Objective Type Questions
for
DESIGN OF MACHINE ELEMENTS
(Third Edition)

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1 INTRODUCTION

1.1 Rowing boat is produced by

(a) design by drawing (b) design by craft evolution

(c) design synthesis (d) simultaneous design

1.2 Musical instruments is produced by

(a) design by drawing (b) design by craft evolution

(c) design synthesis (d) simultaneous design

1.3 Gearbox is produced by

(a) design by drawing (b) design by craft evolution

(c) design synthesis (d) simultaneous design

1.4 Modern automobile car is produced by

(a) sequential design (b) design by craft evolution

(c) design synthesis (d) simultaneous design

1.5 Standardization deals with the characteristics of product that include

(a) dimensions of machine elements

(b) method of testing the product

(c) composition and properties of engineering materials

(d) all the three

1.6 The types of standards used in design office are

(a) standards prepared by Bureau of Indian Standards (BIS)

(b) standards prepared by International Standards Organization (ISO)

(c) standards prepared by professional bodies like American Gear Manufacturing Association (AGMA)

(d) all the three

1.7 The basic series of preferred numbers are,

(a) R5, R10, R20, R40 and R80

(b) R10, R20, R30, R40 and R50

(c) R5, R10, R15, R20 and R25

(d) none of the above

1.8 Series factor for R20 series is,

(a) $\sqrt[10]{20}$ (b) $\sqrt{20}$ (c) $\sqrt[20]{10}$ (d) $\sqrt[3]{20}$

1.9 The external appearance is important in

(a) consumer durables like refrigerators and audiovisual equipment

(b) industrial products like cranes and hoists

(c) machine elements like gearbox, coupling or pressure vessel

(d) none of the above

1.10 The job of industrial designer is

(a) to carry out detailed stress analysis of the product

(b) to design industrial products like cranes and hoists

(c) to create aesthetically forms and shapes for the products

(d) none of the above

1.11 The meaning of blue colour is

(a) the component is hot

(b) the component is cold

(c) the component is safe

(d) there is possible danger

1.12 The meaning of orange colour is

(a) the component is hot

(b) the component is cold

(c) the component is safe

(d) there is possible danger

1.13 Ergonomic deals with

(a) design of controls

(b) design of displays

(c) energy expenditure in hand and foot operations

(d) all the three

1.14 Speedometer is a

(a) display giving quantitative measurements

(b) display giving state of affair

(c) display indicating predetermined settings

(d) none of above

1.15 Moving scale or moving dial is used for

(a) display giving quantitative measurements

(b) display giving state of affair

(c) display indicating predetermined settings

(d) none of above

1.16 The height of letter or number on indicators should be equal to or more than,

(a) $\left(\frac{\text{reading distance}}{10}\right)$ (b) $\left(\frac{\text{reading distance}}{20}\right)$

(c) $\left(\frac{\text{reading distance}}{100}\right)$ (d) $\left(\frac{\text{reading distance}}{200}\right)$

1.17 When large force is required to operate, the type of control used is

(a) knobs and switches (b) levers and wheels

(c) push buttons (d) none of above

1.18 In concurrent engineering, design and manufacturing are

(a) sequentially considered (b) simultaneously considered

(c) separately considered (d) none of above

Answers

1.1 (b)	1.2 (b)	1.3 (a)	1.4 (d)	1.5 (d)
1.6 (d)	1.7 (a)	1.8 (c)	1.9 (a)	1.10 (c)
1.11 (b)	1.12 (d)	1.13 (d)	1.14 (a)	1.15 (a)
1.16 (d)	1.17 (b)	1.18 (b)		

2 ENGINEERING MATERIALS

2.1 Which of the following materials has maximum strength

(a) grey cast iron (b) plain carbon steel

(c) alloy steel (d) aluminium alloy

2.2 Which of the following materials has maximum ductility

(a) grey cast iron (b) mild steel

(c) alloy steel (d) high carbon steel

2.3 Grey cast iron contains

(a) less than 0.3 % carbon (b) 0.3 to 0.5 % carbon

(c) 0.5 to 1.4 % carbon (d) 3 to 4 % carbon

2.4 Mild steel contains

(a) less than 0.3 % carbon (b) 0.3 to 0.5 % carbon

(c) 0.5 to 1.4 % carbon (d) 3 to 4 % carbon

2.5 Medium carbon steel contains

(a) less than 0.3 % carbon (b) 0.3 to 0.5 % carbon

(c) 0.5 to 1.4 % carbon (d) 3 to 4 % carbon

2.6 Steels used for welded assemblies are

(a) medium carbon steel (b) mild steel

(c) high carbon steel (d) alloy steel

2.7 Steels used for automobile bodies and hoods are

(a) medium carbon steel (b) mild steel

(c) high carbon steel (d) alloy steel

2.8 Steels used for helical springs are

(a) medium carbon steel (b) mild steel

(c) high carbon steel (d) alloy steel

2.9 Material used for machine tool beds is

(a) cast iron (b) mild steel

(c) high carbon steel (d) alloy steel

2.10 Material used for bearing bushes is

(a) phosphor bronze (b) gunmetal

(c) babbitt (d) any one of above

2.11 Material used for self-lubricated bearing is

(a) Acetal (b) Polyurethane

(c) Polytetrafluroethylene (Teflon) (d) any one of above

2.12 Die cast parts are used when

(a) material of the parts has low melting point

(b) parts have small size

(c) parts are made on large scale

(d) all three objectives are desired

2.13 Fibres used for fibre-reinforced-plastics are made of

(a) steel wires (b) hemp

(c) glass and carbon (d) asbestos

2.14 Synthetic rubber is used for

(a) V belt (b) gasket

(c) seals (d) all three parts

2.15 Toughness of steel is increased by adding,

(a) nickel (b) chromium

(c) sulphur (d) tungsten

2.16 Wear resistance of steel is increased by adding,

(a) nickel (b) chromium

(c) sulphur (d) none of the above

2.17 Hardness of steel is increased by adding,

(a) nickel (b) molybdenum

(c) sulphur (d) none of above

2.18 Hardness of steel is increased by adding,

(a) chromium (b) molybdenum

(c) tungsten (d) all of above elements

2.19 In free cutting steels, important alloying element is

(a) nickel (b) chromium

(c) sulphur (d) tungsten

2.20 A cast iron designated by FG300 is,

(a) grey cast iron with carbon content of 3%

(b) grey cast iron with ultimate tensile strength of 300 N/mm^2

(c) grey cast iron with ultimate compressive strength of 300 N/mm^2

(d) grey cast iron with tensile yield strength of 300 N/mm^2

2.21 A cast iron designated by BM350 is,

(a) blackheart malleable cast iron with carbon content of 3.5%

(b) blackheart malleable cast iron with ultimate tensile strength of 350 N/mm^2

(c) blackheart malleable cast iron with ultimate compressive strength of 350 N/mm^2

(d) blackheart malleable cast iron with tensile yield strength of 350 N/mm^2

2.22 Plain carbon steels are designated by,

(a) tensile strength (b) carbon content

(c) composition of alloying element (d) none of the above

2.23 Plain carbon steel designated by 40C8 means,

(a) plain carbon steel with ultimate tensile strength of 400 N/mm^2 and 0.8% carbon

(b) plain carbon steel with 0.35 to 0.45% carbon and 0.7to 0.9% manganese

(c) plain carbon steel with 0.8% carbon and 4 % manganese

(d) plain carbon steel with 40% carbon and 8% manganese

2.24 Thermosetting plastic is one,

(a) which softens when heated and hardens upon cooling

(b) which once having cured by chemical reaction, does not soften or melt upon subsequent heating

(c) which can be moulded and remoulded repeated

(d) which has linear polymer chain

2.25 Thermoplastic is one,

(a) which softens when heated and hardens upon cooling

(b) which can be moulded and remoulded

(c) which has linear polymer chain

(d) which has all three characteristics

Answers:

2.1 (c)	2.2 (b)	2.3 (d)	2.4 (a)	2.5 (b)
2.6 (b)	2.7 (b)	2.8 (c)	2.9 (a)	2.10 (d)
2.11 (d)	2.12 (d)	2.13 (c)	2.14 (d)	2.15 (a)
2.16 (b)	2.17 (b)	2.18 (d)	2.19 (c)	2.20 (b)
2.21 (b)	2.22 (b)	2.23 (b)	2.24 (b)	2.25 (d)

3 MANUFACTURING CONSIDERATIONS IN DESIGN

3.1 Cylinder blocks of internal combustion engine are made by

(a) forging (b) extrusion

(c) casting (d) drilling and broaching

3.2 Valve bodies of internal combustion engine are made by

(a) forging (b) rolling

(c) casting (d) turning from bar stock

3.3 In forged components,

(a) fiber lines are arranged in a predetermined way

(b) fiber lines of rolled stock are broken

(c) there are no fiber lines

(d) fiber lines are scattered

3.4 In machined components,

(a) fiber lines are arranged in a predetermined way

(b) fiber lines of rolled stock are broken

(c) there are no fiber lines

(d) fiber lines are scattered

3.5 In cast components,

(a) fiber lines are arranged in a predetermined way

(b) fiber lines of rolled stock are broken

(c) there are no fiber lines and grains are scattered

(d) grains are arranged in a predetermined way

3.6 In cast iron components, shrinkage cavities are formed due to

(a) cored holes at the junction of walls

(b) very small fillet radius

(c) concentration of metal at the junction of walls

(d) very thin wall thickness

3.7 Draft angle is provided for forged parts

(a) to arrange fiber lines in a favorable way

(b) to avoid unbalanced forced

(c) for easy removal from die cavities

(d) to reduce stress concentration

3.8 The temperature at which new stress-free grains are formed in the metal is called,

(a) upper critical temperature (b) melting point

(c) recrystallization temperature (d) eutectic temperature

3.9 Hot working of metals is carried out

(a) above the recrystallization temperature

(b) below the recrystallization temperature

(c) at the recrystallization temperature

(d) at higher temperature

3.10 Cold working of metals is carried out

(a) above the recrystallization temperature

(b) below the recrystallization temperature

(c) at the recrystallization temperature

(d) at lower temperature

3.11 Cold working

(a) increases toughness and ductility

(b) reduces residual stresses

(c) increases hardness and strength

(d) produces favorable pattern of fiber lines

3.12 Hot working

(a) increases toughness and ductility

(b) increases surface finish

(c) increases hardness and strength

(d) produces accurate dimensions for the parts

3.13 In unilateral system for tolerances,

(a) tolerances are given on both positive and negative sides of basic size

(b) one tolerance is zero and other tolerance is given on any one side of basic size

(c) one tolerance is zero and other tolerance is given only on higher side of basic size

(d) one tolerance is zero and other tolerance is given only on lower side of basic size

3.14 In bilateral system for tolerances,

(a) tolerances are given on both positive and negative sides of basic size

(b) one tolerance is zero and other tolerance is given on any one side of basic size

(c) one tolerance is zero and other tolerance is given only on higher side of basic size

(d) one tolerance is zero and other tolerance is given only on lower side of basic size

3.15 In clearance fit,

(a) tolerance zones of hole and shaft overlap

(b) tolerance zone of hole is completely below that of shaft

(c) tolerance zone of hole is entirely above that of shaft

(d) none of the above

3.16 In interference fit,

(a) tolerance zones of hole and shaft overlap

(b) tolerance zone of hole is completely below that of shaft

(c) tolerance zone of hole is entirely above that of shaft

(d) none of the above

3.17 In transition fit,

(a) tolerance zones of hole and shaft overlap

(b) tolerance zone of hole is completely below that of shaft

(c) tolerance zone of hole is entirely above that of shaft

(d) none of the above

3.18 In hole-basis system, the basis hole is one

(a) whose upper deviation is zero

(b) whose upper and lower deviations are zero

(c) whose lower deviation is zero

(d) none of the above

3.19 In shaft-basis system, the basis shaft is one

(a) whose upper deviation is zero

(b) whose upper and lower deviations are zero

(c) whose lower deviation is zero

(d) none of the above

3.20 According to Indian standard, 50 H8-g7 means

(a) upper limit is (50+8) mm and lower limit (50-7) mm

(b) designation of tolerance with basic size 50 mm

(c) designation of fit of two parts with basic size 50 mm

(d) none of above

3.21 According to Indian standard, total number of tolerance grades are

(a) 10 (b) 20 (c) 18 (d) 8

3.22 According to Indian standard, 50 H8-g7 means

(a) tolerance grade for hole is 8 and for shaft is 7

(b) tolerance grade for shaft is 8 and for hole is 7

(c) designation of fit on shaft-basis system

(d) none of above

3.23 The tolerance of grade 8 is obtained by

(a) die casting (b) turning on capstan and turret lathes

(c) grinding (d) sand casting

3.24 The tolerance of grade 16 is obtained by

(a) die casting (b) turning on capstan and turret lathes

(c) grinding (d) sand casting

3.25 The tolerance of grade 6 is obtained by

(a) die casting (b) turning on capstan and turret lathes

(c) grinding (d) sand casting

Answers

3.1 (c)	3.2 (a)	3.3 (a)	3.4 (b)	3.5 (c)
3.6 (c)	3.7 (c)	3.8 (c)	3.9 (a)	3.10 (b)
3.11 (c)	3.12 (a)	3.13 (b)	3.14 (a)	3.15 (c)
3.16 (b)	3.17 (a)	3.18 (c)	3.19 (a)	3.20 (c)
3.21 (c)	3.22 (a)	3.23 (b)	3.24 (d)	3.25 (c)

4 DESIGN AGAINST STATIC LOAD

4.1 Hooke's law holds good up to

(a) breaking point (b) yield point

(c) elastic limit d) plastic limit

4.2 The ratio of tensile stress to tensile strain within elastic limit is called

(a) bulk modulus (b) Poisson's ratio

(c) modulus of rigidity (d) modulus of elasticity

4.3 The ratio of shear stress to shear strain is called

(a) bulk modulus (b) Poisson's ratio

(c) modulus of rigidity (d) modulus of elasticity

4.4 The modulus of elasticity of carbon steel is

(a) 207 000 N/mm^2 (b) 100 000 N/mm^2

(c) 50 000 N/mm^2 (d) 80 000 N/mm^2

4.5 The modulus of elasticity of grey cast iron is

(a) 207 000 N/mm^2 (b) 100 000 N/mm^2

(c) 50 000 N/mm^2 (d) 80 000 N/mm^2

4.6 The modulus of rigidity of carbon steel is

(a) 207 000 N/mm^2 (b) 100 000 N/mm^2

(c) 50 000 N/mm^2 (d) 80 000 N/mm^2

4.7 The modulus of rigidity of grey cast iron is

(a) 207 000 N/mm^2 (b) 100 000 N/mm^2

(c) 40 000 N/mm^2 (d) 80 000 N/mm^2

4.8 The Poisson's ratio of grey cast iron is

(a) 0.40 (b) 0.29 (c) 0.21 (d) 0.75

4.9 The Poisson's ratio of carbon steels is

(a) 0.40 (b) 0.29 (c) 0.21 (d) 0.75

4.10 Within elastic limit, the stress is

(a) equal to strain

(b) inversely proportional to strain

(c) directly proportional to strain

(d) directly proportional to the square of strain

4.11 When a circular shaft is subjected to torque, the torsional shear stress is

(a) maximum at the axis of rotation and zero at the outer surface

(b) uniform from axis of rotation to the outer surface

(c) zero at the axis of rotation and maximum at the outer surface

(d) zero at the axis of rotation and zero at the outer surface and maximum at the mean radius

4.12 When a circular shaft is subjected to torque, the torsional shear stress is

(a) directly proportional to the distance from the axis

(b) inversely proportional to the distance from the axis

(c) proportional to the square of the distance from the axis

(d) constant through out the cross-section

4.13 When the diameter of shaft is doubled, its torque transmitting capacity will increase by

(a) 8 times (b) 2 times (c) 4 times (d) 16 times

4.14 A shaft is transmitting torque M_t and made of material having (τ) as permissible shear stress. The diameter of shaft d is given by,

(a) $\sqrt[3]{\frac{16\ M_t}{\pi\ \tau}}$ (b) $\sqrt[3]{\frac{32\ M_t}{\pi\ \tau}}$ (c) $\sqrt[3]{\frac{64\ M_t}{\pi\ \tau}}$ (d) $\sqrt[3]{\frac{16\ M_t}{\tau}}$

4.15 The torque transmitting capacity of a shaft of diameter d and made of material having (τ) as permissible shear stress is,

(a) $\pi\, d^3\tau$ (b) $\left(\frac{\pi}{16}\right)d^3\tau$ (c) $\left(\frac{\pi}{32}\right)d^3\tau$ (d) $\left(\frac{\pi}{16}\right)d^4\tau$

4.16 When a shaft of diameter d and length l is subjected to torsional moment M_t, then the angle of twist θ in degrees is given by,

(a) $\frac{584\, M_t\, l}{G\ d^4}$ (b) $\frac{M_t\, l}{\pi\ d^4}$ (c) $\frac{584\, M_t\, l}{G\ d^3}$ (d) $\frac{M_t\, G}{d^4\, l}$

4.17 The polar moment of inertia of a hollow shaft, with d_o and d_i as outer and inner diameters respectively, is given by,

(a) $\frac{\pi\ (d_o^4 - d_i^4)}{32}$ (b) $\frac{\pi\ (d_o^3 - d_i^3)}{32}$

(c) $\frac{\pi\ (d_o^2 - d_i^2)}{32}$ (d) $\frac{\pi\ (d_o^4 - d_i^4)}{16}$

4.18 The bending stress induced in a beam is

(a) maximum at the farthest fiber from neutral axis and zero at the neutral axis

(b) uniform through out the cross-section

(c) zero at the farthest fiber from neutral axis and maximum at the neutral axis

(d) zero at the neutral axis and zero at the farthest fiber and maximum at the mean distance

4.19 The neutral axis of a beam is

(a) layer subjected to tensile stress

(b) layer subjected to compressive stress

(c) layer subjected to zero stress

(d) none of the above

4.20 When a hole of diameter D is punched in a plate of thickness t, the force required to punch the hole is given by,

(a) $\pi d\,t\,S_{us}$ (b) $\pi d^2\,t\,S_{us}$ (c) $\frac{\pi}{4}d^2\ t\ S_{us}$ (d) $\frac{\pi}{4}d^2\ S_{us}$

where S_{us} is ultimate shear strength of plate material.

