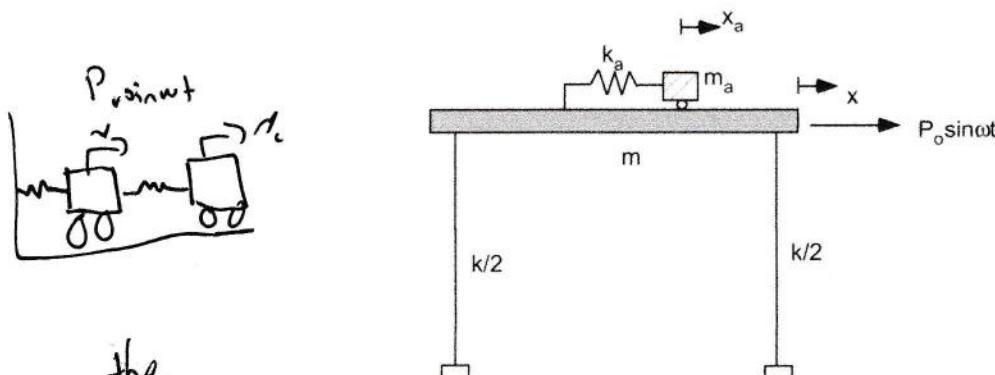


Figure 3.6 Vibration absorber

Using Newtonian Mechanics, we can derive the equations of motion.

$$\begin{bmatrix} m & 0 \\ 0 & m_a \end{bmatrix} \begin{bmatrix} \ddot{x} \\ \ddot{x}_a \end{bmatrix} + \begin{bmatrix} k_1 + k_a & k_a \\ -k_a & k_a \end{bmatrix} \begin{bmatrix} x \\ x_a \end{bmatrix} = \begin{bmatrix} P_0 \sin \omega t \\ 0 \end{bmatrix} \quad (3.171)$$

Let the steady-state solution be of the form

$$\begin{bmatrix} x \\ x_a \end{bmatrix} = \begin{bmatrix} X \\ X_a \end{bmatrix} \sin \omega t \quad (3.172)$$

The corresponding accelerations are

$$\begin{bmatrix} \ddot{x} \\ \ddot{x}_a \end{bmatrix} = -\begin{bmatrix} X \\ X_a \end{bmatrix} \omega^2 \sin \omega t \quad (3.173)$$

Substituting the above expressions in Equation 3.171, we get

$$-\begin{bmatrix} m_1 & 0 \\ 0 & m_a \end{bmatrix} \begin{bmatrix} X \\ X_a \end{bmatrix} \omega^2 \sin \omega t + \begin{bmatrix} k + k_a & -k_a \\ -k_a & k_a \end{bmatrix} \begin{bmatrix} X \\ X_a \end{bmatrix} \sin \omega t = \begin{bmatrix} P_0 \sin \omega t \\ 0 \end{bmatrix} \quad (3.174)$$

Simplifying the above expression, we get

$$\underbrace{\begin{bmatrix} k + k_a - m\omega^2 & -k_a \\ -k_a & k_a - m_a\omega^2 \end{bmatrix}}_{-m^2M+K} \begin{bmatrix} X \\ X_a \end{bmatrix} \sin \omega t = \begin{bmatrix} P_0 \\ 0 \end{bmatrix} \sin \omega t \quad (3.175)$$

which leads us to

Non-trivial solution =

$$\begin{bmatrix} X \\ X_a \end{bmatrix} = (-m^2M+K)^{-1} \begin{bmatrix} P_0 \\ 0 \end{bmatrix} = \begin{bmatrix} k_1 k_a - m m^2 & -k_a \\ -k_a & k_a - m_a \omega^2 \end{bmatrix}^{-1} \begin{bmatrix} P_0 \\ 0 \end{bmatrix} \quad (3.176)$$

Simplifying the above expression leads to the following

$$X = \frac{(k_a - m_a \omega^2) P_0}{(k + k_a - m \omega^2)(k_a - m \omega^2) - k_a^2} \quad (3.177)$$

$$X_a = \frac{k_a P_0}{(k + k_a - m \omega^2)(k_a - m \omega^2) - k_a^2}$$

What happens to X when $k_a/m_a = \omega^2$?

