Appendix A

Table A.1 Displacements in beams

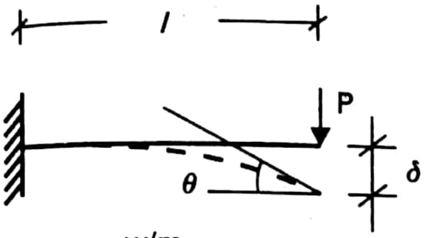
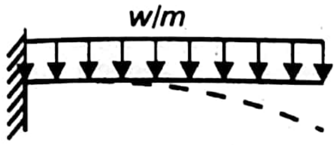
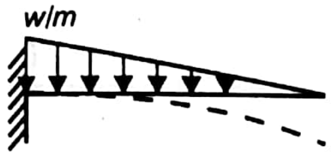
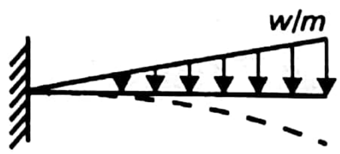
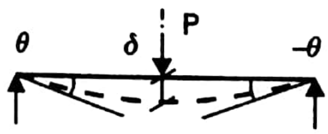
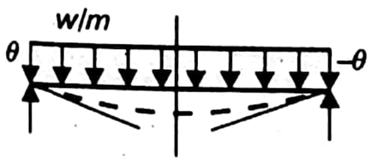
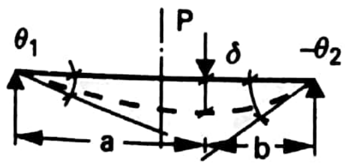
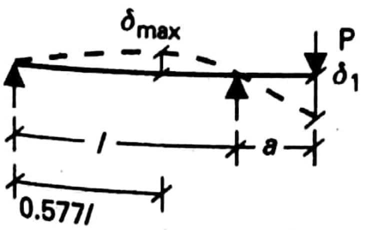
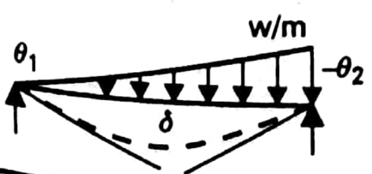
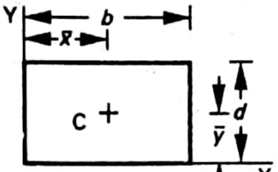
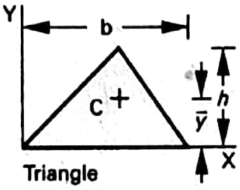
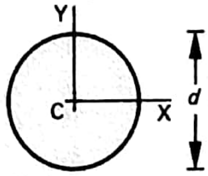
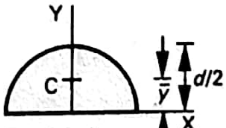
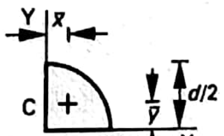
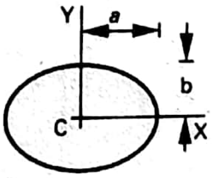
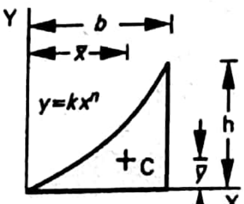
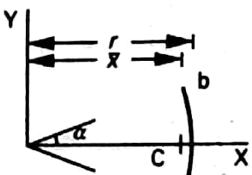
S.No.	Beam	δ	θ
1		$Pl^3/(3EI)$	$-Pl^2/(2EI)$
2		$wl^4/(8EI)$	$-wl^3/(6EI)$
3		$wl^4/(30EI)$	$-5wl^3/(120EI)$
4		$11wl^4/(120EI)$	$-wl^3/(8EI)$
5		$Pl^3/(48EI)$	$+Pl^2/(16EI)$
6		$5wl^4/(384EI)$	$+wl^3/(24EI)$
7		$Pa^2b^2/(3EIl)$	$\theta_1=Pb(l^2-b^2)/(6EIl)$ $\theta_2=Pa(l^2-a^2)/(6EIl)$
8		$\delta_{max}=-Pl^2a/(15.59EI)$ $\delta_1=Pa^2(a+l)/(3EI)$	
9		$2.5wl^4/(384EI)$	$\theta_1=7wl^3/(360EI)$ $\theta_2=wl^3/(45EI)$

Table A.2 Properties of some geometric shapes

S.No	Shape	Area	$\bar{x}$	$\bar{y}$	I_x	$\bar{I}_x$	I_y	$\bar{I}_y$
1	 Rectangle	bd	$b/2$	$d/2$	$bd^3/3$	$bd^3/12$	$b^3d/3$	$b^3d/12$
2	 Triangle	$bh/2$	–	$h/3$	$bh^3/12$	$bh^3/36$	–	–
3	 Circle	$\pi d^2/4$	0	0	$\pi d^4/64$	–	$\pi d^4/64$	–
4	 Semicircle	$\pi d^2/8$	0	$2d/(3\pi)$	$\pi d^4/128$	$0.00686d^4$	$\pi d^4/128$	–
5	 Quarter circle	$\pi d^2/16$	$2d/(3\pi)$	$2d/(3\pi)$	$\pi d^4/256$	$0.00686d^4$	$\pi d^4/256$	$0.00686d^4$
6	 Ellipse	πab	0	0	$\pi ab^3/4$	–	$\pi a^3b/4$	–
7	 Spandrel	$\dfrac{bh}{(n+1)}$	$\dfrac{(n+1)b}{(n+2)}$	$\dfrac{(n+1)h}{(4n+2)}$	$\dfrac{bh^3}{[3(3n+1)]}$	–	$\dfrac{b^3h}{(n+3)}$	–
8	 Circular arc	$2rab$	$\dfrac{r\sin\alpha}{\alpha}$	0	$r^3b[\theta-\sin\theta\cos\theta]$	–	$r^3b[\theta+\sin\theta\cos\theta]$	–

N.B.: I_x, I_y = moments of inertia about X- and Y-axes, respectively

$\bar{I}_x, \bar{I}_y$ = moments of inertia about centroidal X- and Y-axes, respectively

Appendix B Short questions

Students are advised to answer the short questions pertaining to each chapter after these topics are discussed in class. These exercises may be practised periodically at random to recapitulate the subject material, and check the comprehension levels. Students may note that the questions given here are only a few of the possibilities; hundreds more questions may be set on these topics or the same questions may be worded differently. However, answering these questions prepares students to solve such questions in various university and competitive examinations quickly and correctly.

The correct answers are to be chosen from the choices given below each question. There may be more than one correct answer to some questions; credit is given only when all the correct answers are identified. The answers may be checked with the key at the end of the appendix. Students are advised to reason out the correct answers, study the relevant sections of the text, and repeat the test after a few days, if the reasoning is wrong.

Chapter I

I.1 Newton is the force exerted by gravity on a body of mass
a. 1.0 kg **b.** 100.0 g **c.** 100.9 g **d.** 102.0 g

I.2 Flexural rigidity (EI) is expressed in units of
a. kNm^2 **b.** MNm^2 **c.** kNm^{-2} **d.** MNm^4

I.3 Moment is expressed in units of
a. kNm **b.** KNm **c.** kNm^2 **d.** MNm

I.4 Newton is the force exerted on a body of 1.0 kg mass to induce an acceleration of
a. 1.0 mms^{-1} **b.** 1.0 ms^{-1} **c.** 9.81 ms^{-1} **d.** 9.8066 ms^{-1}

I.5 Tests on beams to failure were first conducted by
a. Newton **b.** Bernoulli **c.** Coulomb **d.** Galileo

I.6 The basis for structural mechanics was first formulated by
a. Galileo **b.** Aristotle **c.** Leonardo da Vinci **d.** Bernoulli

I.7 The principles of strength of materials were first formulated by
a. Leonardo da Vinci **b.** Aristotle **c.** Newton **d.** Coulomb

I.8 Elastic behaviour of material was first noted by
a. Young **b.** Hooke **c.** Newton **d.** Galileo

I.9 Elastic limit of materials was first noted by
a. Leonardo da Vinci **b.** Hooke **c.** Archimedes **d.** Young

I.10 Plastic behaviour of materials was first noted by
a. Coulomb **b.** Navier **c.** Newton **d.** Hooke

I.11 The difference between elastic and plastic behaviours of materials was first explained by
a. Bernoulli **b.** Coulomb **c.** Hooke **d.** Navier

I.12 The concepts of tubular design were developed by
a. Hooke **b.** Bernoulli **c.** Coulomb **d.** Fazlur Khan

I.13 A hinged support can transmit
a. one reactive force **b.** two reactive forces **c.** three reactive forces **d.** four reactive forces

I.14 A roller support can transmit
a. one reactive force **b.** two reactive forces **c.** three reactive forces **d.** four reactive forces

I.15 An en castre support can transmit
a. one reactive force **b.** two reactive forces **c.** three reactive forces **d.** four reactive forces

I.16 A fixed support can transmit
a. one reactive force **b.** two reactive forces **c.** three reactive forces **d.** four reactive forces

I.17 An Equivalent system of forces ensures
a. equilibrium conditions **b.** equilibrium and compatibility conditions
c. equilibrium or compatibility conditions **d.** the same effects as the original force system

I.18 Elasto-plastic behaviour was first recognised by
a. Navier **b.** Coulomb **c.** Aristotle **d.** Euler

I.19 Equilibrium conditions $M_A = M_B = \sum F_y = 0$ require that points A and B, about which moments are taken, are
a. on a vertical line **b.** on a horizontal line **c.** concurrent **d.** on an inclined line

I.20 Equilibrium conditions $M_A = M_B = \sum F_x = 0$ require that points A and B, about which moments are taken, are
a. on a vertical line **b.** on a horizontal line **c.** collinear **d.** on an inclined line

I.21 Equilibrium conditions $M_A = M_B = M_C = 0$ require that points A, B and C, about which moments are taken, are
a. on a vertical line **b.** on a horizontal line **c.** collinear **d.** non-collinear

I.22 A free body diagram indicates
a. an equivalent system of forces **b.** internal forces on a body
c. external forces on a body **d.** internal and external forces on a body

I.23 Bending moments occur due to
a. axial forces **b.** transverse forces **c.** eccentric forces **d.** transverse and eccentric forces

I.24 Torsional moments occur due to
a. axial forces **b.** plane forces **c.** eccentric transverse forces **d.** eccentric forces

I.25 Axial forces act
a. on the cross section **b.** on the cross section along the member axis
c. normal to the member axis **d.** normal to the cross section

I.26 Transverse forces on a member induce
a. transverse stresses **b.** bending moments
c. bending and twisting moments **d.** bending or twisting moments

I.27 Strength of materials is the study of the behaviour of
a. structural members **b.** construction materials
c. structural systems **d.** materials and structures

I.28 An elastic body
a. deforms under load **b.** does not deform under load
c. deforms, but regains the original shape and size on removal of load
d. deforms, and does not regain the original shape and size on removal of load

I.29 A plastic body
a. deforms under load **b.** does not deform under load
c. deforms, but regains the original shape and size on removal of load
d. deforms, and does not regain the original shape and size on removal of load

I.30 An elasto-plastic body
a. deforms under load
b. does not deform under load
c. deforms, but regains the original shape and size on removal of load
d. deforms, and does not regain the original shape and size on removal of load

1.31 A rigid body
a. deforms under load b. rotates under load
c. does not deform under load
d. deforms, and does not regain the original shape and size on removal of load

1.32 Indian standards are prepared by the
a. Indian Standards Institution b. Standards Association of India
c. Bureau of Indian Standards d. Indian Standards Bureau

1.33 The standards for the Indian national flag are specified by
a. IS:1–1947 b. IS:1–1968 c. IS:1 000–1998 d. IS:100–1957

1.34 Static equilibrium implies that the
a. forces have zero resultant b. forces have zero resultant and zero moment
c. forces are stationary d. forces are equal

1.35 The following data is not acceptable in SI
a. 2847.8806 b. 2.0×10^5 c. 3 456.981 23 d. 800×10^{-3}

Chapter 1

1.1 Stress is defined as the
a. load per unit deformation b. load per unit area
c. deformation per unit area d. load intensity

1.2 Strain is defined as the
a. load per unit deformation b. deformation per unit load
c. deformation per unit area d. deformation per unit length

1.3 Units of stress are
a. kNm^{-2} b. KNm^{-2} c. MPa d. newtons

1.4 Units of strain are
a. $m\ m^{-1}$ b. mm/m c. Nmm d. mm/mm

1.5 Thermal strain in a body can be expressed as
a. mm/mm b. $\alpha \Delta T$ c. ΔT/m d. $\alpha \Delta T$/mm

1.6 Hooke's law is valid
a. upto the elastic limit b. upto failure c. upto the plastic stage d. upto yield

1.7 At the elastic limit
a. stress is proportional to strain b. strains are fully recoverable
c. load is proportional to strain d. load is proportional to deformation

1.8 Young's modulus is the
a. ratio of strain to stress at the proportionality limit
b. product of strain and stress at the elastic limit
c. ratio of stress to strain at the elastic limit
d. ratio of stress to strain upto the proportionality limit

1.9 Gage length is the
a. length of the test specimen b. deformed length
c. length over which deformation is measured d. length over which strain is measured

1.10 Modulus of elasticity is the
a. ratio of stress to strain within the elastic limit
b. slope of the stress–strain curve within the elastic limit
c. ratio of strain to stress at fracture
d. ratio of stress to strain at the ultimate load

1.11 Secant modulus is the slope of the
a. stress–strain curve at fracture
b. line joining the origin with a point on the stress–strain curve
c. tangent at a point on the curve
d. line joining the origin with the fracture point

1.12 Working stress of mild steel is determined from the
a. upper yield stress **b.** ultimate stress **c.** fracture stress **d.** lower yield stress

1.13 Factor of safety is the ratio of
a. ultimate and working stresses **b.** upper yield and working stresses
c. ultimate and lower yield stresses **d.** lower yield and working stresses

1.14 Strain hardening of a material is
a. increase in stress with strain beyond yield stress
b. increase in strain with stress beyond ultimate stress
c. increase in hardness as strain is increased
d. increase in hardness under repeated loading

1.15 The ratio of the strain at fracture to that at yield for mild steel is approximately
a. 10.0 **b.** 100.0 **c.** 1 000.0 **d.** 250.0

1.16 Young's modulus of a material is an index of its
a. ultimate strength **b.** yield strength **c.** deformability **d.** ductility

1.17 A material with 7.2 percent ultimate strain is
a. ductile **b.** brittle **c.** partly ductile **d.** of intermediate ductility