4.21 A component made of carbon steel is designed on strength basis by

(a) ultimate tensile strength (b) yield strength

(c) modulus of elasticity (d) modulus of rigidity

4.22 A component made of grey cast iron is designed on strength basis by

(a) ultimate tensile strength (b) yield strength

(c) modulus of elasticity (d) modulus of rigidity

4.23 The factor of safety for cast iron component, subjected to static force, is usually

(a) 1.5 to 2 (b) 3 to 5 (c) 1.3 to 1.5 (d) 5 to 10

4.24 The factor of safety for carbon steel component, subjected to static force, is usually

(a) 1.5 to 2 (b) 3 to 5 (c) 1.3 to 1.5 (d) 5 to 10

4.25 The thermal stresses are caused due to

(a) variation in temperature (b) high temperatures

(c) specific heat (d) latent heat

4.26 The residual stresses are caused due to

(a) manufacturing methods like forging

(b) machining methods like milling or grinding

(c) cold working processes like rolling and extrusion

(d) any one of the above

4.27 A cotter joint is used to transmit

(a) axial tensile force only

(b) axial tensile or compressive force

(c) axial compressive force only

(d) combined bending and torsional moment

4.28 The taper on cotter is usually

(a) 1 in 24 (b) 1 in 8 (c) 1 in 100 (d) 1 in 48

4.29 A taper is provided for cotter

(a) to ensure tightness in operating condition

(b) to provide wedge action

(c) to ease the removal of cotter during dismantling

(d) for all three reasons

4.30 The joint between the piston rod and the cross head of steam engine is

(a) knuckle joint (b) universal joint

(c) cotter joint (d) key joint

4.31 Cotter joint is used for the joint between

(a) piston rod and crosshead of steam engine

(b) slide spindle and fork of valve mechanism

(c) piston rod and tail rod or pump rod

(d) for all three applications

4.32 A knuckle joint is used to transmit

(a) axial tensile force only

(b) axial tensile or compressive force

(c) axial compressive force only

(d) combined bending and torsional moment

4.33 The joint in valve mechanism of reciprocating engine is

(a) knuckle joint (b) universal joint

(c) cotter joint (d) key joint

4.34 Knuckle joint is used for the joint between

(a) tie bars in roof trusses (b) links in suspension bridge

(c) fulcrum of lever and support (d) for all three applications

4.35 The pin in knuckle joint is subjected to

(a) double shear stress (b) torsional shear stress

(c) axial tensile stress (d) axial compressive stress

4.36 In lever terminology, ‘leverage’ is the ratio of

(a) load to effort (b) effort to load

(c) load arm to effort arm (d) effort arm to load arm

4.37 In lever terminology, ‘mechanical advantage’ is the ratio of

(a) load to effort (b) effort to load

(c) load arm to effort arm (d) effort arm to load arm

4.38 In levers,

(a) mechanical advantage is more than leverage

(b) mechanical advantage is less than leverage

(c) mechanical advantage is equal to leverage

(d) none of the above

4.39 In ‘first’ type of levers, mechanical advantage is

(a) less than one (b) more than one

(c) equal to one (d) any one of the above

4.40 In ‘second’ type of levers, mechanical advantage is

(a) less than one (b) more than one

(c) equal to one (d) any one of the above

4.41 In ‘third’ type of levers, mechanical advantage is

(a) less than one (b) more than one

(c) equal to one (d) any one of the above

4.42 The rocker arm in internal combustion engine is

(a) first type of lever (b) second type of lever

(c) third type of lever (d) none of the above

4.43 The bell crank lever in centrifugal governor is

(a) first type of lever (b) second type of lever

(c) third type of lever (d) none of the above

4.44 The lever loaded safety valve mounted on boiler is

(a) first type of lever (b) second type of lever

(c) third type of lever (d) none of the above

4.45 The cross-section of lever is

(a) rectangular (b) elliptical

(c) I - section (d) any one of the above

4.46 The cross-section of lever is subjected to

(a) torsional moment (b) axial tensile force

(c) bending moment (d) axial compressive force

4.47 The fulcrum pin of lever is designed on the basis of

(a) torsional moment (b) axial tensile force

(c) bending moment (d) bearing pressure

4.48 Rankine's theory of failure is applicable to

(a) ductile materials (b) elastic materials

(c) brittle materials (d) plastic materials

4.49 Coulomb, Tresca and Guest's theory of failure is applicable to

(a) ductile materials (b) composites

(c) brittle materials (d) non-metals

4.50 Distortion energy theory of failure is applicable to

(a) components made of plain carbon steel

(b) components made of composites

(c) components made of cast iron

(d) components made of non-metals

4.51 According to maximum shear stress theory of failure, the relationship between yield strength in shear (S_{sy}) and tensile yield strength (S_{yt}) is

(a) $S_{sy} = 0.5\ S_{yt}$ (b) $S_{sy} = 0.577\ S_{yt}$

(c) $S_{sy} = 0.75\ S_{yt}$ (d) $S_{sy} = 0.4\ S_{yt}$

4.52 According to distortion energy theory of failure, the relationship between yield strength in shear (S_{sy}) and tensile yield strength (S_{yt}) is

(a) $S_{sy} = 0.5\ S_{yt}$ (b) $S_{sy} = 0.577\ S_{yt}$

(c) $S_{sy} = 0.75\ S_{yt}$ (d) $S_{sy} = 0.4\ S_{yt}$

4.53 For maximum principal stress theory, the shape of the region of safety on σ_1, σ_2 co-ordinate system is

(a) square (b) hexagon

(c) ellipse (d) circle

4.54 For maximum shear stress theory, the shape of the region of safety on σ_1, σ_2 co-ordinate system is

(a) square (b) hexagon

(c) ellipse (d) circle

4.55 For distortion theory, the shape of the region of safety on σ_1, σ_2 co-ordinate system is

(a) square (b) hexagon

(c) ellipse (d) circle

4.56 The maximum bending stress in a curved beam, having symmetrical cross-section, always occurs at

(a) inner fiber (b) outer fiber

(c) centroidal axis (d) neutral axis

4.57 The bending stress in a curved beam is,

(a) zero at the neutral axis (b) zero at the centroidal axis

(c) zero at the inner fiber (d) zero at the outer fiber

4.58 Griffiths' law states that fracture strength of brittle material is

(a) directly proportional to the square root of the crack length

(b) inversely proportional to the square root of the crack length

(c) directly proportional to the square of the crack length

(d) inversely proportional to the square of the crack length

4.59 Fracture mechanics is the science of

(a) predicting the influence of cracks on fatigue fracture of components

(b) predicting the influence of cracks on brittle fracture of components

(c) predicting the influence of cracks on ductile fracture of components

(d) none of the above

Answers

4.1 (c)	4.2 (d)	4.3 (c)	4.4 (a)	4.5 (b)
4.6 (d)	4.7 (c)	4.8 (c)	4.9 (b)	4.10 (c)
4.11 (c)	4.12 (a)	4.13 (a)	4.14 (a)	4.15 (b)
4.16 (a)	4.17 (a)	4.18 (a)	4.19 (c)	4.20 (a)
4.21 (b)	4.22 (a)	4.23 (b)	4.24 (a)	4.25 (a)
4.26 (d)	4.27 (b)	4.28 (a)	4.29 (d)	4.30 (c)
4.31 (d)	4.32 (a)	4.33 (a)	4.34 (d)	4.35 (a)
4.36 (d)	4.37 (a)	4.38 (c)	4.39 (d)	4.40 (b)
4.41 (a)	4.42 (a)	4.43 (a)	4.44 (b)	4.45 (d)
4.46 (c)	4.47 (d)	4.48 (c)	4.49 (a)	4.50 (a)
4.51 (a)	4.52 (b)	4.53 (a)	4.54 (b)	4.55 (c)
4.56 (a)	4.57 (a)	4.58 (b)	4.59 (b)	

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5 DESIGN AGAINST FLUCTUATING LOAD

5.1 Stress concentration factor is defined as

(a) ratio of highest stress near the discontinuity to endurance limit

(b) ratio of highest stress near the discontinuity to yield strength

(c) ratio of highest stress near the discontinuity to nominal stress obtained by elementary equation

(d) ratio of endurance limit to highest stress near the discontinuity

5.2 Stress concentration occurs due to

(a) abrupt change in cross-section

(b) discontinuities in component

(c) internal cracks and flaws

(d) any one of the above

5.3 Stress concentration occurs due to

(a) blow holes (b) keyways and splines

(c) machining scratches (d) any one of the above

5.4 Stress concentration occurs at the contact between

(a) meshing teeth of driving and driven gears

(b) cam and follower

(c) balls and races in ball bearing

(d) any one of the above

5.5 In static loading, the effect of stress concentration is more serious in case of

(a) components made of brittle materials

(b) components made of ductile materials

(c) components made of brittle as well as ductile materials

(d) none of the above

5.6 In cyclic loading, the effect of stress concentration is more serious in case of

(a) components made of brittle materials

(b) components made of ductile materials

(c) components made of brittle as well as ductile materials

(d) none of the above

5.7 The maximum stress concentration factor for a rectangular plate with a transverse hole loaded in tension or compression is

(a) 2 (b) 3 (c) 2.5 (d) 1

5.8 A stress that varies in sinusoidal manner with respect to time from a minimum value to maximum value and which has some mean as well as amplitude value is called

(a) reversed stress (b) fluctuating stress

(c) repeated stress (d) varying stress

5.9 A stress that varies in sinusoidal manner with respect to time from zero to maximum value and which has same values for mean as well as amplitude is called

(a) reversed stress (b) fluctuating stress

(c) repeated stress (d) varying stress

5.10 A stress that varies in sinusoidal manner with respect to time from tensile to compressive (or vice versa) and which zero mean is called

(a) reversed stress (b) fluctuating stress

(c) repeated stress (d) varying stress

5.11 Fatigue failure results due to fluctuating stresses when the stress magnitude is

(a) more than ultimate tensile strength

(b) more than yield strength but lower than ultimate tensile strength

(c) lower than yield strength

(d) none of the above

5.12 The criterion of failure for machine parts subjected to fluctuating stresses is

(a) ultimate tensile strength (b) yield strength

(c) endurance limit (d) modulus of elasticity

5.13 The factor of safety for machine parts subjected to reversed stresses is

(a) ratio of yield strength to maximum stress

(b) ratio of endurance limit to amplitude stress

(c) ratio of ultimate tensile strength to maximum stress

(d) ratio of endurance limit to mean stress

5.14 The approximate relationship between endurance limit of rotating beam specimen (S'_e) and ultimate tensile strength (S_{ut}), in case of steel component, is

(a) $S'_e = 0.4\ S_{ut}$ (b) $S'_e = 0.75\ S_{ut}$

(c) $S'_e = 0.577\ S_{ut}$ (d) $S'_e = 0.5\ S_{ut}$

5.15 The approximate relationship between endurance limit of rotating beam specimen (S'_e) and ultimate tensile strength (S_{ut}), in case of cast iron and cast steel components, is

(a) $S'_e = 0.4\ S_{ut}$ (b) $S'_e = 0.75\ S_{ut}$

(c) $S'_e = 0.577\ S_{ut}$ (d) $S'_e = 0.5\ S_{ut}$

5.16 The relationship between endurance limit of component subjected to fluctuating torsional shear stresses (S_{se}) to endurance limit in revered bending (S_e) is

(a) $S_{se} = 0.4\ S_e$ (b) $S_{se} = 0.75\ S_e$

(c) $S_{se} = 0.577\ S_e$ (d) $S_{se} = \pi\ S_e$

5.17 As the size of the component increases, the endurance limit of the component

(a) increases

(b) decreases

(c) remains same

(d) increases up to the diameter of 50 mm and then decreases

5.18 The surface finish factor for a highly polished component is

(a) 0.89 (b) 1 (c) 0.85 (d) 0.75

5.19 The reliability factor for using 50% reliability in design is

(a) 0.897 (b) 1 (c) 0.868 (d) 0.814

5.20 The endurance limit of the component can be increased by

(a) increasing the size of component

(b) shot peening

(c) increasing the stress concentration

(d) coating

5.21 Cold working

(a) increases fatigue strength

(b) decreases fatigue strength

(c) has no influence on fatigue strength

(d) none of the above

5.22 In order to find the endurance limit, the rotating beam specimen is subjected to

(a) repeated stresses (b) reversed stresses

(c) fluctuating stresses (d) maximum stress

5.23 The notch sensitivity factor (q) is given by

(a) $q = \dfrac{K_f - 1}{K_t - 1}$ (b) $q = \dfrac{K_t - 1}{K_f - 1}$

(c) $q = \dfrac{K_f + 1}{K_t + 1}$ (d) $q = \dfrac{K_t + 1}{K_f + 1}$

where K_t and K_f are theoretical and fatigue stress concentration factors respectively.

Answers:

5.1 (c)	5.2 (d)	5.3 (d)	5.4 (d)	5.5 (a)
5.6 (b)	5.7 (b)	5.8 (b)	5.9 (c)	5.10 (a)
5.11 (c)	5.12 (c)	5.13 (b)	5.14 (d)	5.15 (a)
5.16 (c)	5.17(b)	5.18 (b)	5.19 (b)	5.20 (b)
5.21 (a)	5.22 (b)	5.23 (a)		

6 POWER SCREWS

6.1 Which of the following screw thread is used for jacks, vices and clamps?

(a) square threads (b) trapezoidal threads

(c) buttress threads (d) acme threads

6.2 Which of the following screw thread is used for transmitting power in either direction?

(a) square threads (b) trapezoidal threads

(c) Buttress threads (d) (a) and (b)

6.3 Which of the following screw thread is used for transmitting force in one direction?

(a) square threads (b) trapezoidal threads

(c) Buttress threads (d) V threads

6.4 Which of the following screw thread is adaptable to split type nut?

(a) square threads (b) trapezoidal threads

(c) Buttress threads (d) V threads

6.5 Which of the following screw thread is stronger than other threads?

(a) square threads (b) trapezoidal threads

(c) Buttress threads (d) V threads

6.6 Which of the following screw thread is used for lead screw of lathe?

(a) square threads (b) trapezoidal threads

(c) Buttress threads (d) V threads

6.7 Multiple threads are used for

(a) high efficiency (b) high mechanical advantage

(c) low efficiency for self-locking (d) high load carrying capacity

6.8 Power screws for transmission of power should have

(a) high efficiency (b) low efficiency

(c) self-locking characteristic (d) over hauling characteristic

6.9 The efficiency of square threaded power screw is given by,

(a) $\eta = \dfrac{\tan\alpha}{\tan(\phi + \alpha)}$ (b) $\eta = \dfrac{\tan\alpha}{\tan(\phi - \alpha)}$

(c) $\eta = \dfrac{\tan(\phi+\alpha)}{\tan\alpha}$ (d) $\eta = \dfrac{\tan(\phi-\alpha)}{\tan\alpha}$

where α = helix angle ϕ = friction angle

6.10 For self locking screw

(a) $\phi > \alpha$ (b) $\alpha > \phi$

(c) $\mu < \tan\alpha$ (d) $\mu = \operatorname{cosec}\alpha$

where α = helix angle ϕ = friction angle $\mu =$ coefficient of friction

6.11 For over hauling screw

(a) $\phi > \alpha$ (b) $\alpha > \phi$

(c) $\phi = \alpha$ (d) none of above

where α = helix angle ϕ = friction angle $\mu =$ coefficient of friction

6.12 A screw is said to be self-locking if its efficiency is

(a) equal to 50% (b) more than 50%

(c) less than 50% (d) none of the above

6.13 A screw is said to be over hauling if its efficiency is

(a) equal to 50% (b) more than 50%

(c) less than 50% (d) none of the above

6.14 In design of screw jack from buckling considerations, the end conditions are assumed as

(a) both ends are hinged (b) both ends are fixed

(c) one end fixed and other hinged (d) one end fixed and other free

6.15 A cup is provided in screw jack

(a) to reduce the friction (b) to prevent rotation of load

(c) to increase load capacity (d) To increase efficiency

6.16 The efficiency of square threaded power is maximum when

(a) $\alpha = 60^0 - \phi$ (b) $\alpha = 45^0 - \phi$

(c) $\alpha = \left(45^0 + \frac{\phi}{2}\right)$ (d) $\alpha = \left(45^0 - \frac{\phi}{2}\right)$

where α = helix angle ϕ = friction angle

6.17 The maximum efficiency of square threaded power is

(a) $\frac{1 - \sin\phi}{1 + \sin\phi}$ (b) $\frac{1 + \sin\phi}{1 - \sin\phi}$

(c) $\frac{1 - \tan\phi}{1 + \tan\phi}$ (d) $\frac{1 + \tan\phi}{1 - \tan\phi}$

where ϕ = friction angle

6.18 The maximum efficiency of square threaded power depends upon

(a) lead angle of screw (b) friction angle

(c) pitch of screw (d) nominal diameter of screw

6.19 The efficiency of square threaded power depends upon

(a) mean diameter of screw (b) coefficient of friction

(c) pitch of screw (d) All the above

6.20 The maximum efficiency of square threaded power with friction angle of 30^0 is

(a) 25% (b) 33%

(c) 47% (d) 41%

6.21 By using large thread angle in lifting machines

(a) the mechanical advantage is more

(b) the mechanical advantage is less

(c) the load will be sustained in absence of any effort

(d) the load is easily lifted

6.22 A power screw is specified by

(a) major diameter x pitch (b) major diameter x length

(c) mean diameter x pitch (d) nominal major diameter

Answers:

6.1 (a)	6.2 (d)	6.3 (c)	6.4 (b)	6.5 (c)
6.6 (b)	6.7 (a)	6.8 (a)	6.9 (a)	6.10 (a)
6.11 (b)	6.12 (c)	6.13 (b)	6.14 (d)	6.15 (b)
6.16 (d)	6.17(a)	6.18 (b)	6.19 (d)	6.20 (b)
6.21 (b)	6.22 (a)			

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7 THREADED JOINTS

7.1 Setscrews are

(a) similar to tap bolts except that a greater variety of shapes of heads are available

(b) slotted for screw driver and generally used with a nut

(c) used to prevent relative motion between parts

(d) similar to studs

7.2 A self-locking screw has

(a) fine threads (b) coarse threads

(c) two nuts

(d) coefficient of friction more than tangent of lead angle

7.3 The designation M 36 x 2 means

(a) metric fine threads of 36 mm outside diameter and 2 mm pitch

(b) metric coarse threads of 36 mm outside diameter and 2 mm pitch

(c) metric threads of 36 mm pitch diameter and 2 mm pitch.