When the absorber natural frequency is tuned to the excitation harmonic frequency, the primary system displacement $X = 0$! This means that the absorber cancels out the primary structure motion. The corresponding motion of the absorber is given by

$$X_a = -\frac{P_0}{m_a} \sin \omega t \quad (3.178)$$

The force transferred at the interface spring between the absorber and the primary structure is

$$P_a = k_a X_a = -\frac{k_a P_0}{m_a} \sin \omega t = -\frac{P_0}{m_a} \sin \omega t \quad (3.179)$$

In other words, the absorber produces a reactive force equal in magnitude and opposite in direction to the force applied to the primary structure. In practical situations, both the primary system and absorber have damping. The damping in the absorber is present to reduce excessive displacements in the absorber during resonant conditions. Such a system is more relevant to typical structural applications, and is known as a vibration damper.

Consider now the same system, but with added damping in both the primary and absorber systems.

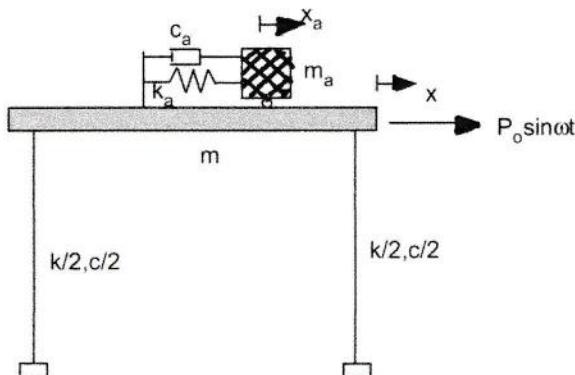


Figure 3.7 Vibration damper

The equations of motion are as follows

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_a \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{x}_a \end{Bmatrix} + \begin{bmatrix} c + c_a & -c_a \\ -c_a & c_a \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{x}_a \end{Bmatrix} + \begin{bmatrix} k + k_a & -k_a \\ -k_a & k_a \end{bmatrix} \begin{Bmatrix} x \\ x_a \end{Bmatrix} = \begin{Bmatrix} P_0 \sin \omega t \\ 0 \end{Bmatrix} \quad (3.180)$$

Since damping is present, both sine and cosine terms need to be present in the solution, and a complex exponential is the simplest analytic form which contains both these harmonics. Let the solution be of the form $j = \sqrt{-1}$

$$\begin{Bmatrix} x \\ x_a \end{Bmatrix} = \begin{Bmatrix} X \\ X_a \end{Bmatrix} (\cos \omega t + j \sin \omega t), \text{ complex plane} = \begin{Bmatrix} X \\ X_a \end{Bmatrix} e^{j\omega t} \quad (3.181)$$

where $j = \sqrt{-1}$. Substituting this in Equation 3.180, we get

$$\underbrace{\begin{bmatrix} (k + k_a - m\omega^2) + (c + c_a)\omega j & -k_a - c_a\omega j \\ -k_a - c_a\omega j & (k_a - m_a\omega^2) + c_a\omega j \end{bmatrix}}_{-\omega^2 M + j\omega C} \begin{Bmatrix} X \\ X_a \end{Bmatrix} e^{j\omega t} = \begin{Bmatrix} P_0 \\ 0 \end{Bmatrix} e^{j\omega t} \quad (3.182)$$

Solving for X and X_a yields

$$X = \frac{[(k_a - m_a\omega^2) + c_a\omega j]P_0}{\det(K - \omega^2 M + \omega j C)} \quad (3.183)$$

$$X_a = \frac{(k_a + c_a\omega j)P_0}{\det(K - \omega^2 M + \omega j C)}$$

Define the following non-dimensional parameters

$$\mu = \frac{m_a}{m} \quad \beta = \frac{\omega_a}{\omega_p} \quad r = \frac{\omega}{\omega_p} \quad (3.184)$$

$$\zeta = \frac{c_a}{2m_a\omega_p} \quad \omega_p = \sqrt{\frac{k}{m}} \quad \omega_a = \sqrt{\frac{k_a}{m_a}}$$

