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47.1 Fundamental Principles

The main purpose of *structural analysis* is to determine forces and deformations of the structure due to applied loads. *Structural design* involves form finding, determination of loadings, and proportioning of structural members and components in such a way that the assembled structure is capable of supporting the loads within the design limit states. An analytical model is an idealization of the actual structure. The structural model should relate the actual behavior to material properties, structural details, loading, and boundary conditions as accurately as is practicable.

Structures often appear in three-dimensional forms. For structures that have a regular layout and are rectangular in shape, subject to symmetric loads, it is possible to idealize them into two-dimensional frames arranged in orthogonal directions. A structure is said to be two-dimensional or planar if all the members lie in the same plane. *Joints* in a structure are those points where two or more members are connected. Beams are members subjected to loading acting transversely to their longitudinal axis and creating flexural bending only. *Ties* are members that are subjected to axial tension only, while struts (columns or posts) are members subjected to resist only axial forces. A structural system in which joints are capable of transferring end moments is called a *frame*. Members in this system are assumed to be capable of resisting bending moments, axial force, and shear force.

Boundary Conditions

A *hinge* or *pinned joint* does not allow translational movements (Fig. 47.1a). It is assumed to be frictionless and to allow rotation of a member with respect to the others. A *roller* permits the attached structural part to rotate freely with respect to the rigid surface and to translate freely in the direction parallel to the surface (Fig. 47.1b). Translational movement in any other direction is not allowed. A *fixed support* (Fig. 47.1c) does not allow rotation or translation in any direction. A *rotational spring* provides some



FIGURE 47.1 Various boundary conditions.

rotational restraint but does not provide any translational restraint (Fig. 47.1d). A *translational spring* can provide partial restraints along the direction of deformation (Fig. 47.1e).

Loads and Reactions

Loads that are of constant magnitude and remain in the original position are called *permanent loads*. They are also referred to as *dead loads*, which may include the self weight of the structure and other loads, such as walls, floors, roof, plumbing, and fixtures that are permanently attached to the structure. Loads that may change in position and magnitude are called *variable loads*. They are commonly referred to as live or imposed loads, which may include those caused by construction operations, wind, rain, earthquakes, snow, blasts, and temperature changes, in addition to those that are movable, such as furniture and warehouse materials.

Ponding loads are due to water or snow on a flat roof that accumulates faster than it runs off. Wind loads act as pressures on windward surfaces and pressures or suctions on leeward surfaces. Impact loads are caused by suddenly applied loads or by the vibration of moving or movable loads. They are usually taken as a fraction of the live loads. Earthquake loads are those forces caused by the acceleration of the ground surface during an earthquake.

A structure that is initially at rest and remains at rest when acted upon by applied loads is said to be in a state of *equilibrium*. The resultant of the external loads on the body and the supporting forces or reactions is zero. If a structure is to be in equilibrium under the action of a system of loads, it must satisfy the six static equilibrium equations:

$$\Sigma F_x = 0, \quad \Sigma F_y = 0, \quad \Sigma F_z = 0$$

$$\Sigma M_x = 0, \quad \Sigma M_y = 0, \quad \Sigma M_z = 0$$
(47.1)

The summation in these equations is for all the components of the forces (F) and of the moments (M) about each of the three axes x, y, and z. If a structure is subjected to forces that lie in one plane, say x-y, the above equations are reduced to:

$$\Sigma F_x = 0, \quad \Sigma F_y = 0, \quad \Sigma M_z = 0$$
 (47.2)

Consider a beam under the action of the applied loads, as shown in Fig. 47.2a. The reaction at support B must act perpendicular to the surface on which the rollers are constrained to roll upon. The support



FIGURE 47.2 Beam in equilibrium.

reactions and the applied loads, which are resolved in vertical and horizontal directions, are shown in Fig. 47.2b.

From geometry, it can be calculated that $B_y = \sqrt{3}B_x 1$. Equation (47.2) can be used to determine the magnitude of the support reactions. Taking moment about B gives

$$10A_v - 346.4 \times 5 = 0$$

from which

 $A_v = 173.2 \text{ kN}$

Equating the sum of vertical forces, ΣF_{v} , to zero gives

 $173.2 + B_v - 346.4 = 0$

and hence we get

 $B_v = 173.2 \text{ kN}$

Therefore

$$B_x = B_y / \sqrt{3} = 100 \text{ kN}.$$

Equilibrium in the horizontal direction, $\Sigma F_x = 0$, gives

 $A_x - 200 - 100 = 0$

and hence

 $A_{x} = 300 \text{ kN}$

There are three unknown reaction components at a fixed end, two at a hinge, and one at a roller. If, for a particular structure, the total number of unknown reaction components equal the number of equations available, the unknowns may be calculated from the equilibrium equations, and the structure is then said to be *statically determinate externally*. Should the number of unknowns be greater than the number of equations available, the structure is *statically indeterminate externally*; if less, it is *unstable externally*. The ability of a structure to support adequately the loads applied to it is dependent not only on the number of reaction components but also on the arrangement of those components. It is possible for a structure to have as many or more reaction components than there are equations available and yet be unstable. This condition is referred to as *geometric instability*.

Principle of Superposition

The principle states that if the structural behavior is linearly elastic, the forces acting on a structure may be separated or divided into any convenient fashion and the structure analyzed for the separate cases. The final results can be obtained by adding up the individual results. This is applicable to the computation of structural responses such as moment, shear, deflection, etc.

However, there are two situations where the principle of superposition cannot be applied. The first case is associated with instances where the geometry of the structure is appreciably altered under load. The second case is in situations where the structure is composed of a material in which the stress is not linearly related to the strain.

47.2 Beams

One of the most common structural elements is a *beam*; it bends when subjected to loads acting transversely to its centroidal axis or sometimes by loads acting both transversely and parallel to this axis. The discussions given in the following subsections are limited to straight beams in which the centroidal axis is a straight line with a shear center coinciding with the centroid of the cross-section. It is also assumed that all the loads and reactions lie in a simple plane that also contains the centroidal axis of the flexural member and the principal axis of every cross-section. If these conditions are satisfied, the beam will simply bend in the plane of loading without twisting.

Relation between Load, Shear Force, and Bending Moment

Shear force at any transverse cross-section of a straight beam is the algebraic sum of the components acting transverse to the axis of the beam of all the loads and reactions applied to the portion of the beam on either side of the cross-section. *Bending moment* at any transverse cross-section of a straight beam is the algebraic sum of the moments, taken about an axis passing through the centroid of the cross-section. The axis about which the moments are taken is, of course, normal to the plane of loading.

When a beam is subjected to transverse loads, there exist certain relationships between load, shear force, and bending moment. Let us consider the beam shown in Fig. 47.3 subjected to some arbitrary loading, p. Let S and M be the shear and bending moment, respectively, for any point m at a distance x, which is measured from A, being positive when measured to the right. Corresponding values of the shear and bending moment at point n at a differential distance dx to the right of m are S + dS and M + dM, respectively. It can be shown, neglecting the second order quantities, that

$$p = \frac{dS}{dx}$$
(47.3)

and





FIGURE 47.3 Beam under arbitrary loading.

Equation (47.3) shows that the rate of change of shear at any point is equal to the intensity of load applied to the beam at that point. Therefore, the difference in shear at two cross-sections C and D is

$$S_{\rm D} - S_{\rm C} = \int_{X_{\rm c}}^{X_{\rm D}} p \, dx$$
 (47.5)

We can write this in the same way for moment as

$$M_{\rm D} - M_{\rm C} = \int_{x_{\rm C}}^{x_{\rm D}} S \, \mathrm{dx}$$
 (47.6)

Shear Force and Bending Moment Diagrams

In order to plot the shear force and bending moment diagrams, it is necessary to adopt a sign convention for these responses. A shear force is considered to be positive if it produces a clockwise moment about a point in the free body on which it acts. A negative shear force produces a counterclockwise moment about the point. The bending moment is taken as positive if it causes compression in the upper fibers of the beam and tension in the lower fiber. In other words, a sagging moment is positive and a hogging moment is negative. The construction of these diagrams is explained with an example given in Fig. 47.4.

Section E of the beam is in equilibrium under the action of applied loads and internal forces acting at E, as shown in Fig. 47.5. There must be an internal vertical force and internal bending moment to maintain equilibrium at section E. The vertical force or the moment can be obtained as the algebraic sum of all forces or the algebraic sum of the moment of all forces that lie on either side of section E.

The shear on a cross-section an infinitesimal distance to the right of point A is +55, and therefore the shear diagram rises abruptly from zero to +55 at this point. In portion AC, since there is no additional load, the shear remains +55 on any cross-section throughout this interval, and the diagram is a horizontal, as shown in Fig. 47.4. An infinitesimal distance to the left of C the shear is +55, but an infinitesimal



FIGURE 47.4 Bending moment and shear force diagrams.



FIGURE 47.5 Internal forces.

distance to the right of this point the concentrated load of magnitude 30 has caused the shear to be reduced to +25. Therefore, at point C, there is an abrupt change in the shear force from +55 to +25. In the same manner, the shear force diagram for portion CD of the beam remains a rectangle. In portion DE, the shear on any cross-section a distance x from point D is

$$S = 55 - 30 - 4x = 25 - 4x$$

which indicates that the shear diagram in this portion is a straight line decreasing from an ordinate of +25 at D to +1 at E. The remainder of the shear force diagram can easily be verified in the same way. It should be noted that, in effect, a concentrated load is assumed to be applied at a point, and hence, at such a point the ordinate to the shear diagram changes abruptly by an amount equal to the load.

In portion AC, the bending moment at a cross-section a distance x from point A is M = 55x. Therefore, the bending moment diagram starts at zero at A and increases along a straight line to an ordinate of +165 at point C. In portion CD, the bending moment at any point a distance x from C is M = 55(x + 3) - 30x. Hence, the bending moment diagram in this portion is a straight line increasing from 165 at C to 265 at D. In portion DE, the bending moment at any point a distance x from D is $M = 55(x + 7) - 30(X + 4) - 4x^2/22$. Hence, the bending moment diagram in this portion is a curve with an ordinate of 265 at D and 343 at E. In an analogous manner, the remainder of the bending moment diagram can easily be constructed.

Bending moment and shear force diagrams for beams with simple boundary conditions and subject to some selected load cases are given in Fig. 47.6.

Fixed-End Beams

When the ends of a beam are held so firmly that they are not free to rotate under the action of applied loads, the beam is known as a built-in or fixed-end beam and it is statically indeterminate. The bending moment diagram for such a beam can be considered to consist of two parts viz. the free bending moment diagram obtained by treating the beam as if the ends are simply supported and the fixing moment diagram resulting from the restraints imposed at the ends of the beam. The solution of a fixed beam is greatly simplified by considering Mohr's principles, which state that:

- 1. The area of the fixing bending moment diagram is equal to that of the free bending moment diagram.
- 2. The centers of gravity of the two diagrams lie in the same vertical line, i.e., are equidistant from a given end of the beam.

The construction of the bending moment diagram for a fixed beam is explained with an example shown in Fig. 47.7. P Q U T is the free bending moment diagram, M_s , and P Q R S is the fixing moment diagram, M_i . The net bending moment diagram, M, is shaded. If A_s is the area of the free bending moment diagram and A_i the area of the fixing moment diagram, then from the first Mohr's principle we have $A_s = A_i$ and

$$\frac{1}{2} \times \frac{Wab}{L} \times L = \frac{1}{2} (M_A + M_B) \times L$$

$$M_A + M_B = \frac{Wab}{L}$$
(47.7)

From the second principle, equating the moment about A of As and Ai, we have

$$M_{A} + 2M_{B} = \frac{Wab}{L^{3}} \left(2a^{2} + 3ab + b^{2} \right)$$
(47.8)



FIGURE 47.6 Shear force and bending moment diagrams for beams with simple boundary conditions subjected to selected loading cases.

Solving Eqs. (47.7) and (47.8) for M_{A} and M_{B} we get

$$M_{A} = \frac{Wab^{2}}{L^{2}}$$
$$M_{B} = \frac{Wa^{2}b}{L^{2}}$$



FIGURE 47.6 (continued).

Shear force can be determined once the bending moment is known. The shear force at the ends of the beam, i.e., at A and B, are

$$S_{A} = \frac{M_{A} - M_{B}}{L} + \frac{Wb}{L}$$
$$S_{B} = \frac{M_{B} - M_{A}}{L} + \frac{Wa}{L}$$

Bending moment and shear force diagrams for fixed-end beams subjected to some typical loading cases are shown in Fig. 47.8.



FIGURE 47.6 (continued).



FIGURE 47.7 Fixed-end beam.

Continuous Beams

Continuous beams like fixed-end beams are statically indeterminate. Bending moments in these beams are functions of the geometry, moments of inertia, and modulus of elasticity of individual members, besides the load and span. They may be determined by Clapeyron's theorem of three moments, the moment distribution method, or the slope deflection method.

An example of a two-span continuous beam is solved by Clapeyron's theorem of three moments. The theorem is applied to two adjacent spans at a time, and the resulting equations in terms of unknown support moments are solved. The theorem states that



FIGURE 47.8 Shear force and bending moment diagrams for built-up beams subjected to typical loading cases.

$$M_{A}L_{1} + 2M_{B}(L_{1} + L_{2}) + M_{C}L_{2} = 6\left(\frac{A_{1}x_{1}}{L_{1}} + \frac{A_{2}x_{2}}{L_{2}}\right)$$
(47.9)

in which M_A , M_B , and M_C are the hogging moment at supports A, B, and C, respectively, of two adjacent spans of length L_1 and L_2 (Fig. 47.9); A_1 and A_2 are the area of bending moment diagrams produced by the vertical loads on the simple spans AB and BC, respectively; x_1 is the centroid of A_1 from A; and x_2 is the distance of the centroid of A_2 from C. If the beam section is constant within a span but remains different for each of the spans Eq. (47.9) can be written as

LOADING	SHEAR FORCE	BENDING MOMENT
$\begin{array}{c c} \downarrow^{P} & \downarrow^{P} & \downarrow^{P} \\ \hline A & C & D & E & B \\ \hline \downarrow_{L/6} \downarrow_{\leftarrow} L/3 \rightarrow \downarrow_{\leftarrow} L/3 \rightarrow \downarrow L/6 & \downarrow_{\leftarrow} \end{array}$	R_A R_B	$M_{A} \underbrace{PL/4}_{M_{A}} = M_{B} = -19PL/72$
$\begin{array}{c} \downarrow^{P} \downarrow^{P} \downarrow^{P} \downarrow^{P} \\ A \xrightarrow{C} \xrightarrow{D} \xrightarrow{E} B \\ \downarrow^{a} \downarrow^{/4} + \mathbf{e} \downarrow/4 + \mathbf{e} \downarrow/4$	R_A R_B $R_B = 3P/2$	$M_{A} = M_{B} = -5PL/16$
$ \begin{array}{c} \downarrow P \downarrow $	R_A R_B	$M_{D} = M_{E} = 5PL/32$ $M_{A} = M_{B} = -11PL/32$

FIGURE 47.8 (continued).



FIGURE 47.9 Continuous beams.

$$M_{A}\frac{L_{1}}{I_{1}} + 2M_{B}\left(\frac{L_{1}}{I_{1}} + \frac{L_{2}}{I_{2}}\right) + M_{C}\frac{L_{2}}{I_{2}} = 6\left(\frac{A_{1}x_{1}}{L_{1}I_{1}} + \frac{A_{2}x_{2}}{L_{2}I_{2}}\right)$$
(47.10)

in which I_1 and I_2 are the moments of inertia of the beam sections in spans L_1 and L_2 , respectively.

Example 47.1

The example in Fig. 47.10 shows the application of this theorem. For spans AC and BC

$$M_{A} \times 10 + 2M_{C} (10 + 10) + M_{B} \times 10 = 6 \left[\frac{\frac{1}{2} \times 500 \times 10 \times 5}{10} + \frac{\frac{1}{2} \times 250 \times 10 \times 5}{10} \right]$$

Since the support at A is simply supported, $M_A = 0$. Therefore,

$$4M_{\rm C} + M_{\rm B} = 1250 \tag{47.11}$$

Considering an imaginary span BD on the right side of B and applying the theorem for spans CB and BD



Spans AC and BC

FIGURE 47.10 Example of a continuous beam.

$$M_{\rm C} \times 10 + 2M_{\rm B}(10) + M_{\rm D} \times 10 = 6 \times \frac{\frac{2}{3} \times 10 \times 5}{10} \times 2$$

$$M_{\rm C} + 2M_{\rm B} = 500 \qquad (\because M_{\rm C} - M_{\rm D})$$
(47.12)

Solving Eqs. (47.11) and (47.12) we get

$$M_{B} = 107.2 \text{ kNm}$$

 $M_{C} = 285.7 \text{ kNm}$

Shear force at A is

$$S_{A} = \frac{M_{A} - M_{C}}{L} + 100 = -28.6 + 100 = 71.4 \text{ kN}$$

Shear force at C is

$$S_{\rm C} = \left(\frac{M_{\rm C} - M_{\rm A}}{L} + 100\right) + \left(\frac{M_{\rm C} - M_{\rm B}}{L} + 100\right)$$
$$= (28.6 + 100) + (17.9 + 100) = 246.5 \,\rm kN$$

Shear force at B is

$$S_{\rm B} = \left(\frac{M_{\rm B} - M_{\rm C}}{L} + 100\right)$$
$$= -17.9 + 100 = 82.1 \,\rm kN$$

The bending moment and shear force diagrams are shown in Fig. 47.10.

Beam Deflection

There are several methods for determining beam deflections: (1) moment area method, (2) conjugate beam method, (3) virtual work, and (4) Castigliano's second theorem, among others.

The elastic curve of a member is the shape the neutral axis takes when the member deflects under load. The inverse of the radius of curvature at any point of this curve is obtained as

$$\frac{1}{R} = \frac{M}{EI}$$
(47.13)

in which M is the bending moment at the point and EI the flexural rigidity of the beam section. Since the deflection is small, 1/R is approximately taken as d^2y/dx^2 , and Eq. (47.13) may be rewritten as:

$$M = EI \frac{d^2 y}{d x^2}$$
(47.14)

In Eq. (47.14), y is the deflection of the beam at distance x measured from the origin of coordinate. The change in slope in a distance dx can be expressed as M dx/EI, and hence the slope in a beam is obtained as

$$\theta_{\rm B} - \theta_{\rm A} = \int_{\rm A}^{\rm B} \frac{M}{\rm EI} d\mathbf{x}$$
(47.15)

Equation (47.15) may be stated: the change in slope between the tangents to the elastic curve at two points is equal to the area of the M/EI diagram between the two points.

Once the change in slope between tangents to the elastic curve is determined, the deflection can be obtained by integrating further the slope equation. In a distance dx the neutral axis changes in direction by an amount d θ . The deflection of one point on the beam with respect to the tangent at another point due to this angle change is equal to $d\delta = x d\theta$, where x is the distance from the point at which deflection is desired to the particular differential distance.

To determine the total deflection from the tangent at one point, A, to the tangent at another point, B, on the beam, it is necessary to obtain a summation of the products of each $d\theta$ angle (from A to B) times the distance to the point where deflection is desired, or

$$\delta_{\rm B} - \delta_{\rm A} = \int_{\rm A}^{\rm B} \frac{Mx \, dx}{\rm EI} \tag{47.16}$$

The deflection of a tangent to the elastic curve of a beam with respect to a tangent at another point is equal to the moment of M/EI diagram between the two points, taken about the point at which deflection is desired.

Moment Area Method

The moment area method is most conveniently used for determining slopes and deflections for beams in which the direction of the tangent to the elastic curve at one or more points is known, such as cantilever beams, where the tangent at the fixed end does not change in slope. The method is applied easily to beams loaded with concentrated loads, because the moment diagrams consist of straight lines. These diagrams can be broken down into single triangles and rectangles. Beams supporting uniform loads or uniformly varying loads may be handled by integration. Properties of some of the shapes of M/EI diagrams that designers usually come across are given in Fig. 47.11.

It should be understood that the slopes and deflections obtained using the moment area theorems are with respect to tangents to the elastic curve at the points being considered. The theorems do not directly give the slope or deflection at a point in the beam compared to the horizontal axis (except in one or two special cases); they give the change in slope of the elastic curve from one point to another or the deflection of the tangent at one point with respect to the tangent at another point. There are some special cases in which beams are subjected to several concentrated loads or the combined action of concentrated and uniformly distributed loads. In such cases it is advisable to separate the concentrated loads and uniformly



FIGURE 47.11 Typical M/EI diagram.



FIGURE 47.12 Deflection — simply supported beam under UDL.

distributed loads, and the moment area method can be applied separately to each of these loads. The final responses are obtained by the principle of superposition.

For example, consider a simply supported beam subjected to uniformly distributed load q, as shown in Fig. 47.12. The tangent to the elastic curve at each end of the beam is inclined. The deflection, δ_1 , of the tangent at the left end from the tangent at the right end is found as ql⁴/24EI. The distance from the original chord between the supports and the tangent at the right end, δ_2 , can be computed as ql⁴/48EI.



FIGURE 47.13 Bending of curved beams.

The deflection of a tangent at the center from a tangent at the right end, δ_3 , is determined as ql⁴/128EI. The difference between δ_2 and δ_3 gives the centerline deflection as (5/384) x (ql⁴/EI).

Curved Beams

The beam formulas derived in the previous section are based on the assumption that the member to which bending moment is applied is initially straight. Many members, however, are curved before a bending moment is applied to them. Such members are called curved beams. In the following discussion all the conditions applicable to straight-beam formulas are assumed valid, except that the beam is initially curved.

Let the curved beam DOE shown in Fig. 47.13 be subjected to the load Q. The surface in which the fibers do not change in length is called the neutral surface. The total deformations of the fibers between two normal sections, such as AB and A_1B_1 , are assumed to vary proportionally with the distances of the fibers from the neutral surface. The top fibers are compressed, while those at the bottom are stretched, i.e., the plane section before bending remains plane after bending.

In Fig. 47.13 the two lines AB and A_1B_1 are two normal sections of the beam before the loads are applied. The change in the length of any fiber between these two normal sections after bending is represented by the distance along the fiber between the lines A_1B_1 and A'B'; the neutral surface is represented by NN₁, and the stretch of fiber PP₁ is $P_1P'_1$, etc. For convenience, it will be assumed that line AB is a line of symmetry and does not change direction.

The total deformations of the fibers in the curved beam are proportional to the distances of the fibers from the neutral surface. However, the strains of the fibers are not proportional to these distances because the fibers are not of equal length. Within the elastic limit the stress on any fiber in the beam is proportional to the strain of the fiber, and hence the elastic stresses in the fibers of a curved beam are not proportional to the distances of the fibers from the neutral surface. The resisting moment in a curved beam, therefore, is not given by the expression σ I/c. Hence the neutral axis in a curved beam does not pass through the centroid of the section. The distribution of stress over the section and the relative position of the neutral axis are shown in Fig. 47.13b; if the beam were straight, the stress would be zero at the centroidal axis and would vary proportionally with the distance from the centroidal axis, as indicated by the dot–dash line in the figure. The stress on a normal section such as AB is called the circumferential stress.

Sign Conventions

The bending moment M is positive when it decreases the radius of curvature and negative when it increases the radius of curvature; y is positive when measured toward the convex side of the beam and negative when measured toward the concave side, that is, toward the center of curvature. With these sign conventions, σ is positive when it is a tensile stress.

Circumferential Stresses

Figure 47.14 shows a free-body diagram of the portion of the body on one side of the section; the equations of equilibrium are applied to the forces acting on this portion. The equations obtained are



FIGURE 47.14 Free-body diagram of curved beam segment.



FIGURE 47.15 Curvature in a curved beam.

$$\Sigma F_{z} = 0 \quad \text{or} \quad \int \sigma da = 0 \tag{47.17}$$

$$\Sigma M_z = 0 \quad \text{or} \quad M = \int y \sigma da$$
 (47.18)

Figure 47.15 represents the part ABB_1A_1 of Fig. 47.13a enlarged; the angle between the two sections AB and A_1B_1 is d θ . The bending moment causes the plane A_1B_1 to rotate through an angle $\Delta d\theta$, thereby changing the angle this plane makes with the plane BAC from d θ to $(d\theta + \Delta d\theta)$; the center of curvature is changed from C to C', and the distance of the centroidal axis from the center of curvature is changed from R to ρ . It should be noted that y, R, and ρ at any section are measured from the centroidal axis and not from the neutral axis.

It can be shown that the bending stress σ is given by the relation

$$\sigma = \frac{M}{aR} \left(1 + \frac{1}{Z} \frac{y}{R+y} \right)$$
(47.19)

in which

$$Z = -\frac{1}{a} \int \frac{y}{R+y} da$$

 σ is the tensile or compressive (circumferential) stress at a point at distance y from the centroidal axis of a transverse section at which the bending moment is M; R is the distance from the centroidal axis of the section to the center of curvature of the central axis of the unstressed beam; a is the area of the cross-section; and Z is a property of the cross-section, the values of which can be obtained from the expressions for various areas given in Fig. 47.17. Detailed information can be obtained from Seely and Smith (1952).

Example 47.2

The bent bar shown in Fig. 47.16 is subjected to a load P = 1780 N. Calculate the circumferential stress at A and B, assuming that the elastic strength of the material is not exceeded. We know from Eq. (47.19)

$$\sigma = \frac{P}{a} + \frac{M}{aR} \left(1 + \frac{1}{Z} \frac{y}{R+y} \right)$$

in which a = the area of rectangular section ($40 \times 12 = 480 \text{ mm}^2$)

R = 40 mm $y_A = -20$ $y_B = +20$ P = 1780 N M = -1780 × 120 = -213,600 N mm.

From Table 47.2.1, for rectangular section

$$Z = -1 + \frac{R}{h} \left[\log_{e} \frac{R+c}{R-c} \right]$$

h = 40 mm
c = 20 mm

Hence,



FIGURE 47.16 Bent bar.

FIGURE 47.17 Analytical expressions for Z.

Therefore

$$\sigma_{\rm A} = \frac{1780}{480} + \frac{-213600}{480 \times 40} \left(1 + \frac{1}{0.0986} \ \frac{-20}{40 - 20} \right) = 105.4 \text{ N/mm}^2 \text{ (tensile)}$$

$$\sigma_{\rm B} = \frac{1780}{480} + \frac{-213600}{480 \times 40} \left(1 + \frac{1}{0.0986} \ \frac{20}{40 + 20} \right) = -45 \text{ N/mm}^2 \text{ (compressive)}$$

47.3 Trusses

A structure that is composed of a number of members pin-connected at their ends to form a stable framework is called a truss. If all the members lie in a plane, it is a planar truss. It is generally assumed that loads and reactions are applied to the truss only at the joints. The centroidal axis of each member is straight, coincides with the line connecting the joint centers at each end of the member, and lies in a plane that also contains the lines of action of all the loads and reactions. Many truss structures are three-dimensional in nature. However, in many cases, such as bridge structures and simple roof systems, the three-dimensional framework can be subdivided into planar components for analysis as planar trusses



FIGURE 47.17 (continued).

without seriously compromising the accuracy of the results. Figure 47.18 shows some typical idealized planar truss structures.

There exists a relation between the number of members, m, the number of joints, j, and the reaction components, r. The expression is



FIGURE 47.18 Typical planar trusses.

$$m = 2j - r$$
 (47.20)

which must be satisfied if it is to be statically determinate internally. r is the least number of reaction components required for external stability. If m exceeds (2j - r), then the excess members are called redundant members, and the truss is said to be statically indeterminate.

For a statically determinate truss, member forces can be found by using the method of equilibrium. The process requires repeated use of free-body diagrams from which individual member forces are determined. The *method of joints* is a technique of truss analysis in which the member forces are determined by the sequential isolation of joints — the unknown member forces at one joint are solved and become known for the subsequent joints. The other method is known as *method of sections*, in which equilibrium of a part of the truss is considered.

Method of Joints

An imaginary section may be completely passed around a joint in a truss. The joint has become a free body in equilibrium under the forces applied to it. The equations $\Sigma H = 0$ and $\Sigma V = 0$ may be applied to the joint to determine the unknown forces in members meeting there. It is evident that no more than two unknowns can be determined at a joint with these two equations.

Example 47.3

A truss shown in Fig. 47.19 is symmetrically loaded and is sufficient to solve half the truss by considering joints 1–5. At joint 1, there are two unknown forces. Summation of the vertical components of all forces at joint 1 gives

$$135 - F_{12} \sin 45^\circ = 0$$

which in turn gives the force in members 1 and 2, $F_{12} = 190$ kN (compressive). Similarly, summation of the horizontal components gives

$$F_{13} - F_{12} \cos 45^\circ = 0$$





FIGURE 47.19 Example of the method of joints, planar truss.

Substituting for F_{12} gives the force in member 1–3 as

 $F_{13} = 135$ kN (tensile)

Now, joint 2 is cut completely, and it is found that there are two unknown forces F_{25} and F_{23} . Summation of the vertical components gives

$$F_{12} \cos 45^{\circ} - F_{23} = 0$$

Therefore

$$F_{23} = 135$$
 kN (tensile)

Summation of the horizontal components gives

$$F_{12} \sin 45^{\circ} - F_{25} = 0$$

and hence

$$F_{25} = 135$$
 kN (compressive)

After solving for joints 1 and 2, one proceeds to take a section around joint 3 at which there are now two unknown forces viz. F_{34} and F_{35} . Summation of the vertical components at joint 3 gives

$$F_{23} - F_{35}\sin 45^\circ - 90 = 0$$

Substituting for F_{23} , one obtains $F_{35} = 63.6$ kN (compressive). Summing the horizontal components and substituting for F_{13} one gets

$$-135 - 45 + F_{34} = 0$$

Therefore,

$$F_{34} = 180 \text{ kN} \text{ (tensile)}$$

The next joint involving two unknowns is joint 4. When we consider a section around it, the summation of the vertical components at joint 4 gives

$$F_{45} = 90 \text{ kN} \text{ (tensile)}$$

Now, the forces in all the members on the left half of the truss are known, and by symmetry the forces in the remaining members can be determined. The forces in all the members of a truss can also be determined by using the method of sections.

Method of Sections

In this method, an imaginary cutting line called section is drawn through a stable and determinate truss. Thus, a section divides the truss into two separate parts. Since the entire truss is in equilibrium, any part of it must also be in equilibrium. Either of the two parts of the truss can be considered, and the three equations of equilibrium $\Sigma F_x = 0$, $\Sigma F_y = 0$, and $\Sigma M = 0$ can be applied to solve for member forces.

Example 47.3 above (Fig. 47.20) is once again considered. To calculate the force in members 3–5, F_{35} , section AA should be run to cut members 3–5 as shown in the figure. It is required only to consider the equilibrium of one of the two parts of the truss. In this case, the portion of the truss on the left of the section is considered. The left portion of the truss as shown in Fig. 47.20 is in equilibrium under the action of the forces viz. the external and internal forces. Considering the equilibrium of forces in the vertical direction, one can obtain

$$135 - 90 + F_{35} \sin 45^\circ = 0$$

Therefore, F₃₅ is obtained as

$$F_{35} = -45\sqrt{2} \text{ kN}$$

The negative sign indicates that the member force is compressive. The other member forces cut by the section can be obtained by considering the other equilibrium equations viz. $\Sigma M = 0$. More sections can be taken in the same way to solve for other member forces in the truss. The most important advantage of this method is that one can obtain the required member force without solving for the other member forces.

Compound Trusses

A compound truss is formed by interconnecting two or more simple trusses. Examples of compound trusses are shown in Fig. 47.21. A typical compound roof truss is shown in Fig. 47.21a in which two simple trusses are interconnected by means of a single member and a common joint. The compound truss shown in Fig. 47.21b is commonly used in bridge construction, and in this case, three members are used to interconnect two simple trusses at a common joint. There are three simple trusses interconnected at their common joints, as shown in Fig. 47.21c.



FIGURE 47.20 Example of the method of sections, planar truss.



FIGURE 47.21 Compound truss.

The method of sections may be used to determine the member forces in the interconnecting members of compound trusses, similar to those shown in Fig. 47.21a and b. However, in the case of a cantilevered truss the middle simple truss is isolated as a free-body diagram to find its reactions. These reactions are reversed and applied to the interconnecting joints of the other two simple trusses. After the interconnecting forces between the simple trusses are found, the simple trusses are analyzed by the method of joints or the method of sections.

47.4 Frames

Frames are statically indeterminate in general; special methods are required for their analysis. Slope deflection and moment distribution methods are two such methods commonly employed. Slope deflection is a method that takes into account the flexural displacements such as rotations and deflections and involves solutions of simultaneous equations. Moment distribution, on the other hand, involves successive cycles of computation, each cycle drawing closer to the "exact" answers. The method is more labor intensive but yields accuracy equivalent to that obtained from the "exact" methods.

Slope Deflection Method

This method is a special case of the stiffness method of analysis. It is a convenient method for performing hand analysis of small structures.

Let us consider a prismatic frame member AB with undeformed position along the x axis deformed into configuration p, as shown in Fig. 47.22. Moments at the ends of frame members are expressed in terms of the rotations and deflections of the joints. It is assumed that the joints in a structure may rotate or deflect, but the angles between the members meeting at a joint remain unchanged. The positive axes, along with the positive member-end force components and displacement components, are shown in the figure.



FIGURE 47.22 Deformed configuration of a beam.

The equations for end moments may be written as

$$M_{AB} = \frac{2 \text{EI}}{\ell} (2\theta_A + \theta_B - 3\psi_{AB}) + M_{FAB}$$

$$M_{BA} = \frac{2 \text{EI}}{\ell} (2\theta_B + \theta_A - 3\psi_{AB}) + M_{FBA}$$
(47.21)

in which M_{FAB} and M_{FBA} are fixed-end moments at supports A and B, respectively, due to the applied load. ψ_{AB} is the rotation as a result of the relative displacement between member ends A and B given as

$$\Psi_{AB} = \frac{\Delta_{AB}}{1} = \frac{\Psi_A + \Psi_B}{1}$$
(47.22)

where Δ_{AB} is the relative deflection of the beam ends. y_A and y_B are the vertical displacements at ends A and B. Fixed-end moments for some loading cases may be obtained from Fig. 47.8. The slope deflection equations in Eq. (47.21) show that the moment at the end of a member is dependent on member properties EI, length l, and displacement quantities. The fixed-end moments reflect the transverse loading on the member.

Frame Analysis Using Slope Deflection Method

The slope deflection equations may be applied to statically indeterminate frames with or without side sway. A frame may be subjected to side sway if the loads, member properties, and dimensions of the frame are not symmetrical about the centerline. Application of the slope deflection method can be illustrated with the following example.

Example 47.4

Consider the frame shown in Fig. 47.23 subjected to side sway Δ to the right of the frame. Equation (47.21) can be applied to each of the members of the frame as follows:

Member AB:

$$\begin{split} M_{AB} &= \frac{2 \text{EI}}{6} \bigg(2 \theta_A + \theta_B - \frac{3 \Delta}{20} \bigg) + M_{FAB} \\ M_{BA} &= \frac{2 \text{EI}}{20} \bigg(2 \theta_B + \theta_A - \frac{3 \Delta}{20} \bigg) + M_{FBA} \\ \theta_A &= 0, \qquad \qquad M_{FAB} = M_{FBA} = 0 \end{split}$$



FIGURE 47.23 Example of the slope deflection method.

Hence

$$M_{AB} = \frac{2EI}{6} \left(\theta_B - 3\psi \right) \tag{47.23}$$

$$M_{BA} = \frac{2 EI}{20} \left(2\theta_{B} - 3\psi \right) \tag{47.24}$$

in which

$$\psi = \frac{\Delta}{6}$$

Member BC:

$$M_{BC} = \frac{2EI}{9} \left(2\theta_B + \theta_C - 3 \times 0 \right) + M_{FBC}$$
$$M_{CB} = \frac{2EI}{9} \left(2\theta_C + \theta_B - 3 \times 0 \right) + M_{FCB}$$
$$M_{FBC} = -\frac{180 \times 3 \times 6^2}{9^2} = -240 \text{ ft-kips}$$
$$M_{FCB} = -\frac{180 \times 3^2 \times 6}{9^2} = 120 \text{ ft-kips}$$

Hence

$$M_{BC} = \frac{2EI}{9} \left(2\theta_{B} + \theta_{C} \right) - 240 \tag{47.25}$$

$$M_{CB} = \frac{2\mathrm{EI}}{9} \left(2\theta_C + \theta_B \right) + 89 \tag{47.26}$$

Member CD:

 $\theta_{\rm D} =$

$$M_{CD} = \frac{2EI}{9} \left(2\theta_C + \theta_D - \frac{3\Delta}{30} \right) + M_{FCD}$$
$$M_{DC} = \frac{2EI}{9} \left(2\theta_D + \theta_C - \frac{3\Delta}{30} \right) + M_{FDC}$$
$$0, \qquad M_{FCD} = M_{FDC} = 0$$

Hence

$$M_{CD} = \frac{2 \text{EI}}{9} \left(2\theta_{C} - \frac{1}{3} \times 6\psi \right) = \frac{2 \text{EI}}{9} \left(2\theta_{C} - 2\psi \right)$$
(47.27)

$$M_{DC} = \frac{2EI}{9} \left(\theta_{C} - \frac{1}{3} \times 6\psi \right) = \frac{2EI}{9} \left(\theta_{C} - 2\psi \right)$$
(47.28)

Considering moment equilibrium at joint B

$$\Sigma M_{B} = M_{BA} + M_{BC} = 0$$

Substituting for $M_{\mbox{\tiny BA}}$ and $M_{\mbox{\tiny BC}}$, one obtains

$$\frac{\mathrm{EI}}{9} \left(10\theta_{B} + 2\theta_{C} - 9\psi \right) = 240$$

or

$$110\,\theta_{\rm B} + 2\theta_{\rm C} - 9\psi = \frac{2160}{\rm EI} \tag{47.29}$$

Considering moment equilibrium at joint C

 $\Sigma M_{\rm C} = M_{\rm CB} + M_{\rm CD} = 0$

$$\frac{2\mathrm{EI}}{9} (4\theta_{\mathrm{C}} + \theta_{\mathrm{B}} - 2\psi) = -120$$

or

 $\theta_{\rm B} + 4\theta_{\rm C} - 2\psi = -\frac{540}{\rm EI} \tag{47.30}$

For summation of base shears equal to zero, we have

 $\Sigma H = H_A + H_D = 0$

or

$$\frac{M_{AB} + M_{BA}}{6} + \frac{M_{CD} + M_{DC}}{9} = 0$$

Substituting for $M_{AB},\,M_{BA},\,M_{CD}$ and M_{DC} and simplifying

$$2\theta_{\rm B} + 12\theta_{\rm C} - 70\psi = 0 \tag{47.31}$$

Solution of Eqs. (47.29) to (47.31) results in

$$\theta_{\rm B} = \frac{342.7}{\rm EI}$$
$$\theta_{\rm C} = \frac{-169.1}{\rm EI}$$

and

$$\Psi = \frac{103.2}{EI}$$
(47.32)

Substituting for θ_B , θ_C , and ψ from Eq. (47.32) into Eqs. (47.23) to (47.28) we get

 $M_{AB} = 11.03 \text{ kNm}$ $M_{BA} = 125.3 \text{ kNm}$ $M_{BC} = -125.3 \text{ kNm}$ $M_{CB} = 121 \text{ kNm}$ $M_{CD} = -121 \text{ kNm}$ $M_{DC} = -83 \text{ kNm}$

Moment Distribution Method

The moment distribution method involves successive cycles of computation, each cycle drawing closer to the "exact" answers. The calculations may be stopped after two or three cycles, giving a very good approximate analysis, or they may be carried out to whatever degree of accuracy is desired. Moment distribution remains the most important hand-calculation method for the analysis of continuous beams and frames, and it may be solely used for the analysis of small structures. Unlike the slope deflection method, this method does require the solution to simultaneous equations.

The terms constantly used in moment distribution are fixed-end moments, the unbalanced moment, distributed moments, and carryover moments. When all of the joints of a structure are clamped to prevent any joint rotation, the external loads produce certain moments at the ends of the members to which they are applied. These moments are referred to as *fixed-end moments*. Initially the joints in a structure are considered to be clamped. When the joint is released, it rotates if the sum of the fixed-end moments at the joint is not zero. The difference between zero and the actual sum of the end moments is the *unbalanced moment*. The unbalanced moment causes the joint to rotate. The rotation twists the ends of the members, and resisting moments are built up in the members as they are twisted. Rotation continues until equilibrium is reached — when the resisting moments equal the unbalanced moment — at which time the sum of the moments at the joint is equal to zero. The distributed moments. The unbalanced moments are the *distributed moments*. The distributed moments are the *distributed moments*. The distributed moments are built up in the members as they are twisted. Rotation continues until equilibrium is reached — when the resisting moments equal the unbalanced moment — at which time the sum of the moments at the joint is equal to zero. The moments developed in the members resisting rotation are the *distributed moments*. The distributed moments in the ends of the member cause moments in the other ends, which are assumed fixed; these are the *carryover moments*.

Sign Convention

The moments at the end of a member are assumed to be positive when they tend to rotate the member clockwise about the joint. This implies that the resisting moment of the joint would be counterclockwise. Accordingly, under a gravity loading condition the fixed-end moment at the left end is assumed as counterclockwise (–ve) and at the right end as clockwise (+ve).