(d) metric threads of 36 mm core diameter and 2 mm pitch

7.4 The designation M 20 means

(a) metric coarse threads of 20 mm outside diameter

(b) metric fine threads of 20 mm outside diameter

(c) metric threads of 20 mm core diameter

(d) metric threads of 20 mm pitch diameter

7.5 The largest diameter of external or internal screw thread is called

(a) major diameter (b) minor diameter

(c) pitch diameter (d) none of the above

7.6 The pitch diameter of external or internal screw thread is

(a) largest diameter (b) smallest diameter

(c) effective diameter (d) mean diameter

7.7 A screw is specified by

(a) major diameter (b) minor diameter

(c) pitch diameter (d) mean diameter

7.8 A washer is specified by

(a) outer diameter (b) inner diameter

(c) thickness (d) mean diameter

7.9 Machine bolts are

(a) through bolts with rough shank and used with nut

(b) are used to prevent relative motion between two parts

(c) similar to stud

(d) turned into a threaded hole in one of the parts

7.10 Jam nut is a locking device in which

(a) a smaller nut is tightened against main nut creating friction at the contacting surface

(b) a split pin is passed through diametrically opposite slots in nut and a hole in bolt and the two ends of split pin are separated and bent back on nut

(c) a slot is cut in the middle of nut along the length and a cap screw is provided to tighten the two parts of nut separated by slot

(d) an elastic piece is tightened in the nut by a setscrew

7.11 Castle nut is a locking device in which

(a) a smaller nut is tightened against main nut creating friction at the contacting surface

(b) a split pin is passed through diametrically opposite slots in nut and a hole in bolt and the two ends of split pin are separated and bent back on nut

(c) a slot is cut in the middle of nut along the length and a cap screw is provided to tighten the two parts of nut separated by slot

(d) an elastic piece is tightened in the nut by a setscrew

7.12 Split nut is a locking device in which

(a) a smaller nut is tightened against main nut creating friction at the contacting surface

(b) a split pin is passed through diametrically opposite slots in nut and a hole in bolt and the two ends of split pin are separated and bent back on nut

(c) a slot is cut in the middle of nut along the length and a cap screw is provided to tighten the two parts of nut separated by slot

(d) an elastic piece is tightened in the nut by a setscrew

7.13 An ordinary bolt is converted into bolt of uniform strength by

(a) reducing the shank diameter of bolt to the core diameter of threads

(b) drilling a hole all along the length of shank in unthreaded portion

(c) increasing the stress in the shank portion of bolt

(d) using all three methods

7.14 The coupler of turnbuckle has

(a) right hand threads on both ends

(b) left hand threads on both ends

(c) left hand threads on one end and right hand threads on other end

(d) no threads

7.15 Eyebolts are used

(a) to prevent relative motion between two parts

(b) to absorb shocks and vibrations

(c) to lift and transport machines and heavy objects on shop floor

(d) to prevent loosening of threads in bolted assembly

7.16 A stud is

(a) screw with long threads (b) screw with circular head

(c) screw with hexagonal head (d) headless screw with threads on both sides

7.17 When a nut is tightened by placing a washer below it, the shank of bolt is subjected to

(a) direct shear stress (b) torsional shear stress

(c) tensile stress (d) compressive stress

7.18 While designing screw threads, adequate length of engaged threads between the screw and nut is provided so as to prevent failure of threads due to

(a) direct shear stress (b) torsional shear stress

(c) tensile stress (d) compressive stress

7.19 When a nut is tightened by placing a washer below it, the threads of bolt are subjected to

(a) direct shear stress (b) torsional shear stress

(c) tensile stress (d) compressive stress

7.20 When the shear strength of nut is equal to the tensile strength of bolt, the height of nut (h) should be

(a) $h = 0.5\ d_c$ (b) $h = 0.25\ d_c$

(c) $h = 0.75\ d_c$ (d) $h = d_c$

where d_c = core diameter of threads

7.21 When the shear strength of nut is half the tensile strength of bolt, the height of nut (h) should be

(a) $h = 0.5\ d_c$ (b) $h = 0.25\ d_c$

(c) $h = 0.75\ d_c$ (d) $h = d_c$

where d_c = core diameter of threads

7.22 Standard coarse threads are equally strong in failure by shear and failure by tension, if the height of nut is

(a) h = 0.4 d (b) h = 0.6 d

(c) h = 0.5 d (d) h = d

where d is nominal diameter of threads

7.23 Allen key is used to tighten

(a) cap screw with hexagonal head

(b) cap screw with filister head

(c) cap screw with hexagonal socket head

(d) cap screw with flat head

7.24 The resultant axial force on bolt depends upon

(a) initial tension (b) external applied load

(c) stiffness of bolt and parts held by bolt (d) all three factors

7.25 If the stiffness of parts held together by bolt is too small compared with the stiffness of bolt (soft gasket), then the resultant load on the bolt is equal to

(a) initial tension

(b) external load

(c) sum of initial tension and external load

(d) higher of initial tension and external load

7.26 If the stiffness of parts held together by bolt is too large compared with the stiffness of bolt (hard gasket), then the resultant load on the bolt is equal to

(a) initial tension

(b) external load

(c) sum of initial tension and external load

(d) higher of initial tension and external load

7.27 The connecting rod bolts are tightened up so that the initial tightening stress

(a) approaches yield point

(b) approaches endurance limit

(c) approaches (yield point stress/factor of safety)

(d) approaches (endurance limit stress/factor of safety)

7.28 The connecting rod bolts are tightened up with the initial tension greater than external load so that

(a) failure of bolt will be static

(b) the resultant load on bolt will not be affected by external cyclic load

(c) the bolt will not fail by fatigue although the external load is fluctuating

(d) All the three

7.29 The connecting rod bolts of internal combustion engines have their shank diameter reduced at some places along the length in order to

(a) reduce weight

(b) reduce inertia forces

(c) increase shock absorbing capacity

(d) none of the above

7.30 The shock absorbing capacity of a bolt can be increased by

(a) increasing shank diameter

(b) making shank diameter equal to core diameter of threads

(c) using castle nut in place of ordinary hexagonal nut

(d) using a washer

7.31 The shock absorbing capacity of a bolt can be increased by

(a) increasing stress in shank

(b) increasing stress in core diameter of threads

(c) decreasing stress in shank

(d) decreasing stress in core diameter of threads

7.32 The resilience of a bolt can be increased by

(a) increasing shank diameter

(b) increasing length of shank portion of bolt

(c) increasing core diameter of threads

(d) using a washer

7.33 The inner diameter of washer is

(a) equal to the nut size

(b) more than the nut size

(c) less than the nut size

(d) independent of the nut size

Answers:

7.1 (c)	7.2 (d)	7.3 (a)	7.4 (a)	7.5 (a)
7.6 (c)	7.7 (a)	7.8 (b)	7.9 (a)	7.10 (a)
7.11 (b)	7.12 (c)	7.13 (d)	7.14 (c)	7.15 (c)
7.16 (d)	7.17 (c)	7.18 (a)	7.19 (a)	7.20 (a)
7.21 (b)	7.22 (a)	7.23 (c)	7.24 (d)	7.25 (c)
7.26 (d)	7.27 (a)	7.28 (d)	7.29 (c)	7.30 (b)
7.31 (a)	7.32 (b)	7.33 (b)		

8 WELDED & RIVETED JOINTS

Welded joints:

8.1 In fusion welding process

(a) only heat is used (b) only pressure is used

(c) combination of heat and pressure is used

(d) all three methods are used

8.2 The principle of applying heat and pressure is used in

(a) spot welding (b) seam welding

(c) electric resistance welding (d) all three methods

8.3 In automobile-body work, the type of welding generally used is

(a) gas welding (b) electric arc welding

(c) electric resistance welding (d) thermit welding

8.4 The weakest plane in a fillet weld is

(a) the throat (b) side parallel to the force

(c) smaller of two sides (d) side normal to the force

8.5 The cross-section of a standard fillet weld is a triangle with base angles of

(a) 45^0 and 45^0 (b) 60^0 and 30^0

(c) 50^0 and 40^0 (d) 20^0 and 70^0

8.6 The size of a fillet weld is given by,

(a) throat of fillet (b) smaller side of triangle

(c) hypotenuse of triangle (d) bigger side of triangle

8.7 In transverse fillet welded joint, the size of weld is equal to

(a) 0.5 x throat of weld (b) throat of weld

(c) 2 x throat of weld (d) $\sqrt{2}$ x throat of weld

8.8 The transverse fillet welds are designed for

(a) tensile strength (b) shear strength

(c) bending strength (d) compressive strength

8.9 The parallel fillet welds are designed for

(a) tensile strength (b) shear strength

(c) bending strength (d) compressive strength

8.10 In butt joint, the size of weld is equal to

(a) 0.5 x throat of weld (b) throat of weld

(c) 2 x throat of weld (d) $\sqrt{2}$ x throat of weld

8.11 A double fillet welded joint with parallel fillet welds of length l and leg h is subjected to a tensile force of P. The tensile stress in weld is given by,

(a) $\frac{\sqrt{2}\,P}{h\,l}$ (b) $\frac{P}{2hl}$ (c) $\frac{P}{\sqrt{2}\,hl}$ (d) $\frac{2P}{hl}$

8.12 For a parallel load on fillet weld of equal legs, the plane of maximum shear occurs at

(a) 22.5^0 (b) 45^0 (c) 30^0 (d) 60^0

8.13 In fillet welded joint, the throat of weld as compared to the size of weld is

(a) about 0.5 times (b) about 0.707 times

(c) about same size (d) about $\sqrt{2}$ times

8.14 When mild steel components are welded, the ratio of strength of the weld material to that of parent body is

(a) more than one (b) less than one

(c) equal to one (d) none of the three

Riveted joints:

8.15 A rivet is specified by

(a) shank diameter (b) length of rivet

(c) type of head (d) material of rivet

8.16 The holes in the plates for riveted joint are made by

(a) flame cutting (b) turning

(c) punching and drilling (d) reaming

8.17 The diameter of the rivet hole is

(a) equal to nominal diameter of rivet

(b) slightly less than nominal diameter of rivet

(c) slightly more than nominal diameter of rivet

(d) independent of nominal diameter of rivet

8.18 A rivet head used in boilers and pressure vessels is

(a) snap head (b) countersunk head

(c) flat head (d) half countersunk head

8.19 Pan head rivets are used in

(a) ship hulls (b) light sheet metal work

(c) structural work (d) air conditioning ducts

8.20 Flat head rivets are used in

(a) ship hulls (b) light sheet metal work

(c) structural work (d) air conditioning ducts

8.21 Rivets are usually made of

(a) high carbon steel (b) alloy steel

(c) cast iron (d) mild steel

8.22 Rivets are usually made of

(a) conformable material (b) hard material

(c) brittle material (d) ductile material

8.23 According to Unwin's formula, the relationship between the diameter of rivet (d) and thickness of cylinder wall (t) is

(a) $d = 5\sqrt{t}$ (b) $d = 6\sqrt{t}$

(c) $d = 1.6\sqrt{t}$ (d) $d = \sqrt{t}$

8.24 The distance between the edge of plate and the centerline of rivets in the nearest row is called

(a) pitch (b) margin

(c) transverse pitch (d) diagonal pitch

8.25 The distance between the center of one rivet and the center of adjacent rivet in the same row is called

(a) pitch (b) margin

(c) transverse pitch (d) diagonal pitch

8.26 The objective of caulking and fullering is to make the riveted joint,

(a) free from residual stresses (b) leak proof

(c) strong (d) permanent

8.27 The edges of boiler plates for fullering and caulking are beveled at an angle of

(a) 45^0 (b) 60^0 (c) 70^0 to 75^0 (d) 30^0

8.28 The edges of boiler plates are beveled at an angle of 70^0 to 75^0 for

(a) caulking and fullering (b) safety

(c) to facilitate riveting (d) to reduce stress concentration

8.29 A lap joint is always subjected to

(a) bending moment (b) torsional moment

(c) tensile force (d) compressive force

8.30 In single riveted lap joint, the rivet is subjected to

(a) double shear (b) single shear

(c) either single or double shear (d) tensile stress

8.31 In double-strap single-riveted butt joint, the rivet is subjected to

(a) double shear (b) single shear

(c) either single or double shear (d) tensile stress

8.32 The shear resistance of one rivet in double shear is

(a) 2.5 times its resistance in single shear

(b) two times its resistance in single shear

(c) 1.875 times its resistance in single shear

(d) 1.5 times its resistance in single shear

8.33 Which of the double-strap butt joint used in boiler shell has highest efficiency,

(a) single-riveted (b) double-riveted

(c) triple-riveted (d) quadruple-riveted

8.34 The strength of solid plate per pitch length is given by

(a) $p t \sigma_t$ (b) $p d \sigma_t$

(c) $p(d-t)\sigma_t$ (d) $2 p t \sigma_t$

where p = pitch of rivets d = diameter of rivet

t = plate thickness σ_t = tensile stress in the plate

8.35 The purpose of longitudinal butt joint in boiler shell is

(a) to make cylindrical ring from steel plate

(b) to increase the length of boiler shell by connecting one ring to another

(c) to make diameter and length of boiler shell

(d) to connect openings to shell

8.36 The purpose of circumferential lap joint in boiler shell is

(a) to make cylindrical ring from steel plate

(b) to increase the length of boiler shell by connecting one ring to another

(c) to make diameter and length of boiler shell

(d) to connect openings to shell

8.37 Lowest value of joint efficiency is assumed in case of

(a) single riveted butt joint

(b) double riveted lap joint

(c) double riveted butt joint

(d) single riveted lap joint

Answers:

8.1 (a)	8.2 (d)	8.3 (c)	8.4 (a)	8.5 (a)
8.6 (b)	8.7 (d)	8.8 (a)	8.9 (b)	8.10 (b)
8.11 (c)	8.12 (b)	8.13 (b)	8.14 (a)	8.15 (a)
8.16 (c)	8.17 (c)	8.18 (a)	8.19 (a)	8.20 (b)
8.21 (d)	8.22 (d)	8.23 (b)	8.24 (b)	8.25 (a)
8.26 (b)	8.27 (c)	8.28 (a)	8.29 (a)	8.30 (b)
8.31 (a)	8.32 (c)	8.33 (d)	8.34 (a)	8.35 (a)
8.36 (b)	8.37 (d)			

9 SHAFTS, KEYS AND COUPLINGS

9.1 Two shafts A and B are made of same material. The diameter of shaft A is twice that of B. The torque transmitted by shaft A will be

(a) twice that of B (b) four times that of B

(c) eight times that of B (d) sixteen times that of B

9.2 Two shafts A and B are made of same material. The diameters of shaft A and B are d_A and d_B respectively. The ratio of torque transmitted by shaft A to that of shaft B will be

(a) $\left(\frac{d_A}{d_B}\right)$ (b) $\left(\frac{d_A}{d_B}\right)^2$ (c) $\left(\frac{d_A}{d_B}\right)^3$ (d) $\left(\frac{d_A}{d_B}\right)^4$

9.3 The shafts will have same strength on the basis of torsional rigidity, if

(a) diameter and length of both shafts is same

(b) material of both shafts is same

(c) angle of twist for both shafts is same

(d) all of above conditions are satisfied

9.4 The angle of twist for a transmission shaft is inversely proportional to

(a) shaft diameter

(b) (shaft diameter)^2

(c) (shaft diameter)^3

(d) (shaft diameter)^4

9.5 A transmission shaft subjected to pure bending moment should be designed on the basis of

(a) maximum principal stress theory

(b) maximum shear stress theory

(c) distortion energy theory

(d) Goodman or Soderberg diagrams

9.6 The maximum shear stress induced in a transmission shaft, subjected to bending stress (σ_b) and torsional shear stress (τ), is given by,

(a) $\sqrt{\left(\frac{\sigma_b}{2}\right)^2 + (\tau)^2}$ (b) $\sqrt{(\sigma_b)^2 + (\tau)^2}$

(c) $\frac{\sigma_b}{2} + \frac{1}{2}\sqrt{\left(\frac{\sigma_b}{2}\right)^2 + (\tau)^2}$ (d) none of above

9.7 Which is the correct statement,

(a) cold rolling produces stronger shafts than hot rolling

(b) hot rolling produces stronger shafts than cold rolling

(c) cold rolling and hot rolling produces equally strong shafts

(d) strength of shaft is independent of rolling processes

9.8 Maximum shear stress theory is used for

(a) cast iron shafts (b) steel shafts

(c) flexible shafts (d) plastic shafts

9.9 Maximum principle stress theory is used for

(a) cast iron shafts (b) steel shafts

(c) Aluminum shafts (d) plastic shafts

9.10 A transmission shaft subjected to bending and torsional moments should be designed on the basis of

(a) Rankine theory

(b) Coulomb, Tresca and Guest theory

(c) Huber von Mises theory

(d) Goodman or Soderberg diagrams

9.11 A transmission shaft is subjected to bending moment (M_b) and torsional moment (M_t). The equivalent torsional moment is given by,

