11

Strength of Materials

11.1. Strain is defined as the ratio of
(*a*) change in volume to original volume
(*b*) change in length to original length
(*c*) change in cross-sectional area to original cross-sectional area
(*d*) any one of the above
(*e*) none of the above.

11.2. Hooke's law holds good upto
(*a*) yield point
(*b*) limit of proportionality
(*c*) breaking point
(*d*) elastic limit
(*e*) plastic limit.

11.3. Young's modulus is defined as the ratio of
(*a*) volumetric stress and volumetric strain
(*b*) lateral stress and lateral strain
(*c*) longitudinal stress and longitudinal strain
(*d*) shear stress to shear strain
(*e*) longitudinal stress and lateral strain.

11.4. The unit of Young's modulus is
(*a*) mm/mm (*b*) kg/cm
(*c*) kg (*d*) kg/cm^2
(*e*) kg cm^2.

11.5. Deformation per unit length in the direction of force is known as
(*a*) strain (*b*) lateral strain
(*c*) linear strain (*d*) linear stress
(*e*) unit strain.

11.6. It equal and opposite forces applied to a body tend to elongate it, the stress so produced is called
(*a*) internal resistance
(*b*) tensile stress (*c*) transverse stress
(*d*) compressive stress
(*e*) working stress.

11.7. The materials having same elastic properties in all directions are called
(*a*) ideal materials
(*b*) uniform materials
(*c*) isotropic materials
(*d*) paractical materials
(*e*) elastic materials.

11.8. A thin mild steel wire is loaded by adding loads in equal increments till it breaks. The extensions noted with increasing loads will behave as under
(*a*) uniform throughout
(*b*) increase uniformly
(*c*) first increase and then decrease
(*d*) increase uniformly first and then increase rapidly
(*e*) increase rapidly first and then uniformly.

11.9. Fig 11.1 shows the relative positions of graphs obtained between stress and strain

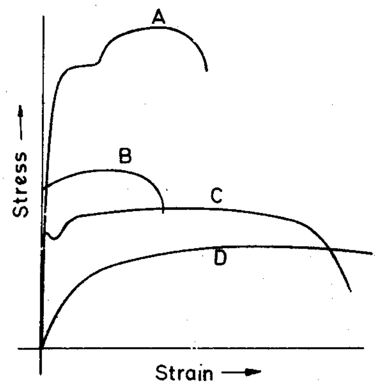

Fig. 11.1.

for four different materials, *viz.*, soft brass; cold rolled steel; low carbon steel; and medium carbon steel, heat treated
In Fig. 11.1, curve A is for
(*a*) soft brass (*b*) cold rolled steel
(*c*) low carbon steel
(*d*) medium–carbon steel heat treated
(*e*) none of the above.

11.10. In Fig. 11.1, curve *B* is for
(*a*) soft brass (*b*) cold rolled steel
(*c*) low carbon steel
(*d*) medium-carbon steel, heat treated
(*e*) none of the above.

11.11. In Fig. 11.1, curve *C* is for
(*a*) soft brass (*b*) cold rolled steel
(*c*) low carbon steel
(*d*) medium-carbon steel, heat treated
(*e*) none of the above.

11.12. In Fig. 11.1, curve *D* is for
(*a*) soft brass
(*b*) cold rolled steel
(*c*) low carbon steel
(*d*) medium-carbon steel, heat treated
(*e*) none of the above.

11.13. Modulus of rigidity is defined as the ratio of
(*a*) longitudinal stress and longitudinal strain
(*b*) volumetric stress and volumetric strain
(*c*) lateral stress and lateral strain
(*d*) shear stress and shear strain
(*e*) linear stress and lateral strain.

11.14. If the radius of wire stretched by a load is doubled, then its Young's modulus will be
(*a*) doubled (*b*) halved
(*c*) become four times
(*d*) become one-fourth
(*e*) remain unaffected.

11.15. The ultimate tensile stress of mild steel compared to ultimate compressive stress is
(*a*) same (*b*) more
(*c*) less
(*d*) more or less depending on other factors
(*e*) unpredictable.