Fixed-End Moments

Fixed-end moments for several cases of loading may be found in Fig. 47.8. Application of moment distribution may be explained with reference to a continuous beam example, as shown in Fig. 47.24. Fixed-end moments are computed for each of the three spans. At joint B the unbalanced moment is obtained and the clamp is removed. The joint rotates, thus distributing the unbalanced moment to the B ends of spans BA and BC in proportion to their distribution factors. The values of these distributed moments are carried over at one half rate to the other ends of the members. When equilibrium is reached,



FIGURE 47.24 Example of a continuous beam by moment distribution.

joint B is clamped in its new rotated position and joint C is released afterwards. Joint C rotates under its unbalanced moment until it reaches equilibrium, the rotation causing distributed moments in the C ends of members CB and CD and their resulting carryover moments. Joint C is now clamped and joint B is released. This procedure is repeated again and again for joints B and C, the amount of unbalanced moment quickly diminishing, until the release of a joint causes negligible rotation. This process is called moment distribution.

The stiffness factors and distribution factors are computed as follows:

$$DF_{BA} = \frac{K_{BA}}{\Sigma K} = \frac{I/20}{I/20 + I/30} = 0.6$$
$$DF_{BC} = \frac{K_{BC}}{\Sigma K} = \frac{I/30}{I/20 + I/30} = 0.4$$
$$DF_{CB} = \frac{K_{CB}}{\Sigma K} = \frac{I/30}{I/30 + I/25} = 0.45$$
$$DF_{CD} = \frac{K_{CD}}{\Sigma K} = \frac{I/25}{II/30 + I/25} = 0.55$$

The fixed-end moments are

$$M_{FAB} = -50;$$
 $M_{FBC} = -150;$ $M_{FCD} = -104$
 $M_{FRA} = 50;$ $M_{FCB} = 150;$ $M_{FDC} = 104$

When a clockwise couple is applied near the end of a beam, a clockwise couple of half the magnitude is set up at the far end of the beam. The ratio of the moments at the far and near ends is defined as the *carryover factor*, 0.5 in the case of a straight prismatic member. The carryover factor was developed for carrying over to fixed ends, but it is applicable to simply supported ends, which must have final moments of zero. It can be shown that the beam simply supported at the far end is only three fourths as stiff as the one that is fixed. If the stiffness factors for end spans that are simply supported are modified by three fourths, the simple end is initially balanced to zero and no carryovers are made to the end afterward. This simplifies the moment distribution process significantly.



FIGURE 47.25 Example of a nonsway frame by moment distribution.

Moment Distribution for Frames

Moment distribution for frames without side sway is similar to that for continuous beams. The example shown in Fig. 47.25 illustrates the applications of moment distribution for a frame without side sway.

$$DF_{BA} = \frac{EI/20}{\frac{EI}{20} + \frac{EI}{20} + \frac{2EI}{20}} = 0.25$$

Similarly,

$$\begin{split} DF_{BE} &= 0.5; & DF_{BC} &= 0.25 \\ M_{FBC} &= 0100; & M_{FCB} &= 100 \\ M_{FRF} &= 50; & M_{FFR} &= -50 \end{split}$$

Structural frames are usually subjected to side sway in one direction or the other, due to asymmetry of the structure and eccentricity of loading. The sway deflections affect the moments, resulting in an unbalanced moment. These moments could be obtained for the deflections computed and added to the originally distributed fixed-end moments. The sway moments are distributed to columns. Should a frame have columns all of the same length and the same stiffness, the side sway moments will be the same for each column. However, should the columns have differing lengths or stiffnesses, this will not be the case. The side sway moments should vary from column to column in proportion to their I/l² values.

The frame in Fig. 47.26 shows a frame subjected to sway. The process of obtaining the final moments is illustrated for this frame.

The frame sways to the right, and the side sway moment can be assumed in the ratio

$$\frac{400}{20^2}:\frac{300}{20^2} \quad (\text{or}) \quad 1:0.75$$



FIGURE 47.26 Example of a sway frame by moment distribution.

Final moments are obtained by adding distributed fixed-end moments and 13.06/2.99 times the distributed assumed side sway moments.

Method of Consistent Deformations

This method makes use of the principle of deformation compatibility to analyze indeterminate structures. It employs equations that relate the forces acting on the structure to the deformations of the structure. These relations are formed so that the deformations are expressed in terms of the forces, and the forces become the unknowns in the analysis.

Let us consider the beam shown in Fig. 47.27a. The first step, in this method, is to determine the degree of indeterminacy or the number of redundants that the structure possesses. As shown in the figure, the beam has three unknown reactions, R_A , R_C , and M_A . Since there are only two equations of equilibrium available for calculating the reactions, the beam is said to be indeterminate to the first degree. Restraints that can be removed without impairing the load-supporting capacity of the structure are referred to as *redundants*.

Once the number of redundants are known, the next step is to decide which reaction is to be removed in order to form a determinate structure. Any one of the reactions may be chosen to be the redundant, provided that a stable structure remains after the removal of that reaction. For example, let us take the reaction R_C as the redundant. The determinate structure obtained by removing this restraint is the cantilever beam shown in Fig. 47.27b. We denote the deflection at end C of this beam, due to P, by Δ_{CP} The first subscript indicates that the deflection is measured at C, and the second subscript indicates that





FIGURE 47.26 (continued).



FIGURE 47.27 Beam with one redundant reaction.

the deflection is due to the applied load P. Using the moment area method, it can be shown that $\Delta_{CP} = 5PL^3/48EI$. The redundant R_C is then applied to the determinate cantilever beam, as shown in Fig. 47.27c. This gives rise to a deflection Δ_{CR} at point C, the magnitude of which can be shown to be $R_CL^3/3EI$.

In the actual indeterminate structure, which is subjected to the combined effects of the load P and the redundant R_C , the deflection at C is zero. Hence the algebraic sum of the deflection Δ_{CP} in Fig. 47.27b and the deflection Δ_{CR} in Fig. 47.27c must vanish. Assuming downward deflections to be positive, we write

$$\Delta_{\rm CP} - \Delta_{\rm CR} = 0 \tag{47.33}$$

or

$$\frac{5PL^3}{48EI} - \frac{R_C L^3}{3EI} = 0$$

from which

$$R_{\rm C} = \frac{5}{16} P$$

Equation (47.33), which is used to solve for the redundant, is referred to as an equation of consistent deformations.

Once the redundant R_c has been evaluated, the remaining reactions can be determined by applying the equations of equilibrium to the structure in Fig. 47.27a. Thus $\Sigma F_v = 0$ leads to

$$R_{A} = P - \frac{5}{16}P = \frac{11}{16}P$$

and $\Sigma M_A = 0$ gives

$$M_{A} = \frac{PL}{2} - \frac{5}{16}PL = \frac{3}{16}PL$$

A free body of the beam, showing all the forces acting on it, is shown in Fig. 47.27d.

The steps involved in the method of consistent deformations follow:

- 1. The number of redundants in the structure are determined.
- 2. Enough redundants to form a determinate structure are removed.
- 3. The displacements that the applied loads cause in the determinate structure at the points where the redundants have been removed are calculated.
- 4. The displacements at these points in the determinate structure, due to the redundants, are obtained.
- 5. At each point where a redundant has been removed, the sum of the displacements calculated in steps 3 and 4 must be equal to the displacement that exists at that point in the actual indeterminate structure. The redundants are evaluated using these relationships.
- 6. Once the redundants are known, the remaining reactions are determined using the equations of equilibrium.

Structures with Several Redundants

The method of consistent deformations can be applied to structures with two or more redundants. For example, the beam in Fig. 47.28a is indeterminate to the second degree and has two redundant reactions. If the reactions at B and C are selected to be the redundants, then the determinate structure obtained by removing these supports is the cantilever beam, shown in Fig. 47.28b. To this determinate structure we apply separately the given load (Fig. 47.28c) and the redundants R_B and R_C , one at a time (Fig. 47.28d and e).

Since the deflections at B and C in the original beam are zero, the algebraic sum of the deflections in Fig. 47.28c, d, and e at these same points must also vanish. Thus



FIGURE 47.28 Beam with two redundant reactions.

$$\Delta_{\rm BP} - \Delta_{\rm BB} - \Delta_{\rm BC} = 0$$

$$\Delta_{\rm CP} - \Delta_{\rm CB} - \Delta_{\rm CC} = 0$$
(47.34)

It is useful in the case of complex structures to write the equations of consistent deformations in the form

$$\Delta_{\rm BP} - \delta_{\rm BB} R_{\rm B} - \delta_{\rm BC} R_{\rm C} = 0$$

$$\Delta_{\rm CP} - \delta_{\rm CB} R_{\rm B} - \delta_{\rm CC} R_{\rm C} = 0$$
(47.35)

in which δ_{BC} , for example, denotes the deflection at B due to a unit load at C in the direction of R_C . Solution of Eq. (47.35) gives the redundant reactions R_B and R_C .

Example 47.5

Determine the reactions for the beam shown in Fig. 47.29, and draw its shear force and bending moment diagrams.

It can be seen from the figure that there are three reactions viz. M_A , R_A , and R_C , one more than that required for a stable structure. The reaction R_C can be removed to make the structure determinate. We know that the deflection at support C of the beam is zero. One can determine the deflection δ_{CP} at C due to the applied load on the cantilever in Fig. 47.29b. In the same way the deflection δ_{CR} at C due to the redundant reaction on the cantilever (Fig. 47.29c) can be determined. The compatibility equation gives



FIGURE 47.29 Example 47.5.

$$\delta_{\rm CP} - \delta_{\rm CR} = 0$$

By moment area method,

$$\begin{split} \delta_{\rm CP} &= \frac{20}{\rm EI} \times 2 \times 1 + \frac{1}{2} \times \frac{20}{\rm EI} \times 2 \times \frac{2}{3} \times 2 + \frac{40}{\rm EI} \times 2 \times 3 + \frac{1}{2} \times \frac{60}{\rm EI} \times 2 \times \left(\frac{2}{3} \times 2 + 2\right) = \frac{1520}{3\rm EI} \\ \delta_{\rm CR} &= \frac{1}{2} \times \frac{4R_{\rm C}}{\rm EI} \times 4 \times \frac{2}{3} \times 4 = \frac{64R_{\rm C}}{3\rm EI} \end{split}$$

Substituting for $\delta_{\mbox{\tiny CP}}$ and $\delta_{\mbox{\tiny CR}}$ in the compatibility equation, one obtains,

$$\frac{1520}{3EI} - \frac{64R_{\rm C}}{3EI} = 0$$

from which

$$R_{c} = 23.75 \text{ kN}^{\uparrow}$$

By using statical equilibrium equations we get

and $M_A = 5$ kNm.

The shear force and bending moment diagrams are shown in Fig. 47.29d.

- 1. Solutions to fix-based portal frames subjected to various loading: Fig. 47.30 shows the bending moment diagram and reaction forces of fix-based portal frames subjected to loading typically encountered in practice. Closed-form solutions are provided for moments and end forces to facilitate a quick solution to the simple frame problem.
- Solutions to pin-based portal frames subjected to various loading: Fig. 47.31 shows the bending moment diagram and reaction forces of pin-based portal frames subjected to loading typically encountered in practice. Closed-form solutions are provided for moments and end forces to facilitate a quick solution to the simple frame problem.






FIGURE 47.30 (continued).

47.5 Plates

Bending of Thin Plates

A plate in which its thickness is small compared to the other dimensions is called a thin plate. The plane parallel to the faces of the plate and bisecting the thickness of the plate, in the undeformed state, is called the middle plane of the plate. When the deflection of the middle plane is small compared with the thickness, h, it can be assumed that

- 1. There is no deformation in the middle plane.
- 2. The normals of the middle plane before bending are deformed into the normals of the middle plane after bending.
- 3. The normal stresses in the direction transverse to the plate can be neglected.



FIGURE 47.30 (continued).

Based on these assumptions, all stress components can be expressed by deflection w of the plate. w is a function of the two coordinates (x, y) in the plane of the plate. This function has to satisfy a linear partial differential equation, which, together with the boundary conditions, completely defines w.

Figure 47.32a shows a plate element cut from a plate whose middle plane coincides with the xy plane. The middle plane of the plate subjected to a lateral load of intensity, q, is shown in Fig. 47.32b. It can be shown, by considering the equilibrium of the plate element, that the stress resultants are given as

$$M_{x} = -D\left(\frac{\partial^{2} w}{\partial x^{2}} + v \frac{\partial^{2} w}{\partial y^{2}}\right)$$

$$M_{y} = -D\left(\frac{\partial^{2} w}{\partial y^{2}} + v \frac{\partial^{2} w}{\partial x^{2}}\right)$$
(47.36)



$$M_B = M_C = -\frac{wL^2}{4\beta} \qquad M_{max} = \frac{wL^2}{8} + M_B$$
$$V_A = V_D = \frac{wL}{2} \qquad H_A = H_D = -\frac{M_B}{h}$$

FIGURE 47.31 Rigid frames with pinned supports.

$$M_{xy} = -M_{yx} = D(1-\nu)\frac{\partial^2 w}{\partial x \partial y}$$
$$V_y = \frac{\partial^3 w}{\partial y^3} + (2-\nu)\frac{\partial^3 w}{\partial y \partial x^2}$$
(47.37)

$$V_{x} = \frac{\partial^{3} w}{\partial x^{3}} + (2 - \nu) \frac{\partial^{3} w}{\partial x \partial y^{2}}$$
(47.38)

$$Q_{y} = -D\frac{\partial}{\partial y} \left(\frac{\partial^{2} w}{\partial x^{2}} + \frac{\partial^{2} w}{\partial y^{2}} \right)$$
(47.39)

$$R = 2D(1-v)\frac{\partial^2 w}{\partial x \partial y}$$
(47.40)

where M_x and M_y = the bending moments per unit length in the x and y directions, respectively M_{xy} and M_{yx} = the twisting moments per unit length

- Q_x and Q_y = the shearing forces per unit length in the x and y directions, respectively V_x and V_y = are the supplementary shear forces in the x and y directions, respectively \dot{R} = the corner force
 - $D = Eh^3/12(1 v^2)$, the flexural rigidity of the plate per unit length
 - E = the modulus of elasticity; and v is Poisson's ratio



$$M_B = \frac{wh^2}{4} \left[-\frac{a}{2b} + 1 \right] \qquad H_D = -\frac{M_C}{h}$$
$$M_C = \frac{wh^2}{4} \left[-\frac{a}{2b} - 1 \right] \qquad H_A = -(wh - H_D)$$
$$V_A = -V_D = -\frac{wh^2}{2L}$$



$$M_B = -M_C = Pa \qquad H_A = H_d = P$$
$$V_A = -V_D = -\frac{2Pa}{L}$$

Moment at Loads = $\pm Pa$

FIGURE 47.31 (continued).

The governing equation for the plate is obtained as

$$\frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} = \frac{q}{D}$$
(47.41)

Any plate problem should satisfy the governing Eq. (47.41) and boundary conditions of the plate.

Boundary Conditions

There are three basic boundary conditions for plates. These are the clamped edge, simply supported edge, and free edge.

Clamped Edge

For this boundary condition, the edge is restrained such that the deflection and slope are zero along the edge. If we consider the edge x = a to be clamped, we have



$$M_B = M_C = -\frac{M_D}{L} \cdot \frac{3}{2b}$$

$$V_A = \frac{Pb}{L} \qquad V_D = \frac{Pa}{L} \qquad H_A = H_D = -\frac{M_B}{h}$$

FIGURE 47.31 (continued).

$$(w)_{x=a} = 0; \quad \left(\frac{\partial w}{\partial x}\right)_{x=a} = 0$$
 (47.42)

Simply Supported Edge

If the edge x = a of the plate is simply supported, the deflection w along this edge must be zero. At the same time this edge can rotate freely with respect to the edge line. This means that

$$(w)_{x=a} = 0; \quad \left(\frac{\partial^2 w}{\partial x^2}\right)_{x=a} = 0$$
 (47.43)

Free Edge

If the edge x = a of the plate is entirely free, there are no bending and twisting moments and also vertical shearing forces. This can be written in terms of w, the deflection, as

$$\left(\frac{\partial^2 w}{\partial x^2} + v \frac{\partial^2 w}{\partial y^2}\right)_{x=a} = 0$$

$$\left(\frac{\partial^3 w}{\partial x^3} + (2 - v) \frac{\partial^3 w}{\partial x \partial y^2}\right)_{x=a} = 0$$
(47.44)







FIGURE 47.32 (a) Plate element. (b) Stress resultants.



FIGURE 47.33 Rectangular plate under sinusoidal loading.

Bending of Rectangular Plates

The plate bending problem may be solved by referring to the differential Eq. (47.41). The solution, however, depends on the loading and boundary conditions. Consider a simply supported plate subjected to a sinusoidal loading, as shown in Fig. 47.33. The differential Eq. (47.41) in this case becomes

$$\frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} = \frac{q_o}{D} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$$
(47.45)

The boundary conditions for the simply supported edges are

w = 0,
$$\frac{\partial^2 w}{\partial x^2} = 0$$
 for x = 0 and x = a
w = 0, $\frac{\partial^2 w}{\partial y^2} = 0$ for y = 0 and y = b
(47.46)

The deflection function becomes

$$w = w_0 \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$$
(47.47)

which satisfies all the boundary conditions in Eq. (47.46). w_0 must be chosen to satisfy Eq. (47.45). Substitution of Eq. (47.47) into Eq. (47.45) gives

$$\pi^4 \left(\frac{1}{a^2} + \frac{1}{b^2} \right)^2 w_0 = \frac{q_o}{D}$$

The deflection surface for the plate can, therefore, be found as

$$w = \frac{q_o}{\pi^4 D\left(\frac{1}{a^2} + \frac{1}{b^2}\right)} \sin\frac{\pi x}{a} \sin\frac{\pi y}{b}$$
(47.48)

Using Eqs. (47.48) and (47.36), we find expression for moments as

$$M_{x} = \frac{q_{o}}{\pi^{2} \left(\frac{1}{a^{2}} + \frac{1}{b^{2}}\right)^{2}} \left(\frac{1}{a^{2}} + \frac{v}{b^{2}}\right) \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$$

$$M_{y} = \frac{q_{o}}{\pi^{2} \left(\frac{1}{a^{2}} + \frac{1}{b^{2}}\right)^{2}} \left(\frac{v}{a^{2}} + \frac{1}{b^{2}}\right) \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$$

$$M_{xy} = \frac{q_{o}(1 - v)}{\pi^{2} \left(\frac{1}{a^{2}} + \frac{1}{b^{2}}\right)^{2}} \cos \frac{\pi x}{a} \cos \frac{\pi y}{b}$$
(47.49)

Maximum deflection and maximum bending moments that occur at the center of the plate can be written by substituting x = a/2 and y = b/2 in Eq. (47.49) as

$$w_{max} = \frac{q_o}{\pi^4 D \left(\frac{1}{a^2} + \frac{1}{b^2}\right)^2}$$

$$\left(M_x\right)_{max} = \frac{q_o}{\pi^2 \left(\frac{1}{a^2} + \frac{1}{b^2}\right)^2} \left(\frac{1}{a^2} + \frac{v}{b^2}\right)$$
(47.50)

$$(M_y)_{max} = \frac{q_o}{\pi^2 \left(\frac{1}{a^2} + \frac{1}{b^2}\right)^2} \left(\frac{\nu}{a^2} + \frac{1}{b^2}\right)$$

If the plate is square, then a = b and Eq. (47.50) becomes

$$w_{max} = \frac{q_o a^4}{4\pi^4 D}$$

$$\left(M_x\right)_{max} = \left(M_y\right)_{max} = \frac{(1+\nu)}{4\pi^2} q_o a^2$$
(47.51)

If the simply supported rectangular plate is subjected to any kind of loading given by

$$\mathbf{q} = \mathbf{q}(\mathbf{x}, \mathbf{y}) \tag{47.52}$$

the function q(x, y) should be represented in the form of a double trigonometric series as

$$q(\mathbf{x}, \mathbf{y}) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} q_{mn} \sin \frac{m\pi x}{a} \sin \frac{n\pi y}{b}$$
(47.53)

in which q_{mn} is given by

$$q_{mn} = \frac{4}{ab} \int_{0}^{a} \int_{0}^{b} q(x, y) \sin \frac{m\pi x}{a} \sin \frac{n\pi y}{b} dxdy$$
(47.54)

From Eqs. (47.45) and (47.52) to (47.54) we can obtain the expression for deflection as

$$w = \frac{1}{\pi^4 D} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{q_{mn}}{\left(\frac{m^2}{a^2} + \frac{n^2}{b^2}\right)^2} \sin\frac{m\pi x}{a} \sin\frac{n\pi y}{b}$$
(47.55)

If the applied load is uniformly distributed of intensity q_o, we have

$$q(x, y) = q_o$$

and from Eq. (47.54) we obtain

$$q_{mn} = \frac{4q_o}{ab} \int_{0}^{a} \int_{0}^{b} \sin\frac{m\pi x}{a} \sin\frac{n\pi y}{b} dx dy = \frac{16q_o}{\pi^2 mn}$$
(47.56)

in which m and n are odd integers. $q_{mn} = 0$ if m or n or both are even numbers. Finally, the deflection of a simply supported plate subjected to a uniformly distributed load can be expressed as

$$w = \frac{16q_{o}}{\pi^{6}D} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{\sin\frac{m\pi x}{a}\sin\frac{n\pi y}{b}}{mn\left(\frac{m^{2}}{a^{2}} + \frac{n^{2}}{b^{2}}\right)^{2}}$$
(47.57)

where m = 1, 3, 5, ...n = 1, 3, 5. ...

The maximum deflection occurs at the center. Its magnitude can be evaluated by substituting x = a/2 and y = b/2 in Eq. (47.57) as

$$w_{max} = \frac{16q_o}{\pi^6 D} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{(-1)\frac{m+n}{2} - 1}{mn\left(\frac{m^2}{a^2} + \frac{n^2}{b^2}\right)^2}$$
(47.58)

Equation (47.58) is a rapid converging series. A satisfactory approximation can be obtained by taking only the first term of the series; for example, in the case of a square plate,

$$w_{max} = \frac{4q_oa^4}{\pi^6 D} = 0.00416 \frac{q_oa^4}{D}$$

Assuming v = 0.3, the maximum deflection can be calculated as

$$w_{max} = 0.0454 \frac{q_o a^4}{E h^3}$$

The expressions for bending and twisting moments can be obtained by substituting Eq. (47.57) into Eq. (47.36). Figure 47.34 shows some loading cases and the corresponding loading functions.

If the opposite edges at x = 0 and x = a of a rectangular plate are simply supported, the solution taking the deflection function as

$$w = \sum_{m=1}^{\infty} Y_m \sin \frac{m\pi x}{a}$$
(47.59)

can be adopted. Equation (47.59) satisfies the boundary conditions w = 0 and $\partial^2 w / \partial x^2 = 0$ on the two simply supported edges. Y_m should be determined such that it satisfies the boundary conditions along the edges y = +b/-2 of the plate shown in Fig. 47.35 and also the equation of the deflection surface

$$\frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} = \frac{q_o}{D}$$
(47.60)

qo being the intensity of the uniformly distributed load.

The solution for Eq. (47.60) can be taken in the form

$$w = w_1 + w_2$$
 (47.61)

for a uniformly loaded simply supported plate. w₁ can be taken in the form

$$w_1 = \frac{q_0}{24D} \left(x^4 - 2ax^3 + a^3 x \right)$$
(47.62)

representing the deflection of a uniformly loaded strip parallel to the x axis. It satisfies Eq. (47.60) and also the boundary conditions along x = 0 and x = a.

The expression w₂ has to satisfy the equation

$$\frac{\partial^4 w_2}{\partial x^4} + 2 \frac{\partial^4 w_2}{\partial x^2 \partial y^2} + \frac{\partial^4 w_2}{\partial y^4} = 0$$
(47.63)

No.	Load $q(x,y) = \sum_{m n}^{\infty} \sum_{n=1}^{\infty} q_{mn} \sin \frac{m\pi x}{a} \sin \frac{n\pi y}{b}$	Expansion Coefficients q _{mn}
1	X b b q _o Z,w Y	$q_{mn} = \frac{16q_o}{\pi^2 mn}$ (m, n = 1, 3, 5,)
2	Z,w Y	$q_{mn} = \frac{-8q_o \cos m\pi}{\pi^2 mn}$ (m, n = 1, 3, 5,)
3	d/2 90 b C/2 C/2 Z,w Y	$p_{mn} = \frac{16q_o}{\pi^2 mn} \sin \frac{m\pi\xi}{a} \sin \frac{n\pi\eta}{b}$ $\times \sin \frac{m\pi c}{2a} \sin \frac{n\pi d}{2b}$ $(m, n = 1, 3, 5,)$
4	X b b b c z,w Y	$q_{mn} = \frac{4q_o}{ab} \sin \frac{m\pi\xi}{a} \sin \frac{n\pi\eta}{b}$ (m, n = 1, 3, 5,)



FIGURE 47.34 Typical loading on plates and loading functions.



FIGURE 47.35 Rectangular plate.

and must be chosen such that Eq. (47.61) satisfies all boundary conditions of the plate. Taking w_2 in the form of series given in Eq. (47.59), it can be shown that the deflection surface takes the form

$$w = \frac{q_{o}}{24D} \left(x^{4} - 2ax^{3} + a^{3}x \right) + \frac{q_{o}a^{4}}{24D} \sum_{m=1}^{\infty} \left(A_{m} \cosh \frac{m\pi y}{a} + B_{m} \frac{m\pi y}{a} \sinh \frac{m\pi y}{a} + C_{m} \sinh \frac{m\pi y}{a} + D_{m} \frac{m\pi y}{a} \cosh \frac{m\pi y}{a} \right) \sin \frac{m\pi x}{a}$$
(47.64)

Observing that the deflection surface of the plate is symmetrical with respect to the x axis, only even functions of y are kept in Eq. (47.64); therefore, $C_m = D_m = 0$. The deflection surface takes the form

$$w = \frac{q_{o}}{24D} \left(x^{4} - 2ax^{3} + a^{3}x\right) + \frac{q_{o}a^{4}}{24D} \sum_{m=1}^{\infty} \left(A_{m} \cosh\frac{m\pi y}{a} + B_{m}\frac{m\pi y}{a} \sinh\frac{m\pi y}{a}\right) \sin\frac{m\pi x}{a} \quad (47.65)$$

Developing the expression in Eq. (47.62) into a trigonometric series, the deflection surface in Eq. (47.65) is written as

$$w = \frac{q_o a^4}{D} \sum_{m=1}^{\infty} \left(\frac{4}{\pi^5 m^5} + A_m \cosh \frac{m \pi y}{a} + B_m \frac{m \pi y}{a} \sin \frac{m \pi y}{a} \right) \sin \frac{m \pi x}{a}$$
(47.66)

Substituting Eq. (47.5.30) in the boundary conditions

$$w = 0, \quad \frac{\partial^2 w}{\partial y^2} = 0$$
 (47.67)

one obtains the constants of integration A_m and B_m , and the expression for deflection may be written as

$$w = \frac{4q_o a^4}{\pi^5 D} \sum_{m=1,3,5...}^{\infty} \frac{1}{m^5} \left(1 - \frac{\alpha_m \tanh \alpha_m + 2}{2\cosh \alpha_m} \cosh \frac{2\alpha_m y}{b} + \frac{\alpha_m}{2\cosh \alpha_m} \frac{2y}{b} \sinh \frac{2\alpha_m y}{b} \right) \sin \frac{m\pi x}{a} \quad (47.68)$$

in which $\alpha_m = m\pi b/2a$.

Maximum deflection occurs at the middle of the plate, x = a/2, y = 0, and is given by

$$w_{max} = \frac{4q_{o}a^{4}}{\pi^{5}D} \sum_{m=1,3,5,\dots}^{\infty} \frac{(-1)\frac{m-1}{2}}{m^{5}} \left(1 - \frac{\alpha_{m} \tanh \alpha_{m} + 2}{2\cosh \alpha_{m}}\right)$$
(47.69)

The solutions of plates with arbitrary boundary conditions are complicated. It is possible to make some simplifying assumptions for plates with the same boundary conditions along two parallel edges in order to obtain the desired solution. Alternately, the energy method can be applied more efficiently to solve plates with complex boundary conditions. However, it should be noted that the accuracy of results depends on the deflection function chosen. These functions must be chosen so that they satisfy at least the kinematics boundary conditions.

Figure 47.36 gives formulas for deflection and bending moments of rectangular plates with typical boundary and loading conditions.



FIGURE 47.36 Typical loading and boundary conditions for rectangular plates.



FIGURE 47.37 (a) Circular plate. (b) Stress resultants.

Bending of Circular Plates

In the case of a symmetrically loaded circular plate, the loading is distributed symmetrically about the axis perpendicular to the plate through its center. In such cases, the deflection surface to which the middle plane of the plate is bent will also be symmetrical. The solution of circular plates can be conveniently carried out by using polar coordinates.

Stress resultants in a circular plate element are shown in Fig. 47.37. The governing differential equation is expressed in polar coordinates as

$$\frac{1}{r}\frac{d}{dr}\left\{r\frac{d}{dr}\left[\frac{1}{r}\frac{d}{dr}\left(r\frac{dw}{dr}\right)\right]\right\} = \frac{q}{D}$$
(47.70)

in which q is the intensity of loading.

In the case of a uniformly loaded circular plate, Eq. (47.70) can be integrated successively and the deflection at any point at a distance r from the center can be expressed as

$$w = \frac{q_0 r^4}{64D} + \frac{C_1 r^2}{4} + C_2 \log \frac{r}{a} + C_3$$
(47.71)

in which q_0 is the intensity of loading and a is the radius of the plate. C_1 , C_2 , and C_3 are constants of integration to be determined using the boundary conditions.

For a plate with clamped edges under uniformly distributed load q_{o} , the deflection surface reduces to

$$w = \frac{q_o}{64D} (a^2 - r^2)^2$$
(47.72)

The maximum deflection occurs at the center, where r = 0, and is given by

$$w = \frac{q_o a^4}{64 D}$$
(47.73)

Bending moments in the radial and tangential directions are respectively given by

$$M_{r} = \frac{q_{o}}{16} \Big[a^{2} (1+\nu) - r^{2} (3+\nu) \Big]$$

$$M_{t} = \frac{q_{o}}{16} \Big[a^{2} (1+\nu) - r^{2} (1+3\nu) \Big]$$
(47.74)

The method of superposition can be applied in calculating the deflections for circular plates with simply supported edges. The expressions for deflection and bending moment are given as

$$w = \frac{q_{o}(a^{2} - r^{2})}{64D} \left(\frac{5 + \nu}{1 + \nu} a^{2} - r^{2} \right)$$

$$w_{max} = \frac{5 + \nu}{64(1 + \nu)} \frac{q_{o}a^{4}}{D}$$

$$M_{r} = \frac{q_{o}}{16} (3 + \nu) (a^{2} - r^{2})$$

$$M_{t} = \frac{q_{o}}{16} [a^{2}(3 + \nu) - r^{2}(1 + 3\nu)]$$
(47.76)

This solution can be used to deal with plates with a circular hole at the center and subjected to concentric moment and shearing forces. Plates subjected to concentric loading and concentrated loading also can be solved by this method. More rigorous solutions are available to deal with irregular loading on circular plates. Once again, the energy method can be employed advantageously to solve circular plate problems. Figure 47.38 gives deflection and bending moment expressions for typical cases of loading and boundary conditions on circular plates.

Strain Energy of Simple Plates

The strain energy expression for a simple rectangular plate is given by

$$U = \frac{D}{2} \iint_{\text{area}} \left\{ \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} \right)^2 - 2(1 - \nu) \left[\frac{\partial^2 w}{\partial x^2} \frac{\partial^2 w}{\partial y^2} - \left(\frac{\partial^2 w}{\partial x \partial y} \right)^2 \right] \right\} dxdy$$
(47.77)

A suitable deflection function w(x, y) satisfying the boundary conditions of the given plate may be chosen. The strain energy, U, and the work done by the given load, q(x, y),

Case No.	Structural System and Static Loading	Deflection and Internal Forc	es
1	$\begin{array}{c} & & \\ & & \\ \bullet^{r} \bullet \bullet^{r} \bullet \\ & \\ & \\ & \\ & \\ & \\ & \\ & \\ \bullet \end{array} \begin{array}{c} & \\ & \\ & \\ & \\ & \\ & \\ & \\ & \\ & \\ & $	$w = \frac{q_0 r_0^4}{64D(1+\nu)} [2(3+\nu) C_1 - (1+\nu) C_0]$ $m_r = \frac{q_0 r_0^2}{16} (3+\nu) C_1$ $m_\theta = \frac{q_0 r_0^2}{16} [2(1-\nu) - (1+3\nu) C_1]$ $q_r = \frac{q_0 r_0}{2} \rho$	$\rho = \frac{r}{r_o}$ $C_o = 1 - \rho^4$ $C_1 = 1 - \rho^2$
2	$q_o q = q_o(1 - \rho)$	$w = \frac{q_{o}r_{o}^{4}}{14400D}B\frac{3(183+43v)}{1+v} - \frac{10(71+29v)}{1+v}\rho$ $(m_{r})_{\rho=0} = (m_{\phi})_{\rho=0} = \frac{q_{o}r_{o}^{4}}{720}(71+29v);$ $(q_{r})_{\rho=1} = -\frac{q_{o}r_{o}}{6}$	$\rho^2 + 225\rho^4 - 64\rho^{5F}$ $\rho = \frac{r}{r_o}$
3		$\begin{split} & w = \frac{q_{o} r_{o}^{4}}{450 \mathrm{D}} \mathrm{B} \frac{3(6+\nu)}{1+\nu} - \frac{5(4+\nu)}{1+\nu} \rho^{2} + 2\rho^{5} \mathrm{F} \\ & (m_{r})_{\rho=0} = (m_{\phi})_{\rho=0} = \frac{q_{o} r_{o}^{2}}{45} (4+\nu); \\ & (q_{r})_{\rho=1} = -\frac{q_{o} r_{o}}{3} \end{split}$	$\rho = \frac{r}{r_o}$
4	$\begin{array}{c} \begin{array}{c} \\ \\ \\ \\ \\ \\ \\ \end{array} \end{array} \xrightarrow{r_o} \\ \hline \\ \\ \\ \\ \\ \end{array} \xrightarrow{r_o} \\ \\ \\ \\ \\ \\ \\ \\ \\ \end{array} \xrightarrow{r_o} \\ \\ \\ \\ \\ \\ \\ \end{array} \xrightarrow{r_o} \\ \\ \\ \\ \\ \\ \\ \\ \\ \\ \\ \\ \\ \\ \\ \\ \\ \\ \\$	$w = \frac{Pr_{0}^{2}}{16\pi D} \frac{B^{3+\nu}}{1+\nu}C_{1} + 2C_{2}F$ $m_{r} = \frac{P}{4\pi}(1+\nu)C_{3}$ $m_{\phi} = \frac{P}{4\pi} [(1-\nu) - (1+\nu)C_{3}]$ $q_{r} = \frac{P}{2\pi r_{0}\rho}$	$C_{1} = 1 - \rho^{2}$ $C_{2} = \rho^{2} \ell n\rho$ $C_{3} = \ell n\rho$ $\rho = \frac{r}{r_{o}}$

FIGURE 47.38 Typical loading and boundary conditions for circular plates.

$$W = -\iint_{area} q(x, y) w(x, y) dx dy$$

can be calculated. The total potential energy is, therefore, given as V = U + W. Minimizing the total potential energy, the plate problem can be solved.

$$\left[\frac{\partial^2 w}{\partial x^2}\frac{\partial^2 w}{\partial y^2} - \left(\frac{\partial^2 w}{\partial x \partial y}\right)^2\right]$$

The term is known as the Gaussian curvature.

If the function w(x, y) = f(x)f(y) (product of a function of x only and a function of y only) and w = 0 at the boundary are assumed, then the integral of the Gaussian curvature over the entire plate equals zero. Under these conditions

$$U = \frac{D}{2} \iint_{\text{area}} \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} \right)^2 dxdy$$

Case No.	Structural System and Static Loading	Deflection and Internal Forces	
5	$(\begin{array}{c c} & & & \\ $	$w = \frac{Mr_o^2}{2D(1+v)C_1}$ $m_r = m_{\phi} = M$ $q_r = 0$ $C_1 = 1 - \rho^2, \ \rho = \frac{r}{r_o}$	
6		$\begin{split} w &= \frac{q_0 r_0^4}{64D} (1 - \rho^2)^2 & q_r = -\frac{q_0 r_0}{2} \rho \\ m_r &= \frac{q_0 r_0^2}{16} [1 + \nu - (3 + \nu)\rho^2] & \rho = \frac{r}{r_0} \\ m_\phi &= \frac{q_0 r_0^2}{16} [1 + \nu - (1 + 3\nu)\rho^2] \end{split}$	
7	$q = q_0(1 - p)$ $q = \frac{q_0(1 - p)}{2r_0}$	$\begin{split} w &= \frac{q_o r_o^4}{14400D} \left(129 - 290\rho^2 + 225\rho^4 - 64\rho^5 \right) \\ (m_r)_{\rho=0} &= (m_{\phi})_{\rho=0} = \frac{29q_o r_o^2}{720} \left(1 + \nu \right) \qquad (q_r)_{\rho=1} = -\frac{q_o r_o}{6} \\ (m_r)_{\rho=1} &= (m_{\phi})_{\rho=1} = -\frac{7q_o r_o^2}{120} \qquad \rho = \frac{r}{r_o} \end{split}$	
8	q_{o}	$\begin{split} w &= \frac{q_0 r_0^4}{450 D} \left(3 - 5 \rho^2 + 2 \rho^5\right) & q_r = -\frac{q_0 r_0}{3} \rho^2 \\ m_r &= \frac{q_0 r_0^2}{45} \left[1 + \nu - (4 + \nu) \rho^3\right] & \rho = \frac{r}{r_0} \\ m_\phi &= \frac{q_0 r_0^2}{45} \left[1 + \nu - (1 + 4\nu) \rho^3\right] \end{split}$	
9		$w = \frac{Pr_{o}^{2}}{16\pi D} (1 - \rho^{2} + 2\rho^{2} \ell n \rho) \qquad q_{r} = -\frac{P}{2\pi r_{o} \rho}$ $m_{r} = -\frac{P}{4\pi} [1 + (1 + \nu) \ell n \rho] \qquad \rho = \frac{r}{r_{o}}$ $m_{\phi} = -\frac{P}{4\pi} [\nu + (1 + \nu) \ell n \rho]$	

FIGURE 47.38 (continued).

If polar coordinates instead of rectangular coordinates are used and axial symmetry of loading and deformation are assumed, the equation for strain energy, U, takes the form

$$U = \frac{D}{2} \iint_{\text{area}} \left\{ \left(\frac{\partial^2 w}{\partial r^2} + \frac{1}{r} \frac{\partial w}{\partial r} \right)^2 - \frac{2(1-\nu)}{r} \frac{\partial w}{\partial r} \frac{\partial^2 w}{\partial r^2} \right\} r dr d\theta$$
(47.78)

and the work done, W, is written as

$$W = -\iint_{\text{area}} qwr dr d\theta \tag{47.79}$$

Detailed treatment of the plate theory can be found in Timoshenko and Woinowsky-Krieger (1959).



FIGURE 47.39 Isosceles triangular plate.

Plates of Various Shapes and Boundary Conditions

Simply Supported Isosceles Triangular Plate Subjected to a Concentrated Load

Plates of shapes other than a circle or rectangle are used in some situations. A rigorous solution of the deflection for a plate with a more complicated shape is likely to be very difficult. Consider, for example, the bending of an isosceles triangular plate with simply supported edges under concentrated load P acting at an arbitrary point (Fig. 47.39). A solution can be obtained for this plate by considering a mirror image of the plate, as shown in the figure. The deflection of OBC of the square plate is identical with that of a simply supported triangular plate OBC. The deflection owing to the force P can be written as

$$w_{1} = \frac{4Pa^{2}}{\pi^{4}D} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{\sin(m\pi x_{1}/a)\sin(n\pi y_{1}/a)}{(m^{2}+n^{2})^{2}} \sin\frac{m\pi x}{a} \sin\frac{n\pi y}{a}$$
(47.80)

Upon substitution of –P for P, $(a - y_1)$ for x_1 , and $(a - x_1)$ for y_1 in Eq. (47.80), we obtain the deflection due to the force –P at A_i:

$$w_{2} = -\frac{4Pa^{2}}{\pi^{4}D} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} (-1)^{m+n} \frac{\sin(m\pi x_{1}/a)\sin(n\pi y_{1}/a)}{(m^{2}+n^{2})^{2}} \sin\frac{m\pi x}{a} \sin\frac{n\pi y}{a}$$
(47.81)

The deflection surface of the triangular plate is then

$$w = w_1 + w_2$$
 (47.82)

Equilateral Triangular Plates

The deflection surface of a simply supported plate loaded by uniform moment M_o along its boundary, and the surface of a uniformly loaded membrane, uniformly stretched over the same triangular boundary, are identical. The deflection surface for such a case can be obtained as

$$w = \frac{M_o}{4aD} \left[x^3 - 3xy^2 - a\left(x^2 + y^2\right) + \frac{4}{27}a^3 \right]$$
(47.83)



FIGURE 47.40 Equilateral triangular plate with coordinate axes.