1.18 Toughness of a material is an index of its
a. strength **b.** ductility **c.** energy absorption **d.** brittlness

1.19 True stress and strain are related to specimen's
a. initial dimensions **b.** dimensions at failure
c. instantaneous dimensions **d.** dimensions at yield

1.20 True strain is not adopted because
a. all materials are not ductile **b.** all materials are not brittle
c. materials are within the elastic limit **d.** materials are in the working stress range

1.21 Plastic range of a material is an index of its
a. ductility **b.** brittleness **c.** strength **d.** toughness

1.22 Proof stress of a material indicates
a. its strength at fracture **b.** its yield stress
c. material without a well defined yield point
d. material without a well defined fracture point

1.23 Isotropic materials have, at any point,
a. same properties in all directions **b.** different properties in all directions
c. equal strength in all directions **d.** equal modulus of elasticity in all directions

1.24 Anisotropy indicates that the material has
a. same properties in orthogonal directions **b.** different properties in all directions
c. different strength in different directions **d.** equal modulus of elasticity in all directions

1.25 Orthotropic materials have
a. different properties in different directions
b. different properties in orthogonal directions
c. different strength in different directions
d. different modulii of elasticity in orthogonal directions

1.26 Principle of superposition is valid for
a. isotropic materials b. linear elastic materials
c. linear elastic and isotropic materials
d. isotropic, homogenous and linearly elastic materials

1.27 Principle of superposition is not valid for
a. orthotropic materials b. materials of non-linear elasticity
c. anisotropic materials d. materials of linear elasticity

1.28 Plastic range of high strength steel, as percentage, is
a. 10.0–15.0 b. 0.10–1.50 c. 30.0–35.0 d. 20.0–35.0

1.29 Plastic range of a material is 0.3 percent; it could be
a. mild steel b. brass c. concrete d. ceramic

1.30 Poisson determined the effects of lateral strain based on
a. experiments b. tests and theory c. molecular theory d. mathematical analysis

1.31 The maximum possible value of Poisson's ratio is
a. 0.10 b. 0.30 c. 1.0 d. 0.50

1.32 Poisson's ratio for materials is in the range of
a. 0.00 to 0.50 b. −0.47 to 0.50 c. −0.50 to 0.50 d. −1.00 to 1.00

1.33 The usual value of factor of safety for steel is
a. 3.0 b. 2.5 c. 1.80 d. 1.50

1.34 The usual value of factor of safety for concrete is
a. 3.0 b. 2.5 c. 1.80 d. 1.50

1.35 Compound bar systems are statically
a. determinate b. indeterminate c. stable d. unstable

1.36 Compatibility conditions deal with
a. force equilibrium b. internal forces c. strains d. displacements

1.37 Apparent modulus of elasticity is
a. $\sum E_i A_i / \sum A_i$ b. $\sum E_i / n$ c. $\sum A_i / \sum A_i E_i$ d. $\sum \sigma_i / \sum \epsilon_i$

1.38 Apparent modulus of elasticity of a composite material is
a. an index of the mean value of elasticity modulii
b. an index of the weighted mean value of elasticity modulii
c. larger than the values of the individual components
d. smaller than the values of the individual components

1.39 Thermal coefficient of expansion is an index of
a. material strength b. Young's modulus of the material
c. thermal sensitivity d. thermal stresses

1.40 Thermal stresses in an unrestrained body
a. are always zero b. are never zero c. may be zero d. are zero in a homogenous body

1.41 Thermal stresses in a restrained body
a. are always zero b. are never zero c. may be zero d. are zero in a homogenous body

1.42 Thermal coefficient of expansion is usually expressed in
a. micro-strains per K b. milli-strains per K c. milli-strains per C d. micro-strains per C

1.43 Bulk modulus of a material is a function of its
a. Young's modulus and Poisson's ratio b. Young's modulus only
c. compressibility d. Poisson's ratio only

1.44 The ratio of bulk modulus to Young's modulus
a. is always greater than 1.0 b. is always less than 1.0
c. varies from 1.0–3.0 d. depends upon the Poisson's ratio

1.45 Constitutive relationship of a material is the
a. stress–strain relationship **b.** ratio of bulk modulus to Young's modulus
c. ratio of Young's modulus to bulk modulus
d. function of Poisson's ratio and Young's modulus

1.46 High tensile strength steels are
a. more ductile than mild steel **b.** less ductile than mild steel
c. stronger than mild steel **d.** less deformable than mild steel for the same stress

1.47 A prismatic body has, over its length,
a. constant cross section **b.** constant cross section and material properties
c. constant section but may have varying material properties
d. constant properties but may have varying cross section

1.48 If a material is incompressible, its
a. Young's modulus is infinite **b.** bulk modulus is zero
c. Poisson's ratio is zero **d.** Poisson's ratio is 0.50

1.49 If the bulk modulus of a material is larger than its Young's modulus, its Poisson's ratio is
a. 0.00 **b.** 0.15 **c.** 0.30 **d.** 0.40

1.50 If the bulk modulus of a material is smaller than its Young's modulus, its Poisson's ratio is
a. 0.50 **b.** 0.15 **c.** 0.38 **d.** 0.40

1.51 An indeterminate system can be analysed by applying the conditions of
a. equilibrium **b.** compatibility and equilibrium
c. compatibility **d.** compatibility or equilibrium

1.52 A non-prismatic bar has
a. constant stress **b.** varying stress
c. constant cross section **d.** varying cross section

1.53 The substance with the largest Young's modulus known is
a. steel **b.** wrought iron **c.** diamond **d.** epoxy

1.54 The substance with the largest Poisson's ratio known is
a. steel **b.** rubber **c.** paraffin **d.** clay

Chapter 2

2.1 Strain energy is a function of
a. body volume and strain **b.** body volume, Young's modulus and strain
c. Young's modulus, strain and cross-sectional area
d. load applied and body volume

2.2 Work done is expressed in units of
a. kNm **b.** kNm^3/m **c.** kNm/m^3 **d.** MNm/m^3

2.3 Strain energy is a measure of
a. work done by applied loads **b.** deformability of the body
c. impact resistance of the body **d.** resilience of the body

2.4 Impact loads on a body
a. increase stresses and strains **b.** increase stresses only
c. increase stresses and deformations **d.** increase deformations but decrease stresses

2.5 The principle of superposition is
a. always valid for impact loads **b.** not valid for impact loads
c. valid for impact loads on prismatic bodies
d. valid for impact loads on linearly elastic bodies

2.6 Resilience modulus of a material indicates its ability
a. to resist deformations b. to resist external loads
c. to resist impact loads d. to regain its original shape on removal of loads

2.7 Toughness modulus of a material indicates its ability
a. to absorb energy before fracture b. to resist external loads
c. to resist impact loads d. to resist fracture

2.8 Resilience modulus is a function of
a. the area of the stress–strain diagram b. Young's modulus and yield stress
c. yield stress and yield strain d. fracture stress and yield strain

2.9 Toughness modulus is a function of
a. Young's modulus and fracture stress b. Young's modulus and yield stress
c. yield stress and yield strain d. the area of the stress–strain diagram

2.10 For large energy absorption, a body should have
a. large volume, large Young's modulus and large yield stress
b. large volume, low Young's modulus and large yield stress
c. large volume, large Young's modulus and low yield stress
d. small volume, low Young's modulus and large yield stress

2.11 Toughness modulus is measured by
a. direct tension test b. Charpy's test c. Izod test d. direct compression test

2.12 A groove on a body
a. increases its resilience b. decreases its resilience
c. increases its toughness d. decreases its toughness

2.13 Impact factor is
a. usually greater than 1.0 b. always greater than 1.0
c. greater than 1.5 d. greater than 2.0

2.14 Impact factor is, compared to static effects, a measure of
a. increase in stresses b. increase in deformations
c. increase in strain energy d. decrease in strength

2.15 Equivalent load of an impact load is the static load that yields the same
a. strain energy b. deformations c. stresses or deformations d. stresses

2.16 Impact factor is significant for
a. static loads b. static and dynamic loads c. dynamic loads d. moving loads

2.17 Stress concentration occurs due to
a. sudden change in cross section b. a discontinuity
c. suddenly applied loads d. yielding of the material

2.18 Impact load increases stresses in
a. steel more than in timber b. timber more than in steel
c. timber more than in concrete d. concrete more than in steel

Chapter 3

3.1 Shear stress acts
a. normal to a surface b. tangential to a surface c. along the body axis d. normal to the body axis

3.2 Pure shear is due to
a. load on a surface b. load normal to a surface c. cutting of a plate d. bending of a bar

3.3 Punching shear is due to
a. load on a surface b. load normal to a surface c. cutting of a plate d. torsion

3.4 Shear stresses are caused by
a. torsion b. axial loads c. bending d. eccentric axial loads

3.5 Shear stresses cause
a. bending of the body b. twisting of the body
c. distortion of the body d. elongation of the body

3.6 Rivets in lap joints are subjected to
a. axial forces only b. axial and shear forces c. double shear d. single shear

3.7 Rivets in butt joints are subjected to
a. bending forces only b. bending and shear forces c. double shear d. single shear

3.8 Welded lap joints are designed for
a. bending forces only b. bending and axial forces c. double shear d. single shear

3.9 Welded butt joints are designed for
a. bending forces only b. bending and shear forces
c. bending and axial forces d. axial forces only

3.10 Complementary shear is
a. equal to the applied shear stress b. double the value of the applied shear
c. equal to the applied shear force d. half of the value of the applied shear

3.11 Shear strain is the
a. change in the right angle of a body
b. ratio of the change in the angle to the original angle
c. ratio of the change in the length to the original length
d. ratio of the change in the length to the original angle

3.12 Rigidity modulus is defined as the ratio of
a. ultimate stress to ultimate strain b. fracture stress to fracture strain
c. shear stress to shear strain d. complementary shear stress to shear strain

3.13 Governing value of a rivet is the
a. maximum spacing allowed b. least strength in shear, tension and bearing
c. minimum spacing allowed d. least strength in shear, tension and crushing

3.14 Bearing stress on a rivet causes its failure by
a. fracture b. crushing c. shearing d. compression

3.15 Pitch of rivets is their
a. spacing b. minimum spacing c. minimum strength d. maximum strength

3.16 Throat of a weld is its
a. mean thickness b. minimum thickness c. maximum thickness d. actual thickness

3.17 Efficiency of a joint is the
a. ratio of the rivet strength to weld strength
b. minimum load it can sustain
c. ratio of the strength of the joint to that of the plates
d. ratio of the strength of the joint to that of the cover plates

3.18 Butt weld is subjected to
a. compression b. tension c. shear d. shear and tension

3.19 Lap weld is subjected to
a. compression and shear b. tension and shear c. shear d. compression and tension

3.20 Bead of a weld is the
a. minimum material deposited at a joint b. material deposited at a joint
c. length of the welded joint d. maximum thickness of the joint

3.21 Shear strain energy is a measure of the
a. material resilience in shear b. material strength in shear
c. material toughness in shear d. stress–strain area in shear

3.22 Rigidity modulus of a material is
a. always greater than Young's modulus b. always smaller than Young's modulus
c. smaller than Young's modulus, if Poisson's ratio is 0.50
d. greater than Young's modulus, if Poisson's ratio is 0.00

Chapter 4

4.1 The maximum shear due to an axial compression of 100.0 MPa is
a. 100.0 MPa b. 150.0 MPa c. 50.0 MPa d. 75.0 MPa

4.2 The maximum direct stress due to a shear stress of 100.0 MPa is
a. + 100.0 MPa b. + 150.0 MPa c. – 50.0 MPa d. – 100.0 MPa

4.3 Principal stress is the
a. maximum direct stress on a plane b. maximum direct stress in a body
c. maximum or minimum direct stress in a body d. maximum shear stress in a body

4.4 Principal planes and planes of maximum shear stress are inclined to each other at
a. 180° b. 90° c. 45° d. 135°

4.5 Mohr's circle is a graphical representation of stresses
a. along a circle b. at a point in a body c. on a plane in a body d. on a circular surface

4.6 If the rigidity modulus of a material is 70.0 GPa, and its Poisson's ratio is zero, its Young's modulus is
a. 100.0 GPa b. 35.0 MPa c. 140.0 MPa d. 35.0 GPa

4.7 The ratio of Young's modulus to rigidity modulus of a material is
a. greater than 1.0 b. less than 1.0 c. 0.50 d. depends upon the Poisson's ratio

4.8 If the Young's modulus and rigidity modulus of a material are 200.0 GPa and 76.0 GPa, respectively, the Poisson's ratio is
a. 0.30 b. 0.27 c. 0.45 d. 0.32

4.9 If the Young's modulus and rigidity modulus of a material are 200.0 GPa and 76.0 GPa, respectively, the bulk modulus is
a. 181.0 GPa b. 138.0 GPa c. 195.0 GPa d. 25.0 GPa

4.10 If the orthogonal stresses at a point are 100.0 and – 70.0 MPa, the radius of the Mohr's circle is
a. 110.0 b. 177.2 c. 52.2 d. 85.0

4.11 Mohr's circle is valid for
a. two-dimensional stress conditions only b. all stress conditions
c. only orthogonal stress conditions d. two- and three-dimensional stress conditions

4.12 Tresca's failure criterion is valid for
a. general stress conditions b. any material c. ductile materials d. brittle materials

4.13 Mises' failure criterion is valid for
a. general stress conditions b. any material c. ductile materials d. brittle materials

4.14 Tresca's and Mises' criteria are not valid for
a. brittle materials b. small differences in stresses
c. ductile materials d. large differences in stresses

4.15 Tresca's criterion is based on the
a. principal stress condition b. principal strain condition
c. principal shear stress condition d. maximum shear stress condition

4.16 Mises' criterion is based on the
a. principal stress condition b. strain energy condition
c. shear strain energy condition d. maximum shear stress condition

4.17 Rankine's criterion is based on the
a. principal stress condition b. principal strain condition
c. principal shear stress condition d. maximum shear stress condition

4.18 Haigh's criterion is based on the
a. maximum strain energy condition b. principal strain condition
c. principal shear stress condition d. maximum shear stress condition

4.19 St. Venant's criterion is based on the
a. maximum strain energy condition b. principal strain condition
c. principal shear stress condition d. maximum shear strain energy condition

4.20 Rankine's theory is applicable to
a. cast iron b. mild steel c. brittle materials d. concrete

4.21 For ductile materials, when the stress difference is small, it is preferable to adopt
a. Mises' criterion b. Tresca's criterion c. Rankine's criterion d. St. Venant's criterion