(a) $\sqrt{(M_b + M_t)}$ (b) $\sqrt{(M_b)^2 + (M_t)^2}$

(c) $[M_b + M_t]$ (d) $M_b + \sqrt{(M_b)^2 + (M_t)^2}$

9.12 A transmission shaft is subjected to bending moment (M_b) and torsional moment (M_t). The equivalent bending moment is given by,

(a) $\sqrt{(M_b + M_t)}$ (b) $\sqrt{(M_b)^2 + (M_t)^2}$

(c) $[M_b + M_t]$ (d) $M_b + \sqrt{(M_b)^2 + (M_t)^2}$

9.13 The function of key is

(a) to connect transmission shaft to a rotating machine elements like gears

(b) to transmit torque from shaft to hub and vice versa

(c) to prevent relative rotational motion between the shaft and the connected element

(d) all of above three functions

9.14 The standard taper for sunk key is

(a) 1 in 100 (b) 1 in 50

(c) 1 in 10 (d) 1 in 1000

9.15 In case of sunk key,

(a) the keyway is cut in the shaft only

(b) the keyway is cut in the hub only

(c) the keyway is cut in both the shaft and the hub

(d) none of the above

9.16 In case of sunk key, power is transmitted by means of,

(a) friction force

(b) shear resistance of key

(c) torsional shear resistance of key

(d) tensile force

9.17 The standard width for square or flat key in terms of shaft diameter (d) is,

(a) d (b) d/2 (c) d/4 (d) d/8

9.18 The standard height for flat key in terms of shaft diameter (d) is,

(a) d (b) d/2 (c) d/4 (d) d/6

9.19 The standard length for square or flat key in terms of shaft diameter (d) is,

(a) d (b) 2d (c) 1.5d (d) 2.5d

9.20 The type of key used when the gear is required to slide on the shaft is

(a) sunk key (b) feather key

(c) Woodruff key (d) Kennedy key

9.21 The key in the form of semi-circular disk of uniform thickness is called,

(a) sunk key (b)feather key

(c) Woodruff key (d) Kennedy key

9.22 The key, which fits in the keyway of hub, only is called,

(a) saddle key (b)feather key

(c) Woodruff key (d) Kennedy key

9.23 In case of saddle key,

(a) the keyway is cut in the shaft only

(b) the keyway is cut in the hub only

(c) the keyway is cut in both the shaft and the hub

(d) none of the above

9.24 In case of saddle key, power is transmitted by means of,

(a) friction force

(b) shear resistance of key

(c) crushing resistance of key

(d) tensile force

9.25 The key, which consists of two square keys, is called,

(a) saddle key (b)feather key

(c) Woodruff key (d) Kennedy key

9.26 Kennedy key is used in

(a) light duty applications (b) heavy duty applications

(c) high speed applications (d) precision equipments

9.27 The compressive stress induced in a square key is,

(a) equal to shear stress (b) four times of shear stress

(c) twice of shear stress (d) half of shear stress

9.28 The keyway,

(a) reduces strength of shaft (b) reduces rigidity of shaft

(c) increases stress concentration (d) all of above

9.29 Splines are used if,

(a) the power to be transmitted is high

(b) the torque to be transmitted is high

(c) the speed is high

(d) there is relative motion between shaft and hub

9.30 Splines are commonly used in

(a) machine tool gear box

(b) automobile gear box

(c) hoist and crane gear box

(d) bicycle

9.31 While designing a shaft, key and hub, care is taken so that

(a) shaft is the weakest component

(b) key is the strongest component

(c) key is the weakest component

(d) the hub is the weakest component

9.32 A flange coupling is used

(a) for intersecting shafts

(b) for collinear shafts

(c) for small shafts rotating at slow speeds

(d) for parallel shafts

9.33 While designing a flange coupling, care is taken so that

(a) shaft is the weakest component

(b) bolts are the weakest component

(c) key is the weakest component

(d) the flange is the weakest component

9.34 A bushed-pin type flange coupling is used

(a) for intersecting shafts

(b) when the shafts are not in exact alignment

(c) for small shafts rotating at slow speeds

(d) for parallel shafts

9.35 A muff coupling is

(a) rigid coupling

(b) flexible coupling

(c) shock absorbing coupling

(d) none of the above

9.36 In case of clamp coupling, power is transmitted by means of,

(a) friction force

(b) shear resistance

(c) crushing resistance

(d) none of the above

Answers:

9.1 (c)	9.2 (c)	9.3 (d)	9.4 (d)	9.5 (a)
9.6 (a)	9.7 (a)	9.8 (b)	9.9 (a)	9.10 (b)
9.11 (b)	9.12 (d)	9.13 (d)	9.14 (a)	9.15 (c)
9.16 (b)	9.17 (c)	9.18 (d)	9.19 (c)	9.20 (b)
9.21 (c)	9.22 (a)	9.23 (b)	9.24 (a)	9.25 (d)
9.26 (b)	9.27 (c)	9.28 (d)	9.29 (d)	9.30 (b)
9.31 (c)	9.32 (b)	9.33 (c)	9.34 (b)	9.35 (a)
9.36 (a)				

10 SPRINGS

10.1 Springs are used,

(a) to absorb shocks and vibrations

(b) to store and release energy

(c) to measure force

(d) for any of the above functions

10.2 The type of spring used to absorb shocks and vibrations in vehicles is,

(a) helical extension spring (b) multi-leaf spring

(c) spiral spring (d) Bellevile (coned disk) spring

10.3 The type of spring used to absorb and release energy in mechanical watches is,

(a) helical extension spring (b) multi-leaf spring

(c) spiral spring (d) helical torsion spring

10.4 The type of spring used in door hinges is,

(a) helical extension spring (b) multi-leaf spring

(c) spiral spring (d) helical torsion spring

10.5 The type of spring used to measure weights in spring balance is,

(a) helical extension spring (b) multi-leaf spring

(c) spiral spring (d) helical torsion spring

10.6 The type of spring used in valve mechanism is,

(a) helical compression spring (b) multi-leaf spring

(c) spiral spring (d) helical torsion spring

10.7 The type of spring used in vehicle clutches is,

(a) helical compression spring (b) Belleville spring

(c) a and b (d) helical torsion spring

10.8 When the helical compression spring is subjected to axial compressive force, the type of stress induced in the spring wire is,

(a) tensile stress (b) compressive stress

(c) bending stress (d) torsional shear stress

10.9 When the helical extension spring is subjected to axial tensile force, the type of stress induced in the spring wire is,

(a) tensile stress (b) compressive stress

(c) bending stress (d) torsional shear stress

10.10 The maximum shear stress in spring wire is induced at

(a) inner surface of the coil

(b) outer surface of the coil

(c) central surface of the coil

(d) end coils

10.11 When the helical torsion spring is subjected to torque, the type of stress induced in the spring wire is,

(a) tensile stress (b) compressive stress

(c) bending stress (d) torsional shear stress

10.12 The leaves of multi-leaf spring are subjected to

(a) tensile stress (b) compressive stress

(c) bending stress (d) torsional shear stress

10.13 The spring operates

(a) within plastic limit (b) within elastic limit

(c) within elasto-plastic limit (d) within visco-elastic limit

10.14 Wahl factor to account for direct shear stress and stress concentration due to curvature for helical springs is given by,

(a) $\frac{4C-1}{4C-4}+\frac{0.615}{C}$ (b) $\frac{4C-1}{4C+4}+\frac{0.615}{C}$

(c) $\frac{4C+1}{4C-4}+\frac{0.615}{C}$ (d) $\frac{4C+1}{4C+4}+\frac{0.615}{C}$

where C is spring index

10.15 Two springs of stiffness k_1 and k_2 are connected in series, the combined stiffness of the connection is given by,

(a) $\frac{k_1 k_2}{k_1+k_2}$ (b) $\frac{k_1 k_2}{k_1-k_2}$ (c) k_1+k_2 (d) $\frac{k_1+k_2}{k_1 k_2}$

10.16 Two springs of stiffness k_1 and k_2 are connected in parallel, the combined stiffness of the connection is given by,

(a) $\frac{k_1 k_2}{k_1+k_2}$ (b) $\frac{k_1 k_2}{k_1-k_2}$ (c) k_1+k_2 (d) $\frac{k_1+k_2}{k_1 k_2}$

10.17 When a helical spring is cut into two halves, the stiffness of each half spring will be,

(a) same as original spring (b) double of original spring

(c) half of original spring (d) one fourth of original spring

10.18 When two concentric springs are made of same material, having same free length and compressed equally by axial load, then the load shared by each spring is proportional to

(a) spring index of each spring

(b) wire diameter of each spring

(c) mean coil diameter of each spring

(d) square of wire diameter of each spring

10.19 The function of automotive multi-leaf spring is

(a) to measure the force

(b) to store and release energy

(c) to absorb shocks and vibrations

(d) to activate the mechanism

10.20 The stiffness of spring is,

(a) deflection per unit of axial force

(b) force per unit cross-sectional area of spring

(c) ratio of mean coil diameter to wire diameter

(d) force required to produce unit deflection

10.21 The spring index is,

(a) ratio of wire diameter to mean coil diameter

(b) force per unit cross-sectional area of spring

(c) ratio of mean coil diameter to wire diameter

(d) force required to produce unit deflection

10.22 The ends of spring, which are in contact with the seat, are,

(a) active coils

(b) inactive coils

(c) transmit maximum force

(d) do not transmit any force

10.23 Concentric springs are used,

(a) to achieve greater load carrying capacity with given space

(b) to achieve fail-safe system

(c) to eliminate surge

(d) any of the above objective

10.24 The type of spring used to achieve greater load carrying capacity within given space is

(a) spiral spring

(b) springs in series

(c) multi-leaf spring

(d) concentric springs

10.25 The type of spring used to achieve any linear or non-linear load-deflection characteristic is

(a) spiral spring

(b) non-ferrous spring

(c) Belleville (coned disk) spring

(d) torsion spring

Answers:

10.1 (d)	10.2 (b)	10.3 (c)	10.4 (d)	10.5 (a)
10.6 (a)	10.7 (c)	10.8 (d)	10.9 (d)	10.10 (a)
10.11 (c)	10.12 (c)	10.13 (b)	10.14 (a)	10.15 (a)
10.16 (c)	10.17 (b)	10.18 (d)	10.19 (c)	10.20 (d)
10.21 (c)	10.22 (b)	10.23 (d)	10.24 (d)	10.25 (c)

11 FRICTION CLUTCHES

11.1 A jaw clutch is

(a) friction clutch (b) disk clutch

(c) positive action clutch (d) cone clutch

11.2 The clutch used in trucks is

(a) centrifugal clutch (b) cone clutch

(c) multi-plate clutch (d) single plate clutch

11.3 The clutch used in scooters is

(a) multi-plate clutch (b) single plate clutch

(c) centrifugal clutch (d) cone clutch

11.4 The friction material of the clutch should have

(a) high coefficient of friction

(b) low coefficient of friction

(c) high surface hardness

(d) high endurance limit strength

11.5 The torque transmitted by single plate clutch, working on uniform wear criterion, is given by,

(a) $0.25\mu P R_{min}$ (b) $0.50\mu P R_{min}$

(c) $0.75\mu P R_{min}$ (d) $\mu P R_{min}$

where,

μ = coefficient of friction

P = axial force required to hold the friction surfaces together

R_{min} = mean radius of friction surfaces

11.6 In case of multi-plate clutch if n_1 is number of disks on driving shaft and n_2 is the number of disks on driven shaft, then the number of pairs of contacting surfaces is given by

(a) $n_1 + n_2$ (b) $n_1 + n_2 + 1$

(c) $n_1 + n_2 - 1$ (d) $n_1 + n_2 - 2$

11.7 The cone clutches have become obsolete because

(a) strict requirement of coaxiality of two shafts

(b) difficult to disengage

(c) difficult construction

(d) none of the above

11.8 In the beginning of engagement of a centrifugal clutch,

(a) the centrifugal force on shoe is slightly more than spring force

(b) the centrifugal force on shoe is equal to spring force

(c) the centrifugal force on shoe is less than spring force

(e) none of the above

11.9 In the running condition, the net force acting on the drum of centrifugal clutch is equal to

(a) the centrifugal force on shoe

(b) the centrifugal force on shoe minus spring force

(c) the centrifugal force on shoe plus spring force

(d) the spring force

11.10 In case of multi-disk clutches, oil is used,

(a) to reduce the friction

(b) to carry away the heat

(c) to lubricate the contacting surfaces

(d) for all above functions

11.11 Torque transmitting capacity of clutch depends upon

(a) coefficient of friction

(b) dimensions of friction lining

(c) axial force provided to engage the clutch

(d) all the above three factors

11.12 The friction moment in a clutch with uniform wear as compared to friction moment with uniform pressure is

(a) more (b) equal

(c) less (d) more or less depending on speed

11.13 The friction radius for new clutch compared to worn out clutch will be

(a) more (b) equal

(c) less (d) more or less depending on size of clutch

11.14 In case of single plate new clutches and brakes, the friction radius is equal to

(a) $\frac{1}{3}\frac{(D^3-d^3)}{(D^2-d^2)}$ (b) $\frac{1}{2}\frac{(D^3-d^3)}{(D^2-d^2)}$

(c) $\frac{1}{4}\frac{(D^3-d^3)}{(D^2-d^2)}$ (d) $\frac{1}{4}(D+d)$

where D and d are outer and inner diameters of friction lining respectively.

11.15 In case of single plate worn out clutches and brakes, the friction radius is equal to

(a) $\frac{1}{3}\frac{(D^3-d^3)}{(D^2-d^2)}$ (b) $\frac{1}{2}\frac{(D^3-d^3)}{(D^2-d^2)}$

(c) $\frac{1}{4}\frac{(D^3-d^3)}{(D^2-d^2)}$ (d) $\frac{1}{4}(D+d)$

where D and d are outer and inner diameters of friction lining respectively.

11.16 In case of cone clutch, the friction torque, as per uniform pressure theory, is equal to

(a) $\frac{1}{3}\frac{\mu P}{\sin\alpha}\frac{(D^3 - d^3)}{(D^2 - d^2)}$ (b) $\frac{1}{2}\frac{\mu P}{\sin\alpha}\frac{(D^3 - d^3)}{(D^2 - d^2)}$

(c) $\frac{1}{4}\frac{\mu P}{\sin\alpha}\frac{(D^3 - d^3)}{(D^2 - d^2)}$ (d) $\frac{1}{4}\frac{\mu P}{\sin\alpha}(D + d)$

where,

μ = coefficient of friction

P = axial force required to hold the friction surfaces together

D = outer diameter of friction lining

d = inner diameter of friction lining

α = semi-cone angle of clutch

11.17 The commonly used angle between leather or asbestos friction lining surface and axis of cone clutch for a cone clutch is

(a) 14.5^0 (b) 20^0 (c)12.5^0 (d) 45^0

11.18 In case of cone clutch, a relatively small axial force can transmit a given torque, if the semi-cone angle is

(a) more (b) equal

(c) less (d) more or less depending on size of clutch

Answers:

11.1 (c)	11.2 (d)	11.3 (a)	11.4 (a)	11.5 (d)
11.6 (c)	11.7 (a)	11.8 (a)	11.9 (b)	11.10 (d)
11.11 (d)	11.12 (c)	11.13 (a)	11.14 (a)	11.15 (d)
11.16 (a)	11.17 (c)	11.18 (c)		

12 BRAKES

12.1 The brake used in railway coaches is

(a) shoe brake (b) block brake

(c) band brake (d) disk brake

12.2 The brake used in most of the automobile vehicles is

(a) internal expanding shoe brake (b) block brake

(c) band brake (d) disk brake

12.3 The brake used in most of motorcycles is

(a) internal expanding brake (b) block brake

(c) band brake (d) disk brake

12.4 The heat generated in brake depends upon

(a) (pv) (b) (p/v) (c) (pv^2) (d) $\frac{1}{2}pv^2$

where p = intensity of normal pressure (N/mm^2)

v = rubbing speed (m/mim)

12.5 The friction material of the brake should have

(a) high coefficient of friction

(b) low coefficient of friction

(c) high surface hardness

(d) high endurance limit strength

12.6 When the frictional force helps to apply the brake, the brake is said to be

(a) partially self energizing (b) self locking

(c) back-stop (d) self acting

12.7 For a block brake with long shoe, the equivalent coefficient of friction is

(a) $\mu\left(\dfrac{4\sin\theta}{2\theta+\sin 2\theta}\right)$ (b) $\mu\left(\dfrac{2\sin\theta}{2\theta+\sin 2\theta}\right)$

(c) $\mu\left(\dfrac{4\sin 2\theta}{2\theta+\sin 2\theta}\right)$ (d) $\mu\left(\dfrac{4\sin\theta}{4\theta+\sin 2\theta}\right)$

where,

μ = coefficient of friction

θ = semi-block angle

12.8 In block brakes, the ratio of shoe width and drum diameter is kept between

(a) 0.1 to 0.25 (b) 0.25 to 0.50

(c) 0.50 to 0.75 (d) 0.75 to 1.0

12.9 The percentage of total brake effort that results from self-energizing action depends upon

(f) the location of brake arm pivot

(g) coefficient of friction

(h) direction of rotation of brake drum

(i) all of the above

12.10 In order to prevent the brake arm from grabbing, the moment of friction force about the brake arm pivot should be

(a) less than the total required braking effort

(b) more than the total required braking effort

(c) equal to the total required braking effort

(d) none of the above

Answers:

12.1 (b)	12.2 (a)	12.3 (d)	12.4 (a)	12.5 (a)
12.6 (a)	12.7 (a)	12.8 (b)	12.9 (d)	12.10 (a)

13 BELT DRIVES

13.1 The power transmitted by belt drive depends upon

(a) belt velocity (b) initial belt tension

(c) arc of contact (d) all of the above

13.2 The suitable material for belt in agricultural machinery is

(a) leather (b) rubber

(c) cotton duck (d) balata gum

13.3 The suitable material for belt used in floor mill is

(a) leather (b) rubber

(c) canvas or cotton duck (d) balata gum

13.4 When the belt speed increases

(a) power transmitted increases

(b) power transmitted decreases

(c) power transmitted increases to a maximum value and then decreases

(d) power transmitted remains the same

13.5 The creep in the belt is due to

(a) effect of temperature on belt

(b) material of belt

(c) unequal extensions in the belt due to tight and slack side tensions

(d) stresses beyond elastic limit of belt material

13.6 The coefficient of friction in belt drive depends upon

(a) material of belt (b) material of pulley

(c) materials of belt and pulley (d) belt velocity

13.7 Which is positive drive?

(a) flat belt drive (b) V belt drive

(c) crossed belt drive (d) timing belt

13.8 Fabric belts are used in industrial applications because

(a) they are cheap

(b) they can work at high temperature

(c) they are unaffected by moisture and humidity

(d) none of the above

13.9 The ratio of belt tensions (P_1 / P_2) considering centrifugal force in flat belt is given by

(a) $\dfrac{P_1 - m v^2}{P_2 - m v^2} = e^{f\alpha}$ (b) $\dfrac{P_1}{P_2} = e^{f\alpha}$

(c) $\dfrac{P_1}{P_2} = e^{-f\alpha}$ (d) $\dfrac{P_1 - m v^2}{P_2 - m v^2} = e^{-f\alpha}$

where m = mass of belt per metre (kg/m)

v = belt velocity (m/s)

f = coefficient of friction

α = angle of wrap (radians)

13.10 The condition for maximum power transmission is that the maximum tension in the flat belt should be equal to

(a) 3 P_c (b) P_c (c) $P_c/3$ (d) 2 P_c

where P_c = tension in belt due to centrifugal force

13.11 When the belt is transmitting maximum power,

(a) the tension in tight side is twice the centrifugal tension

(b) the tension in slack side is equal to the centrifugal tension

(c) the tension in tight side is thrice the centrifugal tension

(d) none of the above

13.12 When the belt is transmitting maximum power, the belt speed should be

(a) $\sqrt{\frac{P_{max}}{2\,m}}$ (b) $\sqrt{\frac{P_{max}}{3\,m}}$ (c) $\sqrt{\frac{P_{max}}{m}}$ (d) $\sqrt{\frac{3m}{P_{max}}}$

where m = mass of belt per metre (kg/m)

P_{max} = maximum permissible tension in belt (N)

13.13 The stress induced in the belt is,

(a) tensile stress (b) compressive stress

(c) direct shear stress (d) torsional shear stress

13.14 In a horizontal flat belt drive, it is customary to use

(a) bottom side of belt as slack side (b) top side of belt as slack side

(c) idler pulley (d) none of the above

13.15 If P_1 and P_2 are the tight and slack side tensions in the belt, then the initial tension P_i (neglecting centrifugal tension) will be equal to

(a) $P_1 - P_2$ (b) $P_1 + P_2$

(c) $\frac{1}{2}(P_1 + P_2)$ (d) $\frac{1}{2}(P_1 + P_2) + P_c$

where P_c is centrifugal tension

13.16 If P_1 and P_2 are the tight and slack side tensions in the belt, then the initial tension P_i (considering centrifugal tension) will be equal to

(a) $P_1 - P_2$ (b) $P_1 + P_2$

(c) $\frac{1}{2}(P_1 + P_2)$ (d) $\frac{1}{2}(P_1 + P_2) + P_c$

where P_c is centrifugal tension

13.17 If P_1 and P_2 are the tight and slack side tensions in the belt, then the initial tension P_i (according to Barth) will be equal to

(a) $\left[\frac{\sqrt{P_1}+\sqrt{P_2}}{2}\right]^2$ (b) $P_1 + P_2$

(c) $\frac{1}{2}(P_1 + P_2)$ (d) $\frac{1}{2}(P_1 + P_2) + P_c$

where P_c is centrifugal tension

13.18 The included angle between the sides of V belt is

(a) 40^0 (b) 45^0 (c) 38^0 (d) 42^0

13.19 The groove angle of pulleys for V belt is

(a) 34^0 to 36^0 (b) 42^0 to 45^0 (c) more than 40^0 (d) 30^0 to 32^0

13.20 The objective of ‘crowning’ of the flat pulleys of belt drive is to

(a) prevent the belt from running off the pulley

(b) increase the power transmission capacity

(c) increase the belt velocity

(d) prevent the belt joint from damaging the belt surface

13.21 The arms of the pulleys for flat belt drive have

(a) elliptical cross-section (b) major axis in plane of rotation

(c) major axis twice the minor axis (d) all the three characteristics

13.22 The objective of idler pulley in belt drive is to

(a) decrease the tendency of belt to slip

(b) increase the power transmission capacity

(c) increase the wrap angle and belt tension

(d) all the above objectives

13.23 In case of V belt drive

(a) the belt should touch the bottom of groove in the pulley

(b) the belt should not touch the bottom of groove in the pulley

(c) the belt should not touch the sides of groove in the pulley

(d) none of the above

13.24 In case of V belt drive, the belt makes contact at

(a) the bottom of groove in the pulley

(b) the bottom and the sides of groove in the pulley

(c) the sides of groove in the pulley

(d) none of the above

13.25 The centrifugal tension in belts

(a) decreases the power transmitted

(b) increases the power transmitted

(c) increase the wrap angle

(d) increases the belt tension without increasing power transmission

13.26 The belt slip occurs due to

(a) heavy load (b) loose belt

(c) driving pulley too small (d) any one of the above

13.27 For same pulley diameters, centre distance, belt speed and belt and pulley materials,

(a) open belt drive transmits more power than crossed belt drive

(b) crossed belt drive transmits more power than open belt drive

(c) open and crossed belt drives transmit same power

(d) power transmission does not depend upon open and crossed types of constructions

13.28 The power transmitted by the belt drive can be increased by

(a) increasing the initial tension in the belt

(b) dressing the belt to increase the coefficient of friction

(c) increasing wrap angle by using idler pulley

(d) all of the above methods

13.29 A V belt designated as B 4430 L_p has

(a) 4430 mm as diameter of small pulley

(b) 4430 mm as nominal pitch length

(c) 4430 mm as diameter of large pulley

(d) 4430 mm as centre distance between pulleys

13.30 In replacing the V belts, a complete set of new belts is used instead of replacing a single damaged belt because

(a) belts are available in sets

(b) only one belt cannot be fitted with other used belts

(c) the new belt will carry more than its share and result in short life

(d) new and old belts will cause vibrations

Answers:

13.1 (d)	13.2 (b)	13.3 (c)	13.4 (c)	13.5 (c)
13.6 (c)	13.7 (d)	13.8 (a)	13.9 (a)	13.10 (a)
13.11 (a)	13.12 (b)	13.13 (a)	13.14 (b)	13.15 (c)
13.16 (d)	13.17 (a)	13.18 (a)	13.19 (a)	13.20 (a)
13.21 (d)	13.22 (d)	13.23 (b)	13.24 (c)	13.25 (d)
13.26 (d)	13.27 (b)	13.28 (d)	13.29 (b)	13.30 (c)

14 CHAIN DRIVES

14.1 In some applications, chain drives are preferred to belt drive because of

(a) compact construction (b) positive transmission

(c) low cost (d) (a) and (b)

14.2 Following type of chain is used in motorcycle

(a) roller (b) silent

(c) link (d) Ewast

14.3 Silent chain is made of

(a) links and blocks

(b) links, pins, bushes and rollers

(c) links

(d) inverted tooth overlapping links

14.4 A chain drives is used for

(a) short distance (b) long distance

(c) medium distance (d) distance is no criterion for chain drive

14.5 The number of teeth on driving sprocket should be more than 17 in order to

(a) reduce wear (b) reduce interference

(c) reduce variation in chain speed (d) reduce undercutting

14.6 The variation in chain speed is due to

(a) chordal action (b) creep

(c) slip (d) backlash

14.7 The number of teeth on sprocket should be odd in order to

(a) reduce polygonal effect

(b) reduce wear

(c) reduce back sliding

(d) evenly distribute wear on all sprocket teeth

14.8 Main types of failure in roller chain are

(a) breakage of link plates (b) wear of rollers and pins

(c) wear of sprocket wheel (d) all of the above

14.9 As compared to belt drive, the chain drive transmits

(a) more power (b) less power

(c) same power (d) none of the above

14.10 The relationship between pitch circle diameter of the sprocket (D), pitch of the chain (p) and number of teeth on sprocket (z) is given by,

(a) $p = D \sin\left(\frac{180}{z}\right)$ (b) $p = D \sin\left(\frac{90}{z}\right)$

(c) $p = D \sin\left(\frac{360}{z}\right)$ (d) $p = D \sin\left(\frac{120}{z}\right)$

14.11 For a chain drive, to have variation of speed less than 1%, the minimum number of teeth on smaller sprocket should be

(a) 15 (b) 17 (c) 20 (d) 24

14.12 The length of the chain in terms of pitches is approximately given by,

(a) $2\left(\frac{a}{p}\right) + \left(\frac{z_1 + z_2}{2}\right) + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \left(\frac{p}{a}\right)$

(b) $2\left(\frac{a}{p}\right) + \left(\frac{z_1 + z_2}{2}\right) - \left(\frac{z_2 - z_1}{2\pi}\right)^2 \left(\frac{p}{a}\right)$

(c) $p\left(\frac{z_1 + z_2}{2}\right)$

(d) $\left(\frac{2a}{p}\right)$

where z_1, z_2 = number of teeth on small and big sprocket

a = center distance

p = chain pitch

Answers:

14.1 (d)	14.2 (a)	14.3 (d)	14.4 (d)	14.5 (c)
14.6 (a)	14.7 (d)	14.8 (d)	14.9 (a)	14.10 (a)
14.11 (d)	14.12 (a)			

15 ROLLING CONTACT BEARINGS

15.1 In radial bearings, the load acts

(a) along the axis of rotation

(b) perpendicular to the axis of rotation

(c) parallel to the axis of rotation

(d) a and c

15.2 In thrust bearings, the load acts

(a) along the axis of rotation

(b) perpendicular to the axis of rotation

(c) parallel to the axis of rotation

(d) a and c

15.3 Antifriction bearings are

(a) oil lubricated bearings

(b) bush bearings

(c) ball and roller bearings

(d) boundary lubricated bearings

15.4 Rolling contact bearings as compared to sliding contact bearings have

(a) lower starting torque

(b) require considerable axial space

(c) generate less noise

(d) costly

15.5 A bearing number XX10 indicates that the bearing is having

(a) bore diameter of 10 mm

(b) bore diameter of 100 mm

(c) bore diameter of 50 mm

(d) outer diameter of 100 mm

15.6 A bearing is designated by the number 410. It means that it is a bearing of

(a) light series with bore diameter of 10 mm

(b) heavy series with bore diameter of 50 mm

(c) medium series with bore diameter of 50 mm

(d) light series with bore diameter of 50 mm

15.7 Stress induced in the balls or rollers of rolling contact bearing is

(a) torsional shear stress

(b) tensile stress

(c) crushing stress

(d) contact stress

15.8 In an application, the bearing is subjected to radial as well as axial loads. Which type of rolling contact bearings you would suggest?

(a) cylindrical roller bearing

(b) needle roller bearing

(c) thrust ball bearing

(d) taper roller bearing

15.9 In an application, the bearing is subjected to radial as well as axial loads. Which type of rolling contact bearings you would suggest?

(a) angular contact bearing

(b) spherical roller bearing

(c) taper roller bearing

(d) any one of above three types

15.10 The rolling contact bearing is known as

(a) sleeve bearing

(b) thin film bearing

(c) antifriction bearing

(d) bush bearing

15.11 The balls of rolling contact bearings are made of

(a) case hardened steel

(b) plain carbon steel

(c) high carbon chromium steel

(d) free cutting steel

15.12 The rollers of rolling contact bearings are made of

(a) case hardened steel

(b) plain carbon steel

(c) high carbon chromium steel

(d) free cutting steel

15.13 The medium series bearings have ………higher load carrying capacity than light series bearing of the same bore diameter.

(a) 30 to 40 per cent

(b) 20 to 30 per cent

(c) 40 to 50 per cent

(d) 10 to 20 per cent

15.14 The heavy series bearings have ………higher load carrying capacity than medium series bearing of the same bore diameter.

(a) 30 to 40 per cent

(b) 20 to 30 per cent

(c) 40 to 50 per cent

(d) 10 to 20 per cent

15.15 Taper roller bearing is used to take

(a) only radial load

(b) only axial load

(c) only torque

(d) both axial and radial loads

15.16 The catalogue life of bearing is

(a) minimum life that 90% of the bearings will reach or exceed

(b) maximum life for 90% of the bearings

(c) average life

(d) median life

15.17 The last two digits of the bearing designation give the bore diameter of rolling contact bearings when multiplies by,

(a) 10

(b) 5

(c) 100

(d) π

15.18 Rolling contact bearings are classified according to

(a) type of rolling element

(b) direction of load

(c) magnitude of load

(d) a and b

15.19 In an application, there is misalignment between the axes of journal and housing bore. Which type of rolling contact bearings you would suggest?

(a) spherical roller bearing

(b) self aligning ball bearing

(c) angular contact bearing

(d) a and b

Answers:

15.1 (b)	15.2 (a)	15.3 (c)	15.4 (a)	15.5 (c)
15.6 (b)	15.7 (d)	15.8 (d)	15.9 (d)	15.10 (c)
15.11 (c)	15.12 (a)	15.13 (a)	15.14 (b)	15.15 (d)
15.16 (a)	15.17 (b)	15.18 (d)	15.19 (d)	

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16 SLIDING CONTACT BEARINGS

16.1 In case of full journal bearing, the angle of contact of the bushing with the journal is

(a) 60^0 (b) 90^0 (c) 180^0 (d) 360^0

16.2 In case of partial bearing, the angle of contact of the bushing with the journal is

(a) 270^0 (b) more than 180^0 (c) less than 180^0 (d) 360^0

16.3 In case of partial bearing, the angle of contact of the bushing with the journal is usually

(a) 120^0 (b) 180^0 (c) 45^0 (d) 360^0

16.4 A zero film bearing is a bearing

(a) where the surfaces of journal and the bearing are separated by a thick film of lubricant

(b) where the surfaces of journal and the bearing are partially separated by a film of lubricant and there is partial metal to metal contact

(c) where the surfaces of journal and the bearing are separated by a film created by elastic deflection of parts

(d) where there is no lubricant

16.5 A thin film bearing is a bearing

(a) where the surfaces of journal and the bearing are completely separated by a film of lubricant

(b) where the surfaces of journal and the bearing are partially separated by a film of lubricant and there is partial metal to metal contact

(c) where the surfaces of journal and the bearing are separated by a film created by elastic deflection of parts

(d) where there is no lubricant

16.6 A thick film bearing is a bearing

(a) where the surfaces of journal and the bearing are completely separated by a film of lubricant

(b) where the surfaces of journal and the bearing are partially separated by a film of lubricant and there is partial metal to metal contact

(c) where the surfaces of journal and the bearing are separated by a film created by elastic deflection of parts

(d) where there is no lubricant

16.7 The length to diameter ratio for a short bearing is

(a) more than 1 (b) less than 1 (c) ∞ (d) 1

16.8 The length to diameter ratio for a long bearing is

(a) more than 1 (b) less than 1 (c) ∞ (d) 1

16.9 The length to diameter ratio for a square bearing is

(a) more than 1 (b) less than 1 (c) ∞ (d) 1

16.10 For hydrodynamic lubrication

(a) there should be relative motion between the surfaces of the journal and the bearing and wedge shaped clearance space

(b) there should be external source like pump to supply lubricant under pressure

(c) there should be elastic deformation of the parts in contact

(d) there should be metal to metal contact

16.11 For hydrostatic lubrication

(a) there should be relative motion between the surfaces of the journal and the bearing and wedge shaped clearance space

(b) there should be external source like pump to supply lubricant under pressure

(c) there should be elastic deformation of the parts in contact

(d) there should be metal to metal contact

16.12 For elastohydrodynamic lubrication

(a) there should be relative motion between the surfaces of the journal and the bearing and wedge shaped clearance space

(b) there should be external source like pump to supply lubricant under pressure

(c) there should be elastic deformation of the parts in contact

(d) there should be metal to metal contact

16.13 Boundary lubricated bearing is

(a) thick film bearing (b) thin film bearing

(c) hydrodynamic bearing (d) hydrostatic bearing

16.14 In hydrodynamic bearing, when the shaft begins to rotate in clockwise direction,

(a) the journal climbs to the right side of the bearing without metal to metal contact

(b) the journal climbs to the left side of the bearing without metal to metal contact

(c) the journal sinks down to the bottom of the bearing

(d) the journal is at the center of the bearing

16.15 In hydrodynamic bearing,

(a) the axis of journal is eccentric with respect to axis of bearing

(b) the axis of journal is concentric with respect to axis of bearing

(c) the axis can be either eccentric or concentric depending upon speed

(d) none of the above

16.16 In hydrostatic bearing,

(a) the axis of journal is eccentric with respect to axis of bearing

(b) the axis of journal is concentric with respect to axis of bearing

(c) the axis can be either eccentric or concentric depending upon speed

(d) none of the above

16.17 In most of internal combustion engines, crankshaft bearings is

(a) hydrodynamic journal bearing (b) hydrostatic journal bearing

(c) ball bearings (d) roller bearings

16.18 The property of the bearing material to yield and adopt its shape to that of journal is called

(a) embeddability (b) conformability

(c) viscosity (d) endurance limit stress

16.19 The property of the bearing material to allow the dust and abrasive particles to get absorbed on the surface of the lining is called