Assumption: that damping in primary system much smaller than damping in primary system
Generally, the primary system damping is small. For the case when $c = 0$, the expression for X in terms of the non-dimensional quantities defined above becomes

$$AF = \left| \frac{X}{X_{st}} \right| = \left| \frac{Xk}{P_0} \right| = \sqrt{\frac{4\zeta^2 r^2 + (r^2 - \beta^2)^2}{4\zeta^2 r^2(r^2 - 1 + \mu r^2)^2 + [\mu r^2 \beta^2 - (r^2 - 1)(r^2 - \beta^2)]^2}} \quad (3.185)$$

In the above equation, the design problem is one of selecting the *optimum* values of the parameters, μ , ζ , β . Closed-form analytical expressions are difficult to obtain and usually one has to resort to numerical means. However, we can make the following substitutions.

M , ζ , and β controllable.

- Practical limit on M

- Control vibration at resonance $r=1$

\hookrightarrow at resonance, $\beta=1$ (assumption) \rightarrow to design dumper, not system

Under these assumptions, the only remaining parameter to be determined is ζ .

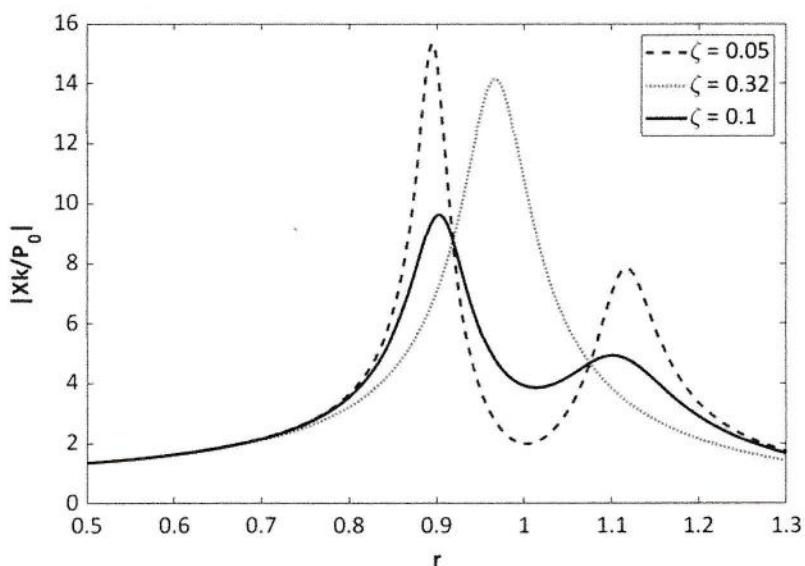


Figure 3.8 Magnitude of the primary structure response as a function of r for $\mu = 1/20, \beta = 1$

As evidenced by Figure 3.8, increasing ζ does not necessarily produce the desired reduction in the amplitude of the primary structure response. At very high values of ζ , the primary and secondary systems lock together, essentially behaving as a single system. The optimal values can be determined through several trials or through numerical approaches. This is beyond the scope of this course.

A relatively simple tuned mass damper (TMD) design can result from setting $r = \beta = 1$. Equation 3.185 then simplifies to:

$$\left| \frac{Xk}{P_0} \right| = \frac{1}{\mu} \sqrt{\frac{4\zeta^2}{1 + 4\zeta^2}} \quad \text{assuming } \beta = r = 1 \quad (3.186)$$

The maximum amplitude of the primary mass is then

$$X = \frac{P_0}{k\mu} \sqrt{\frac{4\zeta^2}{1 + 4\zeta^2}} \quad (3.187)$$

Let's see how we can use the above equation to design a TMD. First, recognize that μ is usually a small number, the mass of the damper is between 0.5-5% of the primary mass. For design, usually a target system damping is used to start the process. This is defined as the *equivalent damping ratio* for a SDOF system without a TMD. Recall from Chapter 2 that the maximum dynamic amplification factor for a harmonically excited damped SDOF systems is given by

$$AF = \frac{x_{max}}{x_{0,r}} = \frac{1}{2\zeta} = \frac{1}{2\zeta_{eq}} \quad (3.188)$$

Therefore,

$$\frac{x_{max}}{x_{0,r}} = \frac{\chi_0 t}{2\zeta_{eq}} = \frac{P_0}{2k\zeta_{eq}} \quad (3.189)$$