4.22 For brittle materials, such as concrete, it is preferable to adopt
a. Mises' criterion b. Haigh's criterion c. Drucker–Prager criterion d. St. Venant's criterion

4.23 Principal planes for stresses and strains are
a. generally the same b. generally different
c. the same, if Poisson's ratio is zero d. always different

4.24 The radius of Mohr's circle for orthogonal strains of 200.0 and −100.0, and shear strain of 150.0 is (all units in micro-strains)
a. 212.13 b. 150.00 c. 167.71 d. 50.00

4.25 The behaviour of a brittle material changes to ductility under
a. biaxial compression b. triaxial compresion c. biaxial tension d. triaxial tension

4.26 Steel fails in brittle mode under
a. biaxial compression b. triaxial compresion c. biaxial tension d. triaxial tension

Chapter 5

5.1 The degree of redundancy of a beam is the number of additional
a. support forces of the beam b. equations required in the analysis
c. statical conditions in the analysis d. conditions in the analysis

5.2 A redundant beam has
a. too many loads b. too many supports
c. too few equilibrium conditions d. too few support reactions

5.3 A hinge may be provided in a structure to relieve
a. moment only b. moment and shear force only
c. moment, shear and axial forces d. any one of moment, shear and axial forces

5.4 A moment hinge releases
a. moment b. moment but transmits shear force
c. moment but transmits shear and axial forces d. shear and axial forces

5.5 A shear hinge releases
a. shear force b. shear force but transmits bending moment
c. shear and axial forces but not bending moment
d. shear force but transmits bending moment and axial force

5.6 An axial force hinge releases
a. axial force b. axial force but transmits shear force and moment
c. moment and shear force but transmits axial force
d. shear and axial forces

5.7 Bending moment is defined as the moment about a section of all the forces
a. on the beam b. to the right of the section
c. to the left of the section d. to the left or right of the section

5.8 Shear force at a section is defined as the sum of all the forces
a. to the left or right of the section b. to the right of the section
c. to the left of the section
d. transverse to the member axis to the left or right of the section

5.9 Axial force at a section is defined as the sum of all the forces
a. along the beam axis to the left or right of the section b. to the right of the section
c. to the left of the section d. to the left or right of the section

5.10 Shear force at a section is the
a. integral of the load function b. differential of the load function
c. differential of the bending moment function d. integral of the bending moment function

5.11 Bending moment at a section is the
a. integral of the load function b. differential of the shear force function
c. differential of the load function d. integral of the shear force function

5.12 Bending moment at a section has the maximum value where
a. the load function has zero value b. shear force is zero
c. shear force is zero or changes sign d. the load has maximum value

5.13 Bending moment function is
a. linear between two point loads b. non-linear between two point loads
c. parabolic in the region of uniformly distributed load d. always non-linear

5.14 Shear force function is
a. linear between two point loads b. parabolic in the region of linearly varying load
c. parabolic in the region of uniformly distributed load
d. constant between two point loads

Chapter 6

6.1 The assumption of sections plane before bending remaining plane after bending implies that the
a. material is linearly elastic b. sections do not warp
c. material is isotropic d. material is homogenous

6.2 The bending theory ignores
a. tensile stresses b. warping effects c. shear stresses d. shear deformations

6.3 Moment of inertia is a measure of the
a. resistance to deformation b. area of cross section
c. strength of extreme fibres d. modulus of elasticity

6.4 The bending theory assumes that the section is
a. rectangular b. symmetric about the vertical axis
c. symmetric about the horizontal axis d. symmetric about both the axes

6.5 A beam subjected to transverse loads develops
a. longitudinal shear stresses only b. transverse shear stresses only
c. longitudinal and transverse shear stresses d. longitudinal or transverse shear stresses

6.6 Neutral axis of a beam develops the
a. maximum shear and bending stresses **b.** minimum bending stresses
c. maximum shear stresses **d.** maximum bending stresses

6.7 Neutral axis of a beam is the location of
a. maximum longitudinal strain **b.** minimum longitudinal strain
c. maximum transverse strain **d.** zero strain

6.8 A beam of section 200.0 mm deep and 100.0 mm wide, and another comprising two segments of 100.0 × 100.0 mm resist the same moment, if the segments are placed
a. side by side and well bonded **b.** one above the other
c. one above the other and well bonded **d.** side by side

6.9 An *I*-section is flexurally efficient because
a. it has large area of cross section **b.** its area is concentrated in the web
c. its area is concentrated away from the neutral axis
d. its area is concentrated near the neutral axis

6.10 For a given area, a circular section is flexurally
a. more efficient than a rectangular section of same depth
b. less efficient than a square section **c.** more efficient than an *I*-section of the same depth
d. less efficient than an *I*-section of the same depth

6.11 Flexural shear stresses generally have the maximum value at the
a. extreme fibres of a section **b.** mid-depth of a section
c. neutral axis of a section **d.** centroid of a section

6.12 Flexural shear stresses do not have the maximum value at the neutral axis for a
a. rectangular section **b.** circular section **c.** I-section **d.** diamond section

6.13 The ratio of maximum to mean values of flexural shear stresses in a rectangular section is
a. 1.5 **b.** 1.25 **c.** 2.0 **d.** 1.75

6.14 The ratio of maximum to mean values of flexural shear stresses in a thin circular section is
a. 1.50 **b.** 1.25 **c.** 2.00 **d.** 1.75

6.15 The ratio of maximum to mean values of flexural shear stresses in a circular section is
a. 1.50 **b.** 1.30 **c.** 1.33 **d.** 1.75

6.16 Euler–Bernoulli's bending theory is applicable to
a. shallow beams **b.** deep beams **c.** curved beams **d.** straight beams

6.17 Euler–Bernoulli's bending theory is applicable to
a. beams of long spans **b.** beams with large deformations
c. beams with large rotations **d.** arches

6.18 If the moment of inertia of a section about its base is 4 000.0 units, its area is 10.0 units and height of neutral axis from the base is 10.0 units, the moment of inertia about the neutral axis is
a. 4 000.0 units **b.** 5 000.0 units **c.** 3 000.0 units **d.** 2 000.0 units

6.19 If the moment of inertia of a section about its base is 4 000.0 units, its area is 10.0 units, height of neutral axis from the base is 10.0 units and the over all depth is 15.0 units, the moment of inertia about the top fibre is
a. 4 000.0 units **b.** 3 250.0 units **c.** 6 250.0 units **d.** 2 000.0 units

6.20 If the moments of inertia of a section about its orthogonal axes are 1 000.0 and 2 000.0 units and its area is 100.0 units, its polar moment of inertia is
a. 3 000.0 units **b.** 3 100.0 units **c.** 5 000.0 units **d.** 4 000.0 units

6.21 If the moments of inertia of a section about its orthogonal axes are 1 000.0 and 2 000.0 units, the product of inertia is 300.0 units and its area is 100.0 units, its major principal moment of inertia is
a. 1 500.0 units **b.** 914.9 units **c.** 500.0 units **d.** 2 083.1 units

6.22 The sum of moments of inertia of a section about any set of orthogonal axes is
a. always constant b. usually constant
c. dependent upon the axes d. dependent upon the material

6.23 The sum of moments of inertia of a section about any set of orthogonal axes is constant because of the
a. assumptions of the bending theory b. theorem of parallel axes
c. theorem of perpendicular axes d. material properties

6.24 The moment capacity of a section depends upon its
a. moment of inertia b. area c. depth and moment of inertia d. elastic modulus

6.25 The flexural rigidity of a beam is the largest, when the flitches are fixed to the
a. top of the section b. soffit of the section
c. top and bottom of the section d. sides of the section

6.26 Flitches are fixed to timber beams in order to increase their
a. moment capacity b. shear capacity c. effective sectional area d. effective depth

6.27 Euler–Bernoulli's bending theory is not applicable to
a. T-sections b. L-sections c. C-sections d. I-sections

6.28 The product of inertia is zero when the section is
a. symmetric about one axis b. symmetric about two axes
c. homogenous and isotropic d. rectangular

6.29 Compared to the section of the weaker material, the larger the modular ratio of a composite section, the
a. larger the shear capacity b. larger the moment capacity
c. smaller the moment capacity d. smaller the flexural rigidity

6.30 Composite beams can be analysed by applying
a. Euler–Bernoulli's bending theory b. equilibrium conditions
c. equivalent loading and compatibility conditions d. equilibrium and compatibility conditions

6.31 Shear connectors in a beam increase
a. only shear capacity b. only moment capacity
c. shear and moment capacities d. effective depth

6.32 The fraction of the shear force sustained by the web of a rolled I-section is about
a. 0.30 b. 0.50 d. 0.70 d. 0.90

6.33 Shear flow in a section indicates
a. shear stress b. shear force per unit width c. shear resistance d. shear force per unit depth

6.34 Shear flow at a point in a section is a function of
a. section profile and shear force b. section profile and its width at the point
c. moment of inertia and shear force d. shear force and section depth

6.35 Shear centre and centroid are different for
a. an I-section b. a channel with horizontal web
c. an angle with vertical leg d. a channel with vertical web

6.36 Euler bending theory is not applicable to
a. an I-section b. a channel with horizontal web
c. an angle with vertical leg d. a channel with vertical web

6.37 Slenderness of a beam is the ratio of its
a. length and minimum radius of gyration b. span and minimum radius of gyration
c. span and depth d. depth and width

6.38 The radius of the Mohr's circle for a section of moments of inertia 1 000.0 and 2 000.0, and product of inertia −500.0 is
a. 1 118.0 **b.** 559.0 **c.** 707.1 **d.** 500.0

6.39 A transformed section has the same moment of inertia as the section of the
a. stronger material **b.** weaker material
c. the material chosen as the basis **d.** beam of the transformed depth

6.40 A transformed section has the same
a. depth as the original section **b.** width as the original section
c. depth and width as the original beam **d.** moment of inertia as the original section

6.41 The product of inertia of a section
a. is always positive **b.** may be negative
c. is always negative **d.** depends upon the axes chosen

6.42 In a composite beam the stronger material sustains higher stress than the weaker material because of
a. strain compatibility at the interface **b.** stress compatibility at the interface
c. larger Young's modulus **d.** the transformed section

6.43 Shear center and geometric centre of a section coincide, if the
a. section is symmetric **b.** load passes through centroid
c. section is symmetric about the load axis
d. section is symmetric about the axis normal to the load

6.44 The shear centre of an angle section is
a. its geometric centre **b.** its leg axis **c.** normal to the leg axis **d.** the junction of its legs

6.45 At the interface of a composite beam, for the two materials the
a. stresses are equal **b.** strains are equal **c.** curvatures are equal **d.** moments are equal

6.46 The curvature of a beam at a section is given by
a. $M/(EI)$ **b.** σ/y **c.** ϵ_{max}/y_{max} **d.** σ/y_{max}

6.47 Slenderness ratio of a beam is restricted in order to prevent
a. buckling about major principal axis **b.** buckling about minor principal axis
c. compression failure **d.** large deformations

6.48 The computed moment of inertia of standard steel sections, based on the mean thicknesses listed, is
a. always more than the listed value **b.** always less than the listed value
c. equal to the listed value **d.** differs from the listed value

6.49 The computed moment of inertia of standard steel sections, based on the mean thicknesses listed, is different from the listed value because of the
a. computational approximations **b.** fillets at the web flange junctions
c. variability of the section along the length
d. differences in the material properties along the span

6.50 The computed moment of inertia of standard steel I-sections, based on the mean thicknesses listed, about the web axis is
a. more than the listed value **b.** less than the listed value
c. equal to the listed value **d.** different from the listed value, depending upon the section

Chapter 7

7.1 Combined stresses on a section induce
a. uniform stress distribution **b.** non-uniform stress distribution
c. tensile stresses **d.** tensile and shear stresses

7.2 Structures are usually subjected to
a. uniform stresses b. non-uniform stresses c. bending stresses d. eccentric loads

7.3 Non-prismatic structures have, along their length,
a. uniform cross section b. uniform moment of inertia
c. non-uniform moment of inertia d. varying cross section

7.4 If a load is applied within the kern of a section, the induced stresses are
a. compressive b. tensile c. uniform d. of the same nature as the applied load

7.5 The kern of a circle is
a. a circle of half the diameter b. a circle of one third the diameter
c. a square of side half the diameter d. within the middle quarter circle

7.6 The kern of a circular ring is
a. a circle within the section b. a circle of one third the diameter
c. a square of side half the diameter d. within the middle quarter circle

7.7 The kern of a rectangle is
a. a symmetric circle of one third the depth
b. of diamond shape within the middle third of the width and depth
c. a symmetric square of one third the width
d. a symmetric rectangle of one third the depth and width

7.8 The kern of an *I*-section is
a. of diamond shape within the section depth
b. of diamond shape within the middle third of the width and depth
c. a symmetric square of one third the width
d. a symmetric rectangle of one third the depth and width

7.9, The kern of a *T*-section is
a. of diamond shape within the section depth
b. of diamond shape within the middle third of the width and depth
c. of symmetric diamond shape of one third the width
d. a symmetric rectangle of one third the depth and width

7.10 Prestressing of a section improves its
a. moment capacity b. axial compressive load capacity
c. shear capacity d. axial tensile load capacity

7.11 Prestressing is adopted for
a. compression members only b. tension members only
c. flexural members only d. any member

Chapter 8

8.1 Torsion equation is valid for
a. any cross section b. prismatic shafts c. circular prismatic sections d. rectangular sections

8.2 Torsion equation can be applied to
a. compound tubes b. isotropic material c. elliptic and circular sections d. rectangular sections

8.3 Torsion equation cannot be applied to non-circular sections because
a. shear stress distribution is not uniform b. shear stress distribution is non-linear
c. torque is not uniform d. they warp

8.4 A section warps when
a. longitudinal strains occur b. longitudinal strains are not uniform
c. a section is unsymmetric d. a section is non-homogenous

8.5 Torsion equation is not valid for
a. curved shafts **b.** hollow shafts **c.** non-metallic shafts **d.** non-prismatic shafts

8.6 A circular shaft under torsional loading is subjected to
a. only shear stresses **b.** shear and axial stresses **c.** only axial stresses **d.** compressive stresses

8.7 A non-circular shaft under torsional loading is subjected to
a. only shear stresses **b.** shear and axial stresses **c.** only axial stresses **d.** compressive stresses

8.8 The torsional rigidity of a thin circular tube is
a. $2\pi R^3t$ **b.** $2\pi R^3tG$ **c.** $2\pi D^3tG$ **d.** πR^3tE