(a) embeddability (b) conformability

(c) viscosity (d) endurance limit stress

16.20 The popular bearing material is babbitt because it has

(a) excellent conformability (b) excellent embeddability

(c) ability to be used in form of thin strip (d) all three charcteristics

16.21 Compared with babbitt, bronze

(a) is cheaper (b) has high strength

(c) can withstand high pressure (d) has all three charcteristics

16.22 The unit of viscosity is

(a) N/mm^2 (b) N-mm (c) $N\text{-}s/mm^2$ (d) N-mm/s

16.23 Petroff's equation is used to find out

(a) load carrying capacity of the bearing

(b) frictional losses in the bearing

(c) unit bearing pressure on the bearing

(d) pressure distribution around the periphery of the journal

16.24 In Petroff's equation, it assumed that the lubricant film is

(a) converging (b) diverging

(c) uniform (d) converging diverging

16.25 If μ = absolute viscosity, N= speed of the journal and p = unit bearing pressure, then the bearing characteristic number is given by

(a) $\frac{\mu N}{p}$ (b) $\frac{\mu p}{N}$ (c) $\frac{p N}{\mu}$ (d) $\frac{\mu}{p N}$

16.26 When the bearing is subjected to fluctuating or impact load the bearing characteristic number should be …….times its minimum value

(a) 5 times (b) 10 times (c) 15 times (d) 20 times

16.27 Sommerfeld number is

(a) similar to bearing characteristic number

(b) similar to Reynold's number

(c) dimensionless parameter that contains all the design parameters

(d) used to find out dynamic load carrying capacity of the hydrodynamic bearing

16.28 As compared with oil with VI=80, a lubricating oil with VI=90

(a) has more viscosity

(b) has less viscosity

(c) has more rate of change of viscosity with respect to temperature

(d) has less rate of change of viscosity with respect to temperature

Answers:

16.1 (d)	16.2 (c)	16.3 (a)	16.4 (d)	16.5 (b)
16.6 (a)	16.7 (b)	16.8 (a)	16.9 (d)	16.10 (a)
16.11 (b)	16.12 (c)	16.13 (b)	16.14 (a)	16.15 (a)
16.16 (b)	16.17 (a)	16.18 (b)	16.19 (a)	16.20 (d)
16.21 (d)	16.22 (c)	16.23 (b)	16.24 (c)	16.25 (a)
16.26 (c)	16.27 (c)	16.28 (d)		

17 SPUR GEARS

17.1 Which of the following type of drives transmit power by friction?

(a) spur gear drive (b) chain drive

(c) worm gear drive (d) belt drive

17.2 When the axes of two shafts are parallel, use

(a) crossed helical gears (b) bevel

(c) worm gears (d) spur or helical gears

17.3 When the axes of two shafts are perpendicular and intersecting, use

(a) spur gears (b) bevel gears

(c) worm gears (d) helical gears

17.4 When the axes of two shafts are perpendicular and non-intersecting, use

(a) spur gears (b) bevel gears

(c) worm gears (d) helical gears

17.5 When the axes of two shafts are non-parallel and non-intersecting, use

(a) helical gears (b) crossed helical gears

(c) straight bevel gears (d) spiral bevel gears

17.6 When the velocity ratio is high and space is limited, use

(a) spur gears (b) bevel gears

(c) worm gears (d) helical gears

17.7 Which of the following type of gears are free from axial thrust?

(a) herringbone gears (b) Bevel gears

(c) worm gears (d) Helical gears

17.8 Which of the following type of gears are used for noiseless operation?

(a) spur gears (b) bevel gears

(c) worm gears (d) helical gears

17.9 Which of the following type of gears provide maximum velocity ratio?

(a) spur gears (b) bevel gears

(c) worm gears (d) helical gears

17.10 Cycloidal tooth gears are used in

(a) automobile gearbox

(b) machine tool gearbox

(c) spring driven watches and clocks

(c) materials handling equipment

17.11 In case of cycloidal tooth gears, which of the following statement is correct?

(a) simple to manufacture

(b) pressure angle remains constant

(c) no interference occurs between meshing teeth

(d) none of the above

17.12 The interference in cycloidal tooth gears

(a) depends upon pressure angle

(b) depends upon number of teeth

(c) is completely absent

(c) is maximum

17.13 Involute profile is widely used for gear tooth because

(a) pressure angle remains constant

(b) face and flank form a continuous curve

(c) involute rack has straight sided teeth

(d) all the above factors

17.14 An automobile gearbox has

(a) simple gear train (b) compound gear train

(c) epicyclic gear train (d) none of the above

17.15 The main function of an automobile gearbox is

(a) to reduce speed (b) to increase speed

(c) to provide variable speeds (d) to increase power

17.16 Maximum efficiency of a pair of spur gears is

(a) 99 % (b) 80 % (c) 50 % (d) 92 %

17.17 Maximum gear ratio for a pair of spur gears is

(a) 10 (b) 3 (c) 100 (d) 6

17.18 For transmitting power, a simple gear train consisting of 11 gears is used. Then the driver and driven shafts will be rotating in

(a) same direction (b) opposite direction

(c) same or opposite direction (d) none of the above

17.19 For transmitting power, a simple gear train consisting of 12 gears is used. Then the driver and driven shafts will be rotating in

(a) same direction (b) opposite direction

(c) same or opposite direction (d) none of the above

17.20 In metric system, the size of the gear tooth is specified by,

(a) circular pitch (b) diametral pitch

(c) module (d) pitch circle diameter

17.21 All dimensions for a standard gear system can be defined in terms of,

(a) module and number of teeth

(b) pressure angle and number of teeth

(c) module

(d) circular and diametral pitch

17.22 The portion of the gear tooth between the pitch circle and dedendum circle is called

(a) top land (b) face

(c) flank (d) bottom land

17.23 The portion of the gear tooth between the pitch circle and outer circle is called

(a) top land (b) face

(c) flank (d) bottom land

17.24 The thickness of gear tooth is measured

(a) along the pitch circle (b) along the base circle

(c) along the addendum circle (d) along the root circle

17.25 The tooth thickness of an involute gear in terms of module 'm' is

(a) 1.157m (b) 1.5m (c) 2m (d) 1.5708m

17.26 The working depth for a pair of gears is given by

(a) addendum plus dedendum (b) twice the addendum

(c) twice the dedendum (d) dedendum minus addendum

17.27 The whole depth of for a pair of gears is given by

(a) addendum plus dedendum (b) twice the addendum

(c) twice the dedendum (d) dedendum minus addendum

17.28 The outside diameter of an involute gear is equal to pitch circle diameter plus

(a) addendum plus dedendum (b) twice the addendum

(c) twice the dedendum (d) dedendum minus addendum

17.29 The curves that satisfy fundamental law of gearing are,

(a) cycloid (b) spiral

(c) involute (d) a and c

17.30 The curve traced by a point on a line as it rolls on a circle is called

(a) cycloid (b) epicycloid

(c) involute (d) hypocycloid

17.31 In which type of gears, slight variation in center distance does not affect velocity ratio?

(a) cycloid (b) epicycloid

(c) involute (d) hypocycloid

17.32 The pressure angle in involute teeth gears

(a) remains constant (b) often changes

(c) rarely changes (d) unpredictable

17.33 Which of the following types of gear is free from interference?

(a) cycloidal gears (b) spiral gears

(c) involute gears (d) stub gears

17.34 A rack is a gear with,

(a) infinite number of teeth (b) infinite module

(c) infinite circular pitch (d) none of the above

17.35 The clearance between meshing teeth is,

(a) radial distance between dedendum circle and addendum circle

(b) dedendum minus addendum

(c) radial distance between pitch circle and base circle

(d) radial distance between dedendum circle and pitch circle

17.36 In involute gear teeth, the base circle must be

(a) under the root circle (b) under the pitch circle

(c) above the pitch circle (d) at the root circle

17.37 The angle through which a gear turns from the beginning of contact of a pair of teeth until the contact arrives at the pitch point is known as

(a) angle of contact (b) angle of recess

(c) angle of approach (d) angle of action

17.38 Spur gears used for machine tool gearboxes must have the contact ratio

(a) equal to 1 (b) less than 1

(c) more than 1.4 (d) equal to 2

17.39 The pitch circle diameter and number of teeth in a spur gear are d' and z respectively. The module m is defined as

(a) $\left(\frac{\pi\, d'}{z}\right)$ (b) $\left(\frac{z}{d'}\right)$ (c) $\left(\frac{d'}{z}\right)$ (d) (d'z)

17.40 The product of diametral pitch P and circular pitch p is equal to

(a) $\frac{1}{\pi}$ (b) π (c) $\frac{\pi}{2}$ (d) 1

17.41 Backlash is

(a) sum of the clearances of pinion and gear teeth

(b) the amount by which the width of a tooth space exceeds the thickness of meshing tooth on pitch circle

(c) difference between the pitch circles of meshing gears

(d) difference between the dedendum and addendum of gear tooth

17.42 Compared with full depth teeth, stub tooth has

(a) short addendum and short dedendum

(b) long addendum and long dedendum

(c) same addendum and same dedendum

(d) short addendum and same dedendum

17.43 Stub tooth has

(a) longer than standard whole depth

(b) shorter than standard whole depth

(c) standard whole depth

(d) non-standard whole depth

17.44 Stub tooth are provided for

(a) high velocity ratios

(b) high power transmitting capacity

(c) low noise

(d) reducing number of teeth for compact construction

17.45 Reducing pressure angle on gears results in

(a) weaker teeth (b) stronger teeth

(c) high velocity ratio (d) high efficiency

17.46 In case of spur gears, increasing pressure angle results in

(a) weaker teeth (b) bigger size of gear

(c) higher pitch line velocity (d) wider base and stronger teeth

17.47 The gear tooth system that will transmit very high load is

(a) 20^0 full depth involute (b) 20^0 full stub involute

(c) 14.5^0 full depth involute (d) 14.5^0 stub involute

17.48 For 20^0 full depth involute tooth system, the minimum number of teeth on pinion for avoiding interference is

(a) 20 (b) 14 (c) 17 (d) 25

17.49 The minimum number of teeth on standard gear with pressure angle (α) is given by,

(a) $\frac{2}{\sin^2\alpha}$ (b) $2\sin^2\alpha$ (c) $\frac{\sin^2\alpha}{2}$ (d) $2\sin\alpha$

17.50 Addendum of a cycloidal gear tooth is

(a) epicycloid (b) hypocycloid

(c) cycloid (d) involutes

17.51 Which of the following gear ratios does not require hunting tooth?

(a) 52:26 (b) 48:24 (c) 50:25 (d) 51:25

17.52 In case of spur gears, Lewis form factor depends upon

(a) module (b) number of teeth

(c) pressure angle (d) b and c

17.53 Beam strength of gear tooth is

(a) maximum tangential force that the tooth can transmit without bending failure

(b) maximum bending stress that the tooth can transmit without failure

(c) maximum tangential force that the tooth can transmit without pitting failure

(d) maximum contact stress that the tooth can transmit without failure

17.54 Wear strength of gear tooth is

(a) maximum tangential force that the tooth can transmit without bending failure

(b) maximum bending stress that the tooth can transmit without failure

(c) maximum tangential force that the tooth can transmit without pitting failure

(d) maximum contact stress that the tooth can transmit without failure

17.55 Lewis equation of gear tooth is based on

(a) maximum crushing stress in gear tooth

(b) maximum bending stress in gear tooth

(c) maximum shear stress in gear tooth

(d) maximum contact stress in gear tooth

17.56 In Lewis equation, gear tooth is considered as

(a) simply supported beam (b) cantilever beam

(c) curved beam (d) none of the above

17.57 Buckingham's equation of gear tooth is based on

(a) maximum crushing stress in gear tooth

(b) maximum bending stress in gear tooth

(c) maximum shear stress in gear tooth

(d) maximum contact stress in gear tooth

17.58 Permissible bending stress for gear tooth is usually taken as

(a) one third of ultimate tensile strength

(b) one half of ultimate tensile strength

(c) one third of tensile yield strength

(d) one third of surface endurance strength

17.59 Dynamic force on gear tooth is induced due to

(a) inaccuracies of tooth profile and errors in tooth spacing

(b) misalignment in bearings

(c) elasticity of parts and inertia of rotating masses

(d) a , b and c

17.60 Dynamic force on gear tooth depends upon

(a) pitch line velocity

(b) sum of errors between two meshing teeth

(c) modulus of elasticity of pinion and gear materials

(d) a , b and c

17.61 The number of grades used to specify the quality of gears is

(a) 10 (b) 3 (c) 12 (d) 6

17.62 Static force on gear tooth is due to

(a) power transmitted by gears

(b) misalignment in bearings

(c) acceleration of gears

(d) bearing reactions

17.63 According to Lewis equation

(a) pinion is always weaker than gear

(b) pinion is weaker than gear if made of same material

(c) gear is weaker than pinion if made of same material

(d) none of the above

17.64 Beam strength of gear tooth should be

(a) less than effective load consisting of static and dynamic load

(b) more than effective load consisting of static and dynamic load

(c) more than wear strength of gear tooth

(d) more than load due to power transmission

17.65 Wear strength of gear tooth should be

(a) less than effective load consisting of static and dynamic load

(b) more than effective load consisting of static and dynamic load

(c) more than beam strength of gear tooth

(d) more than load due to power transmission

17.66 Surface endurance strength of gear tooth depends upon

(a) surface finish of gear tooth

(b) ultimate tensile strength of gear materials.

(c) surface hardness of gear tooth

(d) modulus of elasticity of gear materials

17.67 Non-metallic gears have

(a) low noise and low load carrying capacity

(b) low noise and high load carrying capacity.

(c) high noise and high load carrying capacity

(d) high noise and low load carrying capacity

Answers:

17.1 (d)	17.2 (d)	17.3 (b)	17.4 (c)	17.5 (b)
17.6 (c)	17.7 (a)	17.8 (d)	17.9 (c)	17.10 (c)
17.11 (c)	17.12 (c)	17.13 (d)	17.14 (b)	17.15 (c)
17.16 (a)	17.17 (a)	17.18 (a)	17.19 (b)	17.20 (c)
17.21 (a)	17.22 (c)	17.23 (b)	17.24 (a)	17.25 (d)
17.26 (b)	17.27 (a)	17.28 (b)	17.29 (d)	17.30 (c)
17.31 (c)	17.32 (a)	17.33 (a)	17.34 (a)	17.35 (b)
17.36 (b)	17.37 (c)	17.38 (c)	17.39 (c)	17.40 (b)
17.41 (b)	17.42 (a)	17.43 (b)	17.44 (d)	17.45 (a)
17.46 (d)	17.47 (b)	17.48 (c)	17.49 (a)	17.50 (a)
17.51 (d)	17.52 (d)	17.53 (a)	17.54 (c)	17.55 (b)
17.56 (b)	17.57 (d)	17.58 (a)	17.59 (d)	17.60 (d)
17.61 (c)	17.62 (a)	17.63 (b)	17.64 (b)	17.65 (b)
17.66 (c)	17.67 (a)			

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18 HELICAL GEARS

18.1 The initial contact in helical gears is

(a) point (b) line (c) surface (d) unpredictable

18.2 If z is the actual number of teeth on a helical gear and ψ is the helix angle for the teeth, then the formative number of teeth is given by,

(a) $z \cos^3 \Psi$ (b) $\frac{z}{\cos^3 \Psi}$

(c) $z \cos^2 \Psi$ (d) $\frac{z}{\cos^2 \Psi}$

18.3 In helical gears, the distance between two similar points on adjacent teeth, measured in a plane perpendicular to tooth element, is called

(a) normal circular pitch (b) transverse circular pitch

(c) axial pitch (d) diametral pitch

18.4 In a pair of helical gears, the pinion has right hand helical teeth. The hand of helix for mating gear should be,

(a) right hand (b) left hand

(c) right or left depending upon direction of rotation

(d) none of the above

18.5 The range of helix angle for a single helical gear is,

(a) 15^0 to 25^0 (b) 20^0 to 45^0

(c) 10^0 to 15^0 (d) 5^0 to 10^0

18.6 The range of helix angle for a double helical gear is,

(a) 15^0 to 25^0 (b) 20^0 to 45^0

(c) 10^0 to 15^0 (d) 5^0 to 10^0

18.7 Which of the following gears have zero axial thrust?

(a) herringbone gears (b) bevel gears

(c) worm gears (d) helical gears

18.8 Compared with spur gears, helical gears

(a) run more smoothly

(b) run with noise and vibrations

(c) consume less power

(d) run exactly alike

18.9 In a pair of crossed helical gears, both gears have right hand helical teeth with 45^0 helix angle. The angle between the shafts of two gears is

(a) 45^0 (b) 90^0 (c) 22.5^0 (d) 0^0

18.10 In a pair of crossed helical gears, both gears have opposite hand helical teeth with 45^0 helix angle. The angle between the shafts of two gears is

(a) 45^0 (b) 90^0 (c) 22.5^0 (d) 0^0

Answers:

18.1 (a)	18.2 (b)	18.3 (a)	18.4 (b)	18.5 (a)
18.6 (b)	18.7 (a)	18.8 (a)	18.9 (b)	18.10 (d)

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19 BEVEL GEARS

19.1 When two bevel gears having equal teeth and equal pitch angle are used to transmit power between shafts that are intersecting at right angle, they are called,

(a) spiral bevel gears (b) Miter gears

(c) straight bevel gears (d) hypoid gears

19.2 When two identical bevel gears are used to transmit power between shafts that are intersecting at right angle, they are called,

(a) spiral bevel gears (b) Miter gears

(c) straight bevel gears (d) hypoid gears

19.3 Miter gears are

(a) spur gears with same number of teeth

(b) helical gears with same number of teeth

(c) bevel gears with same number of teeth mounted on perpendicular shafts

(d) spiral bevel gears with zero spiral angle

19.4 When bevel gears are used to transmit power between shafts that are intersecting at an angle greater than 90^0, and if the pitch angle of one of the gears is 90^0, they are called,

(a) spiral bevel gears (b) crown gears

(c) straight bevel gears (d) hypoid gears

19.5 In case of skew bevel gears, the axes of shafts are

(a) parallel

(b) intersecting

(c) non-parallel and non-intersecting and the teeth are straight

(d) non-parallel and non-intersecting and the teeth are curved

19.6 In case of external bevel gears,

(a) the pitch angle is more than 90^0.