Equating Eq. 3.187 and Eq. 3.189,

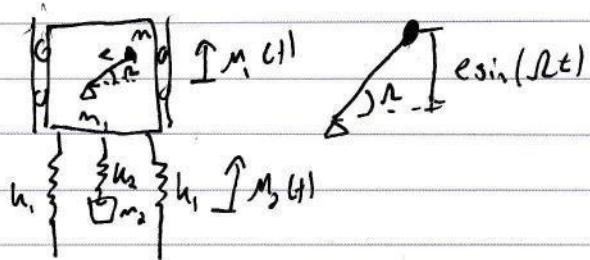
$$\zeta_{eq} = \frac{\mu}{2} \sqrt{1 + \frac{1}{4\zeta^2}} \quad (3.190)$$

As an example, if the **desired effective damping is 10%**, and the mass ratio is selected to be $\mu = 0.02$ based on practical constraints, then ζ can be calculated according to the above equation, yielding $\zeta = 0.05$.

$$\zeta \uparrow$$

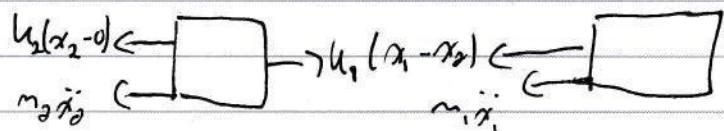
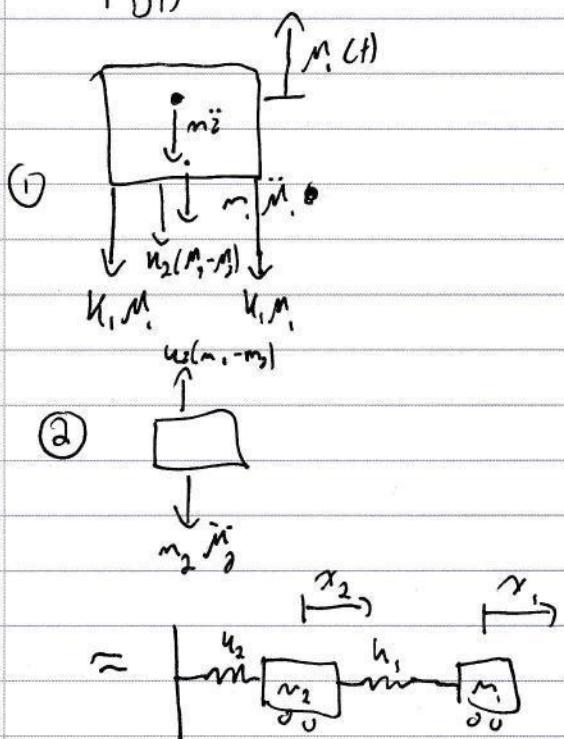
$\zeta - \text{in damper}$

(1)



for rotating mass, $\ddot{z}(t) = \dot{m}_1(t) + e \sin \omega t$
 $\ddot{z}(t) = \ddot{m}_1(t) - e \omega^2 \sin(\omega t)$

FBD



$$\textcircled{1} \quad \sum F_y = 0, \quad -m_1 \ddot{M}_1 - 2h_1 M_1 - h_2 (M_1 - M_2) - m_2 \ddot{z} = 0$$

$$= -m_1 M_1 - 2h_1 M_1 - h_2 M_1 + h_2 M_2 - m_2 \ddot{z} - m_2 (\ddot{M}_1) - e R^2 \sin 2t \\ = (m_1 + m_2) \ddot{M}_1 + 2(h_1 + h_2) M_1 - h_2 M_2 = m_2 R^2 \sin 2t$$

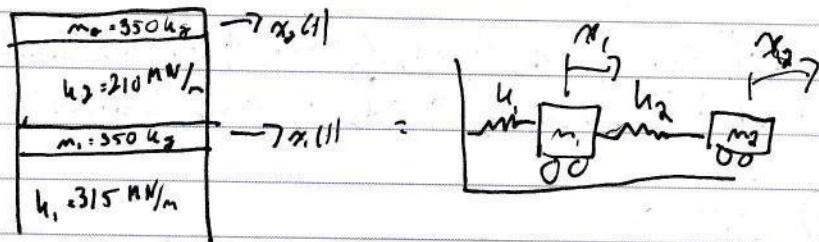
$$\textcircled{2} \quad \sum F_y = 0$$

$$m_2 \ddot{M}_2 - h_2 (M_1 - M_2) = 0, \quad m_2 \ddot{M}_2 - h_2 m_1 + h_2 m_2 = 0$$