8.9 A thin circular tube is likely to fail
a. in shear **b.** in tension **c.** in compression **d.** by buckling

8.10 The ratio of torsional rigidities of a closed tube and an open tube of same dimensions is
a. 1.0 **b.** < 1.0 **c.** > 1.0 **d.** >> 1.0

8.11 The stresses in a closed tube are smaller than those in an open tube of the same dimensions because of
a. shear flow **b.** buckling **c.** compression **d.** tension

8.12 For the same area and depth, the ratio of torsional rigidities of a flanged section and rectangular section is
a. 1.0 **b.** < 1.0 **c.** > 1.0 **d.** >> 1.0

8.13 The equivalent torsion of a combined torsion and bending moment system induces the same
a. deformation **b.** shear stress **c.** bending stress **d.** shear and bending stresses

8.14 The equivalent moment of a combined torsion and bending moment system induces the same
a. deformation **b.** shear stress **c.** bending stress **d.** shear and bending stresses

8.15 The average diameter, when adopted in the analysis of a non-prismatic circular shaft, leads to deformations that are
a. approximately correct **b.** too small **c.** dependent upon the shaft length **d.** too large

8.16 The shear stress trajectories in a shaft subjected to torsion are
a. straight lines **b.** straight lines and circles **c.** helical **d.** parabolic

8.17 The direct stress trajectories in a shaft subjected to torsion are
a. straight lines **b.** straight lines and circles **c.** helical **d.** parabolic

8.18 The maximum power a shaft can transmit is proportional to
a. shaft length **b.** shaft diameter **c.** cube of shaft diameter **d.** allowable stress

8.19 Non-circular shafts under torsion are subjected to
a. warping and longitudinal stresses **b.** warping but no longitudinal stresses
c. longitudinal stresses but no warping **d.** no longitudinal stresses and no warping

8.20 For a given cross-sectional area, torsional stresses are the least in
a. an I-section **b.** a square section **b.** an elliptical section **d.** a circular section

8.21 For a given cross sectional area, torsional stresses are the least in a hollow shaft of
a. rectangular section **b.** square section **c.** elliptical section **d.** circular section

8.22 The intermediate webs are free from torsional stresses in
a. two-cell sections **b.** three-cell sections
c. multi-cell sections **d.** symmetric multi-cell sections

Chapter 9

9.1 The governing differential equation of a beam is the relationship between its
a. deflection and applied load **b.** deflection and bending moment
c. bending moment and flexural rigidity **d.** bending moment and shear force

9.2 At a simple support
a. bending moment and slope are zero
b. bending moment is zero but not the slope
c. shear force is zero but not the bending moment
d. bending moment and deflection are zero

9.3 At a fixed support
a. bending moment and slope are zero
b. slope is zero but not bending moment
c. shear force is zero but not bending moment
d. slope, and deflection are zero

9.4 At the free end of a beam
a. bending moment and shear force are zero
b. bending moment is zero but not the slope
c. shear force is zero but not the bending moment
d. bending moment and deflection are zero

9.5 At the section of symmetry
a. bending moment and slope are zero
b. slope is zero but not the bending moment
c. shear force is zero but not the bending moment
d. bending moment and deflection are zero

9.6 At the section of antimetry
a. bending moment and slope are zero
b. bending moment is zero but not the slope
c. shear force is zero but not the bending moment
d. bending moment and deflection are zero

9.7 The maximum deflection in a cantilever of 3.0 m span with a load of 100.0 kN at the free end is
a. 1 350.0/(EI) b. 900.0/(EI) c. 450.0/(EI) d. 500.0/(EI)

9.8 The maximum slope in a simple beam of 10.0 m span subjected to a u.d.l. of 12.0 kN/m is
a. 1 350.0/(EI) b. 900.0/(EI) c. 450.0/(EI) d. 500.0/(EI)

9.9 Compared to a non-prismatic beam, a similar beam of its mean cross section deflects
a. more b. approximately the same c. less d. depends upon the beam profile

9.10 Compared to a non-prismatic beam, a similar beam of its minimum cross section deflects
a. more b. approximately the same c. less d. depends upon the beam profile

9.11 Compared to a non-prismatic beam, a similar beam of its maximum cross section deflects
a. more b. approximately the same c. less d. depends upon the beam profile

9.12 Elastic curve of a beam is its
a. stress–strain curve b. load–displacement curve
c. deflection curve d. bending moment diagram

9.13 Maxwell–Betti's theorem states that
a. $P_i\delta_{ij} = P_j\delta_{ji}$ b. $\delta_{ij} = \delta_{ji}$ c. $P_j\delta_{ij} = P_i\delta_{ji}$ d. $P\delta = M\theta$

9.14 Maxwell's theorem states that
a. $P_i\,\delta_{ij} = P_j\,\delta_{ji}$ b. $P_j\,\delta_{ij} = P_i\,\delta_{ji}$ c. $\delta_{ij} = \delta_{ji}$ d. $P\delta = M\theta$

9.15 Curvature of a beam at a section is
a. $M/(EI)$ b. reciprocal of the radius of curvature
c. $(EI)/M$ d. first differential of the elastic curve

9.16 Bending theory assumes that the radius of curvature of a beam is large compared to its
a. strains b. deflections c. depth d. span

9.17 A symmetric uniformly distributed load of the same magnitude as a central point load induces
a. larger stresses b. smaller deflections c. smaller curvature d. larger shear forces

9.18 Macaulay's parantheses imply that the expression within is
a. always positive b. zero, if negative c. always zero d. zero, if positive

9.19 Moment area method is valid for
a. simple beams b. determinate beams
c. continuous beams d. beams with continuous elastic curve

9.20 Moment area method yields the
a. deflections at a section b. elastic curve equation
c. bending moment at a section d. slopes at supports

9.21 The second moment area theorem yields
a. only deflections at a section b. slopes and deflections at a section
c. bending moment at a section d. only slopes at supports

9.22 The first moment area theorem yields
a. only deflections at a section b. slopes and deflections at a section
c. only slopes at a section d. only slopes at supports

9.23 In the conjugate beam method
a. spans are transformed b. supports and loading are transformed
c. spans and supports are transformed d. loading and displacements are transformed

9.24 In the conjugate beam method, an intermediate shear hinge is transformed to
a. an intermediate support b. a moment hinge c. continuity with a moment d. a shear hinge

9.25 In the conjugate beam method, an intermediate moment hinge is transformed to
a. an intermediate support b. a moment hinge c. continuity with a moment d. a shear hinge

9.26 In the conjugate beam method, an intermediate support is transformed to
a. an intermediate support b. a moment hinge c. continuity with a moment d. a shear hinge

9.27 In the conjugate beam method, a free end is transformed to
a. a free end b. a fixed support c. a simple support d. a shear hinge

9.28 Shear component of beam deflections is large in
a. shallow beams b. I-sections c. deep beams d. T-beams

9.29 Shear deformations can be considered in the
a. moment area method b. conjugate beam method
c. energy method d. direct integration method

9.30 Energy method are applied in order to determine
a. elastic curves b. only slope at a section
c. slope and deflection at a section d. deflections only in indeterminate beams

Chapter 10

10.1 The maximum slope in a fixed beam occurs
a. at its supports b. at its mid-span section
c. near the point of inflection d. at its quarter span section

10.2 The ratio of maximum deflections in a fixed beam and a similar simple beam under u.d.l. is
a. 0.50 b. 0.20 c. 5.0 d. 2.0

10.3 The ratio of maximum deflections in a fixed beam and a similar simple beam under a central point load is
a. 0.20 b. 0.25 c. 3.0 d. 4.0

10.4 Points of inflection indicate the sections with
a. maximum curvature b. minimum curvature c. zero curvature d. changed curvature

10.5 Points of contraflexure are the sections with
a. zero bending moments b. zero shear forces c. zero curvature d. changed curvature

10.6 Indeterminate beams
a. are continuous b. have fixed supports
c. cannot be analysed d. require compatibility conditions

10.7 Compared to determinate beams, indeterminate beams
a. have smaller deflections b. are sensitive to loading
c. are sensitive to support sinking d. have larger stresses

10.8 Sinking of supports in indeterminate beams
a. increases support moments and reduces span moments
b. reduces support moments and increases span moments
c. redistributes moments
d. increases span and support moments

10.9 The method of flexibility coefficients cannot be applied to
a. determinate beams b. indeterminate beams
c. non-prismatic beams d. beams with intermediate hinges

10.10 The theorem of three moments cannot be applied to
a. determinate beams b. indeterminate beams
c. non-prismatic beams d. beams with intermediate hinges

10.11 The method of flexibility coefficients can be applied to
a. determinate beams b. indeterminate beams
c. non-prismatic beams d. beams with intermediate hinges

10.12 Compatibility conditions pertain to
a. loading on a beam b. bending moments c. displacements d. shear forces

10.13 The method of flexibility coefficients formulates
a. equilibrium conditions b. compatibility conditions
c. flexibility conditions d. loading conditions

10.14 Clapeyron's method formulates
a. moment conditions b. loading conditions
c. equilibrium conditions d. compatibility conditions

10.15 The reaction at a support that sinks
a. increases b. decreases c. depends upon the spans d. depends upon the load

10.16 The effects due to sinking of supports in indeterminate beams depend upon
a. applied loads b. spans c. beam flexural rigidity d. degree of indeterminacy

10.17 The effects of support sinking can be ignored in
a. determinate beams b. indeterminate beams c. long spans d. short spans

Chapter 11

11.1 Arches are more efficient than beams because of
a. smaller deflections b. larger sections c. smaller bending moments d. axial forces

11.2 The prime requirement of arches is
a. curved profile b. large depth and curved profile
c. large curvature d. strong abutments and curved profile

11.3 Eddy's theorem is applicable to
a. parabolic arches only b. circular arches only c. any arch d. masonry arches only

11.4 Linear arch is advantageous for
a. distributed loads b. point loads c. radial loads d. horizontal loads

11.5 The bending moments in an arch depend upon its
a. span b. loading c. profile d. loading and profile

11.6 Line of thrust in an arch depends upon
a. span b. loading c. profile d. loading and profile

11.7 Pressure line of an arch is a hypothetical arch of the same span and loading in which
a. bending moments are zero b. deflections are zero
c. shear forces are small d. axial thrust is zero

11.8 A parabolic arch under part u.d.l. has at any section zero
a. bending moment **b.** radial shear force **c.** shear force **d.** axial force

11.9 A beam is a
a. horizontal member **b.** flexural member **c.** compression member **d.** inclined member

11.10 A column is a
a. horizontal member **b.** inclined member **c.** compression member **d.** vertical member

11.11 Bay of a frame is the space between two adjacent
a. beams **b.** columns **c.** floors **d.** beams or columns

11.12 Stôrey of a frame is the space between two adjacent
a. beams **b.** columns **c.** floors **d.** beams or columns

Chapter 12

12.1 A truss is a system of linear members with
a. rigid joints **b.** pin joints **c.** welded joints **d.** bolted joints

12.2 A panel of a truss is the space between any two
a. members **b.** joints **c.** lower chord joints **d.** upper chord joints

12.3 A truss comprises
a. triangular spaces **b.** rectangular spaces **c.** any shapes **d.** usually triangular spaces

12.4 A truss member is subjected to
a. axial tension **b.** axial compression or tension
c. axial compression **d.** axial forces and bending moment

12.5 A triangle is the favourable geometric figure for truss systems because it is
a. pin jointed **b.** rigid **c.** strong **d.** flexible

12.6 A triangle is the favourable geometric figure for
a. simple trusses **b.** plane trusses **c.** space trusses **d.** any truss

12.7 The largest force in a pitched roof truss occurs in
a. middle panel members **b.** end panel members **c.** web members **d.** support members

12.8 The largest force in a truss with parallel chords occurs in
a. middle panel members **b.** end panel members **c.** web members **d.** support members

12.9 Members with zero forces are always connected to
a. joints with vertical loads **b.** unloaded joints
c. joints with inclined loads **d.** one unloaded joint

12.10 Hinged joints of trusses ensure
a. axial forces in members **b.** simple fabrication
c. load transfer to joints **d.** load transfer to members

12.11 The primary characteristic of a truss is
a. its triangular configuration **b.** loads at the joints
c. axial forces in members **d.** hinged joints

12.12 Tension coefficient of a member indicates
a. tension in the member **b.** force in a member
c. force per unit deformation **d.** force per unit length

12.13 Horizontal component of the force in a member is the product of
a. force and cosine of the angle
b. force and the horizontal projection of length
c. force and the vertical projection of length
d. tension coefficient and horizontal projection of length

12.14 The method of tension coefficients is based upon
a. equilibrium conditions b. method of joints c. compatibility conditions d. method of sections

12.15 Pitch of a truss is the
a. ratio of height to span b. height c. ratio of height to half span d. spacing of trusses

12.16 The degree of redundancy of plane trusses is given by
a. $m + r - 2j$ b. $m + r - 3j$ c. $3j - m - r$ d. $2j - m - r$

12.17 The degree of redundancy of space trusses is given by
a. $m - r - 3j$ b. $m + r - 3j$ c. $2j - m - r$ d. $3j - m - r$

12.18 If $(2j - m - r)$ is greater than zero for a plane truss, the truss is
a. redundant b. determinate c. stable d. unstable

12.19 If $(m + r - 2j)$ is greater than zero for a plane truss, the truss is
a. redundant b. determinate c. stable d. unstable

12.20 If $(3j - m - r)$ is greater than zero for a space truss, the truss is
a. redundant b. determinate c. stable d. unstable

12.21 If $(3j - m - r)$ is less than zero for a space truss, the truss is
a. redundant b. determinate c. stable d. unstable

12.22 The forces in collinear members at the joint of a plane truss are equal, if the joint is
a. not loaded b. not loaded and has only three members
c. loaded d. loaded and has only three members

12.23 The force in a member is equal to the applied load, if the load is along the member axis and the other two members are
a. collinear b. collinear and not loaded c. non-collinear d. non-collinear and loaded

12.24 The force in a member is equal to the applied load, if the load is along the member axis and the other three members are
a. collinear b. collinear and not loaded c. non-coplanar d. coplanar

12.25 The force in the non-coplanar member at the joint of a space truss is zero, if the joint is
a. not loaded b. not loaded and has only three members
c. loaded d. loaded and has only four members

12.26 If all the members, except one, are coplanar at the joint of a truss, the force in the non-coplanar member is zero if the
a. joint is not loaded b. forces at the joint are also coplanar
c. forces at the joint are coplanar d. forces at the joint are collinear