11.16. Tensile strength of a material is obtained by dividing the maximum load during the test by the
(*a*) area at the time of fracture
(*b*) original cross-sectional area
(*c*) average of (*a*) and (*b*)
(*d*) minimum area after fracture
(*e*) none of the above.

11.17. The impact strength of a material is an index of its
(*a*) toughness (*b*) tensile strength
(*c*) capability of being cold worked
(*d*) hardness (*e*) fatigue strength.

11.18. The Young's modulus of a wire is defined as the stress which will increase the length of wire compared to its original length
(*a*) half (*b*) same amount
(*c*) double (*d*) one-fourth
(*e*) four times.

11.19. Percentage reduction of area in performing tensile test on cast iron may be of the order of
(*a*) 50% (*b*) 25%
(*c*) 0% (*d*) 15%
(*e*) 60%.

11.20. The intensity of stress which causes unit strain is called
(*a*) unit stress (*b*) bulk modulus
(*c*) modulus of rigidity
(*d*) modulus of elasticity
(*e*) principal stress.

11.21. True stress-strain curve for materials is plotted between
(*a*) load/original cross-sectional area and change in length/original length
(*b*) load/instantaneous cross-sectional area and $\log_e \left(\dfrac{\text{original area}}{\text{instantaneous area}} \right)$
(*c*) load/instantaneous cross-sectional area and change in length/original length
(*d*) load/instantaneous area and instantaneous area/original area
(*e*) none of the above.

11.22. During a tensile test on a specimen of 1 cm^2 cross-section, maximum load observed was 8 tonnes and area of cross-section at neck was 0.5 cm^2. Ultimate tensile strength of specimen is
(*a*) 4 tonnes/cm^2
(*b*) 8 tonnes/cm^2 (*c*) 16 tonnes/cm^2
(*d*) 22 tonnes/cm^2
(*e*) none of the above.

11.23. For steel, the ultimate strength in shear as compared to in tension is nearly
(*a*) same (*b*) half
(*c*) one-third (*d*) two-third
(*e*) one-fourth.

11.24. Which of the following has no unit
(*a*) kinematic viscosity
(*b*) surface tension
(*c*) bulk modulus
(*d*) strain (*e*) elasticity.

11.25. Which is the false statement about true stress-strain method
(*a*) It does not exist
(*b*) It is more sensitive to changes in both metallurgical and mechanical conditions
(*c*) It gives a more accurate picture of the ductility
(*d*) It can be correlated with stress-strain values in other tests like torsion, impact, combined stress tests etc.
(*e*) It can be used for compression tests as well.

11.26. In a tensile test on mild steel specimen, the breaking stress as compared to ultimate tensile stress is
(*a*) more (*b*) less
(*c*) same
(*d*) more/less depending on composition
(*e*) may have any value.

11.27. If a part is constrained to move and heated, it will develop
(*a*) principal stress
(*b*) tensile stress
(*c*) compressive stress
(*d*) shear stress (*e*) no stress.

11.28. Which of the following materials is most elastic
(*a*) rubber (*b*) plastic
(*c*) brass (*d*) steel
(*e*) glass.

11.29. The value of modulus of elasticity for mild steel is of the order of
(*a*) 2.1×10^5 kg/cm^2
(*b*) 2.1×10^6 kg/cm^2
(*c*) 2.1×10^7 kg/cm^2
(*d*) 0.1×10^6 kg/cm^2
(*e*) 3.8×10^6 kg/cm^2.

11.30. The value of Poisson's ratio for steel is between
(*a*) 0.01 to 0.1 (*b*) 0.23 to 0.27
(*c*) 0.25 to 0.33 (*d*) 0.4 to 0.6
(*e*) 3 to 4.

11.31. The buckling load for a given material depends on
(*a*) slenderness ratio and area of cross-section
(*b*) Poisson's ratio and modulus of elasticity
(*c*) slenderness ratio and modulus of elasticity
(*d*) slenderness ratio, area of cross-section and modulus of elasticity
(*e*) Poisson's ratio and slenderness ratio.