(b) the pitch angle is 90^0.

(c) the pitch angle is less than 90^0.

(d) pitch angle has no effect on type.

19.7 In case of internal bevel gears,

(a) the pitch angle is more than 90^0.

(b) the pitch angle is 90^0.

(c) the pitch angle is less than 90^0.

(d) pitch angle has no effect on type.

19.8 In case of hypoid gears, the axes of shafts are

(a) parallel

(b) intersecting

(c) non-parallel and non-intersecting and the teeth are straight

(d) non-parallel and non-intersecting and the teeth are curved

19.9 In case of zerol gears, the axes of shafts are

(a) intersecting and teeth are straight

(b) intersecting and teeth are curved

(c) non-parallel and non-intersecting and the teeth are straight

(d) non-parallel and non-intersecting and the teeth are curved

19.10 Two bevel gears with 25 and 50 teeth are in mesh with each other. The pitch angle of pinion is

(a) $\tan^{-1}(0.5)$ (b) $\tan^{-1}(2.0)$

(c) $\sin^{-1}(0.5)$ (d) $\sin^{-1}(2.0)$

19.11 Two bevel gears with 25 and 50 teeth are in mesh with each other. The pitch angle of gear is

(a) $\left[\frac{\pi}{2} - \tan^{-1}(0.5)\right]$ (b) $\sin^{-1}(0.5)$

(c) $\tan^{-1}(0.5)$ (d) $\frac{\pi}{2} - \sin^{-1}(0.5)$

19.12 The face angle of a bevel gear is equal to

(a) pitch angle + addendum angle

(b) pitch angle - addendum angle

(c) axial pitch

(d) diametral pitch

19.13 The root angle of a bevel gear is equal to

(a) pitch angle + addendum angle

(b) pitch angle - addendum angle

(c) pitch angle + dedendum angle

(d) pitch angle - dedendum angle

19.14 The relationship between actual number of teeth (z), formative number of teeth (z') and pitch angle (γ) is given by,

(a) $z' = \left(\frac{z}{\cos\gamma}\right)$ (b) $z' = z\cos\gamma$

(c) $z' = \left(\frac{z}{\cos^3\gamma}\right)$ (d) $z' = z\cos^3\gamma$

19.15 The bevel factor for a bevel gear with cone distance A_0 and face width b is given by,

(a) $\left[1 - \frac{b}{A_0}\right]$ (b) $\left[\frac{b}{A_0}\right]$

(c) $\left[\frac{1}{3}\right]$ (d) $\left[1 - \frac{b}{2A_0}\right]$

19.16 In a concrete mixer, the bevel gears for rotating the drum are generally made by

(a) casting (b) forging

(c) hobbing (d) shaping

Answers:

19.1 (b)	19.2 (b)	19.3 (c)	19.4 (b)	19.5 (c)
19.6 (c)	19.7 (a)	19.8 (d)	19.9 (b)	19.10 (a)
19.11 (a)	19.12 (a)	19.13 (d)	19.14 (a)	19.15 (a)
19.16 (a)				

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20 WORM GEARS

20.1 Worm gears are widely used when

(a) velocity ratio is high

(b) space is limited

(c) axes of shafts are non-intersecting

(d) all the three

20.2 The lead angle of worm is given by,

(a) $\tan^{-1}\left(\frac{l}{\pi d_1}\right)$ (b) $\tan^{-1}\left(\frac{z_1}{q}\right)$

(c) $\tan^{-1}(\pi\ m\ z_1)$ (d) a and b

20.3 The number of starts on worm for a velocity ratio of 40 is

(a) single (b) double (c) triple (d) quadruple

20.4 The axial component of resultant force on worm wheel is equal to

(a) tangential component on worm

(b) radial component on worm

(c) axial component on worm

(d) non of the above

20.5 A pair of worm gears is designated as (1/30/10/8). The center distance between the worm and worm wheel is

(a) 160 mm (b) 30 mm (c) 80 mm (d) 96 mm

20.6 A pair of worm gears is designated as (1/30/10/8). The velocity ratio is

(a) 160 (b) 30 (c) 80 (d) 96

20.7 The main advantage of worm gear drive is

(a) ease of manufacturing (b) high velocity ratio

(c) low power loss (d) low cost

20.8 To make the worm gear drive reversible, it is necessary to increase

(a) center distance (b) number of starts

(c) diametral quotient (d) velocity ratio

20.9 For proper meshing of the worm and worm wheel, the axial pitch of the worm compared with circular pitch of worm wheel should be

(a) more (b) equal (c) less (d) non of the above

20.10 A worm gear drive consists of double start worm meshing with a 50 teeth worm wheel. The velocity ratio is

(a) 25 (b) 100 (c) 50 (d) 75

Answers:

20.1 (d)	20.2 (d)	20.3 (a)	20.4 (a)	20.5 (a)
20.6 (b)	20.7 (b)	20.8 (b)	20.9 (b)	20.10 (a)

21 FLYWHEEL

21.1 A flywheel is used,

(a) to store and release energy when required during the work cycle

(b) to reduce the power capacity of electric motor

(c) to reduce the amplitude of speed fluctuations

(d) any one of above functions

21.2 A flywheel is used,

(a) to limit the fluctuations of speed during each cycle

(b) to control the mean speed of the engine

(c) to maintain a constant speed

(d) to come into action when the speed varies

21.3 A flywheel is made of,

(a) cast iron

(b) high strength steels

(c) graphite-fiber reinforced polymer(GFRP)

(d) any one of the above materials

21.4 The maximum fluctuation of speed of flywheel is,

(a) difference between maximum and minimum speeds during the cycle

(b) difference between maximum and mean speeds during the cycle

(c) difference between mean and minimum speeds during the cycle

(d) mean of maximum and minimum speeds during the cycle

21.5 The coefficient of fluctuation of speed of flywheel is,

(a) ratio of maximum and minimum speeds during the cycle

(b) ratio of maximum fluctuation of speed to the mean speed

(c) ratio of mean speed to maximum fluctuation of speed during the cycle

(d) sum of maximum fluctuation of speed and the mean speed

21.6 The coefficient of fluctuation of speed of flywheel is given by,

(a) $\frac{\omega_{max} - \omega_{min}}{\omega}$ (b) $\frac{2(\omega_{max} - \omega_{min})}{(\omega_{max} + \omega_{min})}$

(c) $\frac{2(n_{max} - n_{min})}{(n_{max} + n_{min})}$ (d) any one of the above expressions

where ω_{max}, ω_{min}, and ω are maximum, minimum and mean speed of flywheel respectively.

21.7 The maximum fluctuation of energy of flywheel is,

(a) difference between maximum and minimum kinetic energy during the cycle

(b) difference between maximum and mean kinetic energy during the cycle

(c) difference between mean and minimum kinetic energy during the cycle

(d) mean of maximum and minimum kinetic energy during the cycle

21.8 The coefficient of fluctuation of energy of flywheel is,

(a) ratio of maximum fluctuation of energy to work done per cycle

(b) ratio of to work done per cycle to maximum fluctuation of energy

(c) difference between maximum and minimum kinetic energy during the cycle

(d) ratio of maximum and minimum kinetic energy during the cycle

21.9 The rim of the flywheel is subjected to,

(a) direct tensile stress and bending stress

(b) torsional shear stress and bending stress

(c) direct shear stress and bending stress

(d) compressive stress and bending stress

21.10 The spokes of the flywheel are subjected to,

(a) direct shear stress (b) torsional shear stress

(c) tensile stress (d) compressive stress

21.11 The tensile stress in the rim of the flywheel is given by,

(a) $\rho v^2 / 2$ (b) $\pi \rho v^2$

(c) ρv^2 (d) $2\rho v^2$

where ρ is mass density and v is linear velocity at the mean radius of the rim.

21.12 The cross-section of the flywheel arm is,

(a) I section (b) rectangular

(c) elliptical (d) circular

21.13 For finding out the bending moment for the arm (spoke) of flywheel, the arm is assumed as,

(a) a cantilever beam fixed at the rim and subjected to tangential force at the hub

(b) a simply supported beam fixed at hub and rim and carrying uniformly distributed load

(c) a cantilever beam fixed at the hub and subjected to tangential force at the rim

(d) a fixed beam fixed at hub and rim and carrying uniformly distributed load

21.14 The hub diameter of the flywheel is taken as,

(a) 2.5(shaft diameter) (b) 1.5(shaft diameter)

(c) 4(shaft diameter) (d) 2(shaft diameter)

Answers:

21.1 (d)	21.2 (a)	21.3 (d)	21.4 (a)	21.5 (b)
21.6 (d)	21.7 (a)	21.8 (a)	21.9 (a)	21.10 (c)
21.11 (c)	21.12 (c)	21.13 (c)	21.14 (d)	

22 CYLINDERS AND PRESSURE VESSELS

22.1 A cylinder is considered as thin cylinder when the ratio of inner diameter to the wall thickness is,

(a) more than 15 (b) less than 15

(c) equal to 15 (d) none of these criteria

22.2 A cylinder is considered as thick cylinder when the ratio of inner diameter to the wall thickness is,

(a) more than 15 (b) less than 15

(c) equal to 15 (d) none of these criteria

22.3 In thin cylinders, the longitudinal stress is,

(a) 2(circumferential stress) (b) 1/2(circumferential stress)

(c) 1/4(circumferential stress) (d) 4(circumferential stress)

22.4 The thickness of thin cylinder is determined on the basis of,

(a) radial stress (b) longitudinal stress

(c) circumferential stress (d) principal shear stress

22.5 The thickness of thin cylindrical shell is given by,

(a) $\frac{P_i\ D_i}{2\ \sigma_t}$ (b) $\frac{P_i\ D_i}{4\ \sigma_t}$ (c) $\frac{2P_i\ D_i}{\sigma_t}$ (d) $\frac{4P_i\ D_i}{\sigma_t}$

where D_i, $P_{i,}$ and σ_t are inner diameter, internal pressure and permissible tensile stress respectively.

22.6 The thickness of thin spherical shell is given by,

(a) $\frac{P_i\ D_i}{2\ \sigma_t}$ (b) $\frac{P_i\ D_i}{4\ \sigma_t}$ (c) $\frac{2P_i\ D_i}{\sigma_t}$ (d) $\frac{4P_i\ D_i}{\sigma_t}$

where D_i, $P_{i,}$ and σ_t are inner diameter, internal pressure and permissible tensile stress respectively.

22.7 In thick cylinders, the tangential stress across the thickness of cylinder

(a) remains uniform throughout

(b) varies from internal pressure at the inner surface to zero at the outer surface

(c) varies from maximum value at the inner surface to minimum value at the outer surface

(d) varies from maximum value at the outer surface to minimum value at the inner surface

22.8 In thick cylinders, the radial stress across the thickness of cylinder

(a) remains uniform throughout

(b) varies from internal pressure at the inner surface to zero at the outer surface

(c) varies from maximum value at the inner surface to minimum value at the outer surface

(d) varies from maximum value at the outer surface to minimum value at the inner surface

22.9 In thick cylinders, the axial stress across the thickness of cylinder

(a) remains uniform throughout

(b) varies from internal pressure at the inner surface to zero at the outer surface

(c) varies from maximum value at the inner surface to minimum value at the outer surface

(d) varies from maximum value at the outer surface to minimum value at the inner surface

22.10 In thin cylinders, the tangential stress across the thickness of cylinder

(a) remains uniform throughout

(b) varies from internal pressure at the inner surface to zero at the outer surface

(c) varies from maximum value at the inner surface to minimum value at the outer surface

(d) varies from maximum value at the outer surface to minimum value at the inner surface

22.11 According to Lame's equation, the thickness of cylindrical shell is given by,

(a) $\frac{P_i \ D_i}{2 \ \sigma_t}$

(b) $t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma_t + (1-2\mu)P_i}{\sigma_t - (1+\mu)P_i}} - 1\right]$

(c) $t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma_t + (1-\mu)P_i}{\sigma_t - (1+\mu)P_i}} - 1\right]$

(d) $t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma_t + P_i}{\sigma_t - P_i}} - 1\right]$

where D_i, $P_{i,}$ σ_t and μ are inner diameter, internal pressure, permissible tensile stress and Poisson's ratio respectively.

22.12 According to Clavarino's equation, the thickness of cylindrical shell is given by,

(a) $\frac{P_i \ D_i}{2 \ \sigma_t}$

(b) $t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma_t + (1-2\mu)P_i}{\sigma_t - (1+\mu)P_i}} - 1\right]$

(c) $t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma_t + (1-\mu)P_i}{\sigma_t - (1+\mu)P_i}} - 1\right]$

(d) $t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma_t + P_i}{\sigma_t - P_i}} - 1\right]$

where D_i, $P_{i,}$ σ_t and μ are inner diameter, internal pressure, permissible tensile stress and Poisson's ratio respectively.

22.13 According to Birnie's equation, the thickness of cylindrical shell is given by,

(a) $\frac{P_i \ D_i}{2\,\sigma_t}$ (b) $t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma_t + (1-2\mu)P_i}{\sigma_t - (1+\mu)P_i}} - 1\right]$

(c) $t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma_t + (1-\mu)P_i}{\sigma_t - (1+\mu)P_i}} - 1\right]$ (d) $t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma_t + P_i}{\sigma_t - P_i}} - 1\right]$

where D_i, $P_{i,}$ σ_t and μ are inner diameter, internal pressure, permissible tensile stress and Poisson's ratio respectively.

22.14 The thickness of thick cylindrical shell with closed ends and made of brittle material is determined by,

(a) Lame's equation (b) Clavarino's equation

(c) Birnie's equation (d) Barlow's equation

22.15 The thickness of thick cylindrical shell with closed ends and made of ductile material is determined by,

(a) Lame's equation (b) Clavarino's equation

(c) Birnie's equation (d) Barlow's equation

22.16 The thickness of thick cylindrical shell with open ends and made of ductile material is determined by,

(a) Lame's equation (b) Clavarino's equation

(c) Birnie's equation (d) Barlow's equation

22.17 The thickness of high-pressure oil and gas pipes is determined by,

(a) Lame's equation (b) Clavarino's equation

(c) Birnie's equation (d) Barlow's equation

22.18 Autofrettage is,

(a) a surface coating process of cylinders for corrosion resistance

(b) a heat treatment process for cylinders to relieve residual stresses

(c) a process of pre-stressing the cylinder to develop residual compressive stress at the inner surface

(d) a surface hardening process of cylinder to improve wear resistance

22.19 Autofrettage is achieved by,

(a) compound cylinder

(b) overloading the cylinder before putting it in service

(c) winding a wire under tension around the cylinder

(d) any one of the above methods

22.20 A gasket is made of,

(a) asbestos or cork (b) lead, copper or aluminum

(c) vulcanized rubber (d) any one of the above

22.21 Welded pressure vessels made of steel plates should be designed according to 'Code for unfired vessel IS-2825' when,

(a) internal pressure is from 1 kgf/cm^2 to 200 kgf/cm^2

(b) internal diameter is less than 150 mm

(c) water container is to be designed with capacities less than 500 litres

(d) steam boilers and nuclear pressure vessels are to be designed

22.22 Class 1 pressure vessels are to be designed according to 'Code for unfired vessel IS-2825' when,

(a) hydrocyanic acid, carbonyl chloride or mustard gas are stored

(b) operating temperature is more than -20^0C

(c) liquefied petroleum gas is stored

(d) thickness of shell is less than 38 mm

22.23 Class 3 pressure vessels are to be designed according to ‘Code for unfired vessel IS-2825’ when,

(a) operating pressure is less than 17.5 kgf/cm^2

(b) operating temperature is more than 0^0C and less than 250^0C

(c) thickness of shell is less than 16 mm

(d) any one of the above

22.24 While designing pressure vessels according to ‘Code for unfired vessel IS-2825’, the design pressure is taken as

(a) 1.05(maximum operating pressure)

(b) 1.5(maximum operating pressure)

(c) 2(maximum operating pressure)

(d) 1.3(maximum operating pressure)

22.25 Weld joint efficiency is maximum when the pressure vessel is welded by

(a) single-welded butt joint with backing strip

(b) single-welded butt joint without backing strip

(c) double-welded butt joint with full penetration

(d) none of the above

22.26 Type of domed heads for the pressure vessel is

(a) hemispherical head (b) semi-ellipsoidal head

(c) torispherical head (d) any one of the above

22.27 The end-closure for tall vertical pressure vessel is

(a) hemispherical head (b) conical head

(c) torispherical head (d) flat head

22.28 The end-closure for tankers of milk, petrol or diesel is

(a) hemispherical head (b) conical head

(c) torispherical head (d) flat head

Answers:

22.1 (a)	22.2 (b)	22.3 (b)	22.4 (c)	22.5 (a)
22.6 (b)	22.7 (c)	22.8 (b)	22.9 (a)	22.10 (a)
22.11 (d)	22.12 (b)	22.13 (c)	22.14 (a)	22.15 (b)
22.16 (c)	22.17 (d)	22.18 (c)	22.19 (d)	22.20 (d)
22.21 (a)	22.22 (a)	22.23 (d)	22.24 (a)	22.25 (c)
22.26 (d)	22.27 (a)	22.28 (c)		