, for \textcircled{1},

$$\begin{bmatrix} m_1 + m_2 & 0 \\ 0 & m_2 \end{bmatrix} \begin{bmatrix} \ddot{M}_1 \\ \ddot{M}_2 \end{bmatrix} + \begin{bmatrix} 2h_1 + h_2 & -h_2 \\ -h_2 & h_2 \end{bmatrix} \begin{bmatrix} M_1 \\ M_2 \end{bmatrix} = \begin{bmatrix} m_2 R^2 \sin 2t \\ 0 \end{bmatrix}$$

Question 2 - 2 story shear building



a) Equation of motion

$$m_1 \ddot{x}_1 + k_1 x_1 - k_2 (x_2 - x_1) = 0, \quad m_2 \ddot{x}_2 - k_2 (x_2 - x_1) + k_2 x_2 = 0$$

$$m_1 \ddot{x}_1 + k_1 x_1 - k_2 (x_2 - x_1) = 0$$

$$m_1 \ddot{x}_1 + (k_1 + k_2) x_1 - k_2 x_2 = 0$$

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

M K

b) Natural frequencies and mode shapes

$$\omega_n^2 = k/m \text{ (SDOF)}$$

$$(-\omega_n^2 m + k) \theta_m = 0$$

$\theta_m \neq 0$, \therefore determine

$$\det |K - \omega_n^2 M| = 0$$

$$10^3 \begin{bmatrix} 350 & 0 \\ 0 & 250 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + 10^6 \begin{bmatrix} 525 & -210 \\ -210 & 210 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

$$\begin{vmatrix} 525000 - 350\omega_n^2 & -210000 \\ -210000 & 210000 - 350\omega_n^2 \end{vmatrix}$$

$$\therefore \omega_n = 300, 1800$$

@ 300, $\therefore \omega_{n,1} = 17.321 \text{ rad/sec}, T_{n,1} = 0.363 \text{ (Fundamental Period)}$

$$@ 1800, \omega_n = 42.426 \text{ rad/sec}, T_{n,2} = 0.1419$$

For Mode 1, plug in 300

$$\begin{vmatrix} 525000 - 350(300) & -210000 \\ -210000 & 210000 - 350(300) \end{vmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = 0, \text{ letting } x_1 = 1$$

$$\phi_{n,1} = \begin{pmatrix} 1/2 \\ 1/2 \end{pmatrix}$$

For Mode 2, letting $\frac{\omega_n^2}{m} = 1800, \phi_{n,2} = \begin{pmatrix} 1/2 \\ -1/2 \end{pmatrix}$

c) Mass normalize the shape modes

$$(\omega \phi_{n,:}^T) M (\omega \phi_{n,:}) = 1$$

$$\omega^2 \phi_{n,:}^T M \phi_{n,:} = 1$$

$$\text{Mode 1}, \omega_1^2 \phi_{n,1}^T M \phi_{n,1} = 1$$

$$\omega_1^2 [1 \ 2] 10^3 \begin{bmatrix} 350 & 0 \\ 0 & 350 \end{bmatrix} \begin{bmatrix} 1 \\ 2 \end{bmatrix} = 1$$

$$1750 \omega_1^2 = 1, \omega_1 = 0.0239$$

$$\text{Mode 2}, \omega_2^2 [1 \ -0.5] 10^3 \begin{bmatrix} 350 & 0 \\ 0 & 350 \end{bmatrix} \begin{bmatrix} 1 \\ -0.5 \end{bmatrix} = 1$$

$$437.5 \omega_2^2 = 1, \omega_2 = 0.0478$$

$$\therefore \phi_n = \begin{bmatrix} \phi_{n,1} & \phi_{n,2} \\ d_1(1) & \omega_2(1) \\ d_1(2) & \omega_2(-0.5) \end{bmatrix}$$

$$= \begin{bmatrix} 0.0239 & 0.0478 \\ 0.0478 & -0.0239 \end{bmatrix}$$