11.32. The total elongation produced in a bar of uniform section hanging vertically downwards due to its own weight is equal to that produced by a weight
(*a*) of same magnitude as that of bar and applied at the lower end
(*b*) half the weight of bar applied at lower end
(*c*) half of the square of weight of bar applied at lower end
(*d*) one-fourth of weight of bar applied at lower end
(*e*) none of the above.

11.33. The property of a material by virtue of which a body returns to its original shape after removal of the load is called
(*a*) plasticity (*b*) elasticity
(*c*) ductility (*d*) malleability
(*e*) resilience.

11.34. The materials which exhibit the same elastic properties in all directions are called
(*a*) homogeneous (*b*) inelastic
(*c*) isotropic (*d*) isentropic
(*e*) visco-elastic.

11.35. The value of Poisson's ratio for cast iron is
(*a*) 0.1 to 0.2 (*b*) 0.23 to 0.27
(*c*) 0.25 to 0.33 (*d*) 0.4 to 0.6
(*e*) 3 to 4.

11.36. The property of a material which allows it to be drawn into a smaller section is called
(*a*) plasticity (*b*) ductility
(*c*) elasticity (*d*) malleability
(*e*) drawability.

11.37. Poisson's ratio is defined as the ratio of
(*a*) longitudinal stress and longitudinal strain
(*b*) longitudinal stress and lateral stress
(*c*) lateral stress and longitudinal stress
(*d*) lateral stress and lateral strain
(*e*) none of the above.

11.38. For which material the Poisson's ratio is more than unity
(*a*) steel (*b*) copper
(*c*) aluminium (*d*) cast iron
(*e*) none of the above.

11.39. The property of a material by virtue of which it can be beaten or rolled into plates is called
(*a*) malleability (*b*) ductility
(*c*) plasticity (*d*) elasticity
(*e*) rollability.

11.40. The change in the unit volume of a material under tension with increase in its Poisson's ratio will
(*a*) increase (*b*) decrease
(*c*) remain same
(*d*) increase initially and then decrease
(*e*) unpredictable.

11.41. The percentage reduction in area of a cast iron specimen during tensile test would be of the order of
(*a*) more than 50%
(*b*) 25—50% (*c*) 10—25%
(*d*) 5—10% (*e*) negligible.

11.42. If a material expands freely due to heating it will develop
(*a*) thermal stresses
(*b*) tensile stress
(*c*) bending
(*d*) compressive stress
(*e*) no stress.

11.43. In a tensile test, near the elastic limit zone, the
(*a*) tensile strain increases more quickly
(*b*) tensile strain decreases more quickly
(*c*) tensile strain increases in proportion to the stress
(*d*) tensile strain decreases in proportion to the stress
(*e*) tensile strain remains constant.

11.44. The stress necessary to initiate yielding is
(*a*) considerably greater than that necessary to continue it
(*b*) considerably lesser than that necessary to continue it
(*c*) greater than that necessary to stop it
(*d*) lesser than that necessary to stop it
(*e*) equal to that necessary to stop it.

11.45. In the tensile test, the phenomenon of slow extension of the material, *i.e.* stress increasing with the time at a constant load is called
(*a*) creeping (*b*) yielding
(*c*) breaking (*d*) plasticity
(*e*) none of the above.

11.46. The stress developed in a material at breaking point in extension is called
(*a*) breaking stress
(*b*) fracture stress
(*c*) yield point stress
(*d*) ultimate tensile stress
(*e*) proof stress.

11.47. Rupture stress is
(*a*) breaking stress
(*b*) maximum load/original cross-sectional area (*A*)
(*c*) load at breaking point/*A*
(*d*) load at breaking point/neck area
(*e*) maximum stress.

11.48. The elasticity of various materials is controlled by its
(*a*) ultimate tensile stress
(*b*) proof stress
(*c*) stress at yield point
(*d*) stress at elastic limit
(*e*) tensile stress.

11.49. The ratio of lateral strain to the linear strain within elastic limit is known as
(*a*) Young's modulus
(*b*) bulk modulus
(*c*) modulus of rigidity
(*d*) modulus of elasticity
(*e*) Poisson's ratio.