If the simply supported plate is subjected to uniform load p_o , the deflection surface takes the form

$$w = \frac{P_o}{64aD} \left[x^3 - 3xy^2 - a(x^2 + y^2) + \frac{4}{27}a^3 \right] \left(\frac{4}{9}a^2 - x^2 - y^2 \right)$$
(47.84)

For the equilateral triangular plate (Fig. 47.40) subjected to a uniform load and supported at the corners, approximate solutions based on the assumption that the total bending moment along each side of the triangle vanishes were obtained by Vijakkhna et al. (1973), who derived the equation for the deflection surface as

$$w = \frac{qa^{4}}{144(1-v^{2})D} \left[\frac{8}{27} (7+v)(2-v) - (7+v)(1-v) \left(\frac{x^{2}}{a} + \frac{y^{2}}{a^{2}} \right) - (5-v)(1+v) \right]$$

$$\left(\frac{x^{3}}{a^{3}} - 3\frac{xy^{2}}{a^{3}} \right) + \frac{9}{4} (1-v^{2}) \left(\frac{x^{4}}{a^{4}} + 2\frac{x^{2}y^{2}}{a^{4}} + \frac{y^{4}}{a^{4}} \right)$$

$$(47.85)$$

The errors introduced by the approximate boundary condition, i.e., the assumption that the total bending moment along each side of the triangle vanishes, are not significant because the boundary condition's influence on the maximum deflection and stress resultants is small for practical design purposes. The value of the twisting moment on the edge at the corner given by this solution is found to be exact.

The details of the mathematical treatment may be found in Vijakkhna et al. (1973, p. 123–128).

Rectangular Plate Supported at Corners

Approximate solutions for rectangular plates supported at the corners and subjected to a uniformly distributed load were obtained by Lee and Ballesteros (1960). The approximate deflection surface is given as

$$w = \frac{qa^{4}}{48(1-\nu^{2})D} \left[\left(10+\nu-\nu^{2}\right) \left(1+\frac{b^{4}}{a^{4}}\right) - 2(7\nu-1)\frac{b^{2}}{a^{2}} + 2\left((1+5\nu)\frac{b^{2}}{a^{2}} - (6+\nu-\nu^{2})\right)\frac{x}{a} + 2\left((1+5\nu) - (6+\nu-\nu^{2})\frac{b^{2}}{a^{2}}\right)\frac{y^{2}}{a^{2}} + (2+\nu-\nu^{2})\frac{x^{4}+y^{4}}{a^{4}} - 6(1+\nu)\frac{x^{2}y^{2}}{a^{4}} \right]$$

$$(47.86)$$

The details of the mathematical treatment may be found in Lee and Ballesteros (1960, p. 206–211).

Orthotropic Plates

Plates of anisotropic materials have important applications owing to their exceptionally high bending stiffness. A nonisotropic or anisotropic material displays direction-dependent properties. Simplest among them are those in which the material properties differ in two mutually perpendicular directions. A material so described is orthotropic, e.g., wood. A number of manufactured materials are approximated as orthotropic. Examples include corrugated and rolled metal sheets, fillers in sandwich plate construction, plywood, fiber reinforced composites, reinforced concrete, and gridwork. The latter consists of two systems of equally spaced parallel ribs (beams), mutually perpendicular and attached rigidly at the points of intersection.

The governing equation for orthotropic plates, similar to that of isotropic plates Eq. (47.86), takes the form

$$D_{x}\frac{\delta^{4}w}{\delta x^{4}} + 2H\frac{\delta^{4}w}{\delta x^{2}\delta y^{2}} + D_{y}\frac{\delta^{4}w}{\delta y^{4}} = q \qquad (47.87)$$

in which

$$D_x = \frac{h^3 E_x}{12}, \quad D_y = \frac{h^3 E_y}{12}, \quad H = D_{xy} + 2G_{xy}, \quad D_{xy} = \frac{h^3 E_{xy}}{12}, \quad G_{xy} = \frac{h^3 G_{yy}}{12}$$

The expressions for D_x , D_y , D_{xy} , and G_{xy} represent the flexural rigidities and the torsional rigidity of an orthotropic plate, respectively. E_x , E_y , and G are the orthotropic plate moduli. Practical considerations often lead to assumptions, with regard to material properties, resulting in approximate expressions for elastic constants. The accuracy of these approximations is generally the most significant factor in the orthotropic plate problem. Approximate rigidities for some cases that are commonly encountered in practice are given in Fig. 47.41.

General solution procedures applicable to the case of isotropic plates are equally applicable to orthotropic plates. Deflections and stress resultants can thus be obtained for orthotropic plates of different shapes with different support and loading conditions. These problems have been researched extensively, and solutions concerning plates of various shapes under different boundary and loading conditions may be found in the references viz. Tsai and Cheron (1968), Timoshenko and Woinowsky-Krieger (1959), Lee et al. (1971), and Shanmugam et al. (1988 and 1989).

47.6 Shells

Stress Resultants in Shell Element

A thin shell is defined as a shell with a relatively small thickness, compared with its other dimensions. The primary difference between a shell and a plate is that the former has a curvature in the unstressed state, whereas the latter is assumed to be initially flat. The presence of initial curvature is of little consequence as far as flexural behavior is concerned. The membrane behavior, however, is affected significantly by the curvature. Membrane action in a surface is caused by in-plane forces. These forces may be primary forces caused by applied edge loads or edge deformations, or they may be secondary forces resulting from flexural deformations.

In the case of the flat plates, secondary in-plane forces do not give rise to appreciable membrane action unless the bending deformations are large. Membrane action due to secondary forces is, therefore, neglected in small deflection theory. In the case of a shell which has an initial curvature, membrane action caused by secondary in-plane forces will be significant regardless of the magnitude of the bending deformations.





A plate is likened to a two-dimensional beam and resists transverse loads by two-dimensional bending and shear. A membrane is likened to a two-dimensional equivalent of the cable and resists loads through tensile stresses. Imagine a membrane with large deflections (Fig. 47.42a), reverse the load and the membrane, and we have the structural shell (Fig. 47.42b), provided that the shell is stable for the type of load shown. The membrane resists the load through tensile stresses, but the ideal thin shell must be capable of developing both tension and compression.



FIGURE 47.42



FIGURE 47.43 Shell element.

Consider an infinitely small shell element formed by two pairs of adjacent planes that are normal to the middle surface of the shell and contain its principal curvatures, as shown in Fig. 47.43a. The thickness of the shell is denoted as h. Coordinate axes x and y are taken tangent at o to the lines of principal curvature, and the axis z normal to the middle surface. r_x and r_y are the principal radii of curvature lying in the xz and yz planes, respectively. The resultant forces per unit length of the normal sections are given as

$$N_{x} = \int_{-h/2}^{h/2} \sigma_{x} \left(1 - \frac{z}{r_{y}}\right) dz, \quad N_{y} = \int_{-h/2}^{h/2} \sigma_{y} \left(1 - \frac{z}{r_{x}}\right) dz$$

$$N_{xy} = \int_{-h/2}^{h/2} \tau_{xy} \left(1 - \frac{z}{r_{y}}\right) dz, \quad N_{yx} = \int_{-h/2}^{h/2} \tau_{yx} \left(1 - \frac{z}{r_{x}}\right) dz \quad (47.88)$$

$$Q_{x} = \int_{-h/2}^{h/2} \tau_{xz} \left(1 - \frac{z}{r_{y}}\right) dz, \quad Q_{y} = \int_{-h/2}^{h/2} \tau_{yx} \left(1 - \frac{z}{r_{x}}\right) dz$$

The bending and twisting moments per unit length of the normal sections are given by

$$M_{x} = \int_{-h/2}^{h/2} \sigma_{x} Z \left(1 - \frac{z}{r_{y}} \right) dz, \qquad M_{y} = \int_{-h/2}^{h/2} \sigma_{y} Z \left(1 - \frac{z}{r_{x}} \right) dz$$

$$M_{xy} = -\int_{-h/2}^{h/2} \tau_{xy} Z \left(1 - \frac{z}{r_{y}} \right) dz, \qquad M_{yx} = \int_{-h/2}^{h/2} \sigma_{yx} Z \left(1 - \frac{z}{r_{x}} \right) dz$$

$$(47.89)$$

It is assumed, in bending of the shell, that linear elements such as AD and BC (Fig. 47.43), which are normal to the middle surface of the shell, remain straight and become normal to the deformed middle surface of the shell. If the conditions of a shell are such that bending can be neglected, the problem of stress analysis is greatly simplified, since the resultant moments (Eq. (47.89)) vanish along with shearing forces Q_x and Q_y in Eq. (47.88). Thus the only unknowns are N_x , N_y , and $N_{xy} = N_{yx}$; these are called membrane forces.

Shells of Revolution

Shells having the form of surfaces of revolution find extensive application in various kinds of containers, tanks, and domes. Consider an element of a shell cut by two adjacent meridians and two parallel circles, as shown in Fig. 47.44. There will be no shearing forces on the sides of the element because of the symmetry of loading. By considering the equilibrium in the direction of the tangent to the meridian and z, two equations of equilibrium are written, respectively, as

$$\frac{d}{d\phi} (N_{\phi} r_0) - N_{\theta} r_1 \cos\phi + Y_{r_1 r_0} = 0$$

$$N_{\phi} r_0 + N_{\theta} r_1 \sin\phi + Z_{r_1 r_0} = 0$$
(47.90)

The forces N_{θ} and N_{ϕ} can be calculated from Eq. (47.90) if the radii r_0 and r_1 and the components Y and Z of the intensity of the external load are given.



FIGURE 47.44 Element from shells of revolution — symmetrical loading.



FIGURE 47.45 Spherical dome.

Spherical Dome

The spherical shell shown in Fig. 47.45 is assumed to be subjected to its own weight; the intensity of the self weight is assumed as a constant value, q_o , per unit area. Considering an element of the shell at an angle φ , the self weight of the portion of the shell above this element is obtained as

$$R = 2\pi \int_{0}^{\phi} a^2 q_0 \sin \phi d\phi$$
$$= 2\pi a^2 q_0 (1 - \cos \phi)$$

Considering the equilibrium of the portion of the shell above the parallel circle, defined by the angle ϕ , we can write,

$$2\pi r_{o} N_{o} \sin \phi + R = 0 \tag{47.91}$$

Therefore,

$$N_{\varphi} = -\frac{aq_{o}(1-\cos\varphi)}{\sin^{2}\varphi} = -\frac{aq_{o}}{1+\cos\varphi}$$

We can write from Eq. (47.90)

$$\frac{N_{\varphi}}{r_1} + \frac{N_{\theta}}{r_2} = -Z \tag{47.92}$$

substituting for N_{ϕ} and z = R in Eq. (47.92)

$$N_{\theta} = -aq_{o} \left(\frac{1}{1 + \cos \phi} - \cos \phi \right)$$

It is seen that the forces N_{ϕ} are always negative. There is thus a compression along the meridians that increases as the angle ϕ increases. The forces N_{θ} are also negative for small angles ϕ . The stresses as calculated above will represent the actual stresses in the shell with great accuracy if the supports are of such a type that the reactions are tangent to the meridians, as shown in Figure 47.45.

Conical Shells

If a force P is applied in the direction of the axis of the cone, as shown in Fig. 47.46, the stress distribution is symmetrical and we obtain





FIGURE 47.46 Conical shell.



$$N_{\phi} = -\frac{P}{2\pi r_0 \cos \alpha}$$

By Eq. (47.92), one obtains $N_{\theta} = 0$.

In the case of a conical surface in which the lateral forces are symmetrically distributed, the membrane stresses can be obtained by using Eqs. (47.91) and (47.92). The curvature of the meridian in the case of a cone is zero, and hence $r_1 = \infty$; Eqs. (47.91) and (47.92) can therefore be written as

$$N_{\phi} = -\frac{R}{2\pi r_0 \sin \phi}$$

and

$$N_{\theta} = -r_2 Z = -\frac{Zr_0}{\sin\phi}$$

If the load distribution is given, N_{ϕ} and N_{θ} can be calculated independently.

For example, a conical tank filled with a liquid of specific weight γ is considered in Fig. 47.47. The pressure at any parallel circle mn is

$$p = -Z = \gamma(d - y)$$

For the tank, $\phi = \alpha + (\pi/2)$ and $r_0 = y \tan \alpha$. Therefore,

$$N_{\theta} = \frac{\gamma (d - y) y \tan \alpha}{\cos \alpha}$$

 N_{θ} is maximum when y = d/2 and hence

$$(N_{\theta})_{max} = \frac{\gamma d^2 \tan \alpha}{4 \cos \alpha}$$

The term R in the expression for N_{ϕ} is equal to the weight of the liquid in the conical part mno, and the cylindrical part must be as shown in Fig. 47.46. Therefore,

$$R = -\left[\frac{1}{3}\pi y^{3} \tan^{2}\alpha + \pi y^{2} \tan^{2}\alpha(d-y)\right]\gamma$$
$$= -\pi\gamma y^{2}\left(d-\frac{2}{3}y\right)\tan^{2}\alpha$$



FIGURE 47.48 Element from shells of revolution — unsymmetrical loading.

Hence,

$$N_{\phi} = \frac{\gamma y \left(d - \frac{2}{3} y \right) \tan \alpha}{2 \cos \alpha}$$

 N_{ϕ} is maximum when $y = \frac{3}{4} d$ and

$$\left(N_{\phi}\right)_{\max} = \frac{3}{16} \frac{d^2 \gamma \tan \alpha}{\cos \alpha}$$

The horizontal component of N_{ϕ} is taken by the reinforcing ring provided along the upper edge of the tank. The vertical components constitute the reactions supporting the tank.

Shells of Revolution Subjected to Unsymmetrical Loading

Consider an element cut from a shell by two adjacent meridients and two parallel circles, as shown in Fig. 47.48. In general cases, shear forces $N_{\phi\theta} = N_{\theta\phi}$ and normal forces N_{ϕ} and N_{θ} will act on the sides of the element. Projecting the forces on the element in the y direction, we obtain the governing equation:

$$\frac{\partial}{\partial \phi} \left(N_{\phi} r_0 \right) + \frac{\partial N_{\theta \phi}}{\partial \theta} r_1 - N_{\theta} r_1 \cos \phi + Y_{r_1 r_0} = 0$$
(47.93)

Similarly the forces in the x direction can be summed up to give

$$\frac{\partial}{\partial \phi} \left(r_0 N_{\phi \theta} \right) + \frac{\partial N_{\theta}}{\partial \theta} r_1 + N_{\theta \phi} r_1 \cos \phi + X_{r_0 r_1} = 0$$
(47.94)

Since the projection of shearing forces on the z axis vanishes, the third equation is the same as Eq. (47.92). The problem of determining membrane stresses under unsymmetrical loading reduces to solving Eqs. (47.92) to (47.94) for given values of the components X, Y, and Z of the intensity of the external load.

Cylindrical Shells

It is assumed that the generator of the shell is horizontal and parallel to the x axis. An element is cut from the shell by two adjacent generators and two cross sections perpendicular to the x axis, and its position is defined by the coordinate x and the angle φ . The forces acting on the sides of the element are shown in Fig. 47.49b.



FIGURE 47.49 Membrane forces on a cylindrical shell element.

The components of the distributed load over the surface of the element are denoted as X, Y, and Z. Considering the equilibrium of the element and summing up the forces in the x direction, we obtain

$$\frac{\partial N_x}{\partial x} r d\phi dx + \frac{\partial N_{\phi x}}{\partial \phi} d\phi dx + Xr d\phi dx = 0$$

The corresponding equations of equilibrium in the y and z directions are given, respectively, as

$$\frac{\partial N_{x\phi}}{\partial x} r d\phi dx + \frac{\partial N_{\phi}}{\partial \phi} d\phi dx + Y r d\phi dx = 0$$

$$N_{\phi} d\phi dx + Z r d\phi dx = 0$$

The three equations of equilibrium can be simplified and represented in the following form:

$$\frac{\partial N_{x}}{\partial x} + \frac{1}{r} \frac{\partial N_{x\phi}}{\partial \phi} = -X$$

$$\frac{\partial N_{x\phi}}{\partial x} + \frac{1}{r} \frac{\partial N_{\phi}}{\partial \phi} = -Y$$

$$N_{\phi} = -Zr$$
(47.95)

In each particular case we readily find the value of N_{ϕ} . Substituting this value in the second of the equations, we then obtain $N_{x\phi}$ by integration. Using the value of $N_{x\phi}$ thus obtained, we find N_x by integrating the first equation.

Symmetrically Loaded Circular Cylindrical Shells

To establish the equations required for the solution of a symmetrically loaded circular cylinder shell, we consider an element, as shown in Figs. 47.49a and 47.50. From symmetry, the membrane shearing forces $N_{x\phi} = N_{\phi x}$ vanish in this case; forces N_{ϕ} are constant along the circumference. From symmetry, only the forces Q_z do not vanish. Considering the moments acting on the element in Fig. 47.50, from symmetry it can be concluded that the twisting moments $M_{x\phi} = M_{\phi x}$ vanish and that the bending moments M_{ϕ} are constant along the circumference. Under such conditions of symmetry, three of the six equations of equilibrium of the element are identically satisfied. We have to consider only the equations, obtained by



FIGURE 47.50 Stress resultants in a cylindrical shell element.

projecting the forces on the x and z axes and by taking the moment of the forces about the y axis. For example, consider a case in which external forces consist only of a pressure normal to the surface. The three equations of equilibrium are

$$\frac{dN}{dx}a dx d\phi = 0$$

$$\frac{dQ_x}{dx}a dx d\phi + N_{\phi} dx d\phi + Za dx d\phi = 0$$

$$\frac{dM_x}{dx}a dx d\phi - Q_x a dx d\phi = 0$$
(47.96)

The first one indicates that the forces N_x are constant, and they are taken equal to zero in the further discussion. If they are different from zero, the deformation and stress corresponding to such constant forces can be easily calculated and superposed on stresses and deformations produced by lateral load. The remaining two equations are written in the simplified form:

$$\frac{dQ_x}{dx} + \frac{1}{a}N_{\varphi} = -Z$$

$$\frac{dM_x}{dx} - Q_x = 0$$
(47.97)

These two equations contain three unknown quantities: N_{φ} , Q_x , and M_x . We need, therefore, to consider the displacements of points in the middle surface of the shell.

The component v of the displacement in the circumferential direction vanishes because of symmetry. Only the components u and w in the x and z directions, respectively, are to be considered. The expressions for the strain components then become

$$\varepsilon_{\rm x} = \frac{{\rm d}u}{{\rm d}x} \qquad \varepsilon_{\varphi} = -\frac{w}{a}$$
(47.98)

By Hooke's law, we obtain

$$N_{x} = \frac{Eh}{1 - \nu^{2}} \left(\varepsilon_{x} + \nu \varepsilon_{\phi} \right) = \frac{Eh}{1 - \nu^{2}} \left(\frac{du}{dx} - \nu \frac{w}{a} \right) = 0$$

$$N_{\phi} = \frac{Eh}{1 - \nu^{2}} \left(\varepsilon_{\phi} + \nu \varepsilon_{x} \right) = \frac{Eh}{1 - \nu^{2}} \left(-\frac{w}{a} + \nu \frac{du}{dx} \right) = 0$$
(47.99)

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From the first of these equation it follows that

and the second equation gives

Considering the bending moments, we conclude from symmetry that there is no change in curvature in the circumferential direction. The curvature in the x direction is equal to $-d^2w/dx^2$. Using the same equations as the ones for plates, we then obtain

 $N_{\varphi} = -\frac{Ehw}{a}$

$$M_{\phi} = vM_{x}$$

$$M_{x} = -D\frac{d^{2}w}{dx^{2}}$$
(47.101)

(47.100)

where

 $D = \frac{Eh^3}{12(1-v^2)}$

is the flexural rigidity per unit length of the shell.

Eliminating Q_x from Eq. (47.97), we obtain

 $\frac{d^2 M_x}{dx^2} + \frac{1}{a} N_{\varphi} = -Z$

from which, by using Eqs. (47.100) and (47.101), we obtain

 $\frac{d^2}{dx^2} \left(D \frac{d^2 w}{dx^2} \right) + \frac{Eh}{a^2} w = Z$ (47.102)

All problems of symmetrical deformation of circular cylindrical shells thus reduce to the integration of Eq. (47.102).

The simplest application of this equation is obtained when the thickness of the shell is constant. Under such conditions Eq. (47.102) becomes

 $D\frac{d^4w}{dx^4} + \frac{Eh}{a^2}w = Z$

using the notation

$$B^{4} = \frac{Eh}{4a^{2}D} = \frac{3(1-v^{2})}{a^{2}h^{2}}$$
(47.103)

Equation (47.103) can be represented in the simplified form

$$\frac{d^4 w}{dx^4} + 4\beta^4 w = \frac{Z}{D}$$
(47.104)

$$\frac{\mathrm{du}}{\mathrm{dx}} = v \frac{\mathrm{w}}{\mathrm{a}}$$

The general solution of this equation is

$$w = e^{\beta x} \left(C_1 \cos\beta x + C_2 \sin\beta x \right) + e^{-\beta x} \left(C_3 \cos\beta x + C_4 \sin\beta x \right) + f(x)$$
(47.105)

Detailed treatment of the shell theory can be obtained from Timoshenko and Woinowsky-Krieger (1959) and Gould (1988).

47.7 Influence Lines

Bridges, industrial buildings with traveling cranes, and frames supporting conveyer belts are often subjected to moving loads. Each member of these structures must be designed for the most severe conditions that can possibly be developed in that member. Live loads should be placed at the positions where they will produce these severe conditions. The critical positions for placing live loads will not be the same for every member. On some occasions it is possible by inspection to determine where to place the loads to give the most critical forces, but on many other occasions it is necessary to resort to certain criteria to find the locations. The most useful of these methods is the influence lines.

An influence line for a particular response such as reaction, shear force, bending moment, and axial force is defined as a diagram, the ordinate to which at any point equals the value of that response attributable to a unit load acting at that point on the structure. Influence lines provide a systematic procedure for determining how the force in a given part of a structure varies as the applied load moves about on the structure. Influence lines of responses of statically determinate structures consist only of straight lines, whereas for statically indeterminate structures they consist of curves. They are primarily used to determine where to place live loads to cause maximum force and to compute the magnitude of those forces. The knowledge of influence lines helps to study the structural response under different moving load conditions.

Influence Lines for Shear in Simple Beams

Figure 47.51 shows influence lines for shear at two sections of a simply supported beam. It is assumed that positive shear occurs when the sum of the transverse forces to the left of a section is in the upward direction or when the sum of the forces to the right of the section is downward. A unit force is placed at various locations, and the shear forces at sections 1-1 and 2-2 are obtained for each position of the unit load. These values give the ordinate of influence line with which the influence line diagrams for shear force at sections 1-1 and 2-2 can be constructed. Note that the slope of the influence line for shear



FIGURE 47.51 Influence line for shear force.



FIGURE 47.52 Influence line for bending moment.

on the left of the section is equal to the slope of the influence line on the right of the section. This information is useful in drawing shear force influence lines in other cases.

Influence Lines for Bending Moment in Simple Beams

Influence lines for bending moment at the same sections, 1-1 and 2-2, of the simple beam considered in Fig. 47.51 are plotted as shown in Fig. 47.52. For a section, when the sum of the moments of all the forces to the left is clockwise or when the sum to the right is counterclockwise, the moment is taken as positive. The values of bending moment at sections 1-1 and 2-2 are obtained for various positions of unit load and plotted as shown in the figure.

It should be understood that a shear or bending moment diagram shows the variation of shear or moment across an entire structure for loads fixed in one position. On the other hand, an influence line for shear or moment shows the variation of that response at one particular section in the structure caused by the movement of a unit load from one end of the structure to the other.

Influence lines can be used to obtain the value of a particular response for which they are drawn when the beam is subjected to any particular type of loading. If, for example, a uniform load of intensity q_o per unit length is acting over the entire length of the simple beam shown in Fig. 47.51, the shear force at section 1-1 is given by the product of the load intensity, q_o , and the net area under the influence line diagram. The net area is equal to 0.3, and the shear force at section 1-1 is therefore equal to 0.3 q_o . In the same way, the bending moment at the section can be found as the area of the corresponding influence line diagram times the intensity of loading, q_o . The bending moment at the section is equal to 0.08 q_ol^2 .

Influence Lines for Trusses

Influence lines for support reactions and member forces may be constructed in the same manner as those for various beam functions. They are useful to determine the maximum load that can be applied to the truss. The unit load moves across the truss, and the ordinates for the responses under consideration may be computed for the load at each panel point. Member force, in most cases, does not need to be calculated for every panel point, because certain portions of influence lines can readily be seen to consist of straight lines for several panels. One method used for calculating the forces in a chord member of a truss is the method of sections, discussed earlier.

The truss shown in Fig. 47.53 is considered for illustrating the construction of influence lines for trusses.

The member forces in U_1U_2 , L_1L_2 , and U_1L_2 are determined by passing section 1-1 and considering the equilibrium of the free-body diagram of one of the truss segments. Unit load is placed at L_1 first, and the force in U_1U_2 is obtained by taking the moment about L_2 of all the forces acting on the righthand segment of the truss and dividing the resulting moment by the lever arm (the perpendicular distance



FIGURE 47.53 Influence line for truss.

of the force in U_1U_2 from L_2). The value thus obtained gives the ordinate of the influence diagram at L_1 in the truss. The ordinate at L_2 , obtained similarly, represents the force in U_1U_2 for a unit load placed at L_2 . The influence line can be completed with two other points, one at each of the supports. The force in the member L_1L_2 due to a unit load placed at L_1 and L_2 can be obtained in the same manner, and the corresponding influence line diagram can be completed. By considering the horizontal component of force in the diagonal of the panel, the influence line for force in U_1L_2 , and U_1L_2 . Influence line ordinates for the force in a chord member of a "curved chord" truss may be determined by passing a vertical section through the panel and taking moments at the intersection of the diagonal and the other chord.

Qualitative Influence Lines

One of the most effective methods of obtaining influence lines is using Müller-Breslau's principle, which states that the ordinates of the influence line for any response in a structure are equal to those of the deflection curve obtained by releasing the restraint corresponding to this response and introducing a corresponding unit displacement in the remaining structure. In this way, the shape of the influence lines for both statically determinate and indeterminate structures can be easily obtained, especially for beams.

To draw the influence lines of a

- 1. support reaction, remove the support and introduce a unit displacement in the direction of the corresponding reaction to the remaining structure, as shown in Fig. 47.54, for a symmetrical overhang beam.
- 2. shear, make a cut at the section and introduce a unit relative translation (in the direction of positive shear) without relative rotation of the two ends at the section, as shown in Fig. 47.55.
- 3. bending moment, introduce a hinge at the section (releasing the bending moment) and apply bending (in the direction corresponding to the positive moment) to produce a unit relative rotation of the two beam ends at the hinged section, as shown in Fig. 47.56.

Influence Lines for Continuous Beams

Using Müller-Breslau's principle, the shape of the influence line of any response of a continuous beam can be sketched easily. One of the methods for beam deflection can then be used for determining the



FIGURE 47.54 Influence line for support reaction.



FIGURE 47.55 Influence line for midspan shear force.



FIGURE 47.56 Influence line for midspan bending moment.



FIGURE 47.57 Influence line for bending moment — two-span beam.

ordinates of the influence line at critical points. Figures 47.57 to 47.59 show the influence lines of the bending moment at various points of two-, three-, and four-span continuous beams.

47.8 Energy Methods

Energy methods are a powerful tool in obtaining numerical solutions of statically indeterminate problems. The basic quantity required is the *strain energy*, or work stored due to deformations, of the structure.

Strain Energy Due to Uniaxial Stress

In an axially loaded bar with a constant cross section the applied load causes normal stress σ_y , as shown in Fig. 47.60. The tensile stress σ_y increases from zero to a value σ_y as the load is gradually applied. The



FIGURE 47.58 Influence line for bending moment — three-span beam.



FIGURE 47.59 Influence line for bending moment — four-span beam.



FIGURE 47.60 Axial loaded bar.

original, unstrained position of any section such as C-C will be displaced by an amount dv. A section D-D located a differential length below C-C will have been displaced by an amount $v + (\partial v/\partial y) dy$. As σ_y varies with the applied load, from zero to σ_y the work done by the forces external to the element can be shown to be

$$dV = \frac{1}{2E}\sigma_y^2 Ady = \frac{1}{2}\sigma_y \varepsilon_y Ady$$
(47.106)

in which A = the area of cross section of the bar

 ε_{y} = the strain in the direction of σ_{y} .



FIGURE 47.61 Beam under arbitrary bending load.

Strain Energy in Bending

It can be shown that the strain energy of a differential volume dxdydz stressed in tension or compression in the x direction only by a normal stress σ_x will be

$$dV = \frac{1}{2E}\sigma_x^2 dxdydz = \frac{1}{2}\sigma_x \varepsilon_x dxdydz$$
(47.107)

When σ_x is the bending stress given by $\sigma_x = My/I$ (see Fig. 47.61), then

$$dV = \frac{1}{2E} \frac{M^2 y^2}{I^2} dx dy dz$$

where I is the moment of inertia of the cross-sectional area about the neutral axis.

The total strain energy of bending of a beam is obtained as

$$V = \iiint_{\text{volume}} \frac{1}{2E} \frac{M^2}{I^2} y^2 \, dz \, dy \, dx$$

where

$$I = \iint_{\text{area}} y^2 \, dz \, dy$$

Therefore

$$V = \int_{\text{length}} \frac{M^2}{2EI} dx$$
 (47.108)

Strain Energy in Shear

Figure 47.62 shows an element of volume dxdydz subjected to shear stress τ_{xy} and τ_{yx} . For static equilibrium, it can readily be shown that

$$\tau_{xy} = \tau_{yy}$$

The shear strain, γ , is defined as AB/AC. For small deformations, it follows that

$$\gamma_{xy} = \frac{AB}{AC}$$

Hence, the angle of deformation, γ_{xy} , is a measure of the shear strain. The strain energy for this differential volume is obtained as



FIGURE 47.62 Shear loading.

$$dV = \frac{1}{2}(\tau_{xy} dz dx)\gamma_{xy} dy = \frac{1}{2}\tau_{xy}\gamma_{xy} dx dy dz$$
(47.109)

Hooke's law for shear stress and strain is

$$\gamma_{xy} = \frac{\tau_{xy}}{G} \tag{47.110}$$

where G is the shear modulus of elasticity of the material. The expression for strain energy in shear reduces to

$$dV = \frac{1}{2G} \tau_{xy}^2 dx dy dz$$
 (47.111)

The Energy Relations in Structural Analysis

The energy relations or laws, such as the law of conservation of energy, the theorem of virtual work, the theorem of minimum potential energy, and the theorem of complementary energy, are of fundamental importance in structural engineering and are used in various ways in structural analysis.

The Law of Conservation of Energy

The law of conservation of energy states that if a structure and the external loads acting on it are isolated so that these neither receive nor give out energy, then the total energy of this system remains constant.

A typical application of the law of conservation of energy can be made by referring to Fig. 47.63, which shows a cantilever beam of constant cross sections subjected to a concentrated load at its end. If only bending strain energy is considered,

$$\frac{P\delta}{2} = \int_{0}^{2} \frac{M^2 dx}{2EI}$$

Substituting M = -Px and integrating along the length gives

$$\delta = \frac{PL^3}{3EI} \tag{47.112}$$



FIGURE 47.63 Cantilever beam.



FIGURE 47.64 Equilibrium of a simple supported beam under loading.



FIGURE 47.65 Simply supported beam under point loading.

The Theorem of Virtual Work

The theorem of virtual work can be derived by considering the beam shown in Fig. 47.64. The full curved line represents the equilibrium position of the beam under the given loads. Assume the beam to be given an additional small deformation consistent with the boundary conditions. This is called a virtual deformation and corresponds to increments of deflection $\Delta_{y1}, \Delta_{y2}, \ldots, \Delta_{yn}$ at loads P_1, P_2, \ldots, P_n , as shown by the dashed line.

The change in potential energy of the loads is given by

$$\Delta(\mathbf{P}.\mathbf{E}.) = \sum_{i=1}^{n} \mathbf{P}_i \Delta \mathbf{y}_i$$
(47.113)

By the law of conservation of energy this must be equal to the internal strain energy stored in the beam. Hence, we may state the theorem of virtual work as: if a body in equilibrium under the action of a system of external loads is given any small (virtual) deformation, then the work done by the external loads during this deformation is equal to the increase in internal strain energy stored in the body.

The Theorem of Minimum Potential Energy

Let us consider the beam shown in Fig. 47.65. The beam is in equilibrium under the action of loads P_1 , P_2 , P_3 , ..., P_i , ..., P_n . The curve ACB defines the equilibrium positions of the loads and reactions. Now apply by some means an additional small displacement to the curve so that it is defined by AC'B. Let y_i be the original equilibrium displacement of the curve beneath a particular load P_i . The additional small displacement is called δ_{yi} . The potential energy of the system while it is in the equilibrium configuration is found by comparing the potential energy of the beam and loads in equilibrium and in the undeflected position. If the change in potential energy of the loads is W and the strain energy of the system is

$$U = W + V$$
 (47.114)

If we neglect the second-order terms, then

$$\delta \mathbf{U} = \delta (\mathbf{W} + \mathbf{V}) = 0 \tag{47.115}$$

The above is expressed as the principle or theorem of minimum potential energy, which can be stated as: if all displacements satisfy given boundary conditions, those that satisfy the equilibrium conditions make the potential energy a minimum.


FIGURE 47.66 Example 47.6.

Castigliano's Theorem

This theorem applies only to structures stressed within the elastic limit, and all deformations must be linear homogeneous functions of the loads.

For a beam in equilibrium, as in Fig. 47.64, the total potential energy is

$$\mathbf{U} = -\left[P_1 \mathbf{y}_1 + P_2 \mathbf{y}_2 + \dots P_j \mathbf{y}_j + \dots P_n \mathbf{y}_n\right] + \mathbf{V}$$
(47.116)

For an elastic system, the strain energy, V, turns out to be one half the change in the potential energy of the loads.

$$V = \frac{1}{2} \sum_{i=1}^{i=n} P_i y_i$$
 (47.117)

Castigliano's theorem results from studying the variation in the strain energy, V, produced by a differential change in one of the loads, say P_i.

If the load P_j is changed by a differential amount δP_j and if the deflections y are linear functions of the loads, then

$$\frac{\partial V}{\partial P_j} = \frac{1}{2} \sum_{i=1}^{i=n} P_i \frac{\partial y_i}{\partial P_j} + \frac{1}{2} y_j = y_j$$
(47.118)

Castigliano's theorem states that the partial derivatives of the total strain energy of any structure with respect to any one of the applied forces is equal to the displacement of the point of application of the force in the direction of the force.

To find the deflection of a point in a beam that is not the point of application of a concentrated load, one should apply a load P = 0 at that point and carry the term P into the strain energy equation. Finally, introduce the true value of P = 0 into the expression for the answer.

Example 47.6

Determination of the bending deflection at the free end of a cantilever, loaded as shown in Fig. 47.66, is required.

Solution:

$$V = \int_{0}^{L} \frac{M^{2}}{2EI} dx$$
$$\Delta = \frac{\partial V}{\partial W_{1}} = \int_{0}^{L} \frac{M}{EI} \frac{\partial M}{\partial W_{1}} dx$$

$$M = W_1 x \qquad 0 < x \frac{L}{2}$$
$$= W_1 x + W_2 \left(x - \frac{\lambda}{2} \right) \qquad \frac{L}{2} < x < L$$
$$\Delta = \frac{1}{EI} \int_{0}^{\lambda/2} W_1 x \times x \, dx + \frac{1}{EI} \int_{\lambda/2}^{\lambda} \left[W_1 x + W_2 \left(x - \frac{P}{2} \right) \right] x \, dx$$
$$= \frac{W_1 \lambda^3}{24EI} + \frac{7W_1 \lambda^3}{24EI} + \frac{5W_2 \lambda^3}{48EI}$$
$$= \frac{W_1 \lambda^3}{3EI} + \frac{5W_2 \lambda^3}{48EI}$$

Castigliano's theorem can be applied to determine deflection of trusses as follows: We know that the increment of strain energy for an axially loaded bar is given as

$$dV = \frac{1}{2E}\sigma_y^2 A dy$$

Substituting $\sigma_y = S/A$, where S is the axial load in the bar, and integrating over the length of the bar, the total strain energy of the bar is given as

$$V = \frac{S^2 L}{2AE}$$
(47.119)

The deflection component Δ_i of the point of application of a load P_i in the direction of P_i is given as

$$\Delta_{i} = \frac{\partial V}{\partial P_{i}} = \frac{\partial}{\partial P_{i}} \sum \frac{S^{2}L}{2AE} = \sum \frac{S \frac{\partial S}{\partial P_{i}}L}{AE}$$

Example 47.7

Determine the vertical deflection at g of the truss subjected to three-point load, as shown in Fig. 47.67. Let us first replace the 20 load at g by P and carry out the calculations in terms of P. At the end, P will be replaced by the actual load of 20.

Member	А	L	S	$\frac{\delta S}{\delta P}$	n	$nS\frac{\delta S}{\delta P}~\frac{L}{A}$
ab	2	25	-(33.3 + 0.83P)	-0.83	2	(691 + 17.2P)
af	2	20	(26.7 + 0.67P)	0.67	2	(358 + 9P)
fg	2	20	(26.7 + 0.67P)	0.67	2	(358 + 9P)
bf	1	15	20	0	2	0
bg	1	25	0.83P	0.83	2	34.4P
bc	2	20	-26.7 - 1.33P	-1.33	2	(710 + 35.4P)
cg	1	15	0	0	1	0
				$\sum \frac{s\frac{\delta}{\delta}}{s}$	$\frac{S}{P}L$	2117 + 105P

Note: n indicates the number of similar members.



FIGURE 47.67 Example 47.7.

With P = 20,

$$\Delta_{g} = \sum \frac{S \frac{\delta S}{\delta P} L}{AE} = \frac{(2117 + 105 \times 20) \times 12}{30 \times 10^{3}} = 1.69$$

Unit Load Method

The unit load method is a versatile tool in the solution of deflections of both trusses and beams. Consider an elastic body in equilibrium under loads P_1 , P_2 , P_3 , P_4 , ... P_n and a load p applied at point O, as shown in Fig. 47.68. By Castigliano's theorem, the component of the deflection of point O in the direction of the applied force p is

$$\delta_{o_p} = \frac{\partial V}{\partial p} \tag{47.120}$$

in which V is the strain energy of the body. It has been shown in Eq. (47.108) that the strain energy of a beam, neglecting shear effects, is given by

$$V = \int_{0}^{1} \frac{M^2}{2EI} dx$$

Also it was shown that if the elastic body is a truss, from Eq. (47.119)

$$V = \sum \frac{S^2 L}{2 A E}$$

For a beam, therefore, from Eq. (47.120)

$$\delta_{o_p} = \int_{L} \frac{M \frac{\partial M}{\partial p} dx}{EI}$$
(47.121)



FIGURE 47.68 Elastic body in equilibrium under load.

and for a truss,

$$\delta_{o_p} = \sum \frac{S \frac{\partial S}{\partial p} L}{AE}$$
(47.122)

The bending moments M and the axial forces S are functions of the load p as well as of the loads P_1 , P_2 , ... P_n . Let a unit load be applied at O on the elastic body and the corresponding moment be *m* if the body is a beam and the forces in the members of the body *u* if the body is a truss. For the body in Fig. 47.68 the moments M and the forces S due to the system of forces P_1 , P_2 , ... P_n and p at O applied separately can be obtained by superposition as

$$\mathbf{M} = \mathbf{M}_{\mathbf{p}} = \mathbf{p}m \tag{47.123}$$

$$S = S_p + pu \tag{47.124}$$

in which M_p and S_p are, respectively, moments and forces produced by $P_1,\,P_2,\,\ldots\,P_n$. Then

$$\frac{\partial M}{\partial p}$$
 = m = moments produced by a unit load at O (47.125)

$$\frac{\partial S}{\partial p} = u = \text{ stresses produced by a unit load at O}$$
 (47.126)

Using Eqs. (47.125) and (47.126) in Eqs. (47.121) and (47.122), respectively,

$$\delta_{o_p} = \int_{L} \frac{Mmdx}{EI}$$
(47.127)

$$\delta_{o_p} = \Sigma \frac{SuL}{AE}$$
(47.128)

Example 47.8

Determine, using the unit load method, the deflection at C of a simple beam of constant cross section loaded as shown in Fig. 47.69a.