12.27 The method of tension coefficients was developed by
a. Clapeyron b. Mohr c. Williot d. Southwell

Chapter 13

13.1 The ratio of longitudinal to hoop stresses in a thin cylinder is
a. > 1.0 b. < 1.0 c. $= 1.0$ d. $\simeq 1.0$

13.2 In a cylinder, plane sections remain plane after loading, if
a. the ends are flat b. the ends are spherical
c. the load is radial d. the load is internal

13.3 The ratio of longitudinal to hoop stresses in a sphere is
a. > 1.0 b. < 1.0 c. $= 1.0$ d. $\simeq 1.0$

13.4 The ratio of the cylinder thickness to end sphere thickness to avoid distortion at the joints is (Poisson's ratio = 0)
a. 2.0 b. 0.5 c. 17/7 d. 7/17

13.5 The volumetric strain of a thin cylinder with flat ends is the sum of
a. longitudinal and hoop strains **b.** longitudinal and diametrical strains
c. hoop strain and twice the longitudinal strain **d.** longitudinal strain and twice the hoop strain

13.6 The volumetric strain of a thin spherical shell is
a. three times the hoop strain **b.** twice the hoop strain
c. four times the diametrical strain **d.** three times the diametrical strain

13.7 One of the assumptions of thick cylinder analysis is that
a. longitudinal strain is constant **b.** longitudinal stress is constant
c. hoop stress is constant **d.** hoop strain is constant

13.8 The maximum hoop stress in a thick cylinder occurs on the inner fibre for
a. internal pressure **b.** external pressure **c.** radial pressure **d.** any loading

13.9 In a thick cylinder the ratio of hoop stress to applied pressure is numerically
a. > 1.0 **b.** < 1.0 **c.** $= 1.0$ **d.** $\simeq 1.0$

13.10 In a thick cylinder the ratio of hoop stress to applied pressure numerically tends to unity when the
a. inner diameter is large **b.** outer diameter is large
c. thickness is small **d.** inner diameter tends to zero

13.11 The ratio of maximum stresses estimated by thick cylinder theory to that by thin cylinder theory is
a. > 1.0 **b.** < 1.0 **c.** $= 1.0$ **d.** $\simeq 1.0$

13.12 Prestressing of thin cylinders
a. decreases hoop stresses but increases longitudinal stresses
b. increases hoop stresses but decreases longitudinal stresses
c. decreases hoop stresses
d. decreases longitudinal stresses

13.13 Diametral interference in compound cylinders is the difference of
a. inner and outer diameters **b.** interface stresses **c.** interface diameters **d.** interface strains

13.14 The maximum hoop stress in a solid shaft is
a. equal to the applied radial stress **b.** twice the applied radial stress
c. Poisson's ratio times the applied radial stress **d.** equal to the longitudinal stress

13.15 Membrane stresses are
a. stresses in a fabric
b. stresses in a thin shell
c. stresses in the neutral plane of a thin shell
d. stresses normal to the neutral plane of a thin shell

13.16 In a spherical shell under a crown load, hoop and longitudinal stresses are
a. equal **b.** equal but of opposite nature
c. unequal **d.** unequal and of the same nature

13.17 In a spherical shell under selfweight, the hoop and longitudinal stresses are
a. equal **b.** equal but of opposite nature
c. unequal **d.** unequal and of the same nature

13.18 In a spherical shell under snow loading, the hoop and longitudinal stresses are
a. equal **b.** equal but of opposite nature
c. unequal **d.** unequal and of the same nature

13.19 In a spherical shell under selfweight, the hoop stresses
a. are constant **b.** are always compressive
c. are always tensile **d.** change sign for large shell angles

13.20 Plane of rupture is the plane in a spherical shell about which
a. hoop stresses change sign from compressive to tensile
b. hoop stresses change sign from tensile to compressive
c. longitudinal stresses change sign
d. longitudinal and hoop stresses change sign

13.21 The ratio of the cylinder thickness to end sphere thickness for equal maximum stresses in them is
a. 2.0 b. 0.5 c. 17/7 d. 7/17

13.22 In a thick cylinder, the hoop stress
a. increases from inner to the outer surfaces b. decreases from inner to the outer surfaces
c. is constant over the thickness d. varies linearly over the thickness

13.23 In a thick cylinder, the sum of hoop and radial stresses is
a. equal to zero b. variable c. constant d. constant for internal pressure

13.24 In a cylinder, circumferential prestressing
a. reduces hoop stresses but increases longitudinal stresses
b. reduces hoop and longitudinal stresses
c. increases hoop stresses but reduces longitudinal stresses
d. reduces hoop stresses

13.25 In a cylinder, longitudinal prestressing
a. reduces hoop stresses but increases longitudinal stresses
b. reduces hoop and longitudinal stresses
c. increases hoop stresses but reduces longitudinal stresses
d. reduces longitudinal stresses

Chapter 14

14.1 Buckling of columns is also known as
a. inelastic stability b. elastic stability c. lateral instability d. longitudinal instability

14.2 Euler crippling load of a column is
a. equal to its compressive strength b. greater than its compressive strength
c. less than its compressive strength d. may exceed its compressive strength

14.3 Slenderness ratio of a column is the ratio of
a. Euler crippling load to its compressive strength
b. its length to the maximum radius of gyration
c. its length to the minimum radius of gyration
d. its length to the minimum lateral dimension

14.4 Euler's curve is the graphical representation of the relationship between
a. slenderness ratio and radius of gyration b. slenderness ratio and crippling load
c. compressive strength and Euler's load d. slenderness ratio and compressive stresss

14.5 Euler's theory is applicable to
a. long columns b. short columns c. any column d. columns of intermediate length

14.6 Columns of intermediate length fail by
a. buckling b. buckling or crushing c. crushing d. buckling and crushing

14.7 Equivalent eccentricity of columns considers
a. initial curvature b. initial eccentricity c. actual eccentricity d. final eccentricity

14.8 Rankine–Gordon's empirical formula considers
a. buckling mode b. crushing mode c. eccentric loading d. buckling and crushing modes

14.9 Johnson's straight line formula is unsafe for
a. small slenderness ratios b. large slenderness ratios c. short columns d. long columns

14.10 Johnson's parabolic formula is unsafe for
a. small slenderness ratios **b.** large slenderness ratios **c.** short columns **d.** long columns

14.11 Rankine–Merchant formula is valid for
a. small slenderness ratios **b.** large slenderness ratios
c. short columns **d.** long columns

14.12 Perry–Robertson's formula considers
a. buckling mode **b.** initial curvature **c.** crushing mode **d.** initial eccentricity and curvature

14.13 Non-prismatic struts can be analysed by
a. Euler's theory **b.** Perry–Robertson's method
c. energy method **d.** Rankine–Gordon's formula

14.14 The ratio of failure loads of a long non-prismatic strut to that of a similar column with the minimum cross section is
a. > 1.0 **b.** < 1.0 **c.** $= 1.0$ **d.** $\simeq 1.0$

14.15 The ratio of failure loads of a long non-prismatic strut to that of a similar column with the maximum cross section is
a. > 1.0 **b.** < 1.0 **c.** $= 1.0$ **d.** $\simeq 1.0$

14.16 Compared to a column with hinged supports, the load capacity of a similar column with partially restrained supports is
a. reduced **b.** increased **c.** unchanged **d.** depends upon the restraint

14.17 IS:800–1984 recommends, for the design of columns, the formula of
a. Euler **b.** Rankine–Gordon **c.** Rankine–Merchant **d.** Perry–Robertson

14.18 Rankine–Gordon's formula is
a. conservative for long columns **b.** unsafe for long columns
c. conservative for columns of intermediate length
d. unsafe for columns of intermediate length

14.19 Johnson's parabolic formula is
a. conservative for long columns **b.** unsafe for long columns
c. conservative for columns of intermediate length
d. unsafe for columns of intermediate length

14.20 Equivalent length of columns with one end fixed and the other end hinged is
a. $0.707\,l$ **b.** $0.500\,l$ **c.** $0.85\,l$ **d.** $2.0\,l$

14.21 Equivalent length of columns with one end fixed and the other end free is
a. $0.707\,l$ **b.** $0.500\,l$ **c.** $0.85\,l$ **d.** $2.0\,l$

14.22 As per IS:800–1984, the equivalent length of columns with one end fixed and the other end hinged is
a. $0.707\,l$ **b.** $0.500\,l$ **c.** $0.85\,l$ **d.** $2.0\,l$

14.23 Equivalent length of columns with both ends fixed is
a. $0.707\,l$ **b.** $0.500\,l$ **c.** $0.85\,l$ **d.** $2.0\,l$

14.24 As per IS:800–1984, the equivalent length of columns with both ends fixed is
a. $0.65\,l$ **b.** $0.50\,l$ **c.** $0.85\,l$ **d.** $2.00\,l$

14.25 Critical length of a column is the length for which
a. buckling does not occur **b.** buckling strength is equal to the crushing strength
c. buckling strength is more than crushing strength
d. buckling strength is less than crushing strength

14.26 Secant formula is applicable to
a. long columns **b.** short columns with large eccentricity
c. long columns with eccentric loading **d.** columns of intermediate length

14.27 Johnson's straight line formula is applicable to
a. long columns b. long columns with eccentric loads
c. short columns d. columns of intermediate length

14.28 Euler's theory is not applicable to
a. long columns b. columns with eccentric loading
c. short columns d. columns of intermediate length

14.29 Bending moment in a long column is
a. Pe b. $P\sin(kl)$ c. $Pe\sec(kl/2)$ d. $Pe\sec(kl)$

Chapter 15

15.1 Springs are elements to
a. absorb energy b. reduce impact c. reduce deflections d. store energy

15.2 The ratio of the deflection of a close coiled helical spring to that of a similar open coiled helical spring
a. > 1.0 b. < 1.0 c. $= 1.0$ d. $\simeq 1.0$

15.3 Close coiled helical springs under axial load are subjected to
a. axial deflection and torsional moments b. axial deflection, rotation and shear stresses
c. axial deflection and shear stresses d. axial deflection and bending stresses

15.4 Open coiled helical springs under an axial load are subjected to
a. axial deflection and torsional moments
b. axial deflection, rotation, and shear and bending stresses
c. axial deflection, rotation and shear stresses
d. axial deflection, rotation and bending stresses

15.5 A close coiled helical spring under axial load is subjected to shear stresses because
a. extension of the spring causes rotation of coils
b. axial load induces direct shear in the spring
c. the spring coils are subjected to shear force
d. the bending of coils causes flexural shear

15.6 Close coiled helical springs under axial torsion are subjected to
a. axial deflection and torsional moments b. rotation and shear stresses
c. rotation and bending stresses d. rotation, torsional and bending stresses

15.7 Open coiled helical springs under torsional moment are subjected to
a. axial deflection and torsional moments
b. axial deflection, rotation, and shear and bending stresses
c. axial deflection, rotation and shear stresses
d. axial deflection, rotation and bending stresses

15.8 Wahl's correction for shear stresses in helical springs considers
a. direct and torsional shear effects
b. direct and torsional shear effects, and bending moments
c. direct shear stress distribution and torsional effects
d. torsional and bending effects

15.9 Wahl's correction is applicable to
a. light springs b. heavy springs c. open coiled springs d. close coiled springs

15.10 Spring index is the ratio of
a. spring diameter to that of the coil wire
b. coil wire diameter to that of the spring

c. applied load and the corresponding deflection
d. deflection and the corresponding applied load

15.11 Flat spiral springs are adopted to
a. store energy b. absorb shocks c. release energy d. absorb impact

15.12 Springs placed side by side and loaded
a. deflect more than individual springs
b. deflect less than individual springs
c. are stiffer than individual springs
d. are less stiff than individual springs

15.13 Springs placed end to end and loaded
a. deflect more than individual springs
b. deflect less than individual springs
c. are stiffer than individual springs
d. are less stiff than individual springs

15.14 Proof load of a leaf spring is the
a. load per unit deflection
b. deflection per unit load
c. load required to flatten the spring
d. load required to cause 2.0 percent permanent deflection

15.15 Strain energy of a helical spring is
a. $[\tau^2/(4G)] \times$ volume b. $P\delta$ c. $[\tau^2/(2G)] \times$ volume d. $[P\delta/2]$

Answers to short questions

The correct choices to the short questions of Appendix B are listed chapter wise in a tabular form. The position of the correct answer indicates the corresponding question. The first row indicates the questions with single digit numbers; subsequent rows pertain to successive groups of tens. Some questions have more than one correct answer.