11.50. The ratio of direct stress to volumetric strain in case of a body subjected to three mutually perpendicular stresses of equal intensity, is equal to
(*a*) Young's modulus
(*b*) bulk modulus
(*c*) modulus of rigidity

(*d*) modulus of elasticity
(*e*) Poisson's ratio.

11.51. The stress at which extension of the material takes place more quickly as compared to the increase in load is called
(*a*) elastic point of the material
(*b*) plastic point of the material
(*c*) breaking point of the material
(*d*) yielding point of the material
(*e*) ultimate point of the material.

11.52. Five specimens of M.S. have their lengths and diameters as l, d; $2l$, $2d$; $3l, 3d$; $4l$, $4d$ and $5l, 5d$. Which of these will have largest extension when the same tension is applied to all of them
(*a*) first (*b*) second
(*c*) third (*d*) fourth
(*e*) fifth.

11.53. The percentage reduction in area in case of cast iron when it is subjected to tensible test is of the order of
(*a*) 0% (*b*) 10%
(*c*) 20% (*d*) 25%
(*e*) 30%.

11.54. In a compression test, the fracture in cast iron specimen would occur along
(*a*) the axis of load
(*b*) perpendicular to the axis of load
(*c*) an oblique plane
(*d*) would not occur
(*e*) none of the above.

11.55. The loss of strength in compression due to overloading is known as
(*a*) hysteresis (*b*) relexation
(*c*) creep
(*d*) Bouschinger effect
(*e*) resilience.

11.56. An elastic rod, 1 m long, of negligible weight hangs downward from a support. In one case a load is applied on rod 20 cm below the support, and in other case the same load is applied at bottom of rod. The reactions at supports will be
(*a*) more in first case
(*b*) same in both the cases
(*c*) more in second case
(*d*) data are not sufficient to determine same
(*e*) none of the above.

11.57. In question 11.56, the elongation in second case compared to first case will be
(*a*) same (*b*) 5 times less
(*c*) 5 times more (*d*) 2.5 times more
(*e*) unpredictable.

11.58. In question 11.56, the internal reaction in bottom 80 cm length will be
(*a*) same in both cases
(*b*) zero in first case
(*c*) different in both cases
(*d*) data are not sufficient to determine same
(*e*) none of the above.

11.59. Flow stress corresponds to
(*a*) fluids in motion
(*b*) breaking point
(*c*) plastic deformation of solids
(*d*) rupture stress (*e*) none of the above.

11.60. When it is indicated that a member is elastic, it means that when force is applied, it will
(*a*) not deform (*b*) be safest
(*c*) stretch (*d*) not stretch
(*e*) none of the above.

11.61. The energy absorbed in a body, when it is strained within the elastic limits, is known as
(*a*) strain energy
(*b*) resilience
(*c*) proof resilience
(*d*) modulus of resilience
(*e*) toughness.

11.62. Resilience of a material is considered when it is subjected to
(*a*) frequent heat treatment
(*b*) fatigue (*c*) creep
(*d*) shock loading (*e*) resonant condition.

11.63. The maximum strain energy that can be stored in a body is known as
(*a*) impact energy
(*b*) resilience
(*c*) proof resilience
(*d*) modulus of resilience
(*e*) toughness.

11.64. The total strain energy stored in a body is termed as
(*a*) resilience (*b*) proof resilience
(*c*) modulus of resilience
(*d*) toughness (*e*) impact energy.

11.65. Proof resilience per unit volume of a material is known as
(*a*) resilience (*b*) proof resilience
(*c*) modulus of resilience
(*d*) toughness (*e*) impact energy.

11.66. Strain energy stored in a body of volume V with stress s due to gradually applied load is
(*a*) $\frac{sE}{V}$ (*b*) $\frac{sE^2}{V}$
(*c*) $\frac{sV^2}{E}$ (*d*) $\frac{s^2V}{2E}$
(*e*) $\frac{sV^2}{2E}$

11.67. The strain energy stored in a body of volume V due to shear stress s_s and shear modulus C is
(*a*) $\frac{s_s^2V}{2C}$ (*b*) $\frac{s_sV^2}{2C}$
(*c*) $\frac{s_s^2V}{C}$ (*d*) $\frac{2C}{s_s^2V}$
(*e*) $\frac{s_sV}{2C}$.