Solution:

The bending moment diagram for the beam due to the applied loading is shown in Fig. 47.69b. A unit load is applied at C, where it is required to determine the deflection, as shown in Fig. 47.69c; the corresponding bending moment diagram is shown in Fig. 47.69d. Now, using Eq. (47.127), we have

$$\delta_{c} = \int_{0}^{L} \frac{Mmdx}{EI}$$
$$= \frac{1}{EI} \int_{0}^{\frac{L}{4}} (Wx) \left(\frac{3}{4}x\right) dx + \frac{1}{EI} \int_{\frac{L}{4}}^{\frac{3L}{4}} \left(\frac{WL}{4}\right) \frac{1}{4} (L-x) dx$$



FIGURE 47.69 Example 47.8.

$$+\frac{1}{\mathrm{EI}}\int_{\frac{3\mathrm{L}}{4}}^{\mathrm{L}}\mathrm{W}(\mathrm{L}-\mathrm{x})\frac{1}{4}(\mathrm{L}-\mathrm{x})\mathrm{d}\mathrm{x}$$
$$=\frac{\mathrm{WL}^{3}}{48\mathrm{EI}}$$

Further details on energy methods in structural analysis may be found in Borg and Gennaro (1959).

47.9 Matrix Methods

In this method, a set of simultaneous equations that describe the load–deformation characteristics of the structure under consideration are formed. These equations are solved using the matrix algebra to obtain the load–deformation characteristics of discrete or finite elements into which the structure has been subdivided. The matrix method is ideally suited for performing structural analysis using a computer. In general, there are two approaches for structural analysis using the matrix analysis. The first is called the flexibility method, in which forces are used as independent variables, and the second is called the stiffness method, which employs deformations as the independent variables. The two methods are also called the force method and the displacement method, respectively.

Flexibility Method

In this method, the forces and displacements are related to one another by using stiffness influence coefficients. Let us consider, for example, a simple beam in which three concentrated loads, W_1 , W_2 , and W_3 , are applied at sections 1, 2, and 3, respectively, as shown in Fig. 47.70. The deflection at section 1, Δ_1 , can be expressed as

$$\Delta_1 = F_{11}W_1 + F_{12}W_2 + F_{13}W_3$$



FIGURE 47.70 Simple beam under concentrated loads.

in which F_{11} , F_{12} , and F_{13} are called flexibility coefficients and are defined as the deflection at section 1 due to unit loads applied at sections 1, 2, and 3, respectively. Deflections at sections 2 and 3 are similarly given as

$$\Delta_2 = F_{21}W_1 + F_{22}W_2 + F_{23}W_3$$

and

$$\Delta_3 = F_{31}W_1 + F_{32}W_2 + F_{33}W_3 \tag{47.129}$$

These expressions are written in matrix form as

$$\begin{cases} \Delta_1 \\ \Delta_2 \\ \Delta_3 \end{cases} = \begin{bmatrix} F_{11} & F_{12} & F_{13} \\ F_{21} & F_{22} & F_{23} \\ F_{31} & F_{32} & F_{33} \end{bmatrix} \begin{cases} W_1 \\ W_2 \\ W_3 \end{cases}$$

or

$$\left\{\Delta\right\} = \left[F\right]\left\{W\right\} \tag{47.130}$$

Matrix [F] is called the flexibility matrix. It can be shown, by applying Maxwell's reciprocal theorem (Borg and Gennaro, 1959), that matrix [F] is a symmetric matrix.

Let us consider a cantilever beam loaded as shown in Fig. 47.71. The first column in the flexibility matrix can be generated by applying a unit vertical load at the free end of the cantilever, as shown in Fig. 47.71b, and making use of the moment area method. We get

$$F_{11} = \frac{8L^3}{3EI}$$
, $F_{21} = \frac{2L^2}{EI}$, $F_{31} = \frac{5L^3}{6EI}$, $F_{41} = \frac{3L^2}{2EI}$

Columns 2, 3, and 4 are similarly generated by applying unit moment at the free end and unit force and unit moment at the midspan, as shown in Figs. 47.71c to e, respectively. Combining the results, the flexibility matrix can be formed as

$$\begin{cases} \Delta_{1} \\ \Delta_{2} \\ \Delta_{3} \\ \Delta_{4} \end{cases} = \frac{1}{\mathrm{EI}} \begin{bmatrix} \frac{8\mathrm{L}^{3}}{3} & 2\mathrm{L}^{2} & \frac{5\mathrm{L}^{3}}{6} & \frac{3\mathrm{L}^{2}}{2} \\ 2\mathrm{L}^{2} & 2\mathrm{L} & \frac{\mathrm{L}^{2}}{2} & \mathrm{L} \\ \frac{5\mathrm{L}^{3}}{6} & \frac{\mathrm{L}^{2}}{2} & \frac{\mathrm{L}^{3}}{3} & \frac{\mathrm{L}^{2}}{2} \\ \frac{3\mathrm{L}^{2}}{2} & \mathrm{L} & \frac{\mathrm{L}^{2}}{2} & \mathrm{L} \end{bmatrix} \begin{bmatrix} W_{1} \\ W_{2} \\ W_{3} \\ W_{4} \end{bmatrix}$$
(47.131)

For a given structure, it is necessary to subdivide the structure into several elements and to form the flexibility matrix for each of the elements. The flexibility matrix for the entire structure is then obtained by combining the flexibility matrices of the individual elements.



FIGURE 47.71 Cantilever beam.

The force transformation matrix relates what occurs in these elements to the behavior of the entire structure. Using the conditions of equilibrium, it relates the element forces to the structure forces. The principle of conservation of energy may be used to generate transformation matrices.

Stiffness Method

In this method, forces and deformations in a structure are related to one another by means of stiffness influence coefficients. Let us consider a simply supported beam subjected to end moments W_1 and W_2 applied at supports 1 and 2, respectively, and let the rotations be denoted as Δ_1 and Δ_2 , as shown in Fig. 47.72. We can now write the expressions for end moments W_1 and W_2 as

$$(a) \qquad (b) \qquad (b) \qquad (c) \qquad (c)$$

...

FIGURE 47.72 Simply supported beam.



in which K_{11} and K_{12} are called stiffness influence coefficients defined as moments at 1 due to unit rotation at 1 and 2, respectively. The above equations can be written in matrix form as

$$\begin{cases} W_1 \\ W_2 \end{cases} = \begin{vmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{vmatrix} \begin{vmatrix} \Delta_2 \\ \Delta_2 \end{vmatrix}$$

or

$$\{W\} = [K]\{\Delta\} \tag{47.133}$$

Matrix [K] is referred to as the stiffness matrix. It can be shown that the flexibility matrix of a structure is the inverse of the stiffness matrix and vice versa. The stiffness matrix of the whole structure is formed out of the stiffness matrices of the individual elements that make up the structure.



FIGURE 47.73 Axially loaded member.

Element Stiffness Matrix

Axially Loaded Member

Figure 47.73 shows an axially loaded member of a constant cross-sectional area with element forces q_1 and q_2 and displacements δ_1 and δ_2 . They are shown in their respective positive directions. With unit displacement $\delta_1 = 1$ at node 1, as shown in Fig. 47.73, axial forces at nodes 1 and 2 are obtained as

$$\mathbf{K}_{11} = \frac{\mathbf{E}\mathbf{A}}{\mathbf{L}}, \quad \mathbf{K}_{21} = -\frac{\mathbf{E}\mathbf{A}}{\mathbf{L}}$$

In the same way, by setting $\delta_2 = 1$, as shown in Fig. 47.73, the corresponding forces are obtained as

$$\mathbf{K}_{12} = -\frac{\mathbf{E}\mathbf{A}}{\mathbf{L}}, \quad \mathbf{K}_{22} = \frac{\mathbf{E}\mathbf{A}}{\mathbf{L}}$$

The stiffness matrix is written as

$$\begin{cases} \mathbf{q}_1 \\ \mathbf{q}_2 \end{cases} = \begin{bmatrix} \mathbf{K}_{11} & \mathbf{K}_{12} \\ \mathbf{K}_{21} & \mathbf{K}_{22} \end{bmatrix} \begin{cases} \boldsymbol{\delta}_1 \\ \boldsymbol{\delta}_2 \end{cases}$$

or

$$\begin{cases} \mathbf{q}_1 \\ \mathbf{q}_2 \end{cases} = \frac{\mathbf{E}\mathbf{A}}{\mathbf{L}} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \begin{cases} \boldsymbol{\delta}_1 \\ \boldsymbol{\delta}_2 \end{cases}$$
(47.134)

Flexural Member

The stiffness matrix for the flexural element can be constructed by referring to Fig. 47.74. The forces and the corresponding displacements viz. the moments, shears, and corresponding rotations and translations at the ends of the member are defined in the figure. The matrix equation that relates these forces and displacements can be written in the form

$$\begin{bmatrix} \mathbf{q}_1 \\ \mathbf{q}_2 \\ \mathbf{q}_3 \\ \mathbf{q}_4 \end{bmatrix} = \begin{bmatrix} K_{11} & K_{12} & K_{13} & K_{14} \\ K_{21} & K_{22} & K_{23} & K_{24} \\ K_{31} & K_{32} & K_{33} & K_{34} \\ K_{41} & K_{42} & K_{43} & K_{44} \end{bmatrix} \begin{bmatrix} \boldsymbol{\delta}_1 \\ \boldsymbol{\delta}_2 \\ \boldsymbol{\delta}_3 \\ \boldsymbol{\delta}_4 \end{bmatrix}$$



FIGURE 47.74 Beam element — stiffness matrix.

The terms in the first column consist of the element forces q_1 through q_4 that result from displacement $\delta_1 = 1$ when $\delta_2 = \delta_3 = \delta_4 = 0$. This means that a unit vertical displacement is imposed at the left end of the member, while translation at the right end and rotation at both ends are prevented, as shown in Fig. 47.74. The four member forces corresponding to this deformation can be obtained using the moment area method.

The change in slope between the two ends of the member is zero, and the area of the M/EI diagram between these points must therefore vanish. Hence

$$\frac{K_{41}L}{2EI} - \frac{K_{21}L}{2EI} = 0$$

$$K_{21} = K_{41}$$
(47.135)

and

The moment of the M/EI diagram about the left end of the member is equal to unity. Hence

$$\frac{\mathrm{K}_{41}\mathrm{L}}{2\mathrm{EI}}\left(\frac{2\mathrm{L}}{3}\right) - \frac{\mathrm{K}_{21}\mathrm{L}}{2\mathrm{EI}}\left(\frac{\mathrm{L}}{3}\right) = 1$$

and in view of Eq. (47.135),

$$K_{41} = K_{21} = \frac{6 \text{ EI}}{L^2}$$

Finally, moment equilibrium of the member about the right end leads to

$$K_{11} = \frac{K_{21} + K_{41}}{L} = \frac{12 \text{ EI}}{L^3}$$

and from equilibrium in the vertical direction we obtain

$$K_{31} = K_{11} = \frac{12EI}{L^3}$$

The forces act in the directions indicated in Fig. 47.74b. To obtain the correct signs, one must compare the forces with the positive directions defined in Fig. 47.74a. Thus

$$K_{11} = \frac{12 \text{ EI}}{L^3}, \quad K_{21} = -\frac{6 \text{ EI}}{L^2}, \quad K_{31} = -\frac{12 \text{ EI}}{L^3}, \quad K_{41} = \frac{6 \text{ EI}}{L^2}$$

The second column of the stiffness matrix is obtained by letting $\delta_2 = 1$ and setting the remaining three displacements equal to zero, as indicated in Fig. 47.74c. The area of the M/EI diagram between the ends of the member for this case is equal to unity, and hence

$$\frac{K_{22}L}{2EI} - \frac{K_{42}L}{2EI} = 1$$

The moment of the M/EI diagram about the left end is zero, so that

$$\frac{\mathrm{K}_{22}\mathrm{L}}{2\mathrm{EI}}\left(\frac{\mathrm{L}}{3}\right) - \frac{\mathrm{K}_{42}\mathrm{L}}{2\mathrm{EI}}\left(\frac{2\mathrm{L}}{3}\right) = 0$$

Therefore, one obtains

$$K_{22} = \frac{4 \text{ EI}}{L}, \quad K_{42} = \frac{2 \text{ EI}}{L}$$

From vertical equilibrium of the member,

$$K_{12} = K_{32}$$

and moment equilibrium about the right end of the member leads to

$$K_{12} = \frac{K_{22} - K_{42}}{L} = \frac{6EI}{L^2}$$

Comparison of the forces in Fig. 47.74c with the positive directions defined in Fig. 47.74a indicates that all the influence coefficients except k_{12} are positive. Thus

$$K_{12} = -\frac{6 \text{ EI}}{L^2}, \quad K_{22} = \frac{4 \text{ EI}}{L}, \quad K_{32} = \frac{6 \text{ EI}}{L^2}, \quad K_{42} = \frac{2 \text{ EI}}{L}$$

Using Figs. 47.74d and e, the influence coefficients for the third and fourth columns can be obtained. The results of these calculations lead to the following element stiffness matrix:

$$\begin{bmatrix} q_{1} \\ q_{2} \\ q_{3} \\ q_{4} \end{bmatrix} = \begin{bmatrix} \frac{12 \text{EI}}{L^{3}} & -\frac{6 \text{EI}}{L^{2}} & -\frac{12 \text{EI}}{L^{3}} & -\frac{6 \text{EI}}{L^{2}} \\ -\frac{6 \text{EI}}{L^{2}} & \frac{4 \text{EI}}{L} & \frac{6 \text{EI}}{L^{2}} & \frac{2 \text{EI}}{L} \\ -\frac{12 \text{EI}}{L^{3}} & \frac{6 \text{EI}}{L^{2}} & \frac{12 \text{EI}}{L^{3}} & \frac{6 \text{EI}}{L^{2}} \\ -\frac{6 \text{EI}}{L^{2}} & \frac{2 \text{EI}}{L} & \frac{6 \text{EI}}{L^{2}} & \frac{12 \text{EI}}{L^{3}} \\ -\frac{6 \text{EI}}{L^{2}} & \frac{2 \text{EI}}{L} & \frac{6 \text{EI}}{L^{2}} & \frac{4 \text{EI}}{L} \end{bmatrix} \begin{bmatrix} \delta_{1} \\ \delta_{2} \\ \delta_{3} \\ \delta_{4} \end{bmatrix}$$
(47.136)



FIGURE 47.75 Beam element with axial force.

Note that Eq. (47.135) defines the element stiffness matrix for a flexural member with constant flexural rigidity EI.

If the axial load in a frame member is also considered, the general form of an element stiffness matrix for an element shown in Fig. 47.75 becomes

$$\begin{bmatrix} q_{1} \\ q_{2} \\ q_{3} \\ q_{4} \\ q_{5} \\ q_{6} \end{bmatrix} = \begin{bmatrix} \frac{EA}{L} & 0 & 0 & -\frac{EA}{L} & 0 & 0 \\ 0 & \frac{12EI}{L^{3}} & -\frac{6EI}{L^{2}} & 0 & -\frac{12EI}{L^{3}} & -\frac{6EI}{L^{2}} \\ 0 & -\frac{6EI}{L^{2}} & \frac{4EI}{L} & 0 & \frac{6EI}{L^{2}} & \frac{2EI}{L} \\ -\frac{EI}{L} & 0 & 0 & \frac{EI}{L} & 0 & 0 \\ 0 & -\frac{12EI}{L^{3}} & \frac{6EI}{L^{2}} & 0 & \frac{12EI}{L^{3}} & \frac{6EI}{L^{2}} \\ 0 & -\frac{6EI}{L^{2}} & \frac{2EI}{L} & 0 & \frac{6EI}{L^{2}} & \frac{4EI}{L} \end{bmatrix} \begin{bmatrix} \delta_{1} \\ \delta_{2} \\ \delta_{3} \\ \delta_{4} \\ \delta_{5} \\ \delta_{6} \end{bmatrix}$$

or

$$[\mathbf{q}] = [\mathbf{k}_c][\boldsymbol{\delta}] \tag{47.137}$$

The member stiffness matrix can be written as

$$K = \begin{bmatrix} \frac{GJ}{L} & 0 & 0 & -\frac{GJ}{L} & 0 & 0 \\ 0 & \frac{12EI_{z}}{L^{3}} & \frac{6EI_{z}}{L^{2}} & 0 & -\frac{12EI_{z}}{L^{3}} & \frac{6EI_{z}}{L^{2}} \\ 0 & \frac{6EI_{z}}{L^{2}} & \frac{4EI_{z}}{L} & 0 & -\frac{6EI_{z}}{L^{2}} & \frac{2EI_{z}}{L} \\ 0 & -\frac{GJ}{L} & 0 & 0 & \frac{GJ}{L} & 0 & 0 \\ 0 & -\frac{12EI_{z}}{L^{2}} & -\frac{6EI_{z}}{L^{2}} & 0 & \frac{12EI_{z}}{L^{3}} & -\frac{6EI_{z}}{L^{2}} \\ 0 & \frac{6EI_{z}}{L^{2}} & \frac{2EI_{z}}{L} & 0 & -\frac{6EI_{z}}{L^{2}} & \frac{4EI_{z}}{L} \end{bmatrix}$$
(47.138)

Structure Stiffness Matrix

Equation (47.137) has been expressed in terms of the coordinate system of the individual members. In a structure consisting of many members there would be as many systems of coordinates as the number of members. Before the internal actions in the members of the structure can be related, all forces and deflections must be stated in terms of one single system of axes common to all — the global axes. The transformation from element to global coordinates is carried out separately for each element, and the resulting matrices are then combined to form the structure stiffness matrix. A separate transformation matrix [T] is written for each element, and a relation of the form



FIGURE 47.76 Grid member.

$$\left[\delta\right]_{n} = \left[T\right]_{n} \left[\Delta\right]_{n} \tag{47.139}$$

is written in which $[T]_n$ defines the matrix relating the element deformations of element n to the structure deformations at the ends of that particular element. The element and structure forces are related in the same way as the corresponding deformations as

$$\left[\mathbf{q}\right]_{\mathbf{n}} = \left[\mathbf{T}\right]_{\mathbf{n}} \left[\mathbf{W}\right]_{\mathbf{n}} \tag{47.140}$$

where $[q]_n$ contains the element forces for element n and $[W]_n$ contains the structure forces at the extremities of the element. The transformation matrix $[T]_n$ can be used to transform element n from its local coordinates to structure coordinates. We know, for an element n, that the force–deformation relation is given as

$$\left[q\right]_{n} = \left[k\right]_{n} \left[\delta\right]_{n}$$

Substituting for $[q]_n$ and $[\delta]_n$ from Eqs. (47.138) and (47.139), one obtains

$$\begin{split} \left[T\right]_{n}\left[W\right]_{n} &= \left[k\right]_{n}\left[T\right]_{n}\left[\Delta\right]_{n} \\ \left[W\right]_{n} &= \left[T\right]_{n}^{-1}\left[k\right]_{n}\left[T\right]_{n}\left[\Delta\right]_{n} \\ &= \left[T\right]_{n}^{T}\left[k\right]_{n}\left[T\right]_{n}\left[\Delta\right]_{n} \\ &= \left[K\right]_{n}\left[\Delta\right]_{n} \end{split} \tag{47.141} \\ \left[K\right]_{n} &= \left[T\right]_{n}^{T}\left[k\right]_{n}\left[T\right]_{n} \end{split}$$

or

$$[K]_n$$
 is the stiffness matrix that transforms any element n from its local coordinate to structure coordinates. In this way, each element is transformed individually from element coordinate to structure coordinate, and the resulting matrices are combined to form the stiffness matrix for the entire structure.

For example, the member stiffness matrix $[K]_n$ in global coordinates for the truss member shown in Fig. 47.76 is given as

$$[K]_{n} = \frac{AE}{L} \begin{bmatrix} \lambda^{2}\mu & \lambda\mu & -\lambda^{2} & -\lambda\mu \\ \lambda\mu & \mu^{2} & -\lambda\mu & -\mu^{2} \\ -\lambda^{2} & -\lambda\mu & \lambda^{2} & \lambda\mu \\ -\lambda\mu & -\mu^{2} & \lambda\mu & \mu^{2} \end{bmatrix}_{\ell}^{i} k$$
(47.142)

in which $\lambda = \cos \phi$ $\mu = \sin \phi$.



FIGURE 47.77 Flexural member in global coordinate.

To construct $[K]_n$ for a given member, it is necessary to have the values of λ and μ for the member. In addition, the structure coordinates i, j, k, and l at the extremities of the member must be known.

The member stiffness matrix $[K]_n$ in structural coordinates for the flexural member shown in Fig. 47.77 can be written as

$$\begin{bmatrix} K \end{bmatrix}_{n} = \begin{bmatrix} \lambda^{2} \frac{AE}{L} + \mu^{2} \frac{12EI}{L^{3}} & & \\ \mu\lambda \left(\frac{AE}{L} - \frac{12EI}{L^{3}}\right) & \mu^{2} \frac{AE}{L} + \lambda^{2} \frac{12EI}{L^{3}} & \text{symmetric} \\ -\mu \left(\frac{6EI}{L^{2}}\right) & \lambda \frac{6EI_{z}}{L^{2}} & \frac{4EI}{L} & 0 \\ 0 \\ -\lambda^{2} \frac{AE}{L} - \mu^{2} \frac{12EI}{L^{3}} & \mu\lambda \left(\frac{AE}{L} - \frac{12EI}{L^{3}}\right) & \mu \left(\frac{6EI}{L^{2}}\right) & \lambda^{2} \frac{AE}{L} + \mu^{2} \frac{12EI}{L^{3}} & 0 & 0 \\ -\mu\lambda \left(\frac{AE}{L} - \frac{12EI}{L^{3}}\right) & -\left(\mu^{2} \frac{AE}{L} + \lambda^{2} \frac{12EI}{L^{3}}\right) & -\lambda \left(\frac{6EI}{L^{2}}\right) & \mu\lambda \left(\frac{AE}{L} - \frac{12EI}{L^{3}}\right) & \left(\mu \frac{AE}{L} + \lambda^{2} \frac{12EI}{L^{3}}\right) \\ \mu \frac{6EI_{z}}{L^{2}} & \lambda \frac{6EI_{z}}{L^{2}} & \frac{2EI}{L} & \mu \left(\frac{6EI}{L^{2}}\right) & -\lambda \frac{6EI}{L^{2}} & \frac{4EI}{L} \end{bmatrix} \end{bmatrix}$$
(47.143)

where $\lambda = \cos \phi$ and $\mu = \sin \phi$.

Example 47.9

Determine the displacement at the loaded point of the truss shown in Fig. 47.78a. Both members have the same area of cross section: A = 3 and $E = 30 \times 10^{3}$.

The details required to form the element stiffness matrix with reference to structure coordinates axes are listed below (see Fig. 47.78b):

Member	Length	φ	λ	μ	i	j	k	1
1	10	90°	0	1	1	2	3	4
2	18.9	32°	0.85	0.53	1	2	5	6

We now use these data in Eq. (47.142) to form $[K]_n$ for the two elements.



FIGURE 47.78 Example 47.9.

For member 1,

$$\frac{AE}{L} = \frac{3 \times 30 \times 10^3}{120} = 750$$

$$\begin{bmatrix} 1 & 2 & 3 & 4 \\ 0 & 0 & 0 & 0 \\ 0 & 750 & 0 & -750 \\ 0 & 0 & 0 & 0 \\ 0 & -750 & 0 & 750 \end{bmatrix} \begin{bmatrix} 1 \\ 2 \\ 3 \\ 4 \end{bmatrix}$$

For member 2,

$$\frac{AE}{L} = \frac{3 \times 30 \times 10^3}{18.9 \times 12} = 397$$

$$[K]_2 = \begin{vmatrix} 286 & 179 & -286 & -179 & | & 1\\ 179 & 111 & -179 & -111 & | & 2\\ -286 & -179 & 286 & 179 & | & 5\\ -179 & -111 & 179 & 111 & | & 6 \end{vmatrix}$$

Combining the element stiffness matrices [K]₁ and [K]₂, one obtains the structure stiffness matrix as follows:

$$\begin{bmatrix} W_1 \\ W_2 \\ W_3 \\ W_4 \\ W_5 \\ W_6 \end{bmatrix} = \begin{bmatrix} 286 & 179 & 0 & 0 & -286 & -179 \\ 179 & 861 & 0 & -750 & -179 & -111 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & -750 & 0 & 750 & 0 & 0 \\ -286 & -179 & 0 & 0 & 286 & 179 \\ -179 & -111 & 0 & 0 & 179 & 111 \end{bmatrix} \begin{bmatrix} \Delta_1 \\ \Delta_2 \\ \Delta_3 \\ \Delta_4 \\ \Delta_5 \\ \Delta_6 \end{bmatrix}$$

The stiffness matrix can now be subdivided to determine the unknowns. Let us consider Δ_1 and Δ_2 , the deflections at joint 2, which can be determined in view of $\Delta_3 = \Delta_4 = \Delta_5 = \Delta_6 = 0$ as follows:

$$\begin{bmatrix} \Delta_1 \\ \Delta_2 \end{bmatrix} = \begin{bmatrix} 286 & 179 \\ 179 & 861 \end{bmatrix}^{-1} \begin{bmatrix} -9 \\ 7 \end{bmatrix}$$

or

$$\Delta_1 = 0.042$$

 $\Delta_2 = 0.0169$

Example 47.10

A simple triangular frame is loaded at the tip by 20 units of force, as shown in Fig. 47.80. Assemble the structure stiffness matrix and determine the displacements at the loaded node.

Member	Length	А	Ι	φ	λ	μ
1	72	2.4	1037	0	1	0
2	101.8	3.4	2933	45°	0.707	0.707



FIGURE 47.79 Example 47.10.

For members 1 and 2 the stiffness matrices in structure coordinates can be written by making use of Eq. (47.143):

$$[K]_{1} = 10^{3} \times \begin{bmatrix} 1 & 2 & 3 & 4 & 5 & 6 \\ 1 & 0 & 0 & -1 & 0 & 0 & 1 \\ 0 & 1 & 36 & 0 & -1 & 36 & 2 \\ 0 & 36 & 1728 & 0 & -36 & 864 & 3 \\ -1 & 0 & 0 & 1 & 0 & 0 & 4 \\ 0 & -1 & -36 & 0 & 1 & -36 & 5 \\ 0 & 36 & 864 & 0 & -36 & 1728 & 6 \end{bmatrix}$$

and

$$[K]_{2} = 10^{3} \times \begin{bmatrix} 1 & 2 & 3 & 7 & 8 & 9 \\ 1 & 0 & -36 & -1 & 0 & -36 & 1 \\ 0 & 1 & 36 & 0 & 1 & 36 & 2 \\ -36 & 36 & 3457 & 36 & -36 & 1728 & 3 \\ -1 & 0 & 36 & 1 & 0 & 36 & 7 \\ 0 & 1 & -36 & 0 & 1 & -36 & 8 \\ -36 & 36 & 1728 & 36 & -36 & 3457 & 9 \end{bmatrix}$$

Combining the element stiffness matrices $[K]_1$ and $[K]_2$, one obtains the structure stiffness matrix as follows:

$$\begin{bmatrix} K \end{bmatrix} = 10^{3} \times \begin{bmatrix} 2 & 0 & -36 & -1 & 0 & 0 & -1 & 0 & -36 \\ 0 & 2 & 72 & 0 & -1 & 36 & 0 & 1 & 36 \\ -36 & 72 & 5185 & 0 & -36 & 864 & 36 & -36 & 1728 \\ -1 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 4 \\ 0 & -1 & -36 & 0 & 1 & -36 & 0 & 0 & 0 \\ 0 & 36 & 864 & 0 & -36 & 1728 & 0 & 0 & 0 \\ 0 & 36 & 864 & 0 & 0 & 0 & 1000 & 0 & 36 \\ -1 & 0 & 36 & 0 & 0 & 0 & 1 & -36 \\ 0 & 1 & -36 & 0 & 0 & 0 & 0 & 1 & -36 \\ -36 & 36 & 1728 & 0 & 0 & 0 & 36 & 36 & 3457 \end{bmatrix} 9$$

The deformations at joints 2 and 3 corresponding to Δ_5 to Δ_9 are zero, since joints 2 and 4 are restrained in all directions. Canceling the rows and columns corresponding to zero deformations in the structure stiffness matrix, one obtains the force–deformation relation for the structure:

$$\begin{bmatrix} F_1 \\ F_2 \\ F_3 \end{bmatrix} = \begin{bmatrix} 2 & 0 & -36 \\ 0 & 2 & 72 \\ -36 & 72 & 5185 \end{bmatrix} \times 10^3 \begin{bmatrix} \Delta_1 \\ \Delta_2 \\ \Delta_3 \end{bmatrix}$$

Substituting for the applied load F $_2 = -20$, the deformations are given as

$$\begin{bmatrix} \Delta_1 \\ \Delta_2 \\ \Delta_3 \end{bmatrix} = \begin{bmatrix} 2 & 0 & -36 \\ 0 & 2 & 72 \\ -36 & 72 & 5185 \end{bmatrix}^{-1} \times 10^3 \begin{bmatrix} 0 \\ -20 \\ 0 \end{bmatrix}$$

or

$$\begin{bmatrix} \Delta_1 \\ \Delta_2 \\ \Delta_3 \end{bmatrix} = \begin{bmatrix} 6.66 \\ -23.334 \\ 0.370 \end{bmatrix} \times 10^3$$

Loading between Nodes

The problems discussed so far have involved concentrated forces and moments applied only to nodes. But real structures are subjected to distributed or concentrated loading between nodes, as shown in Fig. 47.80. Loading may range from a few concentrated loads to an infinite variety of uniform or nonuniform distributed loads. The solution method of matrix analysis must be modified to account for such load cases.

One way to treat such loads in the matrix analysis is to insert artificial nodes, such as p and q, as shown in Fig. 47.80. The degrees of freedom corresponding to the additional nodes are added to the total structure, and the necessary additional equations are written by considering the requirements of equilibrium at these nodes. The internal member forces on each side of nodes p and q must equilibrate the external loads applied at these points. In the case of distributed loads, suitable nodes such as l, m, and n, shown in Fig. 47.80, are selected arbitrarily and the distributed loads are lumped as concentrated loads at these nodes. The degrees of freedom corresponding to the arbitrary and real nodes are treated as unknowns of the problem. There are different ways of obtaining equivalence between the lumped and the distributed loads must be statically equivalent to the distributed loads they replace.

The method of introducing arbitrary nodes is not a very elegant procedure because the number of unknown degrees of freedom make the solution procedure laborious. The approach that is of the most general use with the displacement method is the one employing the related concepts of artificial joint restraint, fixed-end forces, and equivalent nodal loads.





Semirigid End Connection

A rigid connection holds unchanged the original angles between interesting members; a simple connection allows the member end to rotate freely; a semirigid connection possesses a moment resistance intermediate between those of the simple and rigid connections.

A simplified linear relationship between the moment M acting on the connection and the resulting connection rotation ψ in the direction of M is assumed, giving

$$M = R \frac{EI}{L} \psi$$
 (47.144)

where EI and L are the flexural rigidity and length of the member, respectively. The nondimension quantity R, which is a measure of the degree of rigidity of the connection, is called the rigidity index. For a simple connection, R is zero; for a rigid connection, R is infinity. Considering the semirigidity of joints, the member flexibility matrix for flexure is derived as

$$\begin{bmatrix} \phi_1 \\ \phi_2 \end{bmatrix} = \frac{L}{EI} \begin{bmatrix} \frac{1}{3} + \frac{1}{R_1} & -\frac{1}{6} \\ -\frac{1}{6} & \frac{1}{3} + \frac{1}{R_2} \end{bmatrix} \begin{bmatrix} M_1 \\ M_2 \end{bmatrix}$$
(47.145)

or

$$[\phi] = [F][M] \tag{47.146}$$

where ϕ_1 and ϕ_2 are as shown in Fig. 47.81.

For convenience, two parameters are introduced as follows

$$p_1 = \frac{1}{1 + \frac{3}{R_1}}$$

$$p_2 = \frac{1}{1 + \frac{3}{R_2}}$$

where p_1 and p_2 are called the fixity factors. For hinged connections, both the fixity factors, p, and the rigidity index, R, are zero, but for rigid connections, the fixity factor is 1 and the rigidity index is infinity. Since the fixity factor can vary only from 0 to 1.0, it is more convenient to use in the analyses of structures with semirigid connections.



FIGURE 47.81 Flexural member with semirigid end connections.

and

Equation (47.145) can be rewritten to give

$$[F] = \frac{L}{EI} \begin{bmatrix} \frac{1}{3p_1} & -\frac{1}{6} \\ \frac{1}{6} & \frac{1}{3p_2} \end{bmatrix}$$
(47.147)

From Eq. (47.147), the modified member stiffness matrix [K] for a member with semirigid connections expresses the member end moments M_1 and M_2 in terms of the member end rotations ϕ_1 and ϕ_2 as

$$[K] = EI\begin{bmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{bmatrix}$$
(47.148a)

Expressions for K_{11} , $K_{12} = K_{21}$, and K_{22} may be obtained by inverting matrix [F], thus

$$K_{11} = \frac{12/p_1}{4/(p_1 p_2) - 1}$$
(47.148b)

$$K_{12} = K_{21} \frac{6}{4(p_1 p_2) - 1}$$
(47.148c)

$$K_{22} = \frac{12/p_1}{4/(p_1p_2) - 1}$$
(47.148d)

The modified member stiffness matrix [K], as expressed by Eq. (47.148a to d), will be needed in the stiffness method of analysis of frames in which there are semirigid member end connections.

47.10 Finite Element Method

For problems involving complex material properties and boundary conditions, numerical methods are employed to provide approximate but acceptable solutions. Of the many numerical methods developed before and after the advent of computers, the finite element method has proven to be a powerful tool. This method can be regarded as a natural extension of the matrix methods of structural analysis. It can accommodate complex and difficult problems such as nonhomogenity, nonlinear stress–strain behavior, and complicated boundary conditions. The finite element method is applicable to a wide range of boundary value problems in engineering, and it dates back to the mid-1950s with the pioneering work by Argyris (1960), Clough and Penzien (1993), and others. The method was applied first to the solution of plane stress problems and extended subsequently to the solution of plates, shells, and axisymmetric solids.

Basic Principle

The finite element method is based on the representation of a body or a structure by an assemblage of subdivisions called finite elements, as shown in Fig. 47.82. These elements are considered to be connected at nodes. Displacement functions are chosen to approximate the variation of displacements over each finite element. Polynomial functions are commonly employed to approximate these displacements. Equilibrium equations for each element are obtained by means of the principle of minimum potential energy. These equations are formulated for the entire body by combining the equations for the individual elements so that the continuity of displacements is preserved at the nodes. The resulting equations are solved satisfying the boundary conditions in order to obtain the unknown displacements.



FIGURE 47.82 Assemblage of subdivisions.

The entire procedure of the finite element method involves the following steps:

- 1. The given body is subdivided into an equivalent system of finite elements.
- 2. A suitable displacement function is chosen.
- 3. The element stiffness matrix is derived using a variational principle of mechanics, such as the principle of minimum potential energy.
- 4. The global stiffness matrix for the entire body is formulated.
- 5. The algebraic equations thus obtained are solved to determine unknown displacements.
- 6. The element strains and stresses are computed from the nodal displacements.

Elastic Formulation

Figure 47.83 shows the state of stress in an elemental volume of a body under load. It is defined in terms of three normal stress components σ_x , σ_y , and σ_z and three shear stress components τ_{xy} , τ_{yz} , and τ_{zx} . The corresponding strain components are three normal strains ε_x , ε_y , and ε_z and three shear strains γ_{xy} , γ_{yz} , and γ_{zx} . These strain components are related to the displacement components u, v, and w at a point as follows:



FIGURE 47.83 State of stress in an elemental volume.



The relations given in Eq. (47.149) are valid in the case of the body experiencing small deformations. If the body undergoes large or finite deformations, higher order terms must be retained.

The stress-strain equations for isotropic materials may be written in terms of Young's modulus and Poisson's ratio as

$$\sigma_{x} = \frac{E}{1 - v^{2}} [\varepsilon_{x} + v(\varepsilon_{y} + \varepsilon_{z})]$$

$$\sigma_{y} = \frac{E}{1 - v^{2}} [\varepsilon_{y} + v(\varepsilon_{z} + \varepsilon_{x})]$$

$$\sigma_{z} = \frac{E}{1 - v^{2}} [\varepsilon_{z} + v(\varepsilon_{x} + \varepsilon_{y})]$$

$$\tau_{yy} = G\gamma_{yy}, \quad \tau_{yz} = G\gamma_{yy}, \quad \tau_{zy} = G\gamma_{yy},$$
(47.150)



FIGURE 47.84 Plane stress problem.



FIGURE 47.85 Practical examples of plane strain problems.

Plane Stress

When the elastic body is very thin and there are no loads applied in the direction parallel to the thickness, the state of stress in the body is said to be plane stress. A thin plate subjected to in-plane loading, as shown in Fig. 47.84, is an example of a plane stress problem. In this case, $\sigma_z = \tau_{yz} = \tau_{zx} = 0$ and the constitutive relation for an isotropic continuum is expressed as

$$\begin{bmatrix} \sigma_{x} \\ \sigma_{y} \\ \sigma_{xy} \end{bmatrix} = \frac{E}{1 - \nu^{2}} \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \frac{1 - \nu}{2} \end{bmatrix} \begin{bmatrix} \varepsilon_{x} \\ \varepsilon_{y} \\ \gamma_{xy} \end{bmatrix}$$
(47.151)

Plane Strain

The state of plane strain occurs in members that are not free to expand in the direction perpendicular to the plane of the applied loads. Examples of some plane strain problems are retaining walls, dams, long cylinders, tunnels, etc., as shown in Fig. 47.85. In these problems ε_z , γ_{yz} , and γ_{zx} will vanish and hence

$$\sigma_z = v(\sigma_x + \sigma_y)$$

The constitutive relations for an isotropic material is written as

$$\begin{bmatrix} \sigma_{x} \\ \sigma_{y} \\ \tau_{xy} \end{bmatrix} = \frac{E}{(1+\nu)(1-2\nu)} \begin{bmatrix} (1-\nu) & \nu & 0 \\ \nu & (1-\nu) & 0 \\ 0 & 0 & \frac{1-2\nu}{2} \end{bmatrix} \begin{bmatrix} \varepsilon_{x} \\ \varepsilon_{y} \\ \gamma_{xy} \end{bmatrix}$$
(47.152)



FIGURE 47.86 (a) One-dimensional element. (b) Two-dimensional element. (c) Three-dimensional element.

Choice of Element Shapes and Sizes

A finite element generally has a simple one-, two-, or three-dimensional configuration. The boundaries of elements are often straight lines, and the elements can be one-, two-, or three-dimensional, as shown in Fig. 47.86. While subdividing the continuum, one has to decide the number, shape, size, and configuration of the elements in such a way that the original body is simulated as closely as possible. Nodes must be located in locations where abrupt changes in geometry, loading, and material properties occur. A node must be placed at the point of application of a concentrated load because all loads are converted into equivalent nodal-point loads.

It is easy to subdivide a continuum into a completely regular one having the same shape and size. But problems encountered in practice do not involve regular shape; they may have regions of steep gradients of stresses. A finer subdivision may be necessary in regions where stress concentrations are expected in order to obtain a useful approximate solution. Typical examples of mesh selection are shown in Fig. 47.87.

Choice of Displacement Function

Selection of displacement function is the important step in the finite element analysis, since it determines the performance of the element in the analysis. Attention must be paid to select a displacement function that:



FIGURE 47.87 Typical examples of finite element mesh.

- 1. has the number of unknown constants as the total number of degrees of freedom of the element.
- 2. does not have any preferred directions.
- 3. allows the element to undergo rigid-body movement without any internal strain.
- 4. is able to represent states of constant stress or strain.
- 5. satisfies the compatibility of displacements along the boundaries with adjacent elements.

Elements that meet both 3 and 4 are known as complete elements.

A polynomial is the most common form of displacement function. Mathematics of polynomials are easy to handle in formulating the desired equations for various elements and convenient in digital computation. The degree of approximation is governed by the stage at which the function is truncated. Solutions closer to exact solutions can be obtained by including a greater number of terms. The polynomials are of the general form

$$w(x) = a_1 + a_2 x + a_3 x^2 + \dots + a_{n+1} x^n$$
(47.153)

The coefficient a's are known as *generalized displacement amplitudes*. The general polynomial form for a two-dimensional problem can be given as

$$u(x, y) = a_1 + a_2 x + a_3 y + a_4 x^2 + a_5 xy + a_6 y^2 + \dots + a_m y^n$$

$$v(x, y) = a_{m+1} + a_{m+2} + a_{m+3} y + a_{m+4} x^2 + a_{m+5} xy + a_{m+6} y^2 + \dots + a_{2m} y^n$$

in which

$$m = \sum_{i=1}^{n+1} i$$
 (47.154)

These polynomials can be truncated at any desired degree to give constant, linear, quadratic, or higher order functions. For example, a linear model in the case of a two-dimensional problem can be given as

$$u = a_1 + a_2 x + a_3 y$$

$$v = a_4 + a_5 x + a_6 y$$
(47.155)

A quadratic function is given by

$$u = a_1 + a_2 x + a_3 y + a_4 x^2 + a_5 xy + a_6 y^2$$

$$v = a_7 + a_8 x + a_9 y + a_{10} x^2 + a_{11} xy + a_{12} y^2$$
(47.156)

Pascal's triangle, shown below, can be used for the purpose of achieving isotropy, i.e., to avoid displacement shapes that change with a change in the local coordinate system.