Chapter I

	1	2	3	4	5	6	7	8	9	10
0	d	a,b	a,d	b	d	b	a	b	d	a
10	d	d	b	a	c	c	d	a	a,d	b,d
20	d	d	b,c,d	c,d	b,d	b,c,d	a	a,c	a,d	a,d
30	c	c	b	b	a,b					

Chapter 1

	1	2	3	4	5	6	7	8	9	10
0	b,d	d	a,c	a,d	a,b	a	b	d	c,d	a,b
10	b	d	d	a	d	c	d	c	c	c,d
20	a,d	b,c	a,c,d	b,c	b,d	b	b	a	c	c
30	d	b	c	a	b	c,d	a	b	c	d
40	b	a,d	a	d	a	b,c	a,b	a,d	d	b
50	b	d	c	b,c,d						

Chapter 2

	1	2	3	4	5	6	7	8	9	10
0	b	a	a,d	a,c	d	d	a,c,d	b,c	d	b
10	b,c	b,d	b	a,b	d	c,d	a,b	a		

Chapter 3

	1	2	3	4	5	6	7	8	9	10
0	b	a,c	a	a,c	c	b,d	c	d	d	c
10	a	c,d	b,d	b	a	b	c	a,b	c	b
20	a	b								

Chapter 4

	1	2	3	4	5	6	7	8	9	10
0	c	a,d	c	c	b	c	a	d	a	d
10	a	c	c	a,b	d	c	a	a	b	a,c
20	c	c	c	c	a,b	c,d				

Chapter 5

	1	2	3	4	5	6	7	8	9	10
0	b	b,c	d	a,b,c	a,b,d	a,b	b,c,d	d	a	a,c
10	d	b,c	a,c	a,b,d						

Chapter 6

	1	2	3	4	5	6	7	8	9	10
0	b	d	a	b	c	c	d	c	c	b,d
10	c	d	a	c	c	a,d	a	c	b	a
20	d	a	c	c	c	a	b,c	a,d	a,b	d
30	c	d	d	a	c,d	c,d	c	c	c	a
40	d	a,c	c	d	b,c	a,c	b	d	b	a

Chapter 7

	1	2	3	4	5	6	7	8	9	10
0	b	b,c,d	c,d	d	d	a	b	a	a	a,c,d
10	d									

Chapter 8

	1	2	3	4	5	6	7	8	9	10
0	c	b	d	a	a,d	a	a	b	d	d
10	a	b	b	c	b	b	c	c,d	b	d
20	d	d								

Chapter 9

	1	2	3	4	5	6	7	8	9	10
0	b	b,d	b,d	a	b,c	b,d	b	d	d	a
10	c	c	a	c	a,b	d	b,c	a,b	a	a,d
20	b	d	b,d	c	a	b	b	c	c	c

Chapter 10

	1	2	3	4	5	6	7	8	9	10
0	c	b	b	c,d	a,c,d	d	a,c	b,c	a	a,c,d
10	b,c,d	c	b	d	b	b,c,d	a			

Chapter 11

	1	2	3	4	5	6	7	8	9	10
0	c	d	c	b	d	d	a	b	b	c
10	b	c								

Chapter 12

	1	2	3	4	5	6	7	8	9	10
0	b	c,d	a	a,b,c	b	d	b	a,b	d	a
10	c	d	a,d	a,b	a	a	b	d	a	d
20	a	b	a	d	a	a,b	d			

Chapter 13

	1	2	3	4	5	6	7	8	9	10
0	b	a	c	a	d	a	a	d	a	b
10	a	c	c	a	c	b	c	c	d	a
20	a	b	c	d	d					

Chapter 14

	1	2	3	4	5	6	7	8	9	10
0	c	c	c	b	a	b	a	d	a,b	b
10	d	d	c	a	b	b	c	c	b	a
20	d	c	b	a	b	c	b	b,c	c	

Chapter 15

	1	2	3	4	5	6	7	8	9	10
0	a,b,d	b	a,c	b	a,b	c	b	c	b,d	a
10	a,c	b,c	a,d	c	a,d					

Answers to selected problems

Chapter 1

1.28 E = 211.6 GPa and σ_w = 240.2 MPa **1.29** f.o.s = 1.72 and P_a = 1.39 MPa
1.30 a = 182.6 mm and P_a = 415.8 kN **1.31** d = 40.0 mm
1.32 δ = 147.6 mm and f.o.s = 1.68 **1.33** P_a = 6.79 kN
1.34 The stresses are 916.7, 707.4 and 480.0 MPa; P_{max} = 13.35 kN
1.35 d = 33.3 mm (35.0 mm), allowable length = 1 173.6 mm **1.36** 28 strands
1.37 Fig.E 1.1 (a) – 2.07 mm (b) – 0.80 mm (c) + 9.06 mm (d) + 0.30 mm
1.38 Fig.E 1.1 (b) 166.7 mm (c) 2 287.0 mm (d) 1 004.3 mm
1.39 Fig.E 1.1 (b) 11.55 mm and 222.2 mm **1.40** Fig.E 1.1 (d) 22.02 mm, and x = 362.0 mm
1.41 Fig.E 1.1(e) 0.51 mm (f) 0.54 mm **1.42** $\delta = 4Pl/(\pi EDd)$ **1.43** $\delta = wl^2/(2EA)$
1.44 $A(x) = Ae^{wxP/A}$ **1.45** (i) 29.8 revolutions / second (ii) 43.6 revolutions/second

1.46	Fig.E 1.2	(a) – 77.1, + 59.0	(b) – 127.5, + 97.5
	Forces in kN	(c) – 22.7, + 0.0	(d) – 435.3, + 332.9
1.47	Fig.E 1.3	(a) + 1.18 kN in both the wires	
		(b) + 0.66 kN in steel wire and + 1.31 kN aluminium wire	

1.48 (i) 2.14 kN (ii) – 0.28 MPa (iii) – 2.50 mm and + 4.75 mm **1.49** 44.4 percent

1.50	Fig.E 1.4	(a) 6.92 and 55.39	(b) 17.93 and 104. 58
	Forces in kN	(c) 95.75 and 568.13	(d) 25.93 and 207.41
1.51	Fig.E 1.5	(a) 100.0, 150.0 and 100.0	(b) 153.9, 192.3 and 153.9
	Forces in kN	(c) 71.4, 107.4 and 71.4	(d) 152.0, 116.0 and 32.0
1.52	Fig.E 1.5	(a) 58.3, 233.3 and 58.3	(b) 93.8, 312.5 and 93.8
	Forces in kN	(c) 41.7, 166.7 and 41.7	(d) 120.0, 180.0 and 0.0

1.53 Fig.E 1.6 (a) (i) 88.8, 111.1 and 88.8 (ii) 39.8, 198.9 and 39.8 (iii) 128.5, 40.2 and 128.5
Forces in kN (b) (i) 209.1, 302.0 and 209.1 (ii) 87.3, 504.7 and 87.3 (iii) 321.0, 115.9 and 321.0
1.54 81.4 kN, 10.9 percent and 1.1 percent
1.55 (i) σ_{al} = + 13.4 MPa and σ_c = – 5.0 MPa (ii) σ_{al} = – 4.9 MPa and σ_c = + 1.9 MPa
1.56 Fig.E 1.7 stresses in MPa
(a) (i) σ_c = – 58.9 and σ_s = –530.0 (ii) σ_c = + 25.5 and σ_s = +229.70
(b) (i) σ_c = –426.6 and σ_s = – 47.4 (ii) σ_c = +184.9 and σ_s = + 20.5
1.57 Fig.E 1.8 (a) – 17.6, – 39.7 and – 17.6 MPa (b) –176.7, – 46.6 and –176.7 MPa
(c) – 9.2, – 35.0 and – 30.8 MPa (d) –174.0, – 88.9 and –137.1 MPa
1.58 Fig.E 1.8 (b) 1.18 mm
1.59 Fig.E 1.8 (a) – 29.3, – 16.5 and – 29.3 MPa (b) –191.0, – 18.1 and –191.0 MPa
(c) – 20.5, – 13.6 and – 40.9 MPa (d) –205.3, – 34.7 and –160.0 MPa
1.60 Fig.E 1.1 (e) – 75.0 mm^3 (f) + 72.9 mm^3 **1.62** 22.23 m^3 **1.63** 0.012 6 mm^3/m^2
1.64 30.0 GPa, – 28.1 MPa and – 0.000 71 **1.65** 272.0 MPa, 0.001 26 and 63.58 GPa
1.66 σ = 180 000.0 ε (1 – 150.6 ε)

Chapter 2

2.8

	Mild steel	Brass	Aluminium	Cast iron
		0.036	0.026	0.008
Modulus of resilience (Nmm/mm^3)	0.144	25.5	12.0	1.2
Modulus of toughness (Nmm/mm^3)	83.9			

2.9 44.39 Nmm and 147.81 Nmm **2.10** 0.156 Nmm/mm^3, 71.09 Nmm/mm^3 and 4.02 kNm
2.11 2.9 m long bar; 3.6 m long bar
2.12 The impact resistance of the aluminium bar is 2.04 times that of the steel bar
2.13 7.56 MPa and 1.70 mm
2.14 (i) 75.3 MPa, 1.13 mm and 90.5 (ii) 191.8 MPa, 1.80 mm and 58.6 **2.15** 382.33 mm
2.16 374.81 mm **2.17** Bar diameter required (i) 95.37 mm (ii) 95.84 mm
2.18 (i) 35.21 kN (ii) 16.33 kN **2.19** The bar yields but does not break
2.20 401.1 MPa and 83.56 mm **2.22** 212.3 MPa and 106.1 MPa, and 1.91 mm

Chapter 3

3.8 130.7 kN **3.9** 10.0 mm and 1 333.33 MPa **3.10** 20.2 mm **3.11** 124.3 MPa
3.17 (a) 254.6 kN (b) 360.0 kN **3.19** Factor of safety = 1.41
3.20 118.4 MPa in shear and 102.3 MPa in bearing **3.21** 5.56 MPa and 2.025 MN

Chapter 4

4.11

	Principal stresses (MPa)		Principal planes (degrees)	
(i)	+ 512.4	+ 97.6	+ 37.3	+ 127.3
(ii)	+ 309.7	− 419.7	− 16.6	+ 73.4
(iii)	− 97.6	− 512.4	+ 52.7	− 37.3
(iv)	+ 419.7	− 309.7	+ 16.6	+ 106.6

Stress (MPa)
Inclination of plane (degrees)

	45		60	
(i)	+ 105.5	+ 55.5	+ 159.3	+ 147.6
(ii)	− 255.0	− 305.0	− 380.7	− 164.1
(iii)	− 505.07	− 55.0	− 505.7	+ 52.4
(iv)	− 145.0	+ 305.0	+ 34.3	+ 364.1

4.12 $\sigma_1 = +164.9$ MPa, $\sigma_2 = -184.9$ MPa, $\theta_1 = -29.5°$ and $\theta_2 = +60.4°$
$\sigma_{-60} = +74.9$ MPa, $\tau_{-60} = +152.9$ MPa, $\sigma_{30} = -94.9$ MPa and $\tau_{30} = -152.9$ MPa
4.13 (i) 134.2 MPa (ii) 120.4 MPa **4.14** $\sigma_1 = +330.1$ MPa, $\sigma_2 = -530.1$ MPa and $\tau_{max} = 315.1$ MPa
4.15 $\sigma_z = +488.7$ MPa **4.16** $\sigma_1 = +316.6$ MPa, $\sigma_2 = -447.5$ MPa, $\theta_1 = 27.8°$ and $\theta_2 = 117.8°$
4.17 $\tau_{max} = 210.2$ MPa **4.18** $\varepsilon_1 = -2.8 \times 10^{-6}$, $\varepsilon_2 = +5.5 \times 10^{-6}$, $\theta_1 = 14.1°$ and $\theta_2 = 104.1°$
4.19 $\theta_1 = -8.1°$, $\theta_2 = +81.9°$ and $\gamma_{max} = 2.08 \times 10^{-6}$ **4.20** $\varepsilon_1 = +2.96 \times 10^{-6}$, $\varepsilon_2 = -9.06 \times 10^{-6}$

Chapter 5

5.5

Fig. No.	M_{max} (kNm)	M_{min} (kNm)	SF_{max} (kN)	SF_{min} (kN)
E 5.1 (a)	+ 3 100.1	0.0	+ 350.0	− 350.0
(b)	+ 847.2	0.0	+ 126.0	− 574.0
(c)	+ 580.0	0.0	+ 165.1	− 234.2
(d)	+ 450.0	0.0	+ 90.0	− 90.0
(e)	+ 457.1	0.0	+ 57.1	− 42.9
(f)	+ 390.6	0.0	+ 140.6	− 140.6
E 5.2 (a)	0.0	− 445.5	0.0	− 54.0
(b)	0.0	− 123.3	0.0	− 35.0
(c)	+ 25.0	0.0	0.0	0.0

(d)	0.0	− 526.5	+ 83.1	0.0
(e)	0.0	− 293.8	0.0	− 75.0
(f)	0.0	− 240.0	+ 70.0	0.0
E 5.3 (a)	+ 61.0	− 25.0	+ 30.5	− 25.7
(b)	+ 17.2	− 3.8	+ 11.1	− 12.3
(c)	+ 11.2	− 43.2	+ 10.2	− 11.0
(d)	+ 158.8	− 121.0	+ 76.7	− 57.8
(e)	+ 37.7	− 18.2	+ 23.5	− 15.5
(f)	+ 10.1	− 23.0	+ 27.6	− 27.6
E 5.4 (a)	+ 61.9	− 22.1	+ 37.4	− 37.4
(b)	+ 37.5	− 22.8	+ 11.8	− 33.2
E 5.5 (a)	+ 158.2	− 49.6	+ 79.0	− 61.1
(b)	+ 172.2	0.0	+ 39.4	− 39.4
E 5.6 (a)	+ 20.2	− 486.0	+ 112.5	− 22.5
(b)	+ 124.4	− 31.1	+ 25.9	− 51.8
E 5.7 (a)	0.0	− 200.0	0.0	− 25.0
(b)	0.0	− 122.2	0.0	− 41.5
(c)	+ 24.3	− 230.4	+ 16.2	− 60.6
(d)	0.0	− 780.0	0.0	− 170.0
E 5.8 (a)	$+Pa$	0.0	$+P$	$-P$
(b)	$+Pab/l$	$-Pab/l$	$+Pa/l$	$-Pb/l$
(c)	+ 61.2	− 1 540.0	+ 35.0	− 195.0
(d)	$+M$	0.0	0.0	0.0
E 5.9 (a)	$+w\,l_1^2/8$	$-w\,l_1^2/8$	$+w\,l_1/2$	$-w\,l_1/2$
(b)	$+P\,l_1/4$	0.0	$+P/2$	$-P/2$

5.6 Fig.E 5.10(a) Simple beam with a 700.0 kN load over the central 4.57 m

(b) Simple beam with a 100.0 kN load and a 200.0 kNm clockwise moment at 6.0 m from the left support

(c) Cantilever beam with the load varying linearly from 17.5 kN/m at the free end to zero at the fixed end

(d) A cantilever beam with a train of 5.0, 10.0, 15.0, 20.0 and 25.0 kN loads

(e) Simple beam with an overhang on the left; 10.4 kN/m load between the supports and an anti-clockwise moment of 50.0 kNm at the left end

(f) Simple beam with an overhang on the left; the load on the overhang varies from zero at the free end to 3.6 kN/m at the support, and a u.d.l. of 3.6 kN/m between the supports

(g) Simple beam with an overhang of 2.5 m on the right; the load comprises an anti-clockwise moment of 11.25 kNm and a downward load of 15.0 kN at the free end, and an upward load of 14.14 kN on the main span

5.7 Fig.E 5.11(a) Simple beam with overhangs on both sides; the main span carries a u.d.l. of 9.0 kN/m, while that on the overhangs varies linearly from 9.0 kN/m at the supports to zero at the free end

(b) Simple beam with overhangs on both the sides; the main span carries a u.d.l. of 10.0 kN/m over its left half, and a 5.0 kN load at the left end and 6.5 kN at the right end

(c) Two-span continuous beam with a shear hinge at the mid span of the right span; the beam carries a u.d.l. of 4.5 kN/m over the entire left span and over the right half of the right span

(d) Simple beam with an anti-clockwise moment of 486.0 kNm at the left support with a u.d.l. of 12.5 kN/m over the entire span
(e) Cantilever frame fixed at the upper end with u.d.l. of 10.0 kN/m over the upper arm, and a vertical load of 50.0 kN and a horizontal load of 38.0 kN at the free end
(f) Simple beam with upward and downward loads of 100.0 kN each
(g) Simple span with a u.d.l. of 15.0 kN/m over the entire span with an upward load of 165.0 kN at 14.0 m from the left support
(h) Simple span with a u.d.l. of 10.0 kN/m over the left 11.0 m, 8.75 kN/m over the right 8.0 m, a load of 50.0 kN at 14.0 m from the left support and an anti-clockwise moment of 1 540.0 kNm at the left support.