11.68. The stress induced in a body due to suddenly applied load compared to when it is applied gradually is
(*a*) same (*b*) half
(*c*) two times (*d*) four times
(*e*) none of the above.

11.69. The strain energy stored in a body due to suddenly applied load compared to when it is applied gradually is
(*a*) same (*b*) twice
(*c*) four times (*d*) eight times
(*e*) half.

11.70. A material capable of absorbing large amount of energy before fracture is known as
(*a*) ductility (*b*) toughness
(*c*) resilience (*d*) shock proof
(*e*) plasticity.

11.71. Coaxing is the method of increasing
(*a*) strength by reversible cycling
(*b*) corrosion resistance by spraying
(*c*) hardness by surface treatment
(*d*) fatigue resistance by over-stressing the metal by successively increasing loadings
(*e*) creep by head treatment.

11.72. A vertical hanging bar of length l weights w kg/unit length and carries a load W at bottom. The tensile force at distance y from support in the bar will be
(*a*) $W + w(l - y)$ (*b*) W
(*c*) $W + wl$ (*d*) $(w + W)\frac{y}{l}$
(*e*) $W + \frac{(l-y)\,W}{w}$.

11.73. A beam is loaded as cantilever. If the load at the end is increased, the failure will occur
(*a*) in the middle
(*b*) at the tip below the load
(*c*) at the support
(*d*) anywhere
(*e*) none of the above.

11.74. A non-yielding support implies that the
(*a*) support is frictionless
(*b*) support can take any amount of reaction
(*c*) support holds member firmly
(*d*) slope of the beam at the support is zero
(*e*) none of the above.

11.75. The ratio of elongation in a prismatic bar due to its own weight (W) as compared to another similar bar carrying an additional weight (W) will be
(*a*) 1 : 2 (*b*) 1 : 3
(*c*) 1 : 4 (*d*) 1 : 2.5
(*e*) 1 : 2.25.

11.76. Stress concentration in a flat bar having a hole of radius r as compared to a flat bar having same dimension but having circular fillets of radius r at both ends will be

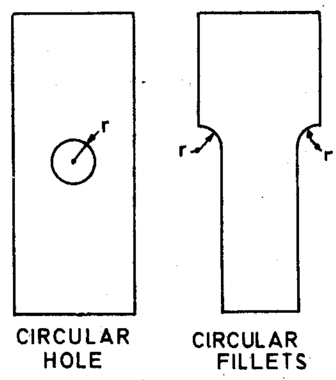

Fig. 11.2.

(*a*) same (*b*) more
(*c*) less (*d*) unpredictable
(*e*) none of the above.

11.77. A steel wire hangs vertically under its own weight. If its density is 8000 kg/m^3 and allowable stress is 2000 kg/cm^2 then how much length it can sustain
(*a*) no limit (*b*) 1250 m
(*c*) 5000 m (*d*) 2500 m
(*e*) 2000 m.

11.78. The dimensions of Young's modulus of elasticity are given by
(*a*) $M^1L^{-1}T^{-2}$ (*b*) $M^1L^{-1}T^{-1}$
(*c*) $M^1L^{-2}T^{-2}$ (*d*) $M^1L^{-1}T^{-3}$
(*e*) $M^1L^{-2}T^{-1}$.

11.79. The yield point in fatigue loading in comparison to yield point in static loading is
(*a*) same (*b*) lower
(*c*) higher (*d*) half
(*e*) twice.

11.80. In a prismatic member made of two materials so joined that they deform equally under axial stress, the unit stresses in two materials are
(*a*) equal
(*b*) proportional to their respective moduli of elasticity
(*c*) inversely proportional to their moduli of elasticity
(*d*) average of the sum of moduli of elasticity
(*e*) none of the above.

11.81. Moment of inertia of an area will be least w.r.t.
(*a*) horizontal axis
(*b*) vertical axis (*c*) bottom most axis
(*d*) central axis (*e*) point of suspension.

11.82. The distance between the centres of two consecutive rivets in the same row is called
(*a*) lead (*b*) lap
(*c*) pitch (*d*) spacing
(*e*) clearance.