Nodal Degrees of Freedom

The deformation of the finite element is specified completely by the nodal displacement, rotations, and/or strains, which are referred to as *degrees of freedom*. Convergence, geometric isotropy, and potential energy function are the factors that determine the minimum number of degrees of freedom necessary for a given element. Additional degrees of freedom beyond the minimum number may be included for any element by adding secondary external nodes, and such elements with additional degrees of freedom are called higher order elements. The elements with more additional degrees of freedom become more flexible.

Isoparametric Elements

The scope of finite element analysis is also measured by the variety of element geometries that can be constructed. Formulation of element stiffness equations requires the selection of displacement expressions with as many parameters as there are node point displacements. In practice, for planar conditions, only the four-sided (quadrilateral) element finds as wide an application as the triangular element. The simplest form of quadrilateral, the rectangle, has four node points and involves two displacement components at each point, for a total of eight degrees of freedom. In this case one would choose four-term expressions for both u and v displacement fields. If the description of the element is expanded to include nodes at the midpoints of the sides, an eight-term expression would be chosen for each displacement component.

The triangle and rectangle can approximate the curved boundaries only as a series of straight line segments. A closer approximation can be achieved by means of *isoparametric* coordinates. These are nondimensionalized curvilinear coordinates whose description is given by the same coefficients as are



FIGURE 47.88 (a) Isoparametric element. (b) Subparametric element. (c) Superparametric element.

employed in the displacement expressions. The displacement expressions are chosen to ensure continuity across element interfaces and along supported boundaries, so that geometric continuity is ensured when the same forms of expressions are used as the basis of description of the element boundaries. The elements in which the geometry and displacements are described in terms of the same parameters and are of the same order are called *isoparametric elements*. The isoparametric concept enables one to formulate elements of any order that satisfy the completeness and compatibility requirements and that have isotropic displacement functions.

Isoparametric Families of Elements

Definitions and Justifications

For example, let u_i represent nodal displacements and x_i represent nodal x coordinates. The interpolation formulas are

$$u = \sum_{i=l}^m N_i u_i \qquad x = \sum_{i=l}^n N_i' x_i$$

where N_i and N are shape functions written in terms of the intrinsic coordinates. The value of u and the value of x at a point within the element are obtained in terms of nodal values of u_i and x_i from the above equations when the (intrinsic) coordinates of the internal point are given. Displacement components v and w in the y and z directions are treated in a similar manner.

The element is *isoparametric* if m = n, $N_i = N$, and the same nodal points are used to define both element geometry and element displacement (Fig. 47.88a); the element is *subparametric* if m > n, the order of N_i is larger than N'_i (Fig. 47.88b); the element is *superparametric* if m < n, the order of N_i is smaller than N'_i (Fig. 47.88c). The isoparametric elements can correctly display rigid-body and constant-strain modes.

Element Shape Functions

The finite element method is not restricted to the use of linear elements. Most finite element codes commercially available allow the user to select between elements with linear or quadratic interpolation functions. In the case of quadratic elements, fewer elements are needed to obtain the same degree of accuracy in the nodal values. Also, the two-dimensional quadratic elements can be shaped to model a curved boundary. Shape functions can be developed based on the following properties:

- 1. Each shape function has a value of 1 at its own node and is zero at each of the other nodes.
- 2. The shape functions for two-dimensional elements are zero along each side that the node does not touch.
- 3. Each shape function is a polynomial of the same degree as the interpolation equation. Shape functions for typical elements are given in Fig. 47.89a and b.

Formulation of Stiffness Matrix

It is possible to obtain all the strains and stresses within the element and to formulate the stiffness matrix and a consistent load matrix once the displacement function has been determined. This consistent load matrix represents the equivalent nodal forces that replace the action of external distributed loads.

Element name	Configuration	DOF	Shape functions
Two-node linear element	η 1 2 (-1,0) (1,0)	+	$N_i = \frac{1}{2}(1 + \xi_0);$ i = 1, 2
Three- node parabolic element	η 1 2 3 (-1,0) (1,0)	+	$\begin{split} N_i &= \frac{1}{2} \xi_0 \; (1 + \xi_0); i = 1, \; 3 \\ N_i &= (1 - \xi^2); i = 2 \end{split}$
Four- node cubic element	η $1 2 3 4$ (-1,0) (1,0)	+	$N_{i} = \frac{1}{16}(1 + \xi_{0})(9\xi^{2} - 1)$ i = 1, 4 $N_{i} = \frac{9}{16}(1 + 9\xi_{0})(1 - \xi^{2})$ i = 2, 3
Five-node quartic element	$ \begin{array}{c} \eta \\ 1 \\ 2 \\ 3 \\ 4 \\ 5 \\ (-1,0) \end{array} \xi $	+	$N_{i} = \frac{1}{6}(1 + \xi_{0}) \{4\xi_{0}(1 - \xi^{2}) + 3\xi_{0}\}$ $i = 1, 5$ $N_{i} = 4\xi_{0}(1 - \xi^{2})(1 + 4\xi_{0})$ $i = 2, 4$ $N_{3} = (1 - 4\xi^{2})(1 - \xi^{2})$

FIGURE 47.89 Shape functions for typical elements.

As an example, let us consider a linearly elastic element of any of the types shown in Fig. 47.90. The displacement function may be written in the form

$${f} = [P]{A}$$
 (47.157)

in which $\{f\}$ may have two components $\{u, v\}$ or simply be equal to w, [P] is a function of x and y only, and $\{A\}$ is the vector of undetermined constants. If Eq. (47.157) is applied repeatedly to the nodes of the element one after the other, we obtain a set of equations of the form

$${D^*} = [C] {A}$$
 (47.158)

in which $\{D^*\}$ is the nodal parameters and [C] is the relevant nodal coordinates. The undetermined constants $\{A\}$ can be expressed in terms of the nodal parameters $\{D^*\}$ as

$$\{A\} = [C]^{-1} \{D^*\}$$
 (47.159)

Serial no.	Element name	Configuration	DOF	Shape functions
1	Four-node plane quadrilateral	(-1,1) $(-1,-1)$ $(-1,-1)$ $(-1,-1)$ $(-1,-1)$ $(-1,-1)$ $(-1,-1)$ $(-1,-1)$ $(-1,-1)$	u, v	$N_{i} = \frac{1}{4}(1 + \xi_{0})(1 + \eta_{0});$ i = 1, 2, 3, 4
2	Eight-node plane quadrilateral	$(-1,1) \underbrace{7}_{(-1,0)} \underbrace{6}_{(-1,-1)} \underbrace{6}_{(-1,-1)$	u, v	$\begin{split} N_i &= \frac{1}{4} (1 + \xi_0) (1 + \eta_0) \\ & (\xi_0 + \eta_0 - 1); \\ i &= 1, 3, 5, 7 \\ N_i &= \frac{1}{2} (1 - \xi^2) (1 + \eta_0) \\ i &= 2, 6 \\ N_i &= \frac{1}{2} (1 - \eta^2) (1 + \xi_0) \\ i &= 4, 8 \end{split}$
3	Twelve-node plane quadrilateral	$(-1,1) \underbrace{(-1,1)}_{(-1,\frac{1}{3})} \underbrace{(-1,1)}_{(1,1)} \underbrace{(-1,\frac{1}{3})}_{(1,1)} \underbrace{(-1,\frac{1}{3})}_{(1,1)} \underbrace{(-1,-\frac{1}{3})}_{(1,1)} \underbrace{(-1,-\frac{1}{3})}_{(1,1)} \underbrace{(-1,-\frac{1}{3})}_{(1,1)} \underbrace{(-1,-\frac{1}{3})}_{(1,1)} \underbrace{(-1,-1)}_{(1,1)} \underbrace{(-\frac{1}{3},-1)}_{(\frac{1}{3},-1)} \underbrace{(-1,-1)}_{(\frac{1}{3},-1)} \underbrace$	u, v	$\begin{split} N_i &= \frac{1}{32}(1+\xi_0)(1+\eta_0) \\ & (-10+9(\xi^2+\eta^2)) \\ i &= 1,4,7,10 \\ \\ N_i &= \frac{9}{32}(1+\xi_0)(1+\eta^2) \\ & (1+9\eta_0) \\ i &= 5,6,11,12 \\ \\ N_i &= \frac{9}{32}(1+\eta_0)(1-\xi^2) \\ & (1+9\xi_0) \\ i &= 2,3,8,9 \end{split}$



Substituting Eq. (47.159) into (47.157)

$${f} = [P][C]^{-1}{D^*}$$
 (47.160)

Constructing the displacement function directly in terms of the nodal parameters, one obtains

$${f} = [L] {D^*}$$
 (47.161)

where $\left[L\right]$ is a function of both (x, y) and $(x, y)_{i,j,m}$ given by

$$[L] = [P][C]^{-1}$$
(47.162)

Serial no.	Element name	Configuration	DOF	Shape functions		
4	Six-node linear quadrilateral	$(-1,1) \xrightarrow{\begin{pmatrix} 0,1 \\ 0,1 \\ 0 \\ -1,-1 \end{pmatrix}} \xrightarrow{\begin{pmatrix} 0,1 \\ 5 \\ -1 \\ -1 \\ 0 \\ -1 \end{pmatrix}} (1,1) \xrightarrow{\xi} (1,-1)$	u, v	$\begin{split} N_i &= \frac{\xi_0}{4} (1 + \xi_0) (1 + \eta_0) \\ &i = 1, 3, 4, 6 \\ N_i &= \frac{1}{2} (1 - \xi^2) (1 + \eta_0) \\ &i = 2, 5 \end{split}$		
5	Eight-node plane quadrilateral	$(-1,-1) \overset{0}{\overset{0}{\underset{(-1,-1)}{3}}} \overset{0}{\underset{(-1,-1)}{3}} \overset{0}{(-1$	u, v	$\begin{split} N_i &= \frac{1}{32} (1 + \xi_0) (-1 + 9\xi^2) \\ & (1 + \eta_0) \\ i &= 1, 4, 5, 8 \\ N_i &= \frac{9}{32} (1 - \xi^2) (1 + 9\xi_0) \\ & (1 + \eta_0) \\ i &= 2, 3, 6, 7 \end{split}$		
6	Seven-node plane quadrilateral	$(-1,1) \circ \underbrace{\begin{array}{c} & & & & \\ & & & \\ (-1,0) & & & \\ & & & \\ (-1,-1) \circ \underbrace{\begin{array}{c} & & & \\ & & & \\ 1 & & & \\ $	u, v	$\begin{split} N_1 &= \frac{1}{4} (1-\xi)(1-\eta) \\ & (1+\xi+\eta) \\ N_2 &= \frac{1}{2} (1-\eta)(1-\xi^2) \\ N_3 &= \frac{\xi}{4} (1+\xi)(1-\eta) \\ N_4 &= \frac{\xi}{4} (1+\xi)(1+\eta) \\ N_5 &= \frac{1}{2} (1+\eta)(1-\xi^2) \\ N_6 &= -\frac{1}{4} (1-\xi)(1+\eta) \\ & (1+\xi-\eta) \\ N_7 &= \frac{1}{2} (1-\xi)(1-\eta^2) \end{split}$		



The various components of strain can be obtained by appropriate differentiation of the displacement function. Thus,

$${\boldsymbol{\varepsilon}} = [B] \{D^*\} \tag{47.163}$$



FIGURE 47.90 Degrees of freedom: (a) triangular plane stress element, (b) triangular bending element.

[B] is derived by differentiating appropriately the elements of [L] with respect to x and y. The stresses { σ } in a linearly elastic element are given by the product of the strain and a symmetrical elasticity matrix [E]. Thus,

 $\{\sigma\} = [E] \{\varepsilon\}$

or

$$[\sigma] = [E][B][D^*]$$
(47.164)

The stiffness and the consistent load matrices of an element can be obtained using the principle of minimum total potential energy. The potential energy of the external load in the deformed configuration of the element is written as

$$W = -\{D^*\}^T \{Q^*\} - \int_a \{f\}^T \{q\} da$$
 (47.165)

In Eq. (47.165) $\{Q^*\}$ represents concentrated loads at nodes, and $\{q\}$ the distributed loads per unit area. Substituting for $\{f\}^T$ from Eq. (47.161), one obtains

$$W = -\{D^*\}^T \{Q^*\} - \{D^*\}^T \int_{a} [L]^T \{q\} da$$
(47.166)

Note that the integral is taken over the area a of the element. The strain energy of the element integrated over the entire volume v, is given as

$$U = \frac{1}{2} \int_{v} \left\{ \varepsilon \right\}^{T} \left\{ \sigma \right\} dv$$

Substituting for $\{\epsilon\}$ and $\{\sigma\}$ from Eqs. (47.163) and (47.164), respectively,

$$\mathbf{U} = \frac{1}{2} \left\{ \mathbf{D}^{\star} \right\}^{\mathrm{T}} \left(\int_{\mathbf{v}} \left[\mathbf{B} \right]^{\mathrm{T}} \left[\mathbf{E} \right] \left[\mathbf{B} \right] \mathrm{d} \mathbf{v} \right) \left\{ \mathbf{D}^{\star} \right\}$$
(47.167)

The total potential energy of the element is

$$V = U + W$$

or

$$V = \frac{1}{2} \{D^*\}^T \left(\int_{v} [B]^T [E] [B] dv \right) \{D^*\} - \{D^*\}^T \{Q^*\} - \{D^*\}^T \int_{a} [L]^T \{q\} da$$
(47.168)

Using the principle of minimum total potential energy, we obtain

$$\left(\int_{v} [B]^{T} [E] [B] dv\right) \left\{ D^{*} \right\} = \left\{ Q^{*} \right\} + \int_{a} [L]^{T} \left\{ q \right\} da$$

or

$$[K]{D^*} = {F^*}$$
(47.169)

where

$$\begin{bmatrix} \mathbf{K} \end{bmatrix} = \int_{\mathbf{v}} \begin{bmatrix} \mathbf{B} \end{bmatrix}^{\mathrm{T}} \begin{bmatrix} \mathbf{E} \end{bmatrix} \begin{bmatrix} \mathbf{B} \end{bmatrix} \mathrm{d}\mathbf{v} \tag{47.170a}$$

and

$$\{F^*\} = \{Q^*\} + \int_a [L]^T \{q\} da \qquad (47.170b)$$

Plates Subjected to In-Plane Forces

The simplest element available in two-dimensional stress analysis is the triangular element. The stiffness and consistent load matrices of such an element will now be obtained by applying the equation derived in the previous section.

Consider the triangular element shown in Fig. 47.90a. There are two degrees of freedom per node and a total of six degrees of freedom for the entire element. We can write

$$\mathbf{u} = \mathbf{A}_1 + \mathbf{A}_2 \mathbf{x} + \mathbf{A}_3 \mathbf{y}$$

and

$$\mathbf{v} = \mathbf{A}_4 + \mathbf{A}_5 \mathbf{x} + \mathbf{A}_6 \mathbf{y}$$

expressed as

$$\left\{ f \right\} = \begin{cases} u \\ v \\ \end{cases} = \begin{bmatrix} 1 & x & y & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & x & y \end{bmatrix} \begin{cases} A_1 \\ A_2 \\ A_3 \\ A_4 \\ A_5 \\ A_6 \\ \end{cases}$$
(47.171)

~

or

$${f} = [P] {A}$$
 (47.172)

Once the displacement function is available, the strains for a plane problem are obtained from

$$\varepsilon_{x} = \frac{\partial u}{\partial x}$$
 $\varepsilon_{y} = \frac{\partial v}{\partial y}$

and

$$\gamma_{xy} = \frac{\partial u}{\partial y} \frac{\partial v}{\partial x}$$

Matrix [B], relating the strains to the nodal displacement $\{D^*\}$, is thus given as

$$\begin{bmatrix} B \end{bmatrix} = \frac{1}{2\Delta} \begin{bmatrix} b_{i} & 0 & b_{j} & 0 & b_{m} & 0\\ 0 & c_{i} & 0 & c_{j} & 0 & c_{m}\\ c_{i} & b_{j} & c_{j} & b_{j} & c_{m} & b_{m} \end{bmatrix}$$
(47.173)

b_i, c_i, etc. are constants related to the nodal coordinates only. The strains inside the element must all be constant and hence the name of the element.

For derivation of strain matrix, only isotropic material is considered. The plane stress and plane strain cases can be combined to give the following elasticity matrix, which relates the stresses to the strains:

$$\begin{bmatrix} \mathbf{E} \end{bmatrix} = \begin{bmatrix} \mathbf{C}_{1} & \mathbf{C}_{1}\mathbf{C}_{2} & \mathbf{0} \\ \mathbf{C}_{1}\mathbf{C}_{2} & \mathbf{C}_{1} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & \mathbf{C}_{12} \end{bmatrix}$$
(47.174)

where

 $C_1 = \overline{E} / (1 - v^2)$ and $C_2 = v$ for plane stress

and

$$C_1 = \frac{\overline{E}(1-\nu)}{(1+\nu)(1-2\nu)}$$
 and $C_2 = \frac{\nu}{(1-\nu)}$ for plane stress

and for both cases,

$$C_{12} = C_1 (1 - C_2)/2$$

and \overline{E} is the modulus of elasticity.

The stiffness matrix can now be formulated according to Eq. (47.170a)

$$[E][B] = \frac{1}{2\Delta} \begin{bmatrix} C_1 & C_1C_2 & 0\\ C_1C_2 & C_1 & 0\\ 0 & 0 & C_{12} \end{bmatrix} \begin{bmatrix} b_i & 0 & b_j & 0 & b_m & 0\\ 0 & c_i & 0 & c_j & 0 & c_m\\ c_i & b_i & c_j & b_j & c_m & b_m \end{bmatrix}$$

where Δ is the area of the element.

The stiffness matrix is given by Eq. (47.10.37a) as

$$[K] = \int_{v} [B]^{T} [E] [B] dv$$

The stiffness matrix has been worked out algebraically to be

$$\begin{split} \left[K \right] = \frac{h}{4\Delta} \begin{bmatrix} C_1 b_i^2 & & \\ + C_{12} c_i^2 & \\ C_1 C_2 b_i c_i & C_1 c_i^2 & \\ + C_{12} b_i c_i & + C_{12} b_i^2 & Symmetrical \\ C_1 b_i b_j & C_1 C_2 b_j c_i & C_1 b_j^2 & \\ + C_{12} c_i c_j & + C_{12} b_i c_j & + C_{12} c_j^2 & \\ C_1 C_2 b_i c_j & C_1 c_i c_j & C_1 C_2 b_j c_j & C_1 c_j^2 & \\ + C_{12} b_j c_i & + C_1 b_i b_j & + C_{12} b_j c_j & + C_{12} b_j^2 & \\ C_1 b_i b_m & C_1 C_2 b_m c_i & C_1 b_j b_m & C_1 C_2 b_m c_j & C_1 b_m^2 & \\ + C_{12} c_i c_m & + C_{12} b_i c_m & + C_{12} c_j c_m & + C_{12} b_j c_m & + C_{12} c_m^2 & \\ C_1 C_2 b_i c_m & C_1 c_i c_m & C_1 C_2 b_j c_m & C_1 c_j c_m & C_1 C_2 b_m c_m & C_1 c_m^2 \\ + C_{12} b_m c_i & + C_{12} b_i b_m & + C_{12} b_m c_j & + C_{12} b_j b_m & + C_{12} b_m c_m & + C_{12} b_m^2 \end{bmatrix} \end{split}$$

Beam Element

The stiffness matrix for a beam element with two degrees of freedom (one deflection and one rotation) can be derived in the same manner as for other finite elements using Eq. (47.170a).

The beam element has two nodes, one at each end, and two degrees of freedom at each node, giving it a total of four degrees of freedom. The displacement function can be assumed as

$$f = w = A_1 + A_2 x + A_3 x^2 + A_4 x^3$$

i.e.,

$$\mathbf{f} = \begin{bmatrix} 1 \mathbf{x} \mathbf{x}^2 \mathbf{x}^3 \end{bmatrix} \begin{cases} \mathbf{A}_1 \\ \mathbf{A}_2 \\ \mathbf{A}_3 \\ \mathbf{A}_4 \end{cases}$$

or

 $f = [P]{A}$

With the origin of the x and y axis at the left-hand end of the beam, we can express the nodal displacement parameters as

$$D_{1}^{*} = (w)_{x=0} = A_{1} + A_{2}(0) + A_{3}(0)^{2} + A_{4}(0)^{3}$$
$$D_{2}^{*} = \left(\frac{dw}{dx}\right)_{x=0} = A_{2} + 2A_{3}(0) + 3A_{4}(0)^{2}$$
$$D_{3}^{*} = (w)_{x=1} = A_{1} + A_{2}(1) + A_{3}(1)^{2} + A_{4}(1)^{3}$$
$$D_{4}^{*} = \left(\frac{dw}{dx}\right)_{x=1} = A_{2} + 2A_{3}(1) + 3A_{4}(1)^{2}$$

 $\left\{ D^{\star}\right\} = \left[C\right] \left\{ A\right\}$

where

 $\{A\} = [C]^{-1} \{D^*\}$

and

$$\begin{bmatrix} C \end{bmatrix}^{-1} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ \frac{-3}{I^2} & \frac{-2}{I} & \frac{3}{I^2} & \frac{-1}{I} \\ \frac{2}{I^3} & \frac{1}{I^2} & \frac{-2}{I^3} & \frac{1}{I^2} \end{bmatrix}$$

Using Eq. (47.177), we obtain

$$[L] = [P][C]^{-1}$$
(47.175)

or

$$\left[C\right]^{-1} = \left[\left(1 - \frac{3x^2}{I^2} + \frac{2x^3}{I^3}\right) \left(x - \frac{2x^2}{I} + \frac{x^3}{I^2}\right) \left|\left(\frac{3x^2}{I^2} - \frac{2x^3}{I^3}\right) \left|\left(-\frac{x^2}{I} + \frac{x^3}{I^2}\right)\right|\right]$$
(47.176)

Neglecting shear deformation

 $\left\{ \epsilon \right\} = -\frac{d^2y}{dx^2}$

Substituting Eq. (47.191) into Eq. (47.176) and the result into Eq. (47.192)

$$\left\{\epsilon\right\} = \left[\left|\frac{6}{I^2} - \frac{12x}{I^3}\right| \frac{4}{I} - \frac{6x}{I^2}\right| - \frac{6}{I^2} + \frac{12x}{I^3}\left|\frac{2}{I} - \frac{6x}{I^2}\right] \left\{D^*\right\}$$

or

 $\left\{ \boldsymbol{\epsilon} \right\} \!=\! \left[\boldsymbol{B} \right] \! \left\{ \boldsymbol{D}^{\star} \right\}$

The moment-curvature relationship is given by

$$M = \overline{EI}\left(-\frac{d^2y}{dx^2}\right)$$

where

 $\bar{\rm E}$ is the modulus of elasticity,

or

$${\boldsymbol{\varepsilon}} = [B] \{D^*\}$$

We know that $\{\sigma\} = [E]\{\epsilon\}$, so we have for the beam element

 $[E] = \overline{E}I$

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The stiffness matrix can now be obtained from Eq. (47.170a) written in the form

$$[K] = \int_{0}^{1} [B]^{T} [d] [B] dx$$

with the integration over the length of the beam. Substituting for [B] and [E], we obtain

$$[k] = \overline{EI} \int_{0}^{1} \left[\frac{\frac{36}{\lambda^{4}} - \frac{144x}{\lambda^{5}} + \frac{144x^{2}}{\lambda^{6}}}{\frac{24}{\lambda^{3}} - \frac{84x}{\lambda^{4}} + \frac{72x^{2}}{\lambda^{5}}}{\frac{16}{\lambda^{2}} - \frac{48x}{\lambda^{3}} + \frac{36x^{2}}{\lambda^{4}}} - \frac{16}{\lambda^{2}} + \frac{14x^{2}}{\lambda^{5}} - \frac{16}{\lambda^{2}} - \frac{48x}{\lambda^{3}} + \frac{36x^{2}}{\lambda^{4}}}{\frac{12}{\lambda^{3}} - \frac{60x}{\lambda^{4}} + \frac{72x^{2}}{\lambda^{5}}}{\frac{16}{\lambda^{2}} - \frac{24}{\lambda^{3}} + \frac{84x}{\lambda^{4}} - \frac{72x^{2}}{\lambda^{5}}}{\frac{36}{\lambda^{4}} - \frac{144x}{\lambda^{5}} + \frac{144x^{2}}{\lambda^{6}}}{\frac{12}{\lambda^{3}} - \frac{60x}{\lambda^{4}} + \frac{72x^{2}}{\lambda^{5}}}{\frac{8}{\lambda^{2}} - \frac{36x}{\lambda^{3}} + \frac{36x^{2}}{\lambda^{4}}} - \frac{-12}{\lambda^{3}} + \frac{60x}{\lambda^{4}} - \frac{72x^{2}}{\lambda^{5}} - \frac{4}{\lambda^{2}} - \frac{24x}{\lambda^{3}} + \frac{36x^{2}}{\lambda^{4}}}{\frac{16}{\lambda^{4}} - \frac{12}{\lambda^{3}} + \frac{60x}{\lambda^{4}} - \frac{72x^{2}}{\lambda^{5}}}{\frac{16}{\lambda^{2}} - \frac{24x}{\lambda^{3}} + \frac{36x^{2}}{\lambda^{4}}}{\frac{16}{\lambda^{4}} - \frac{12}{\lambda^{3}} + \frac{60x}{\lambda^{4}} - \frac{72x^{2}}{\lambda^{5}}}{\frac{16}{\lambda^{2}} - \frac{24x}{\lambda^{3}} + \frac{36x^{2}}{\lambda^{4}}}{\frac{16}{\lambda^{4}} - \frac{12}{\lambda^{3}} + \frac{60x}{\lambda^{4}} - \frac{72x^{2}}{\lambda^{5}}}{\frac{16}{\lambda^{2}} - \frac{24x}{\lambda^{3}} + \frac{36x^{2}}{\lambda^{4}}}{\frac{16}{\lambda^{4}} - \frac{12}{\lambda^{3}} + \frac{60x}{\lambda^{4}} - \frac{72x^{2}}{\lambda^{5}}}{\frac{16}{\lambda^{2}} - \frac{24x}{\lambda^{3}} + \frac{36x^{2}}{\lambda^{4}}}{\frac{16}{\lambda^{4}} - \frac{12}{\lambda^{3}} - \frac{60x}{\lambda^{4}} - \frac{12}{\lambda^{5}} - \frac{$$

or

$$[K] = \overline{EI} \begin{bmatrix} \frac{12}{\lambda^3} & \text{symmetrical} \\ \frac{6}{\lambda^2} & \frac{4}{\lambda} & \\ \frac{-12}{\lambda^3} & \frac{-6}{\lambda^2} & \frac{12}{\lambda^3} \\ \frac{6}{\lambda^2} & \frac{2}{\lambda} & \frac{-6}{\lambda^2} & \frac{4}{\lambda} \end{bmatrix}$$
(47.178)

Plate Element

For the rectangular bending element shown in Fig. 47.91 with three degrees of freedom (one deflection and two rotations) at each node, the displacement function can be chosen as a polynomial with 12 undetermined constants:

$$\{f\} = w = A_1 + A_2 x + A_3 y + A_4 x^2 + A_5 xy + A_6 y^2 + A_7 x^3 + A_8 x^2 y + A_9 xy^2 + A_{10} y^3 + A_{11} x^3 y + A_{12} xy^3$$
(47.179)



FIGURE 47.91 Rectangular bending element.

or

$${f} = {P}{A}$$

The displacement parameter vector is defined as

$$\left\{ \mathbf{D}^{\star} \right\} = \left\{ \mathbf{w}_{i}, \mathbf{\theta}_{xi}, \mathbf{\theta}_{yi} \middle| \mathbf{w}_{j}, \mathbf{\theta}_{xj}, \mathbf{\theta}_{yj} \middle| \mathbf{w}_{k}, \mathbf{\theta}_{xk}, \mathbf{\theta}_{yk} \middle| \mathbf{w}_{\ell}, \mathbf{\theta}_{x\ell}, \mathbf{\theta}_{y\ell} \right\}$$

where

$$\theta_{x} = \frac{\partial w}{\partial y}$$
 and $\theta_{y} = -\frac{\partial w}{\partial x}$

As in the case of beams, it is possible to derive from Eq. (47.179) a system of 12 equations relating $\{D^*\}$ to constants $\{A\}$. The last equation

$$\mathbf{w} = \left[\left[\mathbf{L} \right]_{i} \middle| \left[\mathbf{L} \right]_{j} \middle| \left[\mathbf{L} \right]_{k} \middle| \left[\mathbf{L} \right]_{\ell} \right] \left\{ \mathbf{D}^{*} \right\}$$
(47.180)

The curvatures of the plate element at any point (x, y) are given by

$$\left\{ \epsilon \right\} = \begin{cases} \frac{-\partial^2 w}{\partial x^2} \\ \frac{-\partial^2 w}{\partial y^2} \\ \frac{2\partial^2 w}{\partial x \partial y} \end{cases}$$

By differentiating Eq. (47.180), we obtain

$$\left\{ \boldsymbol{\varepsilon} \right\} = \left[\left[\boldsymbol{B} \right]_{i} \middle| \left[\boldsymbol{B} \right]_{j} \middle| \left[\boldsymbol{B} \right]_{k} \middle| \left[\boldsymbol{B} \right]_{l} \right] \left\{ \boldsymbol{D}^{*} \right\}$$
(47.181)

or

$$\left\{\epsilon\right\} = \sum_{r=i,j,k,l} \left[B\right]_r \left\{D^*\right\}_r \tag{47.182}$$

where

$$\begin{bmatrix} \mathbf{B} \end{bmatrix}_{\mathbf{r}} = \begin{bmatrix} -\frac{\partial^2}{\partial \mathbf{x}^2} [\mathbf{L}]_{\mathbf{r}} \\ -\frac{\partial^2}{\partial \mathbf{y}^2} [\mathbf{L}]_{\mathbf{r}} \\ -\frac{\partial^2}{\partial \mathbf{y}^2} [\mathbf{L}]_{\mathbf{r}} \\ -\frac{\partial^2}{\partial \mathbf{x} \partial \mathbf{y}} [\mathbf{L}]_{\mathbf{r}} \end{bmatrix}$$
(47.183)

and

$$\left\{ D^{*} \right\}_{r} = \left\{ w_{r}, \theta_{xr}, \theta_{yr} \right\}$$
(47.184)

For an isotropic slab, the moment-curvature relationship is given by

$$\left\{\sigma\right\} = \left\{M_{x} \quad M_{y} \quad M_{xy}\right\} \tag{47.185}$$

$$\begin{bmatrix} E \end{bmatrix} = N \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \frac{1-\nu}{2} \end{bmatrix}$$
(47.186)

and

$$N = \frac{Eh^3}{12(1-v^2)}$$
(47.187)

For orthotropic plates with the principal directions of orthotropy coinciding with the x and y axes, no additional difficulty is experienced. In this case we have

$$\begin{bmatrix} \mathbf{E} \end{bmatrix} = \begin{bmatrix} \mathbf{D}_{\mathbf{x}} & \mathbf{D}_{1} & \mathbf{0} \\ \mathbf{D}_{1} & \mathbf{D}_{\mathbf{y}} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & \mathbf{D}_{\mathbf{xy}} \end{bmatrix}$$
(47.188)

where D_x , D_1 , D_y , and D_{xy} are the orthotropic constants used by Timoshenko and Woinowsky-Krieger (1959), and

$$D_{x} = \frac{E_{x} h^{3}}{12(1-v_{x}v_{y})}$$

$$D_{y} = \frac{E_{y} h^{3}}{12(1-v_{x}v_{y})}$$

$$D_{1} = \frac{v_{x}E_{y}h^{3}}{12(1-v_{x}v_{y})} = \frac{v_{y}E_{x}h^{3}}{12(1-v_{x}v_{y})}$$

$$D_{xy} = \frac{Gh^{3}}{12}$$
(47.189)

where E_x , E_y , v_x , v_y , and G are the orthotropic material constants and h is the plate thickness.

Unlike the strain matrix for the plane stress triangle (see Eq. (47.173)), the stress and strain in the present element vary with x and y. In general we calculate the stresses (moments) at the four corners. These can be expressed in terms of the nodal displacements by Eq. (47.164), which, for an isotropic element, take the form

		6p ⁻¹ + 6vp	4vc	-4b	-6vp	2vc	0	$-6p^{-1}$	0	-2b	0	0	0	
$ \begin{cases} \left\{ \boldsymbol{\sigma} \right\}_{i} \\ \left\{ \boldsymbol{\sigma} \right\}_{j} \\ \left\{ \boldsymbol{\sigma} \right\}_{k} \\ \left\{ \boldsymbol{\sigma} \right\}_{r} \end{cases} = \frac{N}{cb} $		6p + 6vp ⁻¹	4c	$-4\nu b$	-6p	2c	0	$-6vp^{-1}$	0	-2vp	0	0	0	
		-(1-v)	$-(1-\nu)\mathbf{b}$	$(1-\nu)c$	(1 - v)	0	-(1-v)c	(1 - v)	$(1-\nu)b$	0	-(1-v)	0	0	
		-6vp	-2vc	0	$6p^{-1} + 6vp$	-4vc	-4b	0	0	0	$-6p^{-1}$	0	-2b	$\int_{D} \left\{ \right\}$
		-6p	-2c	0	$6p + 6\upsilon p^{-1}$	-4c	-4vb	0	0	0	$-6vp^{-1}$	0	-2vb	$\left[\begin{array}{c} 1^{D} \\ - 1 \end{array} \right]_{i}$
	_ N	-(1-v)	0	$(1-\nu)c$	(1 - v)	$-(1-\nu)b$	(1 - v)	0	0	$-(1-\nu)$	(1 - v)b	0	0	$\begin{bmatrix} \left\{ D \right\}_{j} \\ \left\{ D^{*} \right\}_{k} \end{bmatrix} \Sigma$
	cb	-6p ⁻¹	0	2b	0	0	0	$6p^{-1} + 6vp$	4vc	4b	-6vp	2vc	0	
		-6vp ⁻¹	0	2b	0	0	0	$6p + 6\upsilon p^{-1}$	4c	4vb	-6p	2c	0	
		-(1-v)	-(1-v)b	0	(1 - v)	0	0	(1 - v)	$(1-\nu)b$	$(1-\nu)c$	-(1-v)	0	-(1-v)c	[[¹],]
		0	0	0	$-6p^{-1}$	0	2b	-6vp	$-2\nu c$	0	$6p^{-1} + 6\upsilon p$	-4vc	4b	
		0	0	0	$-6vp^{-1}$	0	2vb	-6p	-2c	0	$6p + 6\upsilon p^{-1}$	-4c	4vb	(47, 100)
		-(1-v)	0	0	(1 - v)	-(1-v)b	0	(1 - v)	0	(1 - v)c	-(1-v)	(1 - v)b	-(1-v)c	(4/.190)

The stiffness matrix corresponding to the 12 nodal coordinates can be calculated by

$$[K] = \int_{-b/2}^{b/2} \int_{-c/2}^{c/2} [B]^{T} [E] [B] dx dy$$
(47.191)

For an isotropic element, this gives

$$[K^*] = \frac{N}{15cb} [T] [\overline{k}] [T]$$
(47.192)

where

$$[T] = \begin{bmatrix} [T_s] \text{Submatrices not} \\ [T_s] \text{shown are} \\ [T_s] \text{zero} \\ [T_s] \end{bmatrix}$$
(47.193)
$$[T_s] = \begin{bmatrix} 1 & 0 & 0 \\ 0 & b & 0 \\ 0 & 0 & c \end{bmatrix}$$
(47.194)

and $[\bar{K}]$ is given by the matrix shown in Eq. (47.195).

(47.195)

$$\left[\vec{k} \right] = \begin{bmatrix} 60p^{-2} + 60p^2 \\ -12v + 42 \\ -13v \\ -12v + 42 \\ -12v + 42$$
If the element is subjected to a uniform load in the z direction of intensity q, the consistent load vector becomes

$$\left[Q_{q}^{*}\right] = q \int_{-b/2}^{b/2} \int_{-c/2}^{c/2} \left[L\right]^{T} dx dy$$
(47.196)

where $\{Q_q^*\}$ is 12 forces corresponding to the nodal displacement parameters. Evaluating the integrals in this equation gives

$$\left\{ Q_{q}^{*} \right\} = qcb \begin{cases} 1/4 \\ b/24 \\ -c/24 \\ -b/24 \\ -c/24 \\ -c/24 \\ 1/4 \\ b/24 \\ c/24 \\ -c-- \\ 1/4 \\ -b/24 \\ c/24 \\ c/24 \\ c/24 \\ c/24 \\ c/24 \\ c/24 \end{cases}$$
(47.197)

More details on the finite element method can be found in Desai and Abel (1972) and Ghali and Neville (1978).

47.11 Inelastic Analysis

An Overall View

Inelastic analyses can be generalized into two main approaches. The first approach is known as *plastic hinge analysis*. This analysis assumes that structural elements remain elastic except at critical regions where plastic hinges are allowed to form. The second approach is known as *spread of plasticity analysis*. This analysis follows explicitly the gradual spread of yielding throughout the structure. Material yielding in the member is modeled by discretization of members into several line elements and subdivision of the cross sections into many "fibers." Although the spread of plasticity analysis can predict accurately the inelastic response of the structure, the plastic hinge analysis is considered to be computationally more efficient and less expensive to execute.

If the geometric nonlinear effect is not considered, the plastic hinge analysis predicts the maximum load of the structure corresponding to the formation of a plastic collapse mechanism (Chen and Sohal,

1995). First-order plastic analysis is finding considerable application in continuous beams and low-rise building frames where members are loaded primarily in flexure. For tall building frames and frames with slender columns subjected to side sway, the interaction between yielding and instability may lead to collapse prior to the formation of a plastic mechanism (SSRC, 1988). If an incremental analysis is carried out based on the updated deformed geometry of the structure, the analysis is termed *second order*. The need for a second-order analysis of steel frames is increasing in view of the modern codes and standards that give explicit permission for the engineer to compute load effects from a direct second-order analysis.

This section presents the virtual work principle to explain the fundamental theorems of plastic hinge analysis. Simple and approximate techniques of practical plastic analysis methods are then introduced. The concept of hinge-by-hinge analysis is presented. The more advanced topics, such as second-order elastic-plastic hinge, refined plastic hinge analysis, and spread of plasticity analysis, are covered in the Stability of Structures section.

Ductility

Plastic analysis is strictly applicable for materials that can undergo large deformation without fracture. Steel is one such material, with an idealized stress–strain curve, as shown in Fig. 47.92. When steel is subjected to tensile force, it will elongate elastically until the yield stress is reached. This is followed by an increase in strain without much increase in stress. Fracture will occur at very large deformation. This material idealization is generally known as *elastic-perfectly plastic* behavior. For a compact section, the attainment of initial yielding does not result in failure of a section. The compact section will have reserved plastic strength that depends on the shape of the cross-section. The capability of the material to deform under a constant load without decrease in strength is the *ductility* characteristic of the material.

Redistribution of Forces

The benefit of using a ductile material can be demonstrated from an example of a three-bar system, shown in Fig. 47.93. From the equilibrium condition of the system,

$$2T_1 + T_2 = P \tag{47.198}$$

Assuming the elastic stress-strain law, the displacement and force relationship of the bars may be written as:

$$\delta = \frac{T_1 L_1}{AE} = \frac{T_2 L_2}{AE}$$
(47.199)



FIGURE 47.92 Idealized stress-strain curve.



FIGURE 47.93 Force redistribution in a three-bar system: (a) elastic, (b) partially yielded, (c) fully plastic, (d) loaddeflection curve.

Since $L_2 = L_1/2 = L/2$, Eq. (47.199) can be written as

$$\Gamma_1 = \frac{\Gamma_2}{2} \tag{47.200}$$

 T_1 and T_2 = the tensile forces in the rods L_1 and L_2 = the lengths of the rods A = the cross-section area where

E = the elastic modulus.

Solving Eqs. (47.199) and (47.200) for T_2 :

$$T_2 = \frac{P}{2}$$
 (47.201)

The load at which the structure reaches the first yield (in Fig. 47.93b) is determined by letting $T_2 = \sigma_y A$. From Eq. (47.201),

$$P_v = 2T_2 = 2\sigma_v A \tag{47.202}$$

The corresponding displacement at first yield is

$$\delta_{y} = \varepsilon_{y}L = \frac{\sigma_{y}L}{2E}$$
(47.203)

After bar 2 is yielded, the system continues to take additional load until all the three bars reach their maximum strength of $\sigma_v A$, as shown in Fig. 47.93c. The plastic limit load of the system is thus written as

$$P_{\rm L} = 3\sigma_{\rm v} A \tag{47.204}$$

The process of successive yielding of bars in this system is known as inelastic redistribution of forces. The displacement at the incipient of collapse is

$$\delta_{\rm L} = \varepsilon_{\rm y} L = \frac{\sigma_{\rm y} L}{E} \tag{47.205}$$

Figure 47.93d shows the load-displacement behavior of the system when subjected to increasing force. As load increases, bar 2 will reach its maximum strength first. As it yields, the force in the member remains constant, and additional loads on the system are taken by the less critical bars. The system will eventually fail when all three bars are fully yielded. This is based on an assumption that material strain hardening does not take place.

Concept of Plastic Hinge

A plastic hinge is said to form in a structural member when the cross-section is fully yielded. If material strain hardening is not considered in the analysis, a fully yielded cross-section can undergo indefinite rotation at a constant restraining plastic moment M_p .