Chapter 6

6.9 $I = a^4/12$, where a = length of the side **6.11** $I_x = 636.3 \times 10^6$ mm^4 and $M = 477.2$ kNm
6.12 285.8 mm **6.13** $I_x = 44.44 \times 10^6$ mm^4 and $w = 24.14$ kNm^{-1}
6.14 $I_x = 3.01 \times 10^9$ mm^4, $I_y = 1.893 \times 10^9$ mm^4 and $w = 52.52$ kNm^{-1}
6.15 7.6 percent **6.16** (i) 10.7 percent (ii) 99.9 percent
6.17 One ISWB 450 with 300.0 × 10.9 mm thick plates requires 17.7 percent less material than two ISWB 450, hence is more economical
6.18 $\sigma_t = \pm 11.6$ MPa and $\sigma_s = \pm 156.6$ MPa
6.19 2.44 kNm (i) 4.10 kNm (ii) 0.96 kNm; 3.63 kNm **6.20** 43.73 kNm
6.21 1.02 kNm (i) 38.7 mm (ii) 120.0 mm
6.22 $\sigma_s = +201.1$ MPa and -48.5 MPa, $\sigma_c = -3.6$ MPa and -8.3 MPa
6.23 Maximum allowable load = 73.8 kNm^{-1} **6.24** $\sigma_{max} = 2.74$ MPa and $\sigma_{min} = -2.16$ MPa
6.25 Fig.E 6.9 $l = 13.50$ m Fig.E 6.10 $l = 14.10$ m
6.26 Fig.E 6.11 $\sigma_c = -3.8$ MPa, -6.2 MPa, and $\sigma_s = +159.9$ MPa, -30.9 MPa
Fig.E 6.12 $\sigma_c = -2.3$ MPa, -3.0 MPa, and $\sigma_s = +91.8$ MPa -19.0 MPa
6.27 Fig.E 6.13 $\sigma_{max} = +1.8$ MPa and $\sigma_{min} = -4.1$ MPa
Fig.E 6.14 $\sigma_{max} = +1.8$ MPa and $\sigma_{min} = -3.4$ MPa

6.31

Fig.E	6.1	6.2	6.3	6.4	6.5	6.6
τ_{max} (MPa)	26.1	40.1	66.5	2.6	48.8	23.5

6.32 97.5 percent **6.33** Top flange-web $\tau = 1.14$ MNm^{-1} and bottom flange-web $\tau = 0.88$ MNm^{-1}
6.34 Fig.E 6.9 $\tau_{max} = 0.71$ MPa and Fig.E 6.10 $\tau_{max} = 0.63$ MPa
6.35 Fig.E 6.11 43 studs/m and Fig.E 6.12 24 studs/m
6.36 Fig.E 6.13 $\tau_{max} = 1.88$ MPa and Fig.E 6.14 $\tau_{max} = 0.67$ MPa **6.37** 0.32 mm
6.38 (i) ± 224.0 MPa (ii) 80.0×10^{-6} (iii) 23.33 kNm; ± 280,0 MPa, 80.0×10^{-6}, 23.33 kNm

Chapter 7

7.6 $\sigma_{min} = -22.1$ MPa, $\sigma_{max} = +10.6$ MPa and $P = 164.4$ kN
7.7 The values are from the centroid of the section

Fig. No.	e_x (m)	e_y (m)
E 7.1 (a)	± 0.031	± 0.140
(b)	+ 0.117 – 0.044	± 0.193
(c)	+ 0.039	+ 0.065 – 0.128
(d)	+ 0.113	+ 0.173 – 0.321
(e)	+ 0.073	+ 0.249 – 0.187
(f)	+ 1.237	+ 1.050 – 0.592

7.8 113.6 mm **7.9** D / 4

7.10

Fig. No.	σ_{max} (kPa)	σ_{min} (kPa)
E 7.2 (a)	+ 1.04	− 116.24
(b)	− 38.89	− 79.53
(c)	− 59.78	− 226.84
(d)	+ 16.41	− 213.69

7.11

Fig. No.	σ_{max} (kPa)	σ_{min} (kPa)
E 7.3 (a)	−129.43	− 610.57
(b)	+291.84	−1 956.12
(c)	+276.53	− 863.57
(d)	+1 075.63	−1 484.20

7.12

Fig. No.	σ_{max} (kPa)	σ_{min} (kPa)
E 7.4 (a)	−1 091.85	−3 164.72
(b)	− 840.14	−1 576.09
(c)	− 608.39	−2 493.29
(d)	− 728.07	−1 353.75

7.13 Fig.E 7.5 (a) parallel to the shorter sides = 2.149 MN
parallel to the longer sides = 3.505 MN
(b) parallel to the shorter sides = 1.841 MN
parallel to the longer sides = 2.622 MN

7.14 σ_{max} = + 229.2 MPa, σ_{min} = + 25.0 MPa, E = 118.8 GPa and E = 84.9 GPa, if the eccentricity of the load is ignored

7.15 σ_{max} = − 115.5 kPa, σ_{min} = − 415.3 kPa and the maximum wind force = 3.38 kPa

7.16 σ_{max} = − 195.2 kPa, σ_{min} = − 450.8 kPa and the maximum wind force = 5.05 kPa

7.17 σ_{max} = + 162.5 MPa, σ_{min} = − 27.7 MPa **7.18** σ_{max} = + 533.3 MPa, σ_{min} = − 133.3 MPa

7.19 σ_{max} = + 1.05 MPa, σ_{min} = − 4.05 MPa **7.20** (i) 1.985 MN (ii) 0.496 MN

7.21

Fig. No.	(i) P_{min} (MN)	(ii) w_{max} (kNm^{-1})
E 7.6 (a)	0.567	28.73
(b)	2.703	9.34
(c)	1.599	16.89

7.22 (i) 1.472 MN (ii) 1.442 MN (iii) 1.325 MN (iv) 1.186 MN (v) 0.906 MN (vi) 0.749 MN

Chapter 8

8.12 τ_{max} = 110.0 MPa and θ = 11.3° (i) 5.52 mm (ii) 4.73 mm

8.13 Torsional moment at the origin = $T(l-a)/l$ and Ta/l at the other end **8.14** 211.25 mm

8.15 Torsion in the segments from the left are $[T_1(l-a)-T_2b]/l, -[T_1a+T_2b]/l$ and $T_2-[T_1a+T_2b]/l$

8.16 130.95 mm and 169.96 mm **8.17** 166.91 mm and 10.59°

8.18 (i) 4.04 kNm (ii) 5.22 kNm and (iii) 4.67 kNm

8.19 (i) 188.91 and 157.33 mm (ii) 175.72 and 146.44 mm (iii) 179.91 and 149.97 mm

8.20 2.83 kNm and 9.903 × 10^{-3} radian per metre **8.21** 8.66 kW **8.22** 90.0 mm

8.23 D = 89.9 mm **8.24** 39.79 mm and 49.10 mm **8.25** 41.57 mm and 50.57 mm

8.26 43.34 mm and 159.31 MPa **8.27** 41.66 mm and 191.33 MPa

8.29 0.234 radian and 0.120 radian, 48.7 percent error

8.31 Torsional rigidity in GNmm² Figure E 8.1 (a) 15.04 (b) 4.73 (c) 10.06

8.32 Figure E 8.2 GJ = 0.171 GNm², θ = 0.058 4 × 10^{-3} radian per metre and τ_{max} = 0.18 MPa

8.33 Figure E 8.3 τ_{max} = 28.53 kNm, θ = 2.226 × 10^{-3} radian

8.34 Figure E 8.4 τ_{max} = 0.50 MPa, θ_{max} = 3.782 × 10^{-3} radian

8.35 Figure E 8.5 τ_{max} = 1.96 MPa, θ_{max} = 0.081 × 10^{-3} rad/m
8.36 Figure E 8.6 τ_{max} = 0.52 MPa, θ_{max} = 12.346 × 10^{-3} rad/m
8.37 Figure E 8.7 215.91 Nm and 1.39 mm **8.38** Figure E 8.8 13.61 MNm and 0.497 × 10^{-3} rad/m
8.39 158.5, 124.1, 92.7, 73.6 and 51.0 mm **8.40** 644.3 MPa **8.41** 223.3 mm **8.42** 9.27 kNm
8.43 126.6 mm **8.44** 98.6, 197.3, 394.5, 789.0 and 2 367.2 kW **8.45** Inner diameter = 66.8 mm
8.46 6.875 kNm **8.47** $0.75\,(a/t)^2$; $t/(6a)$ **8.48** 2.04 MPa, 0.277 m **8.49** 69.63 Nm
8.50 4.82 mm **8.51** 348.7 percent, t = 1.21 mm **8.52** 207.2 Nm
8.53 0.89 MPa, 1.033° per metre **8.54** Sleeve thickness = 2.5 mm, efficiency = 92.2 percent
8.55 8 No. 12 mm bolts on 120.0 mm circle
8.56 100.0 mm long key of 20.0 × 20.0 mm section, efficiency = 62.9 percent

Chapter 9

9.8

Fig. No.	$EI\delta_{max}$	Fig. No.	$EI\delta_{max}$
E 9.1	$0.018\,3\,Pl^3$	E 9.4	$wl^4\,(3 - 2\alpha^2)/96$
E 9.2	$0.035\,5\,Pl^3$	E 9.5	$5\,wl^3\,(l + 16)/768$
E 9.3	$wl^4\,(8\alpha - 4\alpha^3 + \alpha^4)/384$	E 9.6	540.0

9.9 δ = 30.81 mm and θ = 0.010 3 radian **9.10** δ = 4.81 mm and θ = 3.844 × 10^{-3} radian
9.11 The required beam size is 116.9 × 233.8 mm (the nearest available size should be adopted)
9.12 δ = 7.73 mm and θ = 2.84 × 10^{-3} radian **9.13** EI = 225.12 MNm^2
9.14 EI = 250.0 MNm^2 **9.17** $\delta_{max} = 4.571 \times 10^{-3} wl^4/(EI)$ at $x = 0.521\,l$
9.18 δ_{min} = − 25.66 mm at 5.774 m from the left support; δ_{max} = + 74.67 mm at the free end
9.19 δ_{max} = + 9.024 mm and δ_{min} = − 9.635 mm
9.20 Prop reactions Fig.E 9.7 (c) 0.633 P (d) 0.594 wl **9.21** $a = 0.378\,l$ **9.22** $a = 0.646\,l$
9.23 $a = 0.456\,l$ **9.24** $a = 0.375\,l$ **9.25** $a = 0.595\,l$
9.26 δ_{max} = + 37.20 mm and δ_{min} = − 31.62 mm
9.27

Fig. No.	$EI\delta_{max}$	$EI\delta_{min}$	Fig. No.	$EI\delta_{max}$	$EI\delta_{min}$
E 9.11	$0.064\,Ml^2$	$-0.179\,Ml^2$	E 9.14	6.522×10^{-3}	–
E 9.12	$5\,M_x l^2/6$	$-17\,M_x l^2/96$	E 9.15	52.85 mm	–
E 9.13	1 914.5	–	E 9.16	235.1 mm	–

9.28 Fig.E 9.13 $\delta_{max} = 3\,768.9/(EI)$ Fig.E 9.14 $\delta_{max} = 5\,wl^4/(384\,EI)$
9.29 Fig.E 9.17 δ_D = 6.4 mm and δ_C = − 3.2 mm Fig.E 9.18 δ_D = 115.2 mm and δ_C = 208.4 mm
9.31 $\delta = 83.313\,Pl/(Ebd)$ **9.32** $\delta = 51.4Pl/(Ebd)$ **9.33** δ_{max} = 13.51 mm, 3.3 percent error
9.34 δ = 38.74 mm, σ_{max} = 84.7 MPa, τ_{max} = 3.5 MPa
9.35 δ = 19.72 mm, σ_{max} = 21.0 MPa, τ_{max} = 11.15 MPa
9.37 Fig.E 9.19 (a) $\delta_{max} = 3\,Pl^3/(16\,EI)$ (b) $\delta_{max} = 144.112/(EI)$ (c) $\delta_{max} = 3\,813.75/(EI)$ (d) $\delta_{max} = 768.231/(EI)$ and $\delta_C = 753.053/(EI)$
9.38 Fig.E 9.20 (a) $\delta_{max} = 0.064\,15Ml^2/(EI)$ at $x = l/\sqrt{3}$ (b) $\delta_{max} = Ml^2\alpha(1+\alpha)/(2EI)$, $\delta_{min} = -Ml^2/(8EI)$

Chapter 10

10.8 $\delta_{max} = Pl^3/(192EI)$ **10.12** $\delta_{max} = 8.537 \times 10^{-3} wl^4/(EI)$ **10.13** $P = wl/2$
10.14 M_{min} = −112.5 kNm **10.15** M_{min} = −319.9 kNm
10.16

	Fixed support moment (kNm)	Inflection points (m) from the fixed end	prop
Fig.E 10.1(a)	− 64.22	2.116	0.908
(b)	+ 90.0	2.167	4.333
(c)	−126.0	2.345	1.045
(d)	+ 93.75	3.167	6.333

10.17 Fig.E 10.2(a) $-[Pa(l-a)/l]$; $[a(l-a)/l]$, $M_{max} = Pa^2/l$
(b) $-wl^2/30, +0.021\ 44wl^2, -wl^2/20, 0.237l, 0.807l$
(c) $-10wl^2/192, +wl^2/12$, $0.223l$ from each end
(d) $-w[4\{(a+c)^3 - a^3\}l - 3\{(a+c)^4 - a^4\}]/(12l^2)$,
$-w[\{(a+c)^2 - a^2\}/(12l^2)][6l^2 + 3\{(a+c)^2 - a^2\} - 8cl]$

10.18 Support moments (i) – 114.89 kNm (ii) – 124.54 kNm

10.20 Fig.E 10.3 Support moments (kNm) from left to right

(a)	0.00	– 82.89	+ 4.00	–192.00
(b)	– 51.40	– 48.83	+ 3.32	–158.18
(c)	– 66.31	–115.76	– 72.74	0.00

10.21 The problem does not admit a feasible solution (reason why)

10.22 Fig.E 10.4

	M_{max}	M_{min}	Points of inflection; distance from the left support
(a)	0.0	$-wl^2/2$	-
(b)	$+wl^2/6$	$-wl^2/3$	$0.423\ l$, $1.578\ l$
(c)	0.0	$-Pl/2$	-
(d)	$-11\ Pl/32$	$+5\ Pl/64$	$0.407\ l$
(e)	$+0.020\ wl^2$	$-0.30\ wl^2$	$0.80\ l$
(f)	– 200.0 kNm	+ 100.0 kNm	1.33 m
(g)	– 84.5 kNm	+ 44.8 kNm	2.31, 7.94, 15.97 m
(h)	– 137.2 kNm	+ 111.8 kNm	5.14, 10.13, 18.76 m

Chapter 11

11.6 $N_D = -687.18$ kN and $S_D = 0.00$

11.7 and 11.8 Reactions (kN)

Figure No.	Left support V	Left support H	Right support V	Right support H
Fig.E 11.1	149.37	265.55	483.47	548.39
E 11.2	167.27	103.63	132.73	103.63
E 11.3	240.62	171.88	309.38	171.88
E 11.4	187.50	62.50	62.50	62.50
E 11.5	225.00	187.50	75.00	187.50
E 11.6	50.62	85.43	151.88	85.43
E 11.7	296.25	545.00	203.75	545.00
E 11.8	508.70	646.74	341.30	646.74

11.11

Figure No.	M_{max} (kNm)	$A.F._{max}$ or $A.F._{min}$ (kN)
Fig.11.9(a)	259.20	– 45.00
(b)	163.20	– 54.50
(c)	18.20	– 48.60
(d)	67.50	54.93
(e)	–250.00	–129.41
(f)	100.00	5.92
(g)	– 80.00	– 25.00
(h)	126.62	–112.50
(i)	202.50	–135.00
(j)	35.00	– 55.83
(k)	330.00	0.00
(l)	270.00	262.50

Chapter 12

Forces in the members from left to right, shown in thick lines, are listed (kN).