11.83. In riveted boiler joints, all stresses, shearing, bearing and tensile are based on the
(*a*) size of rivet
(*b*) size of the drilled or reamed hole
(*c*) average of size of rivet and hole
(*d*) smaller of the two
(*e*) any one of the above.

11.84. The distance between the centres of the rivets in adjacent rows of zig-zag riveted joint is known as
(*a*) pitch (*b*) back pitch
(*c*) diagonal pitch
(*d*) diametral pitch
(*e*) lap.

11.85. Efficiency of a riveted joint is the ratio of its strength (max. load it can resist without failure) to the strength of the unpunched plate in
(*a*) tension (*b*) compression
(*c*) bearing
(*d*) any one of the above
(*e*) none of the above.

11.86. When two plates are butt together and riveted with cover plates with two rows of rivets, the joint is known as
(*a*) lap point
(*b*) butt joint
(*c*) single riveted single cover butt joint
(*d*) double riveted double cover butt joint
(*e*) single riveted double cover butt joint.

11.87. Increase in number of rows of rivets results in
(*a*) decrease in efficiency of joint
(*b*) increase in efficiency of joint
(*c*) no change in efficiency of joint
(*d*) increase/decrease of efficiency of joint dependent upon number of the rivets used
(*e*) none of the above.

11.88. A riveted joint in which every rivet of a row is opposite to other rivet of the outer row, is known as
(*a*) chain riveted joint
(*b*) diamond riveted joint
(*c*) criss-cross riveted joint
(*d*) zig-zag riveted joint
(*e*) none of the above.

11.89. A riveted joint in which the number of rivets decrease from innermost to outermost row is called
(*a*) chain riveted joint
(*b*) diamond riveted joint
(*c*) criss-cross riveted joint
(*d*) zig-zag riveted joint
(*e*) none of the above.

11.90. The diameter of rivets in mm for a plate of thickness 't' mm is taken as
(*a*) t (*b*) $2t$
(*c*) $\sqrt{t}$ (*d*) $1.41\sqrt{t}$
(*e*) $6.05\sqrt{t}$.

11.91. If the rivets in adjacent rows are staggered and the outermost row has only one rivet, the arrangement of the rivets is called
(*a*) chain riveting
(*b*) zig zag riveting
(*c*) diamond riveting
(*d*) criss-cross riveting
(*e*) none of the above.

11.92. Diamond riveted joint can be adopted in the case of following type of joint
(*a*) butt joint (*b*) lap joint
(*c*) double riveted lap joints
(*d*) all types of joints
(*e*) none of the above.

11.93. Rivets are made of following type of material
(*a*) tough (*b*) hard
(*c*) resilient (*d*) ductile
(*e*) malleable.

11.94. A riveted joint in which the spacing of the rivets is staggered in such a way that every rivet is in the middle of the two rivets of the opposite row is known as
(*a*) zig-zag riveted joint
(*b*) diamond riveted joint
(*c*) butt riveted joint
(*d*) chain riveted joint
(*e*) criss-cross riveted joint.

11.95. The weakest section of a diamond riveting is the section which passes through
(*a*) the first row
(*b*) the second row
(*c*) the central row
(*d*) one rivet hole of the end row
(*e*) none of the above.

11.96. If b is the width of a plate joined by diamond riveting of diameter (d), the efficiency of the joint is given by
(*a*) $\frac{b+d}{b}$ (*b*) $\frac{b-d}{b}$
(*c*) $\frac{d-b}{d}$ (*d*) $\frac{b-d}{d}$
(*e*) $\frac{b}{b-d}$.

11.97. In case of an eccentric loading on a bracket subjected to moment (M), the tangential force developed in any rivet, at right angles to its radius vector (r) is
(*a*) $\frac{Mr}{\Sigma r^2}$ (*b*) $\frac{\Sigma r^2}{Mr}$
(*c*) $\frac{Mr^2}{\Sigma r^2}$ (*d*) $\frac{\sqrt{Mr}}{\Sigma r^2}$
(*e*) $\frac{Mr^2}{\Sigma r^2}$.