Most of the plastic analyses assume that plastic hinges are concentrated at zero length plasticity. In reality, the yield zone is developed over a certain length, normally called the *plastic hinge length*, depending on the loading, boundary conditions, and geometry of the section. The hinge lengths of beams (Δ L) with different support and loading conditions are shown in Fig. 47.94a to c. Plastic hinges are developed first at the sections subjected to the greatest moment. The possible locations for plastic hinges to develop are at the points of concentrated loads, at the intersections of members involving a change in geometry, and at the point of zero shear for members under uniform distributed load.

Plastic Moment Capacity

A knowledge of full plastic moment capacity of a section is important in plastic analysis. It forms the basis for limit load analysis of the system. Plastic moment is the moment resistance of a fully yielded cross section. The cross-section must be fully compact in order to develop its plastic strength. The component plates of a section must not buckle prior to the attainment of full moment capacity.

The plastic moment capacity, M_p , of a cross-section depends on the material yield stress and the section geometry. The procedure for the calculation of M_p may be summarized in the following two steps:

- The plastic neutral axis of a cross-section is located by considering equilibrium of forces normal to the cross section. Figure 47.95a shows a cross-section of arbitrary shape subjected to increasing moment. The plastic neutral axis is determined by equating the force in compression (C) to that in tension (T). If the entire cross-section is made of the same material, the plastic neutral axis can be determined by dividing the cross-sectional area into two equal parts. If the cross-section is made of more than one type of material, the plastic neutral axis must be determined by summing the normal force and letting the force equal zero.
- The plastic moment capacity is determined by obtaining the moment generated by the tensile and compressive forces.

Consider an arbitrary section with area 2A and with one axis of symmetry of which the section is strengthened by a cover plate of area a, as shown in Fig. 47.95b. Further assume that the yield strengths of the original section and the cover plate are σ_{vo} and σ_{vc} , respectively. At the full plastic state, the total



FIGURE 47.94 Hinge lengths of beams with different support and loading conditions.



FIGURE 47.95 Cross-section of arbitrary shape subjected to bending.

axial force acting on the cover plate is $a\sigma_{yc}$. In order to maintain equilibrium of force in the axial direction, the plastic neutral axis must shift down from its original position by a', i.e.,

$$\mathbf{a}' = \frac{\mathbf{a}\boldsymbol{\sigma}_{yc}}{2\boldsymbol{\sigma}_{yo}} \tag{47.206}$$

The resulting plastic capacity of the "built-up" section may be obtained by summing the full plastic moment of the original section and the moment contribution by the cover plate. The additional capacity is equal to the moment caused by the cover plate force $a\sigma_{yc}$ and a force due to the fictitious stress $2\sigma_{yo}$



FIGURE 47.96 Plastic moment capacities of sections.

acting on the area a', resulting from the shifting of the plastic neutral axis from the tension zone to the compression zone, as shown in Fig. 47.95c.

Figure 47.96 shows the computation of plastic moment capacity of several shapes of cross-sections. Based on the principle developed in this section, the plastic moment capacities of typical cross-sections may be generated. Additional information for sections subjected to combined bending, torsion, shear, and axial load can be found in Mrazik et al. (1987).

Theory of Plastic Analysis

There are two main assumptions for first-order plastic analysis:

- 1. The structure is made of ductile material that can undergo large deformations beyond elastic limit without fracture or buckling.
- 2. The deflections of the structure under loading are small so that second-order effects can be ignored.

An "exact" plastic analysis solution must satisfy three basic conditions. They are *equilibrium*, *mechanism*, and *plastic moment* conditions. The plastic analysis disregards the continuity condition as required by the elastic analysis of indeterminate structures. The formation of a plastic hinge in members leads to discontinuity of slope. If sufficient plastic hinges are formed to allow the structure to deform into a mechanism, this is a mechanism condition. Since plastic analysis utilizes the limit of resistance of a member's plastic strength, the plastic moment condition is required to ensure that the resistance of the cross-sections is not violated anywhere in the structure. Lastly, the equilibrium condition, which is the same condition to be satisfied in elastic analysis, requires that the sum of all applied forces and reactions be equal to zero and that all internal forces be self-balanced.

When all three conditions are satisfied, the resulting plastic analysis for the limiting load is the "correct" limit load. The collapse loads for simple structures such as beams and portal frames can be solved easily using a direct approach or through visualization of the formation of "correct" collapse mechanism. However, for more complex structures, the exact solution satisfying all three conditions may be difficult to predict. Thus, simple techniques using approximate methods of analysis are often used to assess these solutions. These techniques, named equilibrium and mechanism methods, will be discussed in the subsequent sections.

Principle of Virtual Work

The virtual work principle may be applied to relate a system of forces in equilibrium to a system of compatible displacements. For example, if a structure in equilibrium is given a set of small compatible displacement, then the work done by the external loads on these external displacements is equal to the work done by the internal forces on the internal deformation. In plastic analysis, internal deformations are assumed to be concentrated at plastic hinges. The virtual work equation for hinged structures can be written in explicit form as

$$\sum P_i \delta_j = \sum M_i \theta_j \tag{47.207}$$

where P_i is an external load and M_i is an internal moment at a hinge location. Both P_i and M_i constitute an equilibrium set, and they must be in equilibrium. δ_j is the displacement under point load P_i and in the direction of the load. θ_j is the plastic hinge rotation under the moment M_i . Both δ_j and θ_j constitute a displacement set, and they must be compatible with each other.

Lower Bound Theorem

For a given structure, if there exists any distribution of bending moments in the structure that satisfies both the equilibrium and plastic moment conditions, then the load factor, λ_L , computed from this moment diagram must be equal to or less than the collapse load factor, λ_c , of the structure. The lower bound theorem provides a safe estimate of the collapse limit load, i.e., $\lambda_L \leq \lambda_L$.

Upper Bound Theorem

For a given structure subjected to a set of applied loads, a load factor, λ_u , computed based on an assumed collapse mechanism must be greater than or equal to the true collapse load factor, λ_c . The upper bound theorem, which uses only the mechanism condition, overestimates or equals the collapse limit load, i.e., $\lambda_u \ge \lambda_c$.

Uniqueness Theorem

A structure at collapse has to satisfy three conditions. First, a sufficient number of plastic hinges must be formed to turn the structure, or part of it, into a mechanism; this is called the mechanism condition. Second, the structure must be in equilibrium, i.e., the bending moment distribution must satisfy equilibrium with the applied loads. Finally, the bending moment at any cross-section must not exceed the full plastic value of that cross-section; this is called the plastic moment condition. The theorem simply implies that the collapse load factor, λ_c , obtained from the three basic conditions (mechanism, equilibrium, and plastic moment) has a unique value.

The proof of the three theorems can be found in Chen and Sohal (1995). A useful corollary of the lower bound theorem is that if at a load factor, λ , it is possible to find a bending moment diagram that satisfies both the equilibrium and moment conditions but not necessarily the mechanism condition, then the structure will not collapse at that load factor, unless the load happens to be the collapse load. A corollary of the upper bound theorem is that the true load factor at collapse is the smallest possible one that can be determined from a consideration of all possible mechanisms of collapse. This concept is very useful in finding the collapse load of the system from various combinations of mechanisms. From these, it can be seen that the lower bound theorem is based on the equilibrium approach, while the upper bound technique is based on the mechanism approach. These two alternative approaches to an exact solution, called the *equilibrium method* and the *mechanism method*, will be discussed in the following sections.

Equilibrium Method

The equilibrium method, which employs the lower bound theorem, is suitable for the analysis of continuous beams and frames in which the structural redundancies are not exceeding 2. The procedures of obtaining the equilibrium equations of a statically indeterminate structure and evaluating its plastic limit load are as follows:

To obtain the equilibrium equations of a statically indeterminate structure:

- 1. Select the redundant(s).
- 2. Free the redundants and draw a moment diagram for the determinate structure under the applied loads.
- 3. Draw a moment diagram for the structure due to the redundant forces.
- 4. Superimpose the moment diagrams in steps 2 and 3.
- 5. Obtain the maximum moment at critical sections of the structure utilizing the moment diagram in step 4.

To evaluate the plastic limit load of the structure:

- 6. Select value(s) of redundant(s) such that the plastic moment condition is not violated at any section in the structure.
- 7. Determine the load corresponding to the selected redundant(s).
- 8. Check for the formation of a mechanism. If a collapse mechanism condition is met, then the computed load is the exact plastic limit load. Otherwise, it is a lower bound solution.
- 9. Adjust the redundant(s) and repeat steps 6 to 9 until the exact plastic limit load is obtained.

Example 47.11: Continuous Beam

Figure 47.97a shows a two-span continuous beam that is analyzed using the equilibrium method. The plastic limit load of the beam is calculated based on the step-by-step procedure described in the previous section as follows:

- 1. Select the redundant force as M₁, which is the bending moment at the intermediate support, as shown in Fig. 47.97b.
- 2. Free the redundants and draw a moment diagram for the determinate structure under the applied loads, as shown in Fig. 47.97c.



FIGURE 47.97 Analysis of a two-span continuous beam using equilibrium method.

3. Draw a moment diagram for the structure due to the redundant moment M₁, as shown in Fig. 47.97d.

4. Superimpose the moment diagrams in Fig. 47.97c and d, and the results are shown in Fig. 47.97e.

The moment diagram in Fig. 47.97e is redrawn on a single straight baseline. The critical moment in the beam is

$$M_{cr} = \frac{Pa(L-a)}{L} - \frac{M_{1}a}{L}$$
(47.208)

The maximum moment at critical sections of the structure utilizing the moment diagram in Fig. 47.97e is obtained. By letting $M_{cr} = M_p$, the resulting moment distribution is shown in Fig. 47.97f.

A lower bound solution may be obtained by selecting a value of redundant moment M_1 . For example, if $M_1 = 0$ is selected, the moment diagram is reduced to that shown in Fig. 47.97c. By equating the maximum moment in the diagram to the plastic moment, M_p , we have

$$M_{cr} = \frac{Pa(L-a)}{L} = M_{p}$$
 (47.209)

which gives $P = P_1$ as

$$P_{1} = \frac{M_{p}L}{a(L-a)}$$
(47.210)

The moment diagram in Fig. 47.97c shows a plastic hinge formed at each span. Since two plastic hinges in each span are required to form a plastic mechanism, the load P_1 is a lower bound solution.

However, setting the redundant moment M_1 equal to the plastic moment M_p and letting the maximum moment in Fig. 47.97f equal the plastic moment, we have

$$M_{cr} = \frac{Pa(L-a)}{L} - \frac{M_{p}a}{L} = M_{p}$$
(47.211)

which gives $P = P_2$ as

$$P_{2} = \frac{M_{p}(L+a)}{a(L-a)}$$
(47.212)

Since a sufficient number of plastic hinges has formed in the beams (Fig. 47.97g) to arrive at a collapse mechanism, the computed load, P_2 , is the exact plastic limit load.

Example 47.12: Portal Frame

A pin-based rectangular frame is subjected to vertical load V and horizontal load H, as shown in Fig. 47.98a. All the members of the frame are made of the same section with moment capacity M_p . The objective is to determine the limit value of H if the frame's width-to-height ratio, L/h, is 1.0. Procedure:

The frame has one degree of redundancy. The redundancy for this structure can be chosen as the horizontal reaction at E. Figure 47.98b and c show the resulting determinate frame loaded by the applied loads and redundant force. The moment diagrams corresponding to these two loading conditions are shown in Fig. 47.98d and e.

The horizontal reaction S should be chosen in such a manner that all three conditions — equilibrium, plastic moment, and mechanism — are satisfied. Formation of two plastic hinges is necessary to form a



FIGURE 47.98 Analysis of portal frame using equilibrium method.

mechanism. The plastic hinges may be formed at B, C, and D. Assuming that a plastic hinge is formed at D, as shown in Fig. 47.98e, we have

$$S = \frac{M_p}{h}$$
(47.213)

Corresponding to this value of S, the moments at B and C can be expressed as

$$M_{\rm B} = Hh - M_{\rm p} \tag{47.214}$$

$$M_{\rm C} = \frac{\rm Hh}{2} + \frac{\rm VL}{4} - M_{\rm p} \tag{47.215}$$

The condition for the second plastic hinge to form at B is $|M_B| > |M_C|$. Form Eqs. (47.214) and (47.215) we have

$$Hh - M_p > \frac{Hh}{2} + \frac{VL}{4} - M_p$$
 (47.216)

and

$$\frac{\mathrm{V}}{\mathrm{H}} < \frac{\mathrm{h}}{\mathrm{L}} \tag{47.217}$$

The condition for the second plastic hinge to form at C is $|M_C| > |M_B|$. Form Eqs. (47.214) and (47.215) we have

$$Hh - M_p < \frac{Hh}{2} + \frac{VL}{4} - M_p$$
 (47.218)

and

 $\frac{\mathrm{V}}{\mathrm{H}} > \frac{\mathrm{h}}{\mathrm{L}} \tag{47.219}$

For a particular combination of V, H, L, and h, the collapse load for H can be calculated.

When L/h = 1 and V/H = 1/3, we have

$$M_{\rm B} = Hh - M_{\rm p} \tag{47.220}$$

$$M_{c} = \frac{Hh}{2} + \frac{Hh}{12} - M_{p} = \frac{7}{12}Hh - M_{p}$$
(47.221)

Since $|M_B| > |M_C|$, the second plastic hinge will form at B, and the corresponding value for H is

$$H = \frac{2M_p}{h}$$
(47.222)

When L/h = 1 and V/H = 3, we have

$$M_{\rm B} = Hh - M_{\rm p} \tag{47.223}$$

$$M_{c} = \frac{Hh}{2} + \frac{3}{4}Hh - M_{p} = \frac{5}{4}Hh - M_{p}$$
(47.224)

Since $|M_{c}| > |M_{B}|$, the second plastic hinge will form at C, and the corresponding value for H is

$$H = \frac{1.6M_p}{h}$$
 (47.225)

Mechanism Method

This method, which is based on the upper bound theorem, states that the load computed on the basis of an assumed failure mechanism is never less than the exact plastic limit load of a structure. Thus, it always predicts the upper bound solution of the collapse limit load. It can also be shown that the minimum upper bound is the limit load itself. The procedure of using the mechanism method has the following two steps:

- 1. Assume a failure mechanism and form the corresponding work equation from which an upper bound value of the plastic limit load can be estimated.
- 2. Write the equilibrium equations for the assumed mechanism and check the moments to see whether the plastic moment condition is met everywhere in the structure.

To obtain the true limit load using the mechanism method, it is necessary to determine every possible collapse mechanism, some of which are the combinations of a certain number of independent mechanisms. Once the independent mechanisms have been identified, a work equation may be established for each combination, and the corresponding collapse load is determined. The lowest load among those obtained by considering all the possible combinations of independent mechanisms is the correct plastic limit load.

Independent Mechanisms

The number of possible independent mechanisms, n, for a structure can be determined from the following equation:

$$n = N - R$$
 (47.226)

where N is the number of critical sections at which plastic hinges might form and R is the degrees of redundancy of the structure.

Critical sections generally occur at the points of concentrated loads, at joints where two or more members are meeting at different angles, and at sections where there is an abrupt change in section geometries or properties. To determine the number of redundancies (R) of a structure, it is necessary to free sufficient supports or restraining forces in structural members so that the structure becomes an assembly of several determinate substructures.

Figure 47.99 shows two examples. The cuts that are made in each structure reduce the structural members to either cantilevers or simply supported beams. The fixed-end beam requires a shear force and a moment to restore continuity at the cut section, and thus R = 2. For the two-story frame, an axial force, shear, and moment are required at each cut section for full continuity, and thus R = 12.

Types of Mechanisms

Figure 47.100a shows a frame structure subjected to a set of loading. The frame may fail by different types of collapse mechanisms dependent on the magnitude of loading and the frame's configurations. The collapse mechanisms are:

- 1. Beam: possible mechanisms of this type are shown in Fig. 47.100b.
- 2. Panel: the collapse mode is associated with side sway, as shown in Fig. 47.100c.
- 3. *Gable*: the collapse mode is associated with the spreading of column tops with respect to the column bases, as shown in Fig. 47.100d.



FIGURE 47.99 Number of redundants in: (a) a beam, (b) a frame.



FIGURE 47.100 Typical plastic mechanisms.

- 4. *Joint*: the collapse mode is associated with the rotation of joints of which the adjoining members developed plastic hinges and deformed under an applied moment, as shown in Fig. 47.100e.
- 5. *Combined*: it can be a partial collapse mechanism, as shown in Fig. 47.100f, or it may be a complete collapse mechanism, as shown in Fig. 47.100g.

The principal rule for combining independent mechanisms is to obtain a lower value of collapse load. The combinations are selected in such a way that the external work becomes a maximum and the internal work becomes a minimum. Thus the work equation would require that the mechanism involve as many applied loads as possible and at the same time eliminate as many plastic hinges as possible. This procedure is illustrated in the following example.

Example 47.13: Rectangular Frame

A fixed-end rectangular frame has a uniform section with $M_p = 20$ and carries the load shown in Fig. 47.101. Determine the value of load ratio λ at collapse. Solution:

Number of possible plastic hinges:	N = 5
Number of redundancies:	R = 3
Number of independent mechanisms:	N - R = 2

The two independent mechanisms are shown in Fig. 47.101b and c, and the corresponding work equations are

Panel mechanism: $20\lambda = 4(20) = 80 \implies \lambda = 4$ Beam mechanism: $30\lambda = 4(20) = 80 \Rightarrow \lambda = 2.67$







FIGURE 47.101 Collapse mechanisms of a fixed base portal frame.

The combined mechanisms are now examined to see whether they will produce a lower λ value. It is observed that only one combined mechanism is possible. The mechanism is shown in Fig. 47.101c and involves cancellation of the plastic hinge at B. The calculation of the limit load is described below:

Panel mechanism:	$20\lambda = 4(20)$
Beam mechanism:	$30\lambda = 4(20)$
Addition:	$50\lambda = 8(20)$
Cancel of plastic hinge:	-2(20)
Combined mechanism:	$50\lambda = 6(20) \Rightarrow \lambda = 2.4$

The combined mechanism results in a smaller value for λ , and no other possible mechanism can produce a lower load. Thus, $\lambda = 2.4$ is the collapse load.

Example 47.14: Frame Subjected to Distributed Load

When a frame is subjected to distributed loads, the maximum moment and hence the plastic hinge location is not known in advance. The exact location of the plastic hinge may be determined by writing the work equation in terms of the unknown distance and then maximizing the plastic moment by formal differentiation.

Consider the frame shown in Fig. 47.102a. The side sway collapse mode in Fig. 47.102b leads to the following work equation:

$$4M_{p} = 24(10\theta)$$

which gives

 $M_{p} = 60$

The beam mechanism of Fig. 47.102c gives

$$4M_{p}\theta = \frac{1}{2}(10\theta)32$$

which gives

 $M_{p} = 40$

In fact the correct mechanism is shown in Fig. 47.102d, in which the distance Z from the plastic hinge location is unknown. The work equation is

$$24(10\theta) + \frac{1}{2}(1.6)(20)(z\theta) = M_{p}\left(2 + 2\left(\frac{20}{20-z}\right)\right)\theta$$

which gives

$$M_{p} = \frac{(240 + 16z)(20 - z)}{80 - 2z}$$

To maximize M_p, the derivative of M_p is set to zero, i.e.,

$$(80-2z)(80-32z)+(4800+80z-16z^{2})(2)=0$$



FIGURE 47.102 Portal frame subjected to a combined uniform distributed load and horizontal load. which gives

$$z = 40 - \sqrt{1100} = 6.83$$

and

$$M_{p} = 69.34$$

In practice, uniform load is often approximated by applying several equivalent point loads to the member under consideration. Plastic hinges thus can be assumed to form only at the concentrated load points, and the calculations become simpler when the structural system is getting more complex.

Example 47.15: Gable Frame

The mechanism method is used to determine the plastic limit load of the gable frame shown in Fig. 47.103. The frame is composed of members with a plastic moment capacity of 270 kip-in. The column bases are fixed. The frame is loaded by a horizontal load H and vertical concentrated load V. A graph from which V and H cause the collapse of the frame is to be produced.



FIGURE 47.103 Collapse mechanisms of a fixed base gable frame.

Solution:

Consider the three modes of collapse as follows:

Mechanism 1: plastic hinges form at A, C, D, and E:

The mechanism is shown in Fig. 47.103b. The instantaneous center O for member CD is located at the intersection of AC and $_{\rm E}$ D extended. From similar triangles ACC1 and OCC2, we have

$$\frac{OC_2}{CC_2} = \frac{C_1A}{C_1C}$$

which gives

$$OC_2 = \frac{C_1 A}{C_1 C} CC_2 = \frac{22.5(9)}{18} = 11.25 \text{ft}$$

From triangles ACC' and CC'O, we have

$$AC(\phi) = OC(\theta)$$



FIGURE 47.103 (continued).

which gives

$$\phi = \frac{OC}{AC}\theta = \frac{CC_2}{C_1C}\theta = \frac{9}{8}\theta = \frac{1}{2}\theta$$

Similarly, from triangles ODD' and EDD', the rotation at E is given as

 $DE(\Psi) = OD(\theta)$

which gives

$$\Psi = \frac{\text{OD}}{\text{DE}} \theta = 1.5\theta$$

From the hinge rotations and displacements, the work equation for this mechanism can be written as

$$V(9\theta) + H(13.5\Psi) = M_{p}[\phi + (\phi + \theta) + (\theta + \Psi) + \Psi]$$

Substituting values for ψ and ϕ and simplifying, we have

$$V + 2.25H = 180$$

Mechanism 2: mechanism with hinges at B, C, D, and E:

Figure 47.103c shows the mechanism in which the plastic hinge rotations and displacements at the load points can be expressed in terms of the rotation of member CD about the instantaneous center O.

From similar triangles BCC₁ and OCC₂, we have

$$\frac{OC_2}{CC_2} = \frac{BC_1}{C_1C}$$

which gives

$$OC_2 = \frac{BC_1}{C_1C}CC_2 = \frac{9}{18}(9) = 4.5$$

From triangles BCC' and CC'O, we have

$$BC(\phi) = OC(\theta)$$

which gives

$$\phi = \frac{OC}{BC} \theta = \frac{OC_2}{BC_1} \theta = \frac{4.5}{9} \theta = \frac{1}{2} \theta$$

Similarly, from triangles ODD' and EDD', the rotation at E is given as

$$DE(\Psi) = OD(\theta)$$

which gives

$$\Psi = \frac{\text{OD}}{\text{DE}} \theta = \theta$$

The work equation for this mechanism can be written as

$$V(9\theta) + H(13.5\Psi) = M_{p} \left[\phi + (\phi + \theta) + (\theta + \Psi) + \Psi \right]$$

Substituting values of ψ and ϕ and simplifying, we have

$$V + 1.5H = 150$$

Mechanism 3: mechanism with hinges at A, B, D, and E:

The hinge rotations and displacements corresponding to this mechanism are shown in Fig. 47.103d. The rotation of all hinges is θ . The horizontal load moves by 13.5 θ , but the horizontal load has no vertical displacement. The work equation becomes

$$H(13.5\theta) = M_{p}(\theta + \theta + \theta + \theta)$$

or

$$H = 80$$

The interaction equations corresponding to the three mechanisms are plotted in Fig. 47.104. By carrying out moment checks, it can be shown that mechanism 1 is valid for portion AB of the curve, mechanism 2 is valid for portion BC, and mechanism 3 is valid only when V = 0.





Analysis Aids for Gable Frames

Pin-Based Gable Frames

Figure 47.105a shows a pinned-end gable frame subjected to a uniform gravity load λ wL and a horizontal load λ 1H at the column top. The collapse mechanism is shown in Fig. 47.105b. The work equation is used to determine the plastic limit load. First, the instantaneous center of rotation O is determined by considering similar triangles,

$$\frac{OE}{CF} = \frac{L}{xL} \quad \text{and} \quad \frac{OE}{CF} = \frac{OE}{h_1 + 2xh_2}$$
(47.227)

and

$$OD = OE - h_1 = \frac{(1 - x)h_1 + 2xh_2}{x}$$
(47.228)

From the horizontal displacement of D,

$$\theta \mathbf{h}_{1} = \phi \mathrm{OD} \tag{47.229}$$

of which

$$\phi = \frac{x}{(1-x)+2xk} \Theta \tag{47.230}$$

where $k = h_2/h_1$. From the vertical displacement at C,

$$\beta = \frac{1 - x}{(1 - x) + 2xk} \Theta \tag{47.231}$$

The work equation for the assumed mechanism is

$$\lambda_{1}Hh_{1}\beta + \frac{\lambda wL^{2}}{2}(1-x)\phi = M_{p}(\beta + 2\phi + \theta)$$
(47.232)



FIGURE 47.105 Pinned base gable frame subjected to a combined uniform distributed load and horizontal load. which gives

$$M_{p} = \frac{(1-x)\lambda_{1}Hh_{1} + (1-x)x\lambda wL^{2}/2}{2(1+kx)}$$
(47.233)

Differentiating M_{p} in Eq. (47.233) with respect to x and solving for x,

$$\mathbf{x} = \frac{\mathbf{A} - \mathbf{1}}{\mathbf{k}} \tag{47.234}$$

where

$$A = \sqrt{(1+k)(1-Uk)} \quad \text{and} \quad U = \frac{2\lambda_1 H h_1}{\lambda w L^2}$$
(47.235)

Substituting for \boldsymbol{x} in the expression for \boldsymbol{M}_{p} gives

$$M_{p} = \frac{\lambda w L^{2}}{8} \left[\frac{U(2+U)}{A^{2} + 2A - Uk^{2} + 1} \right]$$
(47.236)



FIGURE 47.106 Analysis chart for pinned base gable frame.

In the absence of horizontal loading, the gable mechanism, as shown in Fig. 47.105c, is the failure mode. In this case, letting H = 0 and U = 0 gives (Horne, 1964):

$$M_{p} = \frac{\lambda w L^{2}}{8} \left[\frac{1}{1+k+\sqrt{1+k}} \right]$$
(47.237)

Equation (47.236) can be used to produce a chart, as shown in Fig. 47.106, by which the value of M_p can be determined rapidly by knowing the values of

$$k = \frac{h_2}{h_1} \quad \text{and} \quad U = \frac{2\lambda_1 H h_1}{\lambda w L^2}$$
(47.238)

Fixed-Base Gable Frames

A similar chart can be generated for fixed-base gable frames, as shown in Fig. 47.107. Thus, if the values of loading, λw and $\lambda_1 H$, and frame geometry, h_1 , h_2 , and L, are known, the parameters k and U can be evaluated and the corresponding value of $M_p/(\lambda w L^2)$ can be read directly from the appropriate chart. The required value of M_p is obtained by multiplying the value of $M_p/(\lambda w L^2)$ by $\lambda w L^2$.

Grillages

Grillage is a type of structure that consists of straight beams lying on the same plane, subjected to loads acting perpendicular to the plane. An example of such a structure is shown in Fig. 47.108. The grillage consists of two equal simply supported beams of span length 2L and full plastic moment M_p . The two beams are connected rigidly at their centers, where a concentrated load W is carried.

The collapse mechanism consists of four plastic hinges formed at the beams adjacent to the point load, as shown in Fig. 47.108. The work equation is

$$WL\theta = 4M_{p}\theta$$



FIGURE 47.107 Analysis chart for fixed gable frame.



FIGURE 47.108 Two-beam grillage system.

of which the collapse load is

$$W = \frac{4M_p}{L}$$

Six-Beam Grillage

A grillage consisting of six beams of span length 4L each and full plastic moment M_p is shown in Fig. 47.109. A total load of 9W acts on the grillage, splitting into concentrated loads W at the nine nodes. Three collapse mechanisms are possible. Ignoring member twisting due to torsional forces, the work equations associated with the three collapse mechanisms are computed as follows: Mechanism 1 (Fig. 47.110a):



FIGURE 47.109 Six-beam grillage system.



FIGURE 47.110 Six-beam grillage system: (a) mechanism 1, (b) mechanism 2, (c) mechanism 3.

Work equation:

$$9 \text{wL}\theta = 12 \text{M}_{\text{p}}\theta$$

of which

$$w = \frac{12}{9} \frac{M_p}{L} = \frac{4M_p}{3L}$$

Mechanism 2 (Fig. 47.110b):

Work equation:

$$wL\theta = 8M_p\theta$$

of which

$$w = \frac{8M_p}{L}$$

Mechanism 3 (Fig. 47.110c):

Work equation:

$$w2L2\theta + 4 \times w2L\theta = M_p(4\theta + 8\theta)$$

of which

 $w = \frac{M_p}{L}$

The lowest upper bound load corresponds to mechanism 3. This can be confirmed by conducting a moment check to ensure that bending moments anywhere are not violating the plastic moment condition. Additional discussion of plastic analysis of grillages can be found in Baker and Heyman (1969) and Heyman (1971).

Vierendeel Girders

Figure 47.111 shows a simply supported girder in which all members are rigidly joined and have the same plastic moment M_p . It is assumed that axial loads in the members do not cause member instability. Two possible collapse mechanisms are considered, as shown in Fig. 47.111b to c.

The work equation for mechanism 1 is

$$W3\theta L = 20M_{p}\theta$$

so that

$$W = \frac{20M_p}{3L}$$

The work equation for mechanism 2 is

$$W3\theta L = 16M_{p}\theta$$

or

$$W = \frac{16M_p}{3L}$$

It can be easily proved that the collapse load associated with mechanism 2 is the correct limit load. This is done by constructing an equilibrium set of bending moments and checking that they are not violating the plastic moment condition.



FIGURE 47.111 Collapse mechanism of a Vierendeel girder.

Hinge-by-Hinge Analysis

Instead of finding the collapse load of the frame, it may be useful to obtain information about the distribution and redistribution of forces prior to reaching the collapse load. Elastic-plastic hinge analysis (also known as hinge-by-hinge analysis) determines the order of plastic hinge formation, the load factor associated with each plastic hinge formation, and member forces in the frame between each hinge formation. Thus the state of the frame can be defined at any load factor rather than only at the state of collapse. This allows a more accurate determination of member forces at the design load level.

Educational and commercial software are now available for elastic-plastic hinge analysis (Chen and Sohal, 1995). The computations of deflections for simple beams and multistory frames can be done using the virtual work method (Chen and Sohal, 1995; ASCE, 1971; Beedle, 1958; Knudsen et al., 1953). The basic assumption of first-order elastic-plastic hinge analysis is that the deformations of the structure are insufficient to alter radically the equilibrium equations. This assumption ceases to be true for slender members and structures, and the method gives unsafe predictions of limit loads.

47.12 Stability of Structures

Stability Analysis Methods

Several stability analysis methods have been utilized in research and practice. Figure 47.112 shows schematic representations of the load-displacement results of a sway frame obtained from each type of analysis to be considered.

Elastic Buckling Analysis

The elastic buckling load is calculated by linear buckling or bifurcation (or eigenvalue) analysis. The buckling loads are obtained from the solutions of idealized elastic frames subjected to loads that do not



FIGURE 47.112 Catagorization of stability analysis methods.

produce direct bending in the structure. The only displacements that occur before buckling occurs are those in the directions of the applied loads. When buckling (bifurcation) occurs, the displacements increase without bound, assuming linearized theory of elasticity and small displacement, as shown by the horizontal straight line in Fig. 47.112. The load at which these displacements occur is known as the buckling load, commonly referred to as the bifurcation load. For structural models that actually exhibit a bifurcation from the primary load path, the elastic buckling load is the largest load that the model can sustain, at least within the vicinity of the bifurcation point, provided that the postbuckling path is in unstable equilibrium. If the secondary path is in stable equilibrium, the load can still increase beyond the critical load value.

Buckling analysis is a common tool for calculations of column effective lengths. The effective length factor of a column member can be calculated using the procedure described later. The buckling analysis provides useful indices of the stability behavior of structures; however, it does not predict actual behavior of all structures, but of idealized structures with gravity loads applied only at the joints.

Second-Order Elastic Analysis

The analysis is formulated based on the deformed configuration of the structure. When derived rigorously, a second-order analysis can include both the member curvature (P- δ) and the side sway (P- Δ) stability effects. The P- δ effect is associated with the influence of the axial force acting through the member displacement with respect to the rotated chord, whereas the P- Δ effect is the influence of the axial force acting through the relative side sway displacements of the member ends. A structural system will become stiffer when its members are subjected to tension. Conversely, the structure will become softer when its members are in compression. Such behavior can be illustrated by a simple model shown in Fig. 47.113. There is a clear advantage for a designer to take advantage of the stiffer behavior of tension structures. However, the detrimental effects associated with second-order deformations due to compression forces must be considered in designing structures subjected to predominant gravity loads.

Unlike the first-order analysis, in which solutions can be obtained in a rather simple and direct manner, the second-order analysis often requires an iterative procedure to obtain solutions. The load-displacement curve generated from a second-order elastic analysis will gradually approach the horizontal straight line, which represents the buckling load obtained from the elastic buckling analysis, as shown in Fig. 47.112.



FIGURE 47.113 Behavior of frame in compression and tension.

Differences in the two limit loads may arise from the fact that the elastic stability limit is calculated for equilibrium based on the deformed configuration, whereas the elastic critical load is calculated as a bifurcation from equilibrium on the undeformed geometry of the frame.

The load-displacement response of many practical structures usually does not involve any bifurcation of the equilibrium path. In some cases, the second-order elastic incremental response may not have yielded any limit. See Chen and Lui (1987) for a basic discussion of these behavioral issues.

Recent works on second-order elastic analysis have been reported in Liew et al. (1991), White and Hajjar (1991), Chen and Lui (1991), and Chen and Toma (1994), among others. Second-order analysis programs that can take into consideration connection flexibility are also available (Chen et al., 1996; Chen and Kim, 1997; Faella et al., 2000).

Second-Order Inelastic Analysis

Second-order inelastic analysis refers to methods of analysis that can capture geometrical and material nonlinearities of the structures. The most rigorous inelastic analysis method is called spread-of-plasticity analysis. It involves discretization of a member into many line segments and the cross-section of each segment into a number of finite elements. Inelasticity is captured within the cross-sections and along the member length. The calculation of forces and deformations in the structure after yielding requires iterative trial-and-error processes because of the nonlinearity of the load–deformation response and the change in the cross section effective stiffness at inelastic regions associated with the increase in the applied loads and the change in structural geometry. Although most spread-of-plasticity analysis methods have been developed for planar analysis (White, 1985; Vogel, 1985), three-dimensional spread-of-plasticity techniques are also available involving various degrees of refinements (Clark, 1994; White, 1988; Wang, 1988; Chen and Atsuta, 1977; Jiang et al, 2002).

The simplest second-order inelastic analysis is the *elastic-plastic hinge* approach. The analysis assumes that the element remains elastic except at its ends, where zero-length plastic hinges are allowed to form. Plastic hinge analysis of planar frames can be found in Orbison (1982), Ziemian et al. (1992a, 1992b), White et al. (1993), Liew et al. (1993), Chen and Toma (1994), Chen and Sohal (1995), and Chen et al. (1996). Advanced analyses of three-dimensional frames are reported in Chen et al. (2000) and Liew et al. (2000). Second-order plastic hinge analysis allows efficient analysis of large-scale building frames. This is particularly true for structures in which the axial forces in the component members are small and the behavior is predominated by bending actions. Although elastic-plastic hinge approaches can provide essentially the same load-displacement predictions as second-order plastic-zone methods for many frame



FIGURE 47.114 Stability equations of a column segment.

problems, they cannot be classified as advanced analysis for use in frame design. Some modifications to the elastic-plastic hinge are required to qualify the methods as advanced analysis; they are discussed later.

Figure 47.112 shows the load-displacement curve (a smooth curve with a descending branch) obtained from the second-order inelastic analysis. The computed limit load should be close to that obtained from the plastic-zone analysis.

Column Stability

Stability Equations

The stability equation of a column can be obtained by considering an infinitesimal deformed segment of the column, as shown in Fig. 47.114. Considering the moment equilibrium about point b, we obtain

$$Q\,dx + P\,dy + M - \left(M + \frac{dM}{dx}dx\right) = 0$$

or, upon simplification,

$$Q = \frac{dM}{dx} - P\frac{dy}{dx}$$
(47.239)

Summing the force horizontally, we can write

$$-Q + \left(Q + \frac{dQ}{dx}dx\right) = 0$$

or, upon simplification,

$$\frac{\mathrm{dQ}}{\mathrm{dx}} = 0 \tag{47.240}$$

Differentiating Eq. (47.239) with respect to x, we obtain

$$\frac{\mathrm{dQ}}{\mathrm{dx}} = \frac{\mathrm{d}^2 \mathrm{M}}{\mathrm{dx}^2} - \mathrm{P}\frac{\mathrm{d}^2 \mathrm{y}}{\mathrm{dx}^2} \tag{47.241}$$

which, when compared with Eq. (47.240), gives

$$\frac{d^2M}{dx^2} - P\frac{d^2y}{dx^2} = 0$$
(47.242)

Since moment M = $-EI(d^2y/dx^4)$, Eq. (47.242) can be written as

$$EI\frac{d^4y}{dx^4} + P\frac{d^2y}{dx^2} = 0$$
 (47.243)

or

$$y^{IV} + k^2 y'' = 0 (47.244)$$

Equation (47.244) is the general fourth-order differential equation that is valid for all support conditions. The general solution to this equation is

$$y = A \sin kx + B \cos kx + Cx + D \qquad (47.245)$$

To determine the critical load, it is necessary to have four boundary conditions: two at each end of the column. In some cases, both geometric and force boundary conditions are required to eliminate the unknown coefficients (A, B, C, and D) in Eq. (47.245).

Column with Pinned Ends

For a column pinned at both ends, as shown in Fig. 47.115a, the four boundary conditions are:

$$y(x=0)=0, M(x=0)=0$$
 (47.246)

$$y(x = L) = 0, \quad M(x = L) = 0$$
 (47.247)

Since M = -EIy'', the moment conditions can be written as

$$y''(0) = 0 \text{ and } y''(x = L) = 0$$
 (47.248)



FIGURE 47.115 Column with: (a) pinned ends, (b) fixed ends, (c) fixed-free ends.

Using these conditions, we have

$$B = D = 0$$
 (47.249)

The deflection function (Eq. 47.245) reduces to

$$y = A\sin kx + Cx \tag{47.250}$$

Using the conditions y(L) = y''(L) = 0, Eq. (47.250) gives

$$A \sin kL + CL = 0$$
 (47.251)

and

$$-Ak^2 \sin kL = 0 \tag{47.252}$$

$$\begin{bmatrix} \sin kL & L \\ -k^2 \sin kL & 0 \end{bmatrix} \begin{bmatrix} A \\ C \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$
(47.253)

If A = C = 0, the solution is trivial. Therefore, to obtain a nontrivial solution, the determinant of the coefficient matrix of Eq. (47.253) must be zero, i.e.,

$$det \begin{vmatrix} \sin kL & L \\ -k^2 \sin kL & 0 \end{vmatrix} = 0$$
(47.254)

or

$$k^{2}L\sin kL = 0$$
 (47.255)

Since k²L cannot be zero, we must have

$$\sin kL = 0$$
 (47.256)

or

$$kL = n\pi, n = 1, 2, 3, \dots$$
 (47.257)

The lowest buckling load corresponds to the first mode obtained by setting n = 1:

$$P_{\rm cr} = \frac{\pi^2 \,\text{EI}}{L^2} \tag{47.258}$$

Column with Fixed Ends

The four boundary conditions for a fixed-end column are (Fig. 47.115b):

$$y(x=0) = y'(x=0) = 0$$
(47.259)

$$y(x = L) = y'''(x = L) = 0$$
(47.260)

Using the first two boundary conditions, we obtain

$$D = -B, \quad C = -Ak$$
 (47.261)

The deflection function (Eq. 47.245) becomes

$$y = A(\sin kx - kx) + B(\cos kx - 1)$$
 (47.262)

Using the last two boundary conditions, we have

$$\begin{bmatrix} \sin kL - kL & \cos kL - 1\\ \cos kL - 1 & -\sin kL \end{bmatrix} \begin{bmatrix} A\\ B \end{bmatrix} = \begin{bmatrix} 0\\ 0 \end{bmatrix}$$
(47.263)

For a nontrivial solution, we must have

$$det \begin{bmatrix} \sin kL - kL & \cos kL - 1\\ \cos kL - 1 & -\sin kL \end{bmatrix} = 0$$
(47.264)

or, after expanding,

$$kL\sin kL + 2\cos kL - 2 = 0 \tag{47.265}$$

Using trigonometric identities $\sin kL = 2 \sin (kL/2) \cos (kL/2)$ and $\cos kL = 1 - 2 \sin^2 (kL/2)$, Eq. (47.265) can be written as

$$\sin\frac{kL}{2}\left(\frac{kL}{2}\cos\frac{kL}{2} - \sin\frac{kL}{2}\right) = 0 \tag{47.266}$$

The critical load for the symmetric buckling mode is $P_{cr} = 4\pi^2 EI/L^2$ by letting sin (kL/2) = 0. The buckling load for the antisymmetric buckling mode is $P_{cr} = 80.8 EI/L^2$ by letting the bracket term in Eq. (47.266) equal zero.