Figure No.	Member forces (kN) 1	2	3	4	Figure No.	Member forces (kN) 1	2	3	4
E 12.1	−106.25	+115.97	+101.88	+ 67.22	E 12.9	−447.92	−198.75	+ 29.87	+143.33
E 12.2	− 21.66	− 30.04	+ 12.03	− 6.00	E 12.10	+512.65	+335.88	−123.74	−654.07
E 12.3	+ 80.00	− 77.78	− 25.00	-	E 12.11	+186.86	+118.79	−535.99	− 25.00
E 12.4	+ 33.33	− 53.33	− 20.00	-	E 12.12	+160.75	−160.75	0.00	−202.50
E 12.5	− 15.00	+ 89.44	− 10.00	-	E 12.13	−202.45	+ 85.51	+ 85.51	-
E 12.6	− 44.79	+136.57	+215.94	-	E 12.14	−359.74	+184.78	+214.95	-
E 12.7	−141.42	0.00	0.00	0.00	E 12.15	+141.58	− 80.39	+ 95.24	−124.63
E 12.8	+509.00	−431.64	+234.00	+ 94.18					

Chapter 13

13.17 Cylinder thickness = 22.0 mm, sphere thickness = 12.0 mm, additional water required = 58.54 litre

13.18 2.4 MPa, 76.7 mm^3 per metre **13.19** 230.5 MPa

13.20 Stress in the tie rods = 239.9 MPa, σ_1 = 225.0 MPa, σ_2 = 84.0 MPa **13.21** 15.75 m

13.22 σ_1 = 188.4 MPa, σ_2 = − 104.0 MPa, τ_{max} = 146.2 MPa

13.23 (i) 4.2 mm (ii) 4.6 mm (iii) 5.8 mm **13.24** 0.22

13.25 σ_1 = 287.0 and 224.0 MPa, σ_2 = 112.0 MPa, σ_1 = 283.5 MPa by thin cylinder theory (mean diameter), Δr = − 0.012 87 mm

13.26 σ_1 = 175.9 and 132.9 MPa, σ_1 = 193.5 MPa by thin cylinder theory (mean diameter)

13.27 Internal pressure = − 2.16 MPa, external pressure = − 1.61 MPa **13.28** (i) 1.32 (ii) 1.13

13.29 t = 0.149 m **13.30** 0.166 m

13.31 Diametral interference = 0.013 mm, final stresses = 22.53, 10.72, and 29.23 and 21.05 MPa; maximum allowable pressure = 4.63 MPa

13.32 0.062 mm **13.33** Radial stress = 103.6 MPa, diametral interference = 0.166 mm, maximum hoop stress in the hub = 147.1 MPa

13.34 (i) σ_Φ = − 0.08 MPa, σ_Θ = + 0.02 MPa (ii) $\sigma_\Phi = -\sigma_\Theta$ = − 0.25 MPa
(iii) σ_Φ = − 0.21 MPa, σ_Θ = + 0.01 MPa

13.35 (i) 45.0 kNm^{-2} (ii) 3.53 MN (iii) 60.0 kNm^{-2}

13.36 Thickness = 44.2 mm for a hemi-spherical dome

Chapter 14

14.10 Critical load in kN

S. No.	End conditions	Length (m) 1.0	3.8	11.0	21.6
1	H–H	1 454.8	184.5	22.0	5.7
2	F–F	1 454.8	738.2	88.1	22.8
3	F–H	1 454.8	376.9	45.8	11.7
4	F–f	666.2	46.1	5.5	1.4

14.11

End conditions	H–H	F–F	F–H	F–f
Critical length (mm)	369.7	739.4	528.4	184.9

14.12 P_c = 2 208.64 kN, l_c = 4.29 m, $P_{6.3}$ = 1 024.04 kN and δ_{max} = 27.94 mm

14.13 $\Delta T = [\pi^2 I/(l^2 \alpha A)]$ **14.14** $\Delta T = [\pi^2 EI/(l^3\alpha)][(1/k_s) + l/(AE)]$
14.15 (i) 338.69 kN (ii) 1 179.75 kN **14.16** (i) 3.013 m (ii) 5.624 m
14.17 4 704.24 kN **14.18** 1 370.16 kN
14.19

End conditions	H–H	F–F	F–H	F–f
Critical deflection (mm)	88.88	10.22	35.31	403.69

14.20 ISWB 600 is a suitable section **14.21** (i) 1.8 mm (ii) 102.3 mm **14.22** 1.94 mm
14.23 $\sigma_{min} = -8.83$ MPa, 1.7 percent error **14.24** (i) 4.18 kNm^{-1} (ii) 8.02 kN
14.25 (i) 21.78 kN (ii) 5.44 kN **14.26** 96.0 kN, 140.0 percent **14.27** 205.0 kN, 412.5 percent

Chapter 15

15.6 $\delta = 44.0$ mm, $\tau = 114.4$ MPa, $k = 5.68$ Nmm^{-1}, $U = 5\ 496.9$ Nmm, $P_{max} = 306.0$ N
15.7 $\phi = 0.945$ radian, $\sigma = 29.8$ MPa **15.8** $P_{max} = 43.96$ mm, $\delta = 142.9$ MPa
15.9 $P_{max} = 604.6$ N, $\delta = 73.8$ mm **15.10** $E = 219.7$ GPa
15.11 $d = 16.1$ mm, $n = 39.13$, $P_c = 480.5$ N
15.12 Problem 15.10 $\tau_{max} = 30.6$ and 32.9 MPa, Problem 15.11 $\tau_{max} = 49.6$ and 50.4 MPa
15.13 63 **15.14** $d = 12.05$ mm, $n = 6.74$
15.15 $\delta = 26.21$ mm, $\phi = 0.053$ radian, $\tau_{max} = 60.3$ MPa, $\sigma_{max} = 69.6$ MPa, $k = 17.2$ Nmm^{-1}
15.16 $\delta = 2.05$ mm, $\Phi = 0.314$ radian, $\tau_{max} = 26.5$ MPa, $\sigma_{max} = 91.8$ MPa
15.17 (i) $k = 5.03$ Nmm^{-1}, $\delta = 99.38$ mm (ii) $k = 45.27$ Nmm^{-1}, $\delta = 11.04$ mm
15.18 $R_1 = 35.63$ mm, $R_2 = 71.25$ mm, $\delta = 36.3$ mm **15.19** $P_{max} = 666.7$ N, $\delta = 163.3$ mm
15.20 9.81, 14.72 and 19.62 mm; 327.3 mm from the heaviest spring **15.21** $k = 2.72$ Nmm^{-1}
15.23 $k = 4.77$ Nmm^{-1} **15.24** Proof load = 24.62 kN, $P = 4.924$ kN, $P_{max} = 4.871$ kN
15.25 $n = 9$, $b = 50.0$ mm, $t = 7.4$ mm, proof load = 3.60 kN, $P_{max} = 3.60$ kN
15.26 Proof load = 1 368.3 N, $P_{max} = 350.0$ N, $P = 228.1$ N
15.27 $n = 9$, $b = 50.0$ mm, $t = 7.4$ mm, proof load = 6 331.6 N, $P_{max} = 1\ 026.8$ N
15.28 $T = 300.0$ Nmm, $U = 11\ 571.4$ Nmm, $n = 12.3$

Glossary of terms

Arch A structural member, curved in elevation, that resists loads predominantly by axial compression.

Axis of a member It is a line joining the centroids of cross sections of a member along its length. A member is idealised as a line in elementary analysis, and is denoted by its axial line.

Beam It is a structural member on which external loads act normal to its axis. A beam resists applied loads predominantly by bending action.

Bending moment Bending moment at a section of a beam is the moment of all the forces, to the left or right of the section, about the section.

Bulk modulus (volumetric modulus) It is the ratio of volumetric stress (applied equally in all directions) to the corresponding volumetric strain within the elastic limit of a material. Bulk modulus of a material is generally expressed in mega newtons per square metre (MNm^{-2}).

Column It is a structural member on which external loads act along the axis. A column resists external loads predominantly by axial reaction.

Compatibility The displacements at various locations on a body are interrelated; this relationship is known as compatibility. Compatibility implies the agreement of the displacements with the conditions at supports or any location on a body.

Deformation Change in the dimensions of a body. Deformation is termed positive if it is elongation and negative if it is shortening of the body. It is generally expressed in millimetres (mm).

Displacement Movement of a body or a point on a body from its original position. The displacement may be due to applied forces or any other cause (temperature variation, for instance). The displacement is termed vertical or horizontal, depending upon its direction. It is expressed usually in millimetres (mm).

Ductility Ductility of a material is its ability to be drawn into thin wires without breaking. It is generally expressed as percentage strain at failure.

Elasticity It is the ability of a material to regain its original shape (and dimensions) completely on the removal of applied load.

Equilibrium (static equilibrium) A body is said to be in equilibrium, when the resultant of all the forces acting upon it is zero (the body remains static).

Factor of safety It is the factor with which the yield (ultimate) strength of a material is divided in order to obtain its working strength.

Force Newton's first law of motion defines force as an entity that changes the state of a body. A force may be applied on a body when it is known as applied or external force. A force may also be developed within a body because of applied forces; it is then known as internal force or a reaction. It is expressed in newtons (N) or its multiples as kilo newtons (kN) and mega newtons (MN).

Frame An assembly of linear elements that forms a stable structural configuration to resist external loads. The members of a frame may be connected through hinged or rigid joints; they are subjected to bending moment and shear force as well as axial force.

Free body diagram An isolated part of a body on which the external forces and the forces acting upon it by the removed part (or parts) are indicated.

Hinge A joint in a structure that cannot resist one of the member forces (bending moment, shear force or axial force; it is accordingly called moment hinge, shear hinge or axial force hinge).

Lever arm It is the perpendicular distance between a force and the point about which moment of the force is required.

Member (beam, column) A body that resists external forces. The length of a member is assumed to be large compared to its lateral dimensions (depth and breadth). It is a prismatic member if its cross section remains constant throughout its length; a member is termed non-prismatic otherwise.

Moment It is the product of force multiplied by lever arm. It is usually expressed in kilo newton metres (kNm) or mega newton metres (MNm).

Moment of inertia The integral

$$I = \int y^2 dA$$

is called the moment of inertia of a section about the specified axis.

Modulus of section The ratio of moment of inertia to the distance of the extreme fibre of the section from the neutral axis.

$$Z = I/y_{\max}$$

Poisson's ratio The ratio of lateral (transverse) strain to the strain in the direction of applied load.

Rigidity modulus (shear modulus) It is the ratio of shear stress applied to the corresponding shear strain within the elastic limit of a material. Shear modulus of a material is generally expressed in mega newtons per square metre (MNm^{-2}).

Shear force Shear force at a section of a beam is the unbalanced force, normal to the beam axis, to the right or left of the section.

Strain Deformation per unit dimension of a body. It is a non-dimensional quantity, being the ratio of change in a dimension to the original dimension of a body. Strain is termed axial (under tensile or compressive force), shear (under shear force), bending (under bending moment) and volumetric (under hydrostatic force).

Strain energy It is the energy stored in a body due to its deformation under external forces. Strain energy is equal to the work done by the external forces applied on the body; strain energy is released when the external forces are removed.

Stress Intensity of force per unit area of the surface on which the force is applied. The stress is termed axial (under tensile or compressive force), shear (under shear force), bending (under bending moment) or volumetric (under hydrostatic force), depending upon the nature of the forces applied. It is generally expressed in newtons per square millimetre (Nmm^{-2}) or in mega newtons per square metre (MNm^{-2}). The latter may also be termed as mega pascals (MPa).

Support A support is the point of contact of a body with another. A body transfers external forces through bending or axial deformations to another body. The points through which the forces are transferred are known as supports. A support is termed 'roller', 'hinged' and 'fixed' as explained in the conventions.

Torsion It is a force that twists a member (or a moment that is applied normal tc the plane of a member).

Truss An assembly of pin-jointed linear elements that forms a stable structural configuration to resist external loads. The members of a truss form triangular subspaces, and are subjected to only axial force under applied loads; the loads are applied only at the joints.

Working strength (allowable or permissible strength) Working strength (or stress) of a material is the maximum allowable stress during the service life of the material.

Yield strength (ultimate strength or stress) Yield strength (or stress) of a material is the stress beyond which the material is no longer elastic.

Young's modulus (modulus of elasticity) It is the ratio of direct stress applied to the corresponding strain within the proportionality limit of a material. Young's modulus has the same dimensions as stress (strain being non-dimensional), and is expressed in giga newtons per square metre (GNm^{-2}) or giga pascals (GPa).