23 MISCELLANEOUS MACHINE ELEMENTS

23.1 An oil seal is used

(a) to prevent leakage of fluid between two part in relative motion

(b) to prevent leakage of fluid between two stationary part

(c) to prevent leakage of fluid between cylinder and cylinder cover

(d) for none of the above

23.2 The part of oil seal, which prevents leakage of fluid, is

(a) casing (b) garter spring

(c) lip (d) none of the above

23.3 Rubber compound that is used for sealing lip is

(a) Nitrile compound (b) Silicon compound

(c) Fluoro elastomer compound (d) any one of the above

23.4 In 6x19 designation of wire rope, the numbers 6 and 19 respectively stand for

(a) number of strands and diameter of wire rope

(b) diameter and length of wire rope

(c) number of strands and number of wires in each strand

(d) thickness and width of cross-section of wire rope

23.5 Which of the following wire ropes will be flexible?

(a) 6x19 (b) 6x7

(c) 6x37 (d) any one of the above

23.6 Which of the following wire ropes will be more wear resistant?

(a) 6x19 (b) 6x7

(c) 6x37 (d) any one of the above

23.7 When the wire rope passes around the sheave or drum, the stress is maximum

(a) in outer wires (b) in inner wires

(c) in central wires (d) none of the above

23.8 The wires in wire rope are subjected to

(a) tensile stress (b) tensile and bending stresses

(c) shear stress (d) crushing stress

23.9 Popular lay for wire rope is

(a) regular lay (b) Lang's lay

(c) any one of the above (d) none of the above

23.10 Wire rope makes contact at

(a) bottom of groove of the pulley

(b) sides of groove of the pulley

(c) bottom and sides of groove of the pulley

(d) any one of the above

23.11 The rope drum for wire rope should have

(a) larger diameter and flat surface

(b) larger diameter and grooves

(c) smaller diameter and grooves

(d) smaller diameter and flat surface

23.12 Wire rope are used in applications with

(a) low speed and low tension (b) low speed and high tension

(c) high speed and high tension (d) high speed and low tension

23.13 A machine component is designed as a column when it is subjected to,

(a) axial tensile force (b) bending moment

(c) axial compressive force (d) torsional moment

23.14 Slenderness ratio is the ratio of,

(a) length of column to least moment of inertia of the cross-section about its axis

(b) length of column to area of the cross-section

(c) length of column to least radius of gyration of the cross-section about its axis

(d) none of these

23.15 Simple compression formula is valid for slenderness ratio

(a) more than 30 (b) less than 30

(c) equal to 30 (d) more than 40

23.16 A column made of ductile material is considered as long column when the slenderness ratio is

(a) more than 80 (b) less than 80

(c) less than 30 (d) more than 100

23.17 A column made of brittle material is considered as long column when the slenderness ratio is

(a) more than 80 (b) less than 80

(c) less than 30 (d) more than 100

23.18 Compression member tends to buckle in the direction of

(a) axis of load

(b) any direction in a plane perpendicular to axis of load

(c) least radius of gyration in a plane perpendicular to axis of load

(d) none of the above

23.19 A column, which can sustain highest buckling load, has

(a) both ends hinged

(b) both ends fixed

(c) one end fixed and other end hinged

(d) one end fixed and other end free

23.20 Which equation is used for long columns?

(a) Euler’s equation

(b) Johnson’s equation

(c) simple compressive stress equation

(d) any one of the above

23.21 Which equation is used for short columns?

(a) Euler’s equation

(b) Johnson’s equation

(c) simple compressive stress equation

(d) any one of the above

23.22 The dividing line between Euler's equation and Johnson's equation is

(a) $\left(\frac{P_{cr}}{A}\right) = \frac{S_{yt}}{2}$ (b) $\left(\frac{l}{k}\right) = 30$

(c) $\left(\frac{l}{k}\right) = 80$ (d) $\left(\frac{P_{cr}}{A}\right) = 100$

where P_{cr} , l, k, A and S_{yt} are critical buckling load, length of column, least radius of gyration, cross-sectional area and yield strength respectively.

23.23 The condition, which gives same value of buckling load by Euler's equation and Johnson's equation, is

(a) $\left(\frac{P_{cr}}{A}\right) = \frac{S_{yt}}{2}$ (b) $\left(\frac{l}{k}\right) = 30$

(c) $\left(\frac{l}{k}\right) = 80$ (d) $\left(\frac{P_{cr}}{A}\right) = 100$

where P_{cr} , l, k, A and S_{yt} are critical buckling load, length of column, least radius of gyration, cross-sectional area and yield strength respectively.

23.24 Euler's equation is valid if

(a) $\left(\frac{P_{cr}}{A}\right) < \frac{S_{yt}}{2}$ (b) $\left(\frac{P_{cr}}{A}\right) > \frac{S_{yt}}{2}$

(c) $\left(\frac{P_{cr}}{A}\right) = \frac{S_{yt}}{2}$ (d) $\left(\frac{P_{cr}}{A}\right) = 100$

where P_{cr} , l, k, A and S_{yt} are critical buckling load, length of column, least radius of gyration, cross-sectional area and yield strength respectively.

23.25 Johnson's equation is valid if

(a) $\left(\frac{P_{cr}}{A}\right) < \frac{S_{yt}}{2}$ (b) $\left(\frac{P_{cr}}{A}\right) > \frac{S_{yt}}{2}$

(c) $\left(\frac{P_{cr}}{A}\right) = \frac{S_{yt}}{2}$ (d) $\left(\frac{P_{cr}}{A}\right) = 100$

where P_{cr} , l, k, A and S_{yt} are critical buckling load, length of column, least radius of gyration, cross-sectional area and yield strength respectively.

23.26 The buckling load depends upon

(a) slenderness ratio (b) cross-sectional area

(c) modulus of elasticity (d) all of the above

23.27 In design of piston rod as column, the end conditions are assumed as

(a) ends are hinged in both planes

(b) ends are hinged in one plane and fixed in perpendicular plane

(c) ends are hinged in both planes

(d) ends are free in one plane and hinged in perpendicular plane

23.28 Which of the following statement is incorrect

(a) solid circular column is less economical than tubular column

(b) radius of gyration of tubular section is same in any direction

(c) tubular column is free to buckle in any direction

(d) none of the above

23.29 Compared with solid section, hollow section column has

(a) high buckling strength (b) low buckling strength

(c) same buckling strength (d) none of the above

Answers:

23.1 (a)	23.2 (c)	23.3 (d)	23.4 (c)	23.5 (c)
23.6 (b)	23.7 (a)	23.8 (b)	23.9 (a)	23.10 (a)
23.11 (b)	23.12 (b)	23.13 (c)	23.14 (c)	23.15 (b)
23.16 (d)	23.17 (a)	23.18 (c)	23.19 (b)	23.20 (a)
23.21 (b)	23.22 (a)	23.23 (a)	23.24 (a)	23.25 (b)
23.26 (d)	23.27 (b)	23.28 (d)	23.29 (a)	

24 STATISTICAL CONSIDERATION IN DESIGN

24.1 Histogram is

(a) a set of rectangles

(b) a line graph of class frequency plotted against class marks

(c) a bell shaped smooth curve

(d) none of the above

24.2 Frequency polygon is

(a) a set of rectangles

(b) a line graph of class frequency plotted against class marks

(c) a bell shaped smooth curve

(d) none of the above

24.3 Central tendency of population is

(a) spread of data or extend to which the observations are scattered

(b) mid point of distribution where most of the data cluster

(c) concentration of data at either low or high end

(d) distribution with sharp peak

24.4 Dispersion of population is

(a) spread of data or extend to which the observations are scattered

(b) mid point of distribution where most of the data cluster

(c) concentration of data at either low or high end

(d) distribution with sharp peak

24.5 Skewness of population is

(a) spread of data or extend to which the observations are scattered

(b) mid point of distribution where most of the data cluster

(c) concentration of data at either low or high end

(d) measure of sharp peak

24.6 Kurtosis of population is

(a) spread of data or extend to which the observations are scattered

(b) mid point of distribution where most of the data cluster

(c) concentration of data at either low or high end

(d) measure of sharp peak

24.7 Central tendency of population is measured in units of

(a) standard deviation (b) arithmetic mean

(c) standard variable (d) square of standard deviation

24.8 Dispersion of population is measured in units of

(a) standard deviation (b) arithmetic mean

(c) geometric mean (d) square of standard deviation

24.9 Standard variable is

(a) square of standard deviation

(b) arithmetic mean

(c) root mean square deviation from the mean

(d) deviation from mean in units of standard deviation

24.10 Standard deviation is

(a) square of standard variable

(b) arithmetic mean

(c) root mean square deviation from the mean

(d) deviation from mean in units of standard deviation

24.11 The area below normal curve from ($Z = -\infty$) to ($Z = +\infty$) is

(a) 1 (b) 0.6827

(c) 0.9545 (d) 0.9973

where Z is standard variable

24.12 The area below normal curve from ($Z = -1$) to ($Z = +1$) is

(a) 1 (b) 0.6827

(c) 0.9545 (d) 0.9973

where Z is standard variable

24.13 The area below normal curve from ($Z = -2$) to ($Z = +2$) is

(a) 1 (b) 0.6827

(c) 0.9545 (d) 0.9973

where Z is standard variable

24.14 The area below normal curve from ($Z = -3$) to ($Z = +3$) is

(a) 1 (b) 0.6827

(c) 0.9545 (d) 0.9973

where Z is standard variable

24.15 When two populations with means μ_X and μ_Y are added, the mean of resultant population is given by,

(a) $(\mu_X + \mu_Y)$ (b) $(\mu_X - \mu_Y)$

(c) $(\mu_X \mu_Y)$ (d) (μ_X / μ_Y)

24.16 When population Y with means μ_Y is subtracted from population X with mean μ_X, the mean of resultant population is given by,

(a) $(\mu_X + \mu_Y)$ (b) $(\mu_X - \mu_Y)$

(c) $(\mu_X \mu_Y)$ (d) (μ_X / μ_Y)

24.17 When two populations with means μ_X and μ_Y are multiplied, the mean of resultant population is given by,

(a) $(\mu_X + \mu_Y)$ (b) $(\mu_X - \mu_Y)$

(c) $(\mu_X \mu_Y)$ (d) (μ_X / μ_Y)

24.18 When population X with mean μ_X, is divided by population Y with mean μ_Y, the mean of resultant population is given by,

(a) $(\mu_X + \mu_Y)$ (b) $(\mu_X - \mu_Y)$

(c) $(\mu_X \mu_Y)$ (d) (μ_X / μ_Y)

24.19 When two populations X and Y are added, the standard deviation of resultant population is given by,

(a) $(\hat{\sigma}_X + \hat{\sigma}_Y)$ (b) $(\hat{\sigma}_X^2 + \hat{\sigma}_Y^2)$

(c) $\sqrt{(\hat{\sigma}_X^2 + \hat{\sigma}_Y^2)}$ (d) $\sqrt{(\hat{\sigma}_X^2 - \hat{\sigma}_Y^2)}$

24.20 When population Y is subtracted from population X, the standard deviation of resultant population is given by,

(a) $(\hat{\sigma}_X + \hat{\sigma}_Y)$ (b) $(\hat{\sigma}_X^2 + \hat{\sigma}_Y^2)$

(c) $\sqrt{(\hat{\sigma}_X^2 + \hat{\sigma}_Y^2)}$ (d) $\sqrt{(\hat{\sigma}_X^2 - \hat{\sigma}_Y^2)}$

24.21 The resultant population is normally distributed,

(a) when populations of two normally distributed random variables are added

(b) when populations of two normally distributed random variables are subtracted

(c) when populations of two normally distributed random variables are multiplied

(d) any one of above

24.22 In statistically controlled system,

(a) variations due to assignable causes are corrected

(b) variations due to chance causes are corrected

(c) variations due to assignable and chance causes are corrected

(d) none of these

24.23 There is no rejection of components when

(a) design tolerance is equal to $(\pm 3\hat{\sigma})$ and the process is centered

(b) design tolerance is slightly more than $(\pm 3\hat{\sigma})$

(c) design tolerance is $(\pm 4\hat{\sigma})$

(d) any one of above

24.24 The reliability of ball bearing selected from manufacture's catalogue is

(a) 90% (b) 50%

(c) 99% (d) more than 90%

Answers:

24.1 (a)	24.2 (b)	24.3 (b)	24.4 (a)	24.5 (c)
24.6 (d)	24.7 (b)	24.8 (a)	24.9 (d)	24.10 (c)
24.11 (a)	24.12 (b)	24.13 (c)	24.14 (d)	24.15 (a)
24.16 (b)	24.17 (c)	24.18 (d)	24.19 (c)	24.20 (c)
24.21 (d)	24.22 (a)	24.23 (d)	24.24 (a)	

25 DESIGN OF I.C. ENGINE COMPONENTS

25.1 I.C. engine cylinder is made of,

(a) cast iron (b) plain carbon steel

(c) alloy steel (d) copper

25.2 The ratio of length of stroke to cylinder bore (l/D) is usually,

(a) 1/2 (b) 5 (c) 1.5 (d) 1/4

25.3 Cylinder thickness is calculated on the basis of,

(a) radial stress (b) residual stress

(c) whipping stress (d) circumferential hoop stress

25.4 The length of cylinder is taken as,

(a) equal to cylinder diameter (b) equal to length of stroke

(c) 1.15 times of stroke length (d) 1.5 times length of piston

25.5 Piston is made of,

(a) alloy steel (b) plain carbon steel

(c) cast iron (d) brass

25.6 The function of piston skirt is

(a) to provide bearing surface for side thrust

(b) to support gas load

(c) to support gudgeon pin

(d) to seal the cylinder and prevent leakage of oil past piston

25.7 The ratio of side thrust on piston to maximum gas force on piston head is,

(a) 1 (b) 0.5 (c) 0.25 (d) 0.1

25.8 For buckling consideration, the end conditions of connecting rod in plane of motion are,

(a) one end is free and the other fixed

(b) both ends are hinged

(c) both ends are fixed

(d) one end is fixed and the other hinged

25.9 The condition for connecting rod to be equally strong in plane of motion (XX) and a plane perpendicular to plane of motion (YY) is,

(a) $I_{xx} = I_{yy}$ (b) $I_{xx} = 4\ I_{yy}$

(c) $4\ I_{xx} = I_{yy}$ (d) $I_{xx} = 2\ I_{yy}$

25.10 Connecting rod is made of,

(a) aluminium (b) medium carbon steel

(c) cast iron (d) babbits

25.11 The ratio of length of connecting rod to the crank radius (L/r) is usually,

(a) 1.5 to 2 (b) 10 to 12 (c) 4 to 5 (d) 1:1

25.12 The bolts for the cap of big end of connecting rod are subjected to

(a) only gas force (b) only inertia force

(d) only side thrust (c) inertia force, gas force and side thrust

25.13 Automotive crankshafts are made by,

(a) casting process (b) machining from rolled stock

(c) drop forging process (d) welding process

25.14 The function of valve gear mechanism is,

(a) to rotate the cam

(b) to reduce the speed of crankshaft

(c) to transmit the power

(d) open and close inlet and exhaust valve

25.15 At the top dead centre position, the crankshaft is subjected to,

(a) maximum torque

(b) maximum bending moment

(c) maximum torsional and bending moment

(d) none of the above

25.16 For maximum torque condition, the crank angle is,

(a) 0^0 from top dead centre for petrol and diesel engines

(b) 33^0 before top dead centre for petrol engine and 1^0 after top dead centre position for diesel engine

(c) 25^0 to 35^0 for petrol engine and 30^0 to 40^0 for diesel engine from top dead centre

(d) 90^0 from top dead centre for petrol and diesel engines

25.17 The area of inlet valve is

(a) equal to the area of exhaust valve

(b) more than the area of exhaust valve

(c) less than the area of exhaust valve

(d) none of the above

25.18 Whipping stress is due to

(a) vibrations of crankshaft

(b) reciprocating motion of piston

(c) inertia force on connecting rod

(d) obliquity of connecting rod

25.19 When the length of connecting rod is small, it results in

(a) greater angular swing and greater side thrust on piston

(b) lesser angular swing and lesser side thrust on piston

(c) more chances of buckling failure

(d) no side thrust on piston

25.20 The design of piston head is based on,

(a) strength and rigidity considerations

(b) bending and torsional moments

(c) buckling consideration

(d) strength and heat transfer considerations

25.21 The spring index for valve spring is usually

(a) 5 (b) 8 (c) 12 (d) 20

25.22 The main objective of providing two concentric valve springs, one inside another, in heavy duty engines is,

(a) to increase force on valve

(b) to eliminate surge

(c) to provide fail safe system

(e) to provide linear force-deflection characteristic

25.23 Push rod is designed on the basis of,

(a) tensile strength (b) compression strength

(c) bending strength (d) buckling strength

25.24 The valve lift depends upon

(a) bore and length of cylinder

(b) length of connecting rod and crank radius

(c) seat angle and diameter of port

(d) length of stroke and length of piston

25.25 Valve springs have

(a) plain ends (b) plain and ground ends

(c) square ends (d) square and ground ends

Answers:

25.1 (a)	25.2 (c)	25.3 (d)	25.4 (c)	25.5 (c)
25.6 (a)	25.7 (d)	25.8 (b)	25.9 (b)	25.10 (b)
25.11 (c)	25.12 (b)	25.13 (c)	25.14 (d)	25.15 (b)
25.16 (c)	25.17 (b)	25.18 (c)	25.19 (a)	25.20 (d)
25.21 (b)	25.22 (b)	25.23 (d)	25.24 (c)	25.25 (d)