Column with One End Fixed and One End Free

The boundary conditions for a fixed-free column are (Fig. 47.115c):

$$y(x=0) = y'(x=0) = 0$$
(47.267)

at the fixed end and

$$y''(x=L)=0$$
 (47.268)

and at the free end. The moment M = EIy''' is equal to zero, and the shear force V = -dM/dx = -EI''' is equal to Py', which is the transverse component of P acting at the free end of the column:

$$V = -EIy''' = Py'$$
 (47.269)

It follows that the shear force condition at the free end has the form

$$\mathbf{y'''} + \mathbf{k}^2 \mathbf{y'} = \mathbf{0} \tag{47.270}$$

Using the boundary conditions at the fixed end, we have

$$B+D=0$$
, and $Ak+C=0$ (47.271)

The boundary conditions at the free end give

A sin
$$kL + B \cos kL = 0$$
, and $C = 0$ (47.272)

In matrix form, Eqs. (47.271) and (47.272) can be written as

$$\begin{bmatrix} 0 & 1 & 1 \\ k & 0 & 0 \\ \sin kL & \cos kL & 0 \end{bmatrix} \begin{bmatrix} A \\ B \\ C \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix}$$
(47.273)

For a nontrivial solution, we must have

$$det \begin{vmatrix} 0 & 1 & 1 \\ k & 0 & 0 \\ \sin kL & \cos kL & 0 \end{vmatrix} = 0$$
(47.274)

The characteristic equation becomes

$$k \cos kL = 0$$
 (47.275)

Since k cannot be zero, we must have $\cos kL = 0$ or

$$kL = \frac{n\pi}{2}$$
 n = 1,3,5,... (47.276)

The smallest root (n = 1) gives the lowest critical load of the column

$$P_{\rm cr} = \frac{\pi^2 EI}{4L^2}$$
(47.277)

The boundary conditions for columns with various end conditions are summarized in Table 47.1.

Column Effective Length Factor

The effective length factor, K, of columns with different end boundary conditions can be obtained by equating the P_{cr} load obtained from the buckling analysis with the Euler load of a pinned-end column of effective length KL:

$$P_{cr} = \frac{\pi^2 EI}{(KL)^2}$$

The effective length factor can be obtained as

$$K = \sqrt{\frac{\pi^2 EI/L^2}{P_{cr}}}$$
(47.278)

The K factor is a factor that can be multiplied to the actual length of the end-restrained column to give the length of an equivalent pinned-end column whose buckling load is the same as that of the endrestrained column. Table 47.1 (AISC, 1993) summarizes the theoretical K factors for columns with different boundary conditions. Also shown in the table are the recommended K factors for design applications. The recommended values for design are equal to or higher than the theoretical values to account for semirigid effects of the connections used in practice.

Stability of Beam-Columns

Figure 47.116a shows a beam-column subjected to an axial compressive force P at the ends, a lateral load w along the entire length, and end moments M_A and M_B . The stability equation can be derived by considering the equilibrium of an infinitesimal element of length ds, as shown in Fig. 47.116b. The cross section forces S and H act in the vertical and horizontal directions.

TABLE 47.1Boundary Conditionsfor Various End Conditions

End Conditions	Boundary Conditions		
Pinned	y = 0,	y'' = 0	
Fixed	y = 0,	y'' = 0	
Guided	$\mathbf{y'} = 0$	y''' = 0	
Free	y‴ = 0,	$\mathbf{y}''' + \mathbf{k}^2 \mathbf{y'} = 0$	

	(a)	(b)	(c)	(d)	(e)	(f)
Buckled shape of column is shown by dashed line						
Theoretical K value	0.5	0.7	1.0	1.0	2.0	2.0
Recommended design value when ideal condi- tions are approximated	0.65	0.80	1.2	1.0	2.10	2.0
End condition code		Rotation fixed and translation fixed Rotation free and translation fixed Rotation fixed and translation free Rotation free and translation free				

 TABLE 47.2
 Comparison of Theoretical and Design K Factors



FIGURE 47.116 Basic differential equation of a beam-column.

Considering the equilibrium of forces,

Horizontal equilibrium:

$$H + \frac{dH}{ds}ds - H = 0 \tag{47.279}$$

Vertical equilibrium:

$$S + \frac{dS}{ds}ds - S + w ds = 0 \tag{47.280}$$

Moment equilibrium:

$$M + \frac{dM}{ds}ds - M - \left(S + \frac{dS}{ds} + S\right)\cos\theta\left(\frac{ds}{2}\right) + \left(H + \frac{dH}{ds}ds + H\right)\sin\theta\left(\frac{ds}{2}\right) = 0$$
(47.281)

Since (dS/ds)ds and (dH/ds)ds are negligibly small compared to S and H, the above equilibrium equations can be reduced to

$$\frac{\mathrm{dH}}{\mathrm{ds}} = 0 \tag{47.282a}$$

$$\frac{\mathrm{dS}}{\mathrm{ds}} + \mathbf{w} = 0 \tag{47.282b}$$

$$\frac{dM}{ds} - S\cos\theta + H\sin\theta = 0 \qquad (47.282c)$$

For small deflections and neglecting shear deformations,

$$ds \cong dx, \quad \cos\theta \cong 1 \quad \sin\theta \cong \theta \cong \frac{dy}{dx}$$
 (47.283)

where y is the lateral displacement of the member. Using the above approximations, Eq. (47.282) can be written as

$$\frac{\mathrm{dM}}{\mathrm{dx}} - \mathrm{S} + \mathrm{H}\frac{\mathrm{dy}}{\mathrm{dx}} = 0 \tag{47.284}$$

Differentiating Eq. (47.284) and substituting Eq. (47.283a and b) into the resulting equation, we have

$$\frac{d^2M}{dx^2} + w + H\frac{d^2y}{dx^2} = 0$$
(47.285)

From elementary mechanics of materials, it can easily be shown that

$$M = -EI \frac{d^2 y}{dx^2}$$
(47.286)

Upon substitution of Eq. (47.286) into Eq. (47.285) and realizing that H = -P, we obtain

$$EI\frac{d^4y}{dx^4} + P\frac{d^2y}{dx^2} = w$$
 (47.287)

The general solution to this differential equation has the form

$$y = A \sin kx + B \cos kx + Cx + D + f(x)$$
(47.288)

where $k = \sqrt{P/EI}$ and f(x) is a particular solution satisfying the differential equation. The constants A, B, C, and D can be determined from the boundary conditions of the beam-column under investigation.



FIGURE 47.117 Beam-column subjects to uniform loading.

Beam-Column Subjected to Transverse Loading

Figure 47.117 shows a fixed-end beam-column with a uniformly distributed load w. The general solution to Eq. (47.287) is

y = A sin kx + B cos x + Cx + D +
$$\frac{W}{2EIk^2}x^2$$
 (47.289)

Using the boundary conditions

$$y_{x=0} = 0$$
 $y'_{x=0} = 0$ $y_{x=L} = 0$ $y'_{x=L} = 0$ (47.290)

in which a prime denotes differentiation with respect to x, it can be shown that

$$A = \frac{wL}{2EIk^3}$$
(47.291a)

$$B = \frac{wL}{2EIk^3 \tan(kL/2)}$$
(47.291b)

$$C = -\frac{wL}{2EIk^2}$$
(47.291c)

$$D = -\frac{wL}{2EIk^{3} \tan(kL/2)}$$
(47.291d)

Upon substitution of these constants into Eq. (47.289), the deflection function can be written as

$$y = \frac{wL}{2EIk^{3}} \left[\sin kx + \frac{\cos kx}{\tan(kL/2)} - kx - \frac{1}{\tan(kL/2)} + \frac{kx^{2}}{L} \right]$$
(47.292)

The maximum moment for this beam-column occurs at the fixed ends and is equal to

$$M_{max} = -EIy''|_{x=0} = -EIy''|_{x=L} = -\frac{wL^2}{12} \left[\frac{3(\tan u - u)}{u^2 \tan u}\right]$$
(47.293)

where u = kL/2.

Since wL²/12 is the maximum first-order moment at the fixed ends, the term in the bracket represents the theoretical moment amplification factor due to the P- δ effect.
Boundary Conditions	Р	Location of M	Moment Amplification Factor
Hinged-hinged	$\frac{\pi^2 EI}{L^2}$	Midspan	$\frac{2(\sec u - 1)}{u^2}$
Hinged-fixed	$\frac{\pi^2 \text{EI}}{\left(0.7 \text{L}\right)^2}$	End	$\frac{2(\tan u - u)}{u^2(1/2u - 1/\tan 2u)}$
Fixed-fixed	$\frac{\pi^2 \operatorname{EI}}{\left(0.5 L\right)^2}$	End	$\frac{3(\tan u - u)}{u^2 \tan u}$
Hinged-hinged	$\frac{\pi^2 \text{EI}}{\text{L}^2}$	Midspan	tanu u
Hinged-fixed	$\frac{\pi^2 \text{EI}}{\left(0.7 L\right)^2}$	End	$\frac{4u(1-\cos u)}{3u^2\cos \left(1/2u-1/\tan 2u\right)}$
Fixed-fixed	$\frac{\pi^2 \operatorname{EI}}{(0.5 \mathrm{L})^2}$	Midspan and end	$\frac{2(1-\cos u)}{u\sin u}$
M _A			MB
		EI = constant	
		L	

TABLE 47.3Theoretical and Design Moment Amplification Factor (u = kL/2 = $1/2 \sqrt{(PL^2 / EI)}$

FIGURE 47.118 Beam-column subjects to end moments.

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For beam-columns with other transverse loading and boundary conditions, a similar approach can be followed to determine the moment amplification factor. Table 47.3 summarizes the expressions for the theoretical and design moment amplification factors for some loading conditions (AISC, 1989).

Beam-Column Subjected to End Moments

Consider the beam-column shown in Fig. 47.118. The member is subjected to an axial force of P and end moments M_A and M_B . The differential equation for this beam-column can be obtained from Eq. (47.287) by setting w = 0:

$$EI\frac{d^4y}{dx^4} + P\frac{d^2y}{dx^2} = 0$$
 (47.294)

The general solution is

$$y = A \sin kx + B \cos kx + Cx + D \qquad (47.295)$$

The constants A, B, C, and D are determined by enforcing the four boundary conditions:

$$y_{x=0} = 0, \quad y''_{x=0} = \frac{M_A}{EI} \quad y_{x=L} = 0 \quad y''_{x=L} = \frac{-M_B}{EI}$$
 (47.296)

to give

$$A = \frac{M_{A} \cos kL + M_{B}}{EIk^{2} \sin kL}$$
(47.297a)

$$B = -\frac{M_A}{EIk^2}$$
(47.297b)

$$C = -\left(\frac{M_{A} + M_{B}}{EIk^{2}L}\right)$$
(47.297c)

$$D = \frac{M_A}{EIk^2}$$
(47.297d)

Substituting Eq. (47.297a to d) into the deflection function Eq. (47.295) and rearranging gives

$$y = \frac{1}{EIk} \left[\frac{\cos kL}{\sin kL} \sin kx - \cos kx - \frac{x}{L} + 1 \right] M_A + \frac{1}{EIk^2} \left[\frac{1}{\sin kL} \sin kx - \frac{x}{L} \right] M_B$$
(47.298)

The maximum moment can be obtained by first locating its position by setting dM/dx = 0 and substituting the result into M = -EIy'' to give

$$M_{max} = \frac{\sqrt{\left(M_{A}^{2} + 2M_{A}M_{B}\cos kL + M_{B}^{2}\right)}}{\sin kL}$$
(47.299)

Assuming that M_B is the larger of the two end moments, Eq. (47.299) can be expressed as

$$M_{max} = M_{B} \left[\frac{\sqrt{\left\{ \left(M_{A}/M_{B}\right)^{2} + 2\left(M_{A}/M_{B}\right)\cos kL + 1\right\}}}{\sin kL} \right]$$
(47.300)

Since M_B is the maximum first-order moment, the expression in brackets is therefore the theoretical moment amplification factor. In Eq. (47.300), the ratio (M_A/M_B) is positive if the member is bent in double (or reverse) curvature, and the ratio is negative if the member is bent in single curvature. A special case arises when the end moments are equal and opposite (i.e., $M_B = -M_A$). By setting $M_B = -M_A = M_0$ in Eq. (47.300), we have

$$M_{max} = M_0 \left[\frac{\sqrt{2(1 - \cos kL)}}{\sin kL} \right]$$
(47.301)

For this special case, the maximum moment always occurs at midspan.

Slope Deflection Equations

The slope deflection equations of a beam-column can be derived by considering the beam-column shown in Fig. 47.118. The deflection function for this beam-column can be obtained from Eq. (47.298) in terms of M_A and M_B as:

$$y = \frac{1}{EIk^2} \left[\frac{\cos kL}{\sin kL} \sin kx - \cos kx - \frac{x}{L} + 1 \right] M_A + \frac{1}{EIk^2} \left[\frac{1}{\sin kL} \sin kx - \frac{x}{L} \right] M_B$$
(47.302)

from which

$$y' = \frac{1}{EIk} \left[\frac{\cos kL}{\sin kL} \cos kx + \sin kx - \frac{1}{kL} \right] M_A + \frac{1}{EIk} \left[\frac{\cos kx}{\sin kL} - \frac{1}{kL} \right] M_B$$
(47.303)

The end rotations θ_A and θ_B can be obtained from Eq. (47.303) as

$$\theta_{A} = y'(x=0) = \frac{1}{EIk} \left[\frac{\cos kL}{\sin kL} - \frac{1}{kL} \right] M_{A} + \frac{1}{EIk} \left[\frac{1}{\sin kL} - \frac{1}{kL} \right] M_{B}$$

$$= \frac{L}{EI} \left[\frac{kL \cos kL - \sin kL}{(kL)^{2} \sin kL} \right] M_{A} + \frac{L}{EI} \left[\frac{kL - \sin kL}{(kL)^{2} \sin kL} \right] M_{B}$$
(47.304)

and

$$\theta_{\rm B} = \mathbf{y}'(\mathbf{x} = \mathbf{L}) = \frac{1}{\mathrm{EIk}} \left[\frac{1}{\sin kL} - \frac{1}{kL} \right] \mathbf{M}_{\rm A} + \frac{1}{\mathrm{EIk}} \left[\frac{\cos kL}{\sin kL} - \frac{1}{kL} \right] \mathbf{M}_{\rm B}$$

$$= \frac{\mathrm{L}}{\mathrm{EI}} \left[\frac{\mathrm{kL} - \sin \mathrm{kL}}{(\mathrm{kL})^2 \sin \mathrm{kL}} \right] \mathbf{M}_{\rm A} + \frac{\mathrm{L}}{\mathrm{EI}} \left[\frac{\mathrm{KL} \cos \mathrm{kL} - \sin \mathrm{kL}}{(\mathrm{kL})^2 \sin \mathrm{kL}} \right] \mathbf{M}_{\rm B}$$
(47.305)

The moment rotation relationship can be obtained from Eqs. (47.304) and (47.305) by arranging M_A and M_B in terms of θ_A and θ_B as:

$$M_{A} = \frac{EI}{L} \left(s_{ii} \theta_{A} + s_{ij} \theta_{B} \right)$$
(47.306)

$$M_{B} = \frac{EI}{L} \left(s_{ji} \theta_{A} + s_{jj} \theta_{B} \right)$$
(47.307)

where

$$s_{ii} = s_{ij} = \frac{kL\sin kL - (kL)^2 \cos kL}{2 - 2\cos kL - kL\sin kL}$$
(47.308)

$$s_{ij} = s_{ji} = \frac{(kL)^2 - kL\sin kL}{2 - 2\cos kL - kL\sin kL}$$
(47.309)

are referred to as the stability functions.

Equations (47.306) and (47.307) are the slope deflection equations for a beam-column that is not subjected to transverse loading and relative joint translation. It should be noted that when P approaches zero, $kL = \sqrt{(P / EI)}L$ approaches zero, and by using the L'Hospital's rule, it can be shown that $s_{ij} = 4$ and $s_{ij} = 2$. Values for s_{ii} and s_{ij} for various values of kL are plotted as shown in Fig. 47.119.

Equations (47.307) and (47.308) are valid if the following conditions are satisfied:

- 1. The beam is prismatic.
- 2. There is no relative joint displacement between the two ends of the member.
- 3. The member is continuous, i.e., there is no internal hinge or discontinuity in the member.
- 4. There is no in-span transverse loading on the member.
- 5. The axial force in the member is compressive.

If these conditions are not satisfied, some modifications to the slope deflection equations are necessary.



FIGURE 47.119 Plot of stability functions.



FIGURE 47.120 Beam-column subjects to end moments and side sway.

Member Subjected to Side Sway

If there is a relative joint translation, Δ , between the member ends, as shown in Fig. 47.120, the slope deflection equations are modified as

$$\begin{split} \mathbf{M}_{\mathrm{A}} &= \frac{\mathrm{EI}}{\mathrm{L}} \bigg[\mathbf{s}_{\mathrm{ii}} \bigg(\mathbf{\theta}_{\mathrm{A}} - \frac{\Delta}{\mathrm{L}} \bigg) + \mathbf{s}_{\mathrm{ij}} \bigg(\mathbf{\theta}_{\mathrm{B}} - \frac{\Delta}{\mathrm{L}} \bigg) \bigg] \\ &= \frac{\mathrm{EI}}{\mathrm{L}} \bigg[\mathbf{s}_{\mathrm{ii}} \,\mathbf{\theta}_{\mathrm{A}} + \mathbf{s}_{\mathrm{ij}} \,\mathbf{\theta}_{\mathrm{B}} - (\mathbf{s}_{\mathrm{ii}} + \mathbf{s}_{\mathrm{ij}}) \frac{\Delta}{\mathrm{L}} \bigg] \end{split} \tag{47.310} \\ \mathbf{M}_{\mathrm{B}} &= \frac{\mathrm{EI}}{\mathrm{L}} \bigg[\mathbf{s}_{\mathrm{ij}} \bigg(\mathbf{\theta}_{\mathrm{A}} - \frac{\Delta}{\mathrm{L}} \bigg) + \mathbf{s}_{\mathrm{ii}} \bigg(\mathbf{\theta}_{\mathrm{B}} - \frac{\Delta}{\mathrm{L}} \bigg) \bigg] \\ &= \frac{\mathrm{EI}}{\mathrm{L}} \bigg[\mathbf{s}_{\mathrm{ij}} \big(\mathbf{\theta}_{\mathrm{A}} + \mathbf{s}_{\mathrm{ii}} \,\mathbf{\theta}_{\mathrm{B}} - \big(\mathbf{s}_{\mathrm{ii}} + \mathbf{s}_{\mathrm{ij}} \big) \frac{\Delta}{\mathrm{L}} \bigg] \end{aligned} \tag{47.311}$$

Member with a Hinge at One End

If a hinge is present at the B end of the member, the end moment there is zero, i.e.,

$$M_{\rm B} = \frac{\rm EI}{\rm L} \left(s_{\rm ij} \theta_{\rm A} + s_{\rm ii} \theta_{\rm B} \right) = 0 \tag{47.312}$$



FIGURE 47.121 Beam column with end springs.

from which

$$\boldsymbol{\theta}_{\mathrm{B}} = -\frac{\mathbf{s}_{\mathrm{ij}}}{\mathbf{s}_{\mathrm{ii}}} \boldsymbol{\theta}_{\mathrm{A}} \tag{47.313}$$

Upon substituting Eq. (47.313) into Eq. (47.310), we have

$$\mathbf{M}_{\mathrm{A}} = \frac{\mathrm{EI}}{\mathrm{L}} \left(\mathbf{s}_{\mathrm{ii}} - \frac{\mathbf{s}_{\mathrm{ij}}^{2}}{\mathbf{s}_{\mathrm{ii}}} \right) \mathbf{\theta}_{\mathrm{A}}$$
(47.314)

If the member is hinged at the A end rather than at the B end, Eq. (47.314) is still valid, provided that the subscript A is changed to B.

Member with End Restraints

If the member ends are connected by two linear elastic springs, as in Fig. 47.121, with spring constants R_{kA} and R_{kB} at the A and B ends, respectively, the end rotations of the linear spring are M_A/R_{kA} and M_B/R_{kB} . If we denote the total end rotations at joints A and B by θ_A and θ_B , respectively, then the member end rotations, with respect to its chord, will be $\theta_A - M_A/R_{kA}$ and $\theta_B - M_B/R_{kB}$. As a result, the slope deflection equations are modified to

$$M_{A} = \frac{EI}{L} \left[s_{ii} \left(\theta_{A} - \frac{M_{A}}{R_{kA}} \right) + s_{ij} \left(\theta_{B} - \frac{M_{B}}{R_{kB}} \right) \right]$$
(47.315)

$$M_{B} = \frac{EI}{L} \left[s_{ij} \left(\theta_{A} - \frac{M_{A}}{R_{kA}} \right) + s_{jj} \left(\theta_{B} - \frac{M_{B}}{R_{kB}} \right) \right]$$
(47.316)

Solving Eqs. (47.315) and (47.316) simultaneously for M_A and M_B gives

$$M_{A} = \frac{EI}{LR^{*}} \left[\left(s_{ii} + \frac{EIs_{ii}^{2}}{LR_{kB}} - \frac{EIs_{ij}^{2}}{LR_{kB}} \right) \theta_{A} + s_{ij} \theta_{B} \right]$$
(47.317)

$$M_{B} = \frac{EI}{LR^{*}} \left[s_{ij} \theta_{A} + \left(s_{ii} + \frac{EIs_{ii}^{2}}{LR_{kA}} - \frac{EIs_{ij}^{2}}{LR_{kA}} \right) \theta_{B} \right]$$
(47.318)

where

$$\mathbf{R}^{*} = \left(1 + \frac{\mathrm{EIs}_{\mathrm{ii}}}{\mathrm{LR}_{\mathrm{kA}}}\right) \left(1 + \frac{\mathrm{EIs}_{\mathrm{ii}}}{\mathrm{LR}_{\mathrm{kB}}}\right) - \left(\frac{\mathrm{EI}}{\mathrm{L}}\right)^{2} \frac{\mathbf{s}_{\mathrm{ij}}^{2}}{\mathbf{R}_{\mathrm{kA}} \mathbf{R}_{\mathrm{kB}}}$$
(47.319)

In writing Eqs. (47.317) and (47.318), the equality $s_{jj} = s_{ii}$ has been used. Note that as R_{kA} and R_{kB} approach infinity, Eqs. (47.317) and (47.318) reduce to Eqs. (47.306) and (47.307), respectively.

Member with Transverse Loading

For members subjected to transverse loading, the slope deflection Eqs. (47.306) and (47.307) can be modified by adding an extra term for the fixed-end moment of the member.

$$\mathbf{M}_{A} = \frac{\mathrm{EI}}{\mathrm{L}} \left(\mathbf{s}_{ii} \,\boldsymbol{\theta}_{A} + \mathbf{s}_{ii} \,\boldsymbol{\theta}_{B} \right) + \mathbf{M}_{\mathrm{FA}} \tag{47.320}$$

$$M_{B} = \frac{EI}{L} \left(s_{ij} \theta_{A} + s_{jj} \theta_{B} \right) + M_{FB}$$
(47.321)

Table 47.4 gives the expressions for the fixed-end moments of five commonly encountered cases of transverse loading. See Chen and Lui (1987, 1991) for more details.

Member with Tensile Axial Force

For members subjected to tensile force, Eqs. (47.306) and (47.307) can be used, provided that the stability functions are redefined as

$$s_{ii} = s_{ij} = \frac{\left(kL\right)^2 \cosh kL - kL \sinh kL}{2 - 2\cosh kL + kL \sinh kL}$$
(47.322)

$$s_{ij} = s_{ji} = \frac{kL \sinh kL - (kL)^2}{2 - 2\cosh kL + kL \sinh kL}$$
(47.323)

Member Bent in Single Curvature with $\theta_{\rm B} = -\theta_{\rm A}$

For the member bent in a single curvature in which $\theta_B = -\theta_A$, the slope deflection equations reduce to

$$M_{A} = \frac{EI}{L} \left(s_{ii} - s_{ij} \right) \theta_{A}$$
(47.324)

$$M_{\rm B} = -M_{\rm A} \tag{47.325}$$

Member Bent in Double Curvature with $\theta_{\rm B} = \theta_{\rm A}$

For the member bent in a double curvature such that $\theta_B = \theta_A$, the slope deflection equations become

$$\mathbf{M}_{\mathrm{A}} = \frac{\mathrm{EI}}{\mathrm{L}} \left(\mathbf{s}_{\mathrm{ii}} - \mathbf{s}_{\mathrm{ij}} \right) \boldsymbol{\theta}_{\mathrm{A}} \tag{47.326}$$

$$M_{\rm B} = M_{\rm A} \tag{47.327}$$

Second-Order Elastic Analysis

There are two methods to incorporate second-order effects, the stability function approach and the geometric stiffness (or finite element) approach. The stability function approach is based on the governing differential equations of the problem, as described above, whereas the stiffness approach is based on an assumed cubic polynomial variation of the transverse displacement along the element length. Therefore, the stability function approach is more exact in terms of representing the member stability behavior. However, the geometric stiffness approach is easier to implement for matrix analysis.



TABLE 47.4 Beam-Column Fixed-End Moments (Chen and Lui, 1991)



FIGURE 47.122 Nodal displacements and forces of a beam-column element.

For either of these approaches, the linearized element stiffness equations may be expressed in either incremental or total force and displacement forms as

$$[K]{d} + {r_f} = {r}$$
(47.328)

where [K] is the element stiffness matrix, $\{d\} = \{d_1, d_2, ..., d_6\}^T$ is the element nodal displacement vector, $\{r_f\} = \{r_{f1}, r_{f2}, ..., r_{f6}\}^T$ is the element fixed-end force vector due to the presence of in-span loading, and $\{r\} = \{r_1, r_2, ..., r_6\}^T$ is the nodal force vector, as shown in Fig. 47.122. If stability function approach is employed, the stiffness matrix of a two-dimensional beam-column element may be written as

$$[K] = \frac{EI}{L} \begin{bmatrix} \frac{A}{I} & 0 & 0 & -\frac{A}{I} & 0 & 0 \\ & \frac{2(S_{ii} + S_{ij}) - (kL)^2}{L^2} & \frac{S_{ii} + S_{ij}}{L} & 0 & \frac{-2(S_{ii} + S_{ij}) + (kL)^2}{L^2} & \frac{S_{ii} + S_{ij}}{L} \\ & S_{ii} & 0 & \frac{-(S_{ii} + S_{ij})}{L} & S_{ij} \\ & & \frac{A}{I} & 0 & 0 \\ & & & \frac{2(S_{ii} + S_{ij}) - (kL)^2}{L^2} & \frac{-(S_{ii} + S_{ij})}{L} \\ & & & S_{ii} \end{bmatrix}$$
(47.329)

where S_{ii} and S_{ij} are the member stiffness coefficients obtained from the elastic beam-column stability functions (Chen and Lui, 1987). These coefficients may be expressed as

$$S_{ii} = \begin{cases} \frac{kL\sin(kL) - (kL)^{2}\cos(kL)}{2 - 2\cos(kL) - kL\sin(kL)} & \text{for } P < 0\\ \frac{(kL)^{2}\cosh(kL) - kL\sin(kL)}{2 - 2\cosh(kL) + kL\sinh(kL)} & \text{for } P > 0 \end{cases}$$
(47.330)

$$S_{ij} = \begin{cases} \frac{(kL)^{2} - kL\sin(kL)}{2 - 2\cos(kL) - \rho\sin(kL)} & \text{for } P < 0\\ \frac{kL\sinh(kL) - (kL)^{2}}{2 - 2\cosh(kL) + \rho\sinh(kL)} & \text{for } P > 0 \end{cases}$$
(47.331)

where $kL = L\sqrt{P/EI}$ and P is positive in compression and negative in tension.

The fixed-end force vector r_f is a 6 × 1 matrix that can be computed from the in-span loading in the beam-column. If curvature shortening is ignored, $r_{f1} = r_{f4} = 0$, $r_{f3} = M_{FA}$, and $r_{f6} = M_{FB}$. M_{FA} and M_{FB} can be obtained from Table 47.4 for different in-span loading conditions. r_{f2} and r_{f5} can be obtained from the equilibrium of forces.

If the axial force in the member is small, Eq. (47.329) can be simplified by ignoring the higher order terms of the power series expansion of the trigonometric functions. The resulting element stiffness matrix becomes:

$$[K] = \frac{EI}{L} \begin{bmatrix} \frac{A}{I} & 0 & 0 & -\frac{A}{I} & 0 & 0 \\ & \frac{12}{L^2} & \frac{6}{L} & 0 & \frac{-12}{L^2} & \frac{6}{L} \\ & & 4 & 0 & \frac{-6}{L} & 2 \\ & & & \frac{A}{I} & 0 & 0 \\ & & & & \frac{12}{L^2} & \frac{-6}{L} \\ & & & & \frac{12}{L^2} & \frac{-6}{L} \\ & & & & & \frac{2L}{I^2} & 0 & \frac{-1}{10} & \frac{-1}{30} \\ & & & & & \frac{6}{5L} & \frac{-1}{10} \\ & & & & & \frac{2L}{15} \end{bmatrix} (47.332)$$

The first term on the right is the first-order elastic stiffness matrix, and the second term is the geometric stiffness matrix, which accounts for the effect of axial force on the bending stiffness of the member. Detailed discussions on the limitation of the geometric stiffness approach versus the stability function approach are given in Liew et al. (2000).

Modifications to Account for Plastic Hinge Effects

There are two commonly used approaches for representing plastic hinge behavior in a second-order elastic-plastic hinge formulation (Chen et al., 1996). The most basic approach is to model the plastic hinge behavior as a "real" hinge for the purpose of calculating the element stiffness. The change in moment capacity due to the change in axial force can be accommodated directly in the numerical formulation. The change in moment is determined in the force recovery at each solution step such that, for continued plastic loading, the new force point is positioned at the strength surface at the current value of the axial force. A detailed description of these procedures is given by Chen and Lui (1991), Chen et al. (1996), and Lee and Basu (1989), among others.

Alternatively, the elastic-plastic hinge model may be formulated based on the "extending and contracting" plastic hinge model. The plastic hinge can rotate and extend or contract for plastic loading and axial force. The formulation can follow the force–space plasticity concept using the normality flow rule relative to the cross section surface strength (Chen and Han, 1988). Formal derivations of the beam-column element based on this approach have been presented by Porter and Powell (1971), Orbison et al. (1982), and Liew et al., (2000), among others.

Modification for End Connections

The moment rotation relationship of the beam-column with end connections at both ends can be expressed as (Eqs. (47.317) and (47.318)):

$$\mathbf{M}_{\mathrm{A}} = \frac{\mathrm{EI}}{\mathrm{L}} \left[\mathbf{s}_{\mathrm{ii}}^{*} \mathbf{\theta}_{\mathrm{A}} + \mathbf{s}_{\mathrm{ij}}^{*} \mathbf{\theta}_{\mathrm{B}} \right]$$
(47.333)

$$\mathbf{M}_{\mathrm{B}} = \frac{\mathrm{EI}}{\mathrm{L}} \left[\mathbf{s}_{ij}^{*} \boldsymbol{\theta}_{\mathrm{A}} + \mathbf{s}_{jj}^{*} \boldsymbol{\theta}_{\mathrm{B}} \right]$$
(47.334)



FIGURE 47.123 Nodal displacements and forces of a beam-column with end connections.

where

$$S_{ii}^{*} = \frac{S_{ii} + \frac{EIS_{ii}^{2}}{LR_{kB}} - \frac{EIS_{ij}^{2}}{LR_{kB}}}{\left[1 + \frac{EIS_{ii}}{LR_{kB}}\right] \left[1 + \frac{EIS_{ij}}{LR_{kB}}\right] - \left[\frac{EI}{L}\right]^{2} \frac{S_{ij}^{2}}{R_{kA}R_{kB}}}$$

$$S_{ii} + \frac{EIS_{ii}^{2}}{LR_{kB}} - \frac{EIS_{ij}^{2}}{R_{kA}R_{kB}}$$
(47.335)

$$S_{jj}^{*} = \frac{S_{ii} + \frac{EIS_{ii}}{LR_{kA}} - \frac{EIS_{ij}}{LR_{kA}}}{\left[1 + \frac{EIS_{ii}}{LR_{kA}}\right] \left[1 + \frac{EIS_{jj}}{LR_{kB}}\right] - \left[\frac{EI}{L}\right]^{2} \frac{S_{ij}^{2}}{R_{kA}R_{kB}}}$$
(47.336)

and

$$S_{ij}^{*} = \frac{S_{ij}}{\left[1 + \frac{EIS_{ii}}{LR_{kA}}\right] \left[1 + \frac{EIS_{jj}}{LR_{kB}}\right] - \left[\frac{EI}{L}\right]^{2} \frac{S_{ij}^{2}}{R_{kA}R_{kB}}}$$
(47.337)

The member stiffness relationship can be written in terms of six degrees of freedom — see the beamcolumn element shown in Fig. 47.123 — as

$$\begin{pmatrix} r_{1} \\ r_{2} \\ r_{3} \\ r_{4} \\ r_{5} \\ r_{6} \end{pmatrix} = \underbrace{EI}_{k} \begin{bmatrix} \frac{A}{I} & 0 & 0 & -\frac{A}{I} & 0 & 0 \\ & \frac{S_{ii}^{*} + 2S_{ij}^{*} + S_{jj}^{*} - (kL)^{2}}{L^{2}} & \frac{S_{ii}^{*} + S_{ij}^{*}}{L} & 0 & \frac{-(S_{ii}^{*} + 2S_{ij}^{*} + S_{jj}^{*}) + (kL)^{2}}{L^{2}} & \frac{S_{ij}^{*} + S_{jj}^{*}}{L} \\ & & \frac{A}{I} & 0 & 0 \\ & & & \frac{S_{ii}^{*} + 2S_{ij}^{*} + S_{jj}^{*} - (kL)^{2}}{L^{2}} & \frac{-(S_{iij}^{*} + S_{jj}^{*})}{L} \\ & & & \frac{S_{ii}^{*} + 2S_{ij}^{*} + S_{jj}^{*} - (kL)^{2}}{L^{2}} & \frac{-(S_{iij}^{*} + S_{jj}^{*})}{L} \\ & & & Sym. \end{bmatrix}$$

$$(47.338)$$

Second-Order Refined Plastic Hinge Analysis

The main limitation of the conventional elastic-plastic hinge approach is that it overpredicts the strength of columns that fail by inelastic flexural buckling. The key reason for this limitation is the modeling of

a member by a perfect elastic element between the plastic hinge locations. Furthermore, the elastic-plastic hinge model assumes that material behavior changes abruptly from the elastic state to the fully yielded state. The element under consideration exhibits a sudden stiffness reduction upon the formation of a plastic hinge. This approach, therefore, overestimates the stiffness of a member loaded into the inelastic range (Liew et al., 1993; White et al., 1991, 1993). This leads to further research and development of an alternative method called the *refined plastic hinge approach*. This approach is based on the following improvements to the elastic-plastic hinge model:

- 1. A column tangent modulus model E_t is used in place of the elastic modulus E to represent the distributed plasticity along the length of a member due to axial force effects. The member inelastic stiffness, represented by the member axial and bending rigidities E_tA and E_tI , is assumed to be the function of axial load only. In other words, E_tA and E_tI can be thought of as the properties of an effective core of the section, considering column action only. The tangent modulus captures the effect of early yielding in the cross-section due to residual stresses, which is believed to be the cause for the low strength of inelastic column buckling. The tangent modulus approach has been previously utilized by Orbison et al. (1982), Liew (1992), and White et al. (1993) to improve the accuracy of the elastic-plastic hinge approach for structures in which members are subjected to large axial forces.
- 2. Distributed plasticity effects associated with flexure are captured by gradually degrading the member stiffness at the plastic hinge locations as yielding progresses under an increasing load as the cross section strength is approached. Several models of this type have been proposed in recent literature based on extensions to the elastic-plastic hinge approach (Powell and Chen, 1986), as well as the tangent modulus inelastic hinge approach (Liew et al., 1993; White et al., 1993). The rationale of modeling stiffness degradation associated with both axial and flexural actions is that the tangent modulus model represents the column strength behavior in the limit of pure axial compression, and the plastic hinge stiffness degradation model represents the beam behavior in pure bending; thus the combined effects of these two approaches should also satisfy the cases in which the member is subjected to combined axial compression and bending.

It has been shown that with the above two improvements, the refined plastic hinge model can be used with sufficient accuracy to provide a quantitative assessment of a member's performance up to failure. Detailed descriptions of the method and discussion of the results generated by the method are given in White et al. (1993) and Chen et al. (1996). Significant work has been done to implement the refined plastic hinge methods for the design of three-dimensional real-size structures (Al-Bermani et al., 1995; Liew et al., 2000).

Second-Order Spread of Plasticity Analysis

Spread of plasticity analyses can be classified into two main types, namely three-dimensional shell element and two-dimensional beam-column approaches. In the three-dimensional spread of plasticity analysis, the structure is modeled using a large number of finite three-dimensional shell elements, and the elastic constitutive matrix, in the usual incremental stress–strain relations, is replaced by an elastic-plastic constitutive matrix once yielding is detected. This analysis approach typically requires numerical integration for the evaluation of the stiffness matrix. Based on a deformation theory of plasticity, the combined effects of normal and shear stresses may be accounted for. The three-dimensional spread-ofplasticity analysis is computational intensive and best suited for analyzing small-scale structures.

The second approach for plastic-zone analysis is based on use of the beam-column theory, in which the member is discretized into many beam-column segments, and the cross section of each segment is further subdivided into a number of fibers. Inelasticity is typically modeled by the consideration of normal stress only. When the computed stresses at the centroid of any fibers reach the uniaxial normal strength of the material, the fiber is considered yielded. Compatibility is treated by assuming that full continuity is retained throughout the volume of the structure in the same manner as for elastic range calculations. Most of the plastic-zone analysis methods developed are meant for planar (two-dimensional) analysis (Chen and Toma, 1994; White, 1985; Vogel, 1985). Three-dimensional plastic-zone techniques are also available involving various degrees of refinements (White, 1988; Wang, 1988).

A plastic-zone analysis, which includes the spread of plasticity, residual stresses, initial geometric imperfections, and any other significant second-order behavioral effects, is often considered to be an exact analysis method. Therefore, when this type of analysis is employed, the checking of member interaction equations is not required. However, in reality, some significant behavioral effects, such as joint and connection performances, tend to defy precise numerical and analytical modeling. In such cases, a simpler method of analysis that adequately captures the inelastic behavior would be sufficient for engineering application. Second-order plastic hinge-based analysis is still the preferred method for advanced analysis of large-scale steel frames.

Three-Dimensional Frame Element

The two-dimensional beam-column formulation can be extended to a three-dimensional space frame element by including additional terms due to shear force, bending moment, and torsion. The following stiffness equation for a space frame element has been derived by Yang and Kuo (1994) by referring to Fig. 47.124:

$$[k_{e}]{d} + [k_{g}]{d} = {{}^{2}f} - {{}^{1}f}$$
(47.339)



FIGURE 47.124 Three-dimensional frame element: (a) nodal degrees of freedom, (b) nodal forces.

where

$${\left\{ \mathbf{d} \right\}}^{\mathrm{T}} = {\left\{ \mathbf{d}_{1}, \mathbf{d}_{2}, \dots, \mathbf{d}_{12} \right\}}$$
 (47.340)

is the displacement vector, which consists of three translations and three rotations at each node, and

$${{}^{i}f}^{T} = {{}^{i}f_{1}, {}^{i}f_{2}, ..., {}^{i}f_{12}}$$
 $i = 1, 2$ (47.341)

is the force vector, which consists of the corresponding nodal forces at configuration i = 1 or i = 2.

The physical interpretation of Eq. (47.339) is as follows: by increasing the nodal forces acting on the element from $\{^{1}f\}$ to $\{^{2}f\}$, further deformations $\{d\}$ may occur with the element, resulting in the motion of the element from a configuration associated with the forces $\{^{1}f\}$ to the new configuration associated with $\{^{2}f\}$. During this process of deformation, the increments in the nodal forces, i.e., $\{^{2}f\} - \{^{1}f\}$, will be resisted not only by the elastic actions generated by the elastic stiffness matrix $[k_e]$ but also by the forces induced by the change in geometry, as represented by the geometric stiffness matrix $[k_e]$.

The only assumption with the incremental stiffness equation is that the strains occurring with each incremental step should be small, so that the approximations implied by the incremental constitutive law are not violated.

The elastic stiffness matrix $[K_e]$ for the space frame element, which has a 12 x 12 dimension, can be derived as

$$\begin{bmatrix} \mathbf{k} \end{bmatrix} = \begin{bmatrix} \begin{bmatrix} \mathbf{k}_1 \end{bmatrix} & \begin{bmatrix} \mathbf{k}_2 \end{bmatrix} \\ \begin{bmatrix} \mathbf{k}_2 \end{bmatrix}^{\mathrm{T}} & \begin{bmatrix} \mathbf{k}_3 \end{bmatrix} \end{bmatrix}$$
(47.342)

where the submatrices are

$$\begin{bmatrix} k_1 \end{bmatrix} = \begin{bmatrix} \frac{EA}{L} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{12EI_x}{L^3} & 0 & 0 & 0 & \frac{6EI_x}{L^2} \\ 0 & 0 & \frac{12EI_y}{L^3} & 0 & -\frac{6EI_y}{L^2} & 0 \\ 0 & 0 & 0 & \frac{GJ}{L} & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{4EI_x}{L} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{4EI_x}{L} \end{bmatrix}$$
(47.343)
$$\begin{bmatrix} -\frac{EA}{L} & 0 & 0 & 0 & 0 & \frac{4EI_x}{L} \\ 0 & 0 & 0 & 0 & 0 & \frac{4EI_x}{L} \end{bmatrix} \\ \begin{bmatrix} -\frac{EA}{L} & 0 & 0 & 0 & 0 & \frac{4EI_x}{L} \\ 0 & 0 & -\frac{12EI_x}{L^3} & 0 & 0 & \frac{6EI_y}{L^2} & 0 \\ 0 & 0 & 0 & -\frac{GJ}{L} & 0 & 0 \\ 0 & 0 & 0 & -\frac{GJ}{L} & 0 & 0 \\ 0 & 0 & \frac{6EI_y}{L^2} & 0 & \frac{2EI_y}{L} & 0 \\ 0 & 0 & -\frac{6EI_x}{L^2} & 0 & 0 & 0 & \frac{2EI_x}{L} \end{bmatrix}$$
(47.344)

$$[k_{3}] = \begin{bmatrix} \frac{EA}{L} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{12EI_{z}}{L^{3}} & 0 & 0 & 0 & -\frac{6EI_{z}}{L^{2}} \\ 0 & 0 & \frac{12EI_{y}}{L^{3}} & 0 & \frac{6EI_{y}}{L^{2}} & 0 \\ 0 & 0 & 0 & \frac{GJ}{L} & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{4EI_{y}}{L} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{4EI_{z}}{L} \end{bmatrix}$$
(47.345)

where I_x , I_y , and I_z = the moments of inertia about the x, y, and z axes

- L = the member length
- E = the modulus of elasticity
- A = the cross-sectional area
- G = the shear modulus
- J = the torsional stiffness

The geometric stiffness matrix for a three-dimensional space frame element can be given as

$$\left[k_{g}\right] = \begin{bmatrix} a & 0 & 0 & 0 & -d & -e & -a & 0 & 0 & 0 & -n & -o \\ b & 0 & d & g & k & 0 & -b & 0 & n & -g & k \\ c & e & h & g & 0 & 0 & -c & o & -h & -g \\ f & i & 1 & 0 & -d & -e & -f & -i & -1 \\ j & 0 & d & -g & h & -i & p & -q \\ m & e & -k & -g & -l & q & r \\ a & 0 & 0 & 0 & n & o \\ b & 0 & -n & g & -k \\ c & -o & h & g \\ f & i & l \\ sym. & & & & j & o \\ m \end{bmatrix}$$
(47.346)

where $a = -f_6 + f_{12}/L^2$; $b = 6f_7/5L$; $c = -f_5 + f_{11}/L^2$; $d = f_5/L$; $e = f_6/L$; $f = f_7/AL$; $g = f_{10}/L$; $h = -f_7/10$; $i = f_6 + f_{12}/6$; $j = 2f_7L/15$; $k = -f_5 + f_{11}/6$; $l = f_{11}/L$; $m = f_{12}/L$; $n = -f_7L/30$; $o = -f_{10}/2$.

Further details can be obtained from Yang and Kuo (1994).

Buckling of Thin Plates

Rectangular Plates

The main difference between columns and plates is that quantities such as deflections and bending moments, which are functions of a single independent variable in columns, become functions of two independent variables in plates. Consequently, the behavior of plates is described by partial differential equations, whereas ordinary differential equations suffice for describing the behavior of columns. A main difference between column and plate buckling is that column buckling terminates the ability of the member to resist the axial load in columns; this is not true for plates. Upon reaching the critical load, the plate continues to resist the increasing axial force, and it does not fail until a load considerably in



FIGURE 47.125 Plate subjected to in-plane forces.

excess of the elastic buckling load is reached. The critical load of a plate is, therefore, not its failure load. Instead, one must determine the load-carrying capacity of a plate by considering its postbuckling strength.

To determine the critical in-plane loading of a plate, a governing equation in terms of biaxial compressive forces N_x and N_y and constant shear force N_{xy} , as shown in Fig. 47.125, can be derived as

$$D\left(\frac{\delta^4 w}{\delta x^4} + 2\frac{\delta^4 w}{\delta x^2 \delta y^2} + \frac{\delta^4 w}{\delta y^4}\right) + N_x \frac{\delta^2 w}{\delta x^2} + N_y \frac{\delta^2 w}{\delta y^2} + 2N_{xy} \frac{\delta^2 w}{\delta x \delta y} = 0$$
(47.347)

The critical load for uniaxial compression can be determined from the differential equation

$$D\left(\frac{\delta^4 w}{\delta x^4} + 2\frac{\delta^4 w}{\delta x^2 \delta y^2} + \frac{\delta^4 w}{\delta y^4}\right) + N_x \frac{\delta^2 w}{\delta x^2} = 0$$
(47.348)

which is obtained by setting $N_x = N_{xy} = 0$ in Eq. (47.347).

For example, in the case of a simply supported plate Eq. (47.348) can be solved to give

$$N_{x} = \frac{\pi^{2} a^{2} D}{m^{2}} \left(\frac{m^{2}}{a^{2}} + \frac{n^{2}}{b^{2}}\right)^{2}$$
(47.349)

The critical value of N_x (i.e., the smallest value) can be obtained by taking n equal to 1. The physical meaning of this is that a plate buckles in such a way that there can be several half-waves in the direction of compression, but only one half-wave in the perpendicular direction. Thus, the expression for the critical value of the compressive force becomes

$$\left(N_{x}\right)_{cr} = \frac{\pi^{2}D}{a^{2}} \left(m + \frac{1}{m} \frac{a^{2}}{b^{2}}\right)^{2}$$
(47.350)

The first factor in this expression represents the Euler load for a strip of unit width and of length a. The second factor indicates in what proportion the stability of the continuous plate is greater than the stability of an isolated strip. The magnitude of this factor depends on the magnitude of the ratio a/b and also on the number m, which is the number of half-waves into which the plate buckles. If a is smaller than b, the second term in the parentheses of Eq. (47.350) is always smaller than the first, and the minimum value of the expression is obtained by taking m = 1, i.e., by assuming that the plate buckles in one half-wave. The critical value of N_x can be expressed as

$$N_{cr} = \frac{k\pi^2 D}{b^2}$$
(47.351)



FIGURE 47.126 Buckling stress coefficients for unaxially compressed plate.

The factor k depends on the aspect ratio a/b of the plate and m. The variation of k with a/b for different values of m can be plotted as shown in Fig. 47.126. The critical value of N_x is the smallest value obtained for m = 1, and the corresponding value of k is 4.0. This formula is analogous to Euler's formula for the buckling of a column.

In the case where the normal forces N_x and N_y and the shearing forces N_{xy} are acting on the boundary of the plate, the same general method can be used. The critical stress for the case of a uniaxially compressed simply supported plate can be written as

$$\sigma_{\rm cr} = 4 \frac{\pi^2 E}{12(1-v^2)} \left(\frac{h}{b}\right)^2 \tag{47.352}$$

The critical stress values for different loading and support conditions can be expressed in the form

$$f_{cr} = k \frac{\pi^2 E}{12(1 - \nu^2)} \left(\frac{h}{b}\right)^2$$
(47.353)

Values of k for plates with several different boundary and loading conditions are given in Fig. 47.127.

Circular Plates

The critical value of the compressive forces N_r uniformly distributed around the edge of a circular plate of radius r_o , clamped along the edge (Fig. 47.128), can be determined by

$$r^{2}\frac{d^{2}\phi}{dr^{2}} + r\frac{d\phi}{dr} - \phi = -\frac{Qr^{2}}{D}$$
(47.354)

in which ϕ is the angle between the axis of revolution of the plate surface and any normal to the plate, r is the distance of any point measured from the center of the plate, and Q is the shearing force per unit of length. When there are no lateral forces acting on the plate, the solution of Eq. (47.5.60) involves a Bessel function of the first order of the first and second kind, and the resulting critical value of N_r is obtained as

$$\left(N_{\rm r}\right)_{\rm cr} = \frac{14.68D}{r_0^2} \tag{47.355}$$

The critical value of N_r for the plate when the edge is simply supported can be obtained in the same way as

$$\left(N_{\rm r}\right)_{\rm cr} = \frac{4.20D}{r_0^2} \tag{47.356}$$



FIGURE 47.127 Values of K for plate with different boundary and loading conditions.



FIGURE 47.128 Circular plate under compressive loading.

Buckling of Shells

If a circular cylindrical shell is uniformly compressed in the axial direction, buckling symmetrical with respect to the axis of the cylinder (Fig. 47.129) may occur at a certain value of the compressive load. The critical value of the compressive force N_{cr} per unit length of the edge of the shell can be obtained by solving the differential equation



FIGURE 47.129 Buckling of a cylindrical shell.

$$D\frac{d^4w}{dx^4} + N\frac{d^2w}{dx^2} + Eh\frac{w}{a^2} = 0$$
(47.357)

in which a is the radius of the cylinder and h is the wall thickness.

Alternatively, the critical force per unit length may also be obtained by using the energy method. For a cylinder of length L, simply supported at both ends, one obtains

$$N_{cr} = D \left(\frac{m^2 \pi^2}{L^2} + \frac{EhL^2}{Da^2 m^2 \pi^2} \right)$$
(47.358)

For each value of m there is a unique buckling mode shape and a unique buckling load. The lowest value is of greatest interest and is thus found by setting the derivative of N_{cr} with respect to L equal to zero for m = 1. With Poisson's ratio equal to 0.3, the buckling load is obtained as

$$N_{cr} = 0.605 \frac{Eh^2}{a}$$
(47.359)

It is possible for a cylindrical shell to be subjected to uniform external pressure or to the combined action of axial and uniform lateral pressure.

47.13 Dynamic Analysis

Equation of Motion

The essential physical properties of a linearly elastic structural system subjected to external dynamic loading are its mass, stiffness properties, and energy absorption capability or damping. The principle of dynamic analysis may be illustrated by considering a simple single-story structure, as shown in Fig. 47.130. The structure is subjected to a time-varying force f(t). k is the spring constant that relates the lateral story deflection x to the story shear force, and the dash pot relates the damping force to the velocity by a damping coefficient c. If the mass, m, is assumed to concentrate at the beam, the structure becomes a single-degree-of-freedom (SDOF) system. The equation of motion of the system may be written as

$$m\ddot{\mathbf{x}} + c\dot{\mathbf{x}} + k\mathbf{x} = \mathbf{f}(\mathbf{t}) \tag{47.360}$$

Various solutions to Eq. (47.360) can give insight into the behavior of the structure under dynamic situations.

Free Vibration

In this case the system is set to motion and allowed to vibrate in the absence of applied force f(t). Letting f(t) = 0, Eq. (47.360) becomes



FIGURE 47.130 (a) One DOF structure. (b) Forces applied to structures.

$$m\ddot{x} + c\dot{x} + kx = 0$$
 (47.361)

Dividing Eq. (47.361) by the mass, m, we have

$$\ddot{\mathbf{x}} + 2\boldsymbol{\xi}\boldsymbol{\omega}\dot{\mathbf{x}} + \boldsymbol{\omega}^2 \mathbf{x} = \mathbf{0} \tag{47.362}$$

where

$$2\xi\omega = \frac{c}{m} \quad \text{and} \quad \omega^2 = \frac{k}{m} \tag{47.363}$$

The solution to Eq. (47.362) depends on whether the vibration is damped or undamped.

Example 47.16: Undamped Free Vibration

In this case, c = 0, and the solution to the equation of motion may be written as

$$\mathbf{x} = \mathbf{A}\sin\omega \mathbf{t} + \mathbf{B}\cos\omega \mathbf{t} \tag{47.364}$$

where $\omega = \sqrt{k/m}$ is the circular frequency. A and B are constants that can be determined by the initial boundary conditions. In the absence of external forces and damping, the system will vibrate indefinitely in a repeated cycle of vibration with an amplitude of

$$X = \sqrt{A^2 + B^2}$$
(47.365)

and a natural frequency of

$$f = \frac{\omega}{2\pi}$$
(47.366)

The corresponding natural period is

$$T = \frac{2\pi}{\omega} = \frac{1}{f}$$
(47.367)

The undamped free vibration motion, as described by Eq. (47.364), is shown in Fig. 47.131.



FIGURE 47.131 Response of undamped free vibration.

Example 47.17: Damped Free Vibration

If the system is not subjected to applied force and damping is presented, the corresponding solution becomes

$$\mathbf{x} = \mathbf{A} \exp(\lambda_1 \mathbf{t}) + \mathbf{B} \exp(\lambda_2 \mathbf{t}) \tag{47.368}$$

where

$$\lambda_1 = \omega \left[-\xi + \sqrt{\xi^2 - 1} \right] \tag{47.369}$$

and

$$\lambda_2 = \omega \left[-\xi - \sqrt{\xi^2 - 1} \right] \tag{47.370}$$

The solution of Eq. (47.368) changes its form with the value of ξ , defined as

$$\xi = \frac{c}{2\sqrt{mk}} \tag{47.371}$$

If $\xi^2 < 1$, the equation of motion becomes

$$\mathbf{x} = \exp(-\xi\omega t) \left(A\cos\omega_{d} t + B\sin\omega_{d} t \right)$$
(47.372)

where ξ_d is the damped angular frequency defined as

$$\boldsymbol{\omega}_{\mathrm{d}} = \sqrt{\left(1 - \boldsymbol{\xi}^2\right)\boldsymbol{\omega}} \tag{47.373}$$

For most building structures ξ is very small (about 0.01), and therefore $\omega_d \approx \omega$. The system oscillates about the neutral position as the amplitude decays with time t. Figure 47.132 illustrates an example of such motion. The rate of decay is governed by the amount of damping present.

If the damping is large, then oscillation will be prevented. This happens when $\xi^2 > 1$; the behavior is referred to as *overdamped*. The motion of such behavior is shown in Fig. 47.133.

Damping with $\xi^2 = 1$ is called critical damping. This is the case where minimum damping is required to prevent oscillation, and the critical damping coefficient is given as

$$c_{\rm cr} = 2\sqrt{km} \tag{47.374}$$

where k and m are the stiffness and mass of the system, respectively.

The degree of damping in the structure is often expressed as a proportion of the critical damping value. Referring to Eqs. (47.371) and (47.375), we have



FIGURE 47.132 Response of damped free vibration.



FIGURE 47.133 Response of free vibration with critical damping.

$$\xi = \frac{c}{c_{\rm cr}} \tag{47.375}$$

 ξ is called the critical damping ratio.

Forced Vibration

If a structure is subjected to a sinusoidal motion such as a ground acceleration of $\ddot{x} = F \sin \omega_f t$, it will oscillate, and after some time the motion of the structure will reach a steady state. For example, the equation of motion due to the ground acceleration (from Eq. (47.362)) is

$$\ddot{\mathbf{x}} + 2\boldsymbol{\xi}\boldsymbol{\omega}\dot{\mathbf{x}} + \boldsymbol{\omega}^2 \mathbf{x} = -F\sin\omega_f \mathbf{t} \tag{47.376}$$

The solution to the above equation consists of two parts: the complimentary solution given by Eq. (47.364) and the particular solution. If the system is damped, oscillation corresponding to the complementary solution will decay with time. After some time the motion will reach a steady state, and the system will vibrate at a constant amplitude and frequency. This motion, which is called force vibration, is described by the particular solution expressed as

$$\mathbf{x} = \mathbf{C}_1 \sin \omega_f \mathbf{t} + \mathbf{C}_2 \cos \omega_f \mathbf{t} \tag{47.377}$$

It can be observed that the steady force vibration occurs at the frequency of the excited force, ω_f , not at the natural frequency of the structure, ω .

Substituting Eq. (47.377) into (47.376), the displacement amplitude can be shown to be

$$X = -\frac{F}{\omega^{2}} \frac{1}{\sqrt{\left[\left\{1 - \left(\frac{\omega_{f}}{\omega}\right)^{2}\right\}^{2} + \left(\frac{2\xi\omega_{f}}{\omega}\right)^{2}\right]}}$$
(47.378)



FIGURE 47.134 Vibration of dynamic amplification factor with frequency ratio.

The term $-F/\omega^2$ is the static displacement caused by the force due to the inertia force. The ratio of the response amplitude relative to the static displacement $-F/\omega^2$ is called the dynamic displacement amplification factor, D, given as

$$D = \frac{1}{\sqrt{\left[\left\{1 - \left(\frac{\omega_{f}}{\omega}\right)^{2}\right\}^{2} + \left(\frac{2\xi\omega_{f}}{\omega}\right)^{2}\right]}}$$
(47.379)

The variation of the magnification factor with the frequency ratio ω_t/ω and damping ratio ξ is shown in Fig. 47.134.

When the dynamic force is applied at a frequency much lower than the natural frequency of the system $(\omega_f/\omega \ll 1)$, the response is quasistatic. The response is proportional to the stiffness of the structure, and the displacement amplitude is close to the static deflection.

When the force is applied at a frequency much higher than the natural frequency ($\omega_f/\omega \ge 1$), the response is proportional to the mass of the structure. The displacement amplitude is less than the static deflection (D < 1).

When the force is applied at a frequency close to the natural frequency, the displacement amplitude increases significantly. The condition at which $\omega_r/\omega = 1$ is known as resonance.

Similarly, the ratio of the acceleration response relative to the ground acceleration may be expressed as

$$D_{a} = \left| \frac{\ddot{x} + \ddot{x}_{g}}{\ddot{x}_{g}} \right| = \sqrt{\frac{1 + \left(\frac{2\xi\omega_{f}}{\omega}\right)^{2}}{\left[\left\{ 1 - \left(\frac{\omega_{f}}{\omega}\right)^{2} \right\}^{2} + \left(\frac{2\xi\omega_{f}}{\omega}\right)^{2} \right]}}$$
(47.380)

 D_{a} is called the dynamic acceleration magnification factor.

Response to Suddenly Applied Load

Consider the spring–mass damper system of which a load P_o is applied suddenly. The differential equation is given by

$$M\ddot{x} + c\dot{x} + kx = P_{o} \tag{47.381}$$

If the system is started at rest, the equation of motion is

$$x = \frac{P_o}{k} \left[1 - \exp(-\xi \omega t) \left\{ \cos \omega_d t + \frac{\xi \omega}{\omega_d} \sin \omega_d t \right\} \right]$$
(47.382)

If the system is undamped, then $\xi = 0$ and $\omega_d = \omega$; we have

$$\mathbf{x} = \frac{\mathbf{P}_{o}}{k} \left[1 - \cos \omega_{d} \mathbf{t} \right] \tag{47.383}$$

The maximum displacement is $2(P_o/k)$, corresponding to $\cos \omega_d t = -1$. Since P_o/k is the maximum static displacement, the dynamic amplification factor is 2. The presence of damping would naturally reduce the dynamic amplification factor and the force in the system.

Response to Time-Varying Loads

Some forces and ground motions that are encountered in practice are rather complex in nature. In general, numerical analysis is required to predict the response of such effects, and the finite element method is one of the most common techniques to be employed in solving such problems.

The evaluation of responses due to time-varying loads can be carried out using the piecewise exact method. In using this method, the loading history is divided into small time intervals. Between these points, it is assumed that the slope of the load curve remains constant. The entire load history is represented by a piecewise linear curve, and the error of this approach can be minimized by reducing the length of the time steps. A description of this procedure is given in Clough and Penzien (1993).

Other techniques employed include Fourier analysis of the forcing function, followed by solution for Fourier components in the frequency domain. For random forces, the random vibration theory and spectrum analysis may be used (Dowrick, 1988; Warburton, 1976).

Multiple Degree Systems

In multiple degree systems, an independent differential equation of motion can be written for each degree of freedom. The nodal equations of a multiple degree system consisting of n degrees of freedom may be written as

$$[m]{\ddot{x}} + [c]{\dot{x}} + [k]{x} = {F(t)}$$
(47.384)

where [m] = a symmetrical n x n matrix of mass

[c] = a symmetrical n x n matrix of damping coefficient

 ${F(t)}$ = the force vector, which is zero in the case of free vibration

Consider a system under free vibration without damping; the general solution of Eq. (47.384) is assumed in the form of

$$\begin{cases} x_1 \\ x_2 \\ \vdots \\ x_n \end{cases} = \begin{bmatrix} \cos(\omega t - \phi) & 0 & 0 & 0 \\ 0 & \cos(\omega t - \phi) & 0 & 0 \\ \vdots & \vdots & \vdots & \vdots \\ 0 & 0 & 0 & \cos(\omega t - \phi) \end{bmatrix} \begin{bmatrix} C_1 \\ C_2 \\ \vdots \\ C_n \end{bmatrix}$$
(47.385)

where angular frequency ω and phase angle ϕ are common to all x's. In this assumed solution, ϕ and C₁, C₂, ... C_n are the constants to be determined from the initial boundary conditions of the motion, and ω is a characteristic value (eigenvalue) of the system.

Substituting Eq. (47.385) into Eq. (47.384) yields

$$\begin{bmatrix} k_{11} - m_{11}\omega^{2} & k_{12} - m_{12}\omega^{2} & \dots & k_{1n} - m_{1n}\omega^{2} \\ k_{21} - m_{21}\omega^{2} & k_{22} - m_{22}\omega^{2} & \dots & k_{2n} - m_{2n}\omega^{2} \\ \vdots & \vdots & \vdots & \vdots \\ k_{n1} - m_{n1}\omega^{2} & k_{n2} - m_{n2}\omega^{2} & \dots & k_{nn} - m_{nn}\omega^{2} \end{bmatrix} \begin{bmatrix} C_{1} \\ C_{2} \\ \vdots \\ C_{n} \end{bmatrix} \cos(\omega t - \phi) = \begin{cases} 0 \\ 0 \\ \vdots \\ 0 \end{cases}$$
(47.386)

or

$$[[k] - \omega^{2}[m]] \{C\} = \{0\}$$
(47.387)

where [k] and [m] are the $n \times n$ matrices, ω^2 and $\cos(\omega t - \phi)$ are scalars, and {C} is the amplitude vector. For nontrivial solution, $\cos(\omega t - \phi) \neq 0$; thus solution to Eq. (47.387) requires the determinant of [[k] – $\omega^2[m]$] = 0. The expansion of the determinant yields a polynomial of n degree as a function of ω^2 , the n roots of which are the eigenvalues $\omega_1, \omega_2, ..., \omega_n$.

If the eigenvalue ω for a normal mode is substituted in Eq. (47.387), the amplitude vector {C} for that mode can be obtained. {C₁}, {C₂}, {C₃}, ... {C_n} are therefore called eigenvectors, the absolute values that must be determined through initial boundary conditions. The resulting motion is a sum of n harmonic motions, each governed by the respective natural frequency ω , written as

$$\{x\} = \sum_{i=1}^{n} \{C_i\} \cos(\omega_i t - \phi_i)$$
(47.388)

Distributed Mass Systems

Although many structures may be approximated by lumped mass systems, in practice all structures are distributed mass systems consisting of an infinite number of particles. Consequently, if the motion is repetitive, the structure has an infinite number of natural frequency and mode shapes. The analysis of a distributed-parameter system is entirely equivalent to that of a discrete system once the mode shapes and frequencies have been determined, because in both cases the amplitudes of the modal response components are used as generalized coordinates in defining the response of the structure.

In principle an infinite number of these coordinates are available for a distributed-parameter system, but in practice only a few modes, usually those of lower frequencies, will provide significant contribute to the overall response. Thus the problem of a distributed-parameter system can be converted to a discrete system form in which only a limited number of modal coordinates is used to describe the response.

Flexural Vibration of Beams

The motion of the distributed mass system is best illustrated by a classical example of a uniform beam with of span length L, a flexural rigidity EI, and a self-weight of m per unit length, as shown in Fig. 47.135a. The beam is free to vibrate under its self-weight. From Fig. 47.135b, dynamic equilibrium of a small beam segment of length dx requires:

$$\frac{\partial V}{\partial x}dx = mdx\frac{\partial^2 y}{\partial t^2}$$
(47.389)

in which

$$\frac{\partial^2 y}{\partial x^2} = \frac{M}{EI}$$
(47.390)



FIGURE 47.135 (a) Beam in flexural vibration. (b) Equilibrium of beam segment in vibration.

and

$$V = -\frac{\partial M}{\partial x}, \qquad \frac{\partial V}{\partial x} = -\frac{\partial^2 M}{\partial x^2}$$
(47.391)

Substituting these equations into Eq. (47.389) leads to the equation of motion of the flexural beam:

$$\frac{\partial^4 y}{\partial x^4} + \frac{m}{EI} \frac{\partial^2 y}{\partial t^2} = 0$$
(47.392)

Equation (47.392) can be solved for beams with given sets of boundary conditions. The solution consists of a family of vibration modes with corresponding natural frequencies. Standard results are available in Table 47.5 to compute the natural frequencies of uniform flexural beams of different supporting conditions. Methods are also available for dynamic analysis of continuous beams (Clough and Penzien, 1993).

Shear Vibration of Beams

Beams can deform by flexure or shear. Flexural deformation normally dominates the deformation of slender beams. Shear deformation is important for short beams or in higher modes of slender beams. Table 47.6 gives the natural frequencies of uniform beams in shear, neglecting flexural deformation. The natural frequencies of these beams are inversely proportional to the beam length L rather than L², and the frequencies increase linearly with the mode number.

Combined Shear and Flexure

The transverse deformation of real beams is the sum of flexure and shear deformations. In general, numerical solutions are required to incorporate both the shear and flexural deformation in the prediction of the natural frequency of beams. For beams with comparable shear and flexural deformations, the following simplified formula may be used to estimate the beam's frequency:

$$\frac{1}{f^2} = \frac{1}{f_f^2} + \frac{1}{f_s^2}$$
(47.393)

where f is the fundamental frequency of the beam and f_f and f_s are the fundamental frequencies predicted by the flexure and shear beam theories, respectively (Rutenberg, 1975).

$f_n = \frac{K_n}{2\pi} \sqrt{\frac{EI}{mL^4}} HZ$ $n = 1, 2, 3$	L = Length (m) EI = Flexural rigidity (Nm ²) M = Mass per unit length (kg/m)				
Boundary conditions	K _n ; n = 1,2,3	Mode shape $y_n\left(\frac{x}{L}\right)$	A _n ; n = 1,2,3		
Pinned - Pinned					
$\begin{array}{c} & & & \\ & & & \\ & & & \\$	(nπ) ²	sin <u>nπx</u> L			
Fixed - Fixed	$22.37 61.67 120.90 199.86 298.55 (2n + 1) \frac{\pi^2}{4};n > 5$	$\cosh \frac{\sqrt{K_n x}}{L} - \cos \frac{\sqrt{K_n x}}{L}$ $- A_n \left(\sin h \frac{\sqrt{K_n x}}{L} - \sin \frac{\sqrt{K_n x}}{L} \right)$	0.98250 1.00078 0.99997 1.00000 0.99999 1.0; n > 5		
Fixed - Pinned	$\begin{array}{c} 15.42\\ 49.96\\ 104.25\\ 178.27\\ 272.03\\ (4n+1)^2 \frac{\pi^2}{4};\\ n>5 \end{array}$	$\cosh \frac{\sqrt{K_n x}}{L} - \cos \frac{\sqrt{K_n x}}{L}$ $- A_n \left(\sin h \frac{\sqrt{K_n x}}{L} - \sin \frac{\sqrt{K_n x}}{L} \right)$	1.00078 1.00000 1.0; n > 3		
Cantilever	$3.5222.0361.69120.90199.86(2n-1)^2 \frac{\pi^2}{4};n > 5$	$\cosh \frac{\sqrt{K_n} x}{L} - \cos \frac{\sqrt{K_n} x}{L}$ $- A_n \left(\sin h \frac{\sqrt{K_n} x}{L} - \sin \frac{\sqrt{K_n} x}{L} \right)$	0.73410 1.01847 0.99922 1.00003 1.0; n > 4		

Natural Frequency of Multistory Building Frames

Tall building frames often deform more in the shear mode than in flexure. The fundamental frequencies of many multistory building frameworks can be approximated by (Housner and Brody, 1963; Rinne, 1952)

$f_{n} = \frac{K_{n}}{2\pi} \sqrt{\frac{KG}{\rho L^{2}}} HZ$	L = Length K = Shear coefficient (Cowper, 1966) G = Shear modulus = E/[2(1 + v)] ρ = Mass density				
Boundary condition	K _n ; n = 1,2,3	Mode shape $y_n K_L^X O$			
Fixed - Free	nπ; n = 1,2,3	cos <u>nπx;</u> n = 1,2,3			
Fixed - Fixed	nπ; n = 1,2,3	sin <u>nπx;</u> n = 1,2,3			

 TABLE 47.6
 Frequencies and Mode Shapes of Beams in Shear Vibration

$$f = \alpha \frac{\sqrt{B}}{H}$$
(47.394)

where α = approximately equal to 11 \sqrt{m} /sec

B = the building width in the direction of vibration

H = the building height

This empirical formula suggests that a shear beam model with f inversely proportional to H is more appropriate than a flexural beam for predicting natural frequencies of buildings.

Portal Frames

A portal frame consists of a cap beam rigidly connected to two vertical columns. The natural frequencies of portal frames vibrating in the fundamental asymmetric and symmetric modes are shown in Tables 47.7

First asymmetric in-plane mode											
First asymmetric in-plane mode $ \begin{array}{c} $						f = E = I = m =	$\frac{\lambda^2}{2\pi L_1^2} \left(\frac{E_1}{n}\right)$ Modulus Area mo Mass pe	II ₁) ^{1/2} HZ of elasti ment of i	city inertia gth		
						λ να	alue				
$\frac{m_1}{m_2}$	$\frac{E_1I_1}{E_2I_2}$		Pir	ned bas	es		Clamped bases				
	1112 $L_{2}12$ L_{1}/L_{2}			3.0	6.0	0.25	0.75	L ₁ /L ₂	3.0	6.0	
0.25	0.25	0.6964	0.9520	1.1124	1.2583	1.3759	0.9953	1.3617	1.6003	1.8270	2.0193
	0.75	0.6108	0.8961	1.0764	1.2375	1.3649	0.9030	1.2948	1.5544	1.7999	2.0051
	1.5	0.5414	0.8355	1.0315	1.2093	1.3491	0.8448	1.2323	1.5023	1.7649	1.9853
	3.0	0.4695	0.7562	0.9635	1.1610	1.3201	0.7968	1.1648	1.4329	1.7096	1.9504
	6.0	0.4014	0.6663	0.8737	1.0870	1.2702	0.7547	1.1056	1.3573	1.6350	1.8946
0.75	0.25	0.8947	1.1740	1.3168	1.4210	1.4882	1.2873	1.7014	1.9262	2.0994	2.2156
	0.75	0.7867	1.1088	1.2776	1.3998	1.4773	1.1715	1.6242	1.8779	2.0733	2.2026
	1.5	0.6983	1.0368	1.2281	1.3707	1.4617	1.0979	1.5507	1.8218	2.0390	2.1843
	3.0	0.6061	0.9413	1.1516	1.3203	1.4327	1.0373	1.4698	1.7454	1.9838	2.1516
	6.0	0.5186	0.8314	1.0485	1.2414	1.3822	0.9851	1.3981	1.6601	1.9072	2.0983
1.5	0.25	1.0300	1.2964	1.4103	1.4826	1.5243	1.4941	1.9006	2.0860	2.2090	2.2819
	0.75	0.9085	1.2280	1.3707	1.4616	1.5136	1.3652	1.8214	2.0390	2.1842	2.2695
	1.5	0.8079	1.1514	1.3203	1.4326	1.4982	1.2823	1.7444	1.9837	2.1515	2.2521
	3.0	0.7021	1.0482	1.2414	1.3821	1.4694	1.2141	1.6583	1.9070	2.0983	2.2206
	6.0	0.6011	0.9279	1.1335	1.3024	1.4191	1.1570	1.5808	1.8198	2.0234	2.1693
3.0	0.25	1.1597	1.3898	1.4719	1.5189	1.5442	1.7022	2.0612	2.1963	2.2756	2.3190
	0.75	1.0275	1.3202	1.4326	1.4981	1.5336	1.5649	1.9834	2.1515	2.2520	2.3070
	1.5	0.9161	1.2412	1.3821	1.4694	1.5182	1.4752	1.9063	2.0982	2.2206	2.2899
	3.0	0.7977	1.1333	1.3024	1.4191	1.4896	1.4015	1.8185	2.0233	2.1693	2.2595
	6.0	0.6838	1.0058	1.1921	1.3391	1.4395	1.3425	1.7382	1.9366	2.0964	2.2094
6.0	0.25	1.2691	1.4516	1.5083	1.5388	1.5545	1.8889	2.1727	2.2635	2.3228	2.3385
	0.75	1.1304	1.3821	1.4694	1.5181	1.5440	1.7501	2.0980	2.2206	2.2899	2.3268
	1.5	1.0112	1.3023	1.4191	1.4896	1.5287	1.6576	2.0228	2.1693	2.2595	2.3101
	3.0	0.8827	1.1919	1.3391	1.4395	1.5002	1.5817	1.9358	2.0963	2.2095	2.2802
	6.0	0.7578	1.0601	1.2277	1.3595	1.4502	1.5244	1.8550	2.0110	2.1380	2.2309

TABLE 47.7 Fundamental Frequencies of Portal Frames in Asymmetrical Mode of Vibration

and 47.8, respectively. The beams in these frames are assumed to be uniform and sufficiently slender, so that shear and axial and torsional deformations can be neglected. The method of analysis of these frames is given in Yang and Sun (1973). The vibration is assumed to be in the plane of the frame, and the results are presented for portal frames with pinned and fixed bases.

If the beam is rigid and the columns are slender and uniform, but not necessarily identical, then the natural fundamental frequency of the frame can be approximated using the following formula (Robert, 1979):

First symmetric in-plane mode									
$E_{2}I_{2}, m_{2}$					f = E = I = m =	$= \frac{\lambda^2}{2\pi L_1^2} \left(\frac{E_1 I}{m_1} \right)$ Modulus o Area mom Mass per o	^{1)^{1/2} HZ f elasticity ent of inerti unit length}	a	
					λ value				
$\left(\frac{m_2}{m_1}\right)^{1/4} \left(\frac{E_2I_2}{E_1I_1}\right)^{3/4}$	$\left(\frac{m_2}{m_1}\right)^{1/4} \left(\frac{E_2 I_2}{E_1 I_1}\right)^{3/4}$					$ \begin{pmatrix} E_1 I_1 \\ E_2 I_2 \\ m_1 \end{pmatrix}^{1/4} \frac{L_2}{L_1} $			
		8.0	4.0	2.0	1.0	0.8	0.4	0.2	
Pinned bases 8	.0	0.4637	0.8735	1.6676	3.1416	3.5954	3.8355	3.8802	
4.0		0.4958	0.9270	1.7394	3.1416	3.4997	3.7637	3.8390	
2.0		0.5273	0.9911	1.8411	3.1416	3.4003	3.6578	3.7690	
1	.0	0.5525	1.0540	1.9633	3.1416	3.3110	3.5275	3.6642	
0	0.8		1.0720	2.0037	3.1416	3.2864	3.4845	3.6240	
0.4		0.5735	1.1173	2.1214	3.1416	3.2259	3.3622	3.4903	
0	.2	0.5819	1.1466	2.2150	3.1416	3.1877	3.2706	3.3663	
Clamped bases 8	.0	0.4767	0.8941	1.6973	3.2408	3.9269	4.6167	4.6745	
4	.0	0.5093	0.9532	1.7847	3.3166	3.9268	4.5321	4.6260	
2	.0	0.5388	1.0185	1.9008	3.4258	3.9268	4.4138	4.5454	
1.0		0.5606	1.0773	2.0295	3.5564	3.9267	4.2779	4.4293	
0	.8	0.5659	1.0932	2.0696	3.5988	3.9267	4.2351	4.3861	
0.4 0.5776 1.1316 2			2.1790	3.7176	3.9267	4.1186	4.2481		
0.2 0.5842 1.1551 2.25				2.2575	3.8052	3.9266	4.0361	4.1276	

TABLE 47.8 Fundamental Frequencies of Portal Frames in Symmetrical Mode of Vibration

$$f = \frac{1}{2\pi} \left[\frac{12\sum E_i I_i}{L^3 (M + 0.37 \sum M_i)} \right]^{1/2} Hz$$
 (47.395)

where M is the mass of the beam, M_i is the mass of the i-th column, and E_iI_i is the flexural rigidity of the i-th column. The summation refers to the sum of all columns, and i must be greater or equal to 2. Additional results for frames with inclined members are discussed in Chang (1978).

Damping

Damping is found to increase with the increasing amplitude of vibration. It arises from the dissipation of energy during vibration. The mechanisms contributing to energy dissipation are material damping, friction at interfaces between components, and energy dissipation due to a foundation interacting with soil, among others. Material damping arises from the friction at bolted connections and frictional interaction between structural and nonstructural elements, such as partitions and cladding.

The amount of damping in a building can never be predicted precisely, and design values are generally derived based on dynamic measurements of structures of a corresponding type. Damping can be measured by the rate of decay of free vibration following an impact, spectral methods based on an analysis of the response to wind loading, and force excitation by a mechanical vibrator at varying frequencies to establish the shape of the steady-state resonance curve. However, these methods may not be easily carried out if several modes of vibration close in frequency are presented.

Table 47.8 gives values of modal damping that are appropriate for use when amplitudes are low. Higher values are appropriate at larger amplitudes where local yielding may develop, e.g., in seismic analysis.

Numerical Analysis

Many less complex dynamic problems can be solved without much difficulty by hand methods. For more complex problems, such as determination of natural frequencies of complex structures, calculation of response due to time-varying loads and response spectrum analysis to determine seismic forces may require numerical analysis. The finite element method has been shown to be a versatile technique for this purpose.

The global equations of an undamped force-vibration motion, in matrix form, may be written as

$$[M]{\ddot{x}} = [K]{\dot{x}} = {F(t)}$$
 (47.396)

where

$$[K] = \sum_{i=1}^{n} [k_i] \qquad [M] = \sum_{i=1}^{n} [m_i] \qquad [F] = \sum_{i=1}^{n} [f_i]$$
(47.397)

are the global stiffness, mass, and force matrices, respectively. $[k_i]$, $[m_i]$, and $[f_i]$ are the stiffness, mass, and force of the i-th element, respectively. The elements are assembled using the direct stiffness method to obtain the global equations such that intermediate continuity of displacements is satisfied at common nodes, and in addition, interelement continuity of acceleration is also satisfied.

Equation (47.396) is the matrix equations discretized in space. To obtain solution of the equation, discretization in time is also necessary. The general method used is called direct integration. There are two methods for direct integration: *implicit* or *explicit*. The first, and simplest, is an explicit method known as the central difference method (Biggs, 1964). The second, more sophisticated but more versatile, is an implicit method known as the Newmark method (Newmark, 1959). Other integration methods are also available in Bathe (1982).

The natural frequencies are determined by solving Eq. (47.396) in the absence of force F(t) as

$$[M]{\ddot{x}} + [K]{x} = 0 (47.398)$$

The standard solution for x(t) is given by the harmonic equation in time

$$\left\{\mathbf{x}(\mathbf{t})\right\} = \left\{\mathbf{X}\right\}\mathbf{e}^{\mathrm{i}\boldsymbol{\omega}\mathbf{t}} \tag{47.399}$$

where {X} is the part of the nodal displacement matrix called natural modes, which are assumed to be independent of time; i is the imaginary number; and ω is the natural frequency.

TABLE 47.9 Typical Structural Damping Values

Structural Type	Damping Value, ξ (%)
Unclad welded steel structures	0.3
Unclad bolted steel structures	0.5
Floor, composite and noncomposite	1.5-3.0
Clad buildings subjected to side sway	1

Differentiating Eq. (47.399) twice with respect to time, we have

$$\ddot{\mathbf{x}}(\mathbf{t}) = \left\{ \mathbf{X} \right\} \left(-\boldsymbol{\omega}^2 \right) \mathbf{e}^{i\boldsymbol{\omega}\mathbf{t}} \tag{47.400}$$

Substituting of Eqs. (47.399) and (47.400) into Eq. (47.398) yields

$$\mathbf{e}^{\mathrm{i}\omega t} \left(\left[\mathbf{K} \right] - \boldsymbol{\omega}^{2} \left[\mathbf{M} \right] \right) \left\{ \mathbf{X} \right\} = \mathbf{0}$$
(47.401)

Since e^{iωt} is not zero, we obtain

$$([K] - \omega^2[M]){X} = 0$$
 (47.402)

Equation (47.402) is a set of linear homogeneous equations in terms of displacement mode $\{X\}$. It has a nontrivial solution if the determinant of the coefficient matrix $\{X\}$ is nonzero; that is

$$[K] - \omega^2 [M] = 0 \tag{47.403}$$

In general, Eq. (47.403) is a set of n algebraic equations, where n is the number of degrees of freedom associated with the problem.

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