11.98. A beam of length l, having uniform load of w kg per unit length, is supported freely at the ends. The bending moment at mid span will be
(*a*) $\frac{wl}{4}$ (*b*) $\frac{wl^2}{2}$
(*c*) $\frac{wl^2}{4}$ (*d*) $\frac{wl^2}{8}$
(*e*) none of the above.

11.99. Twisting couple in a shaft introduces in it
(*a*) bending moment
(*b*) deflection (*c*) shear strain
(*d*) stress (*e*) shear stress.

11.100. A circular bar is subjected to an axial force of 1000 kg. The components of force normal and tangent to a plane inclined at 45° to the axis of bar will be
(*a*) 1003 kg each (*b*) 707 kg each
(*c*) 500 kg each
(*d*) not possible to determine with these data
(*e*) none of the above.

11.101. A rectangular plate 8 cm long and 6 cm wide is subjected to normal forces of 600 kg at 6 cm side and 200 kg at 8 cm side. The normal and tangential components of force on the diagonal plane will be
(*a*) 200 kg and 600 kg
(*b*) 520 kg and 360 kg
(*c*) 800 kg and 400 kg
(*d*) unpredictable
(*e*) none of the above.

11.102. The tensile stress in a conical rod, having diameter D at bottom, d at top, length l and subjected to tensile force F_t at distance x from small end will be
(*a*) $\frac{4F}{\pi D^2}$ (*b*) $\frac{4F}{\pi d^2}$

(c) $\frac{4F}{\pi (D-d)^2}$ (d) $\frac{4Fl^2}{\pi [(D-d)x + ld]^2}$
(e) none of the above.

11.103. The deformation of a bar under its own weight compared to the deformation of same body subjected to a direct load equal to weight of the body is
(a) same (b) double
(c) half (d) four times
(e) one-fourth.

11.104. The extension of a circular bar tapering uniformly from diameter d_1 to d_2 is same as of uniform circular bar of diameter
(a) $\frac{d_1 + d_2}{2}$ (b) $\frac{d_1 - d_2}{2}$
(c) $\sqrt{d_1 d_2}$ (d) $\sqrt{d_1^2 - d_2^2}$
(e) $(d_1 \times d_2)^{3/2}$.

11.105. The extension of a circular bar of length l tapering from d_1 to d_2 and subjected to axial pull F is
(a) $\frac{4Fl}{\pi E d_1 d_2}$ (b) $\frac{4Fl}{\pi d_1 d_2}$
(c) $\frac{4Fl}{\pi E \sqrt{d_1 d_2}}$ (d) $\frac{4Fl}{\pi E (d_1 + d_2)}$
(e) $\frac{Fl}{2\pi E d_1 d_2}$.

11.106. The elongation of a freely hanging uniform steel rope, if its length is doubled will increase in the ratio of
(a) 2 : 1 (b) 1 : 1
(c) 4 : 1 (d) 8 : 1
(e) 16 : 1.

11.107. Young's Modulus is defined as the ratio of
(a) longitudinal stress to longitudinal strain
(b) lateral strain to longitudinal strain
(c) lateral stress to longitudinal strain
(d) longitudinal stress to lateral strain
(e) none of the above.

11.108. If a material is loaded beyond yield point stress
(a) it becomes elastic
(b) it becomes ductile
(c) its resistance to fatigue increases
(d) it loses its tendency to return to 's original shape
(e) it becomes brittle.

11.109. If all the dimensions of a prismatic bar be increased in the ratio of $k : 1$, then maximum stress produced in the bar due to its own weight will increase in the following ratio
(a) $k : 1$ (b) $k^2 : 1$
(c) $k^3 : 1$ (d) $1 : k$
(e) $1 : k^2$.

11.110. Modulus of rigidity is defined as the ratio of
(a) linear stress to longitudinal strain
(b) stress to volumetric strain
(c) shear stress to shear strain
(d) stress to strain
(e) stress to longitudinal strain.

11.111. A body is subjected to three mutually perpendicular stresses of equal intensity. The ratio of direct stress to corresponding volumetric strain is called
(a) Poisson's ratio
(b) bulk modulus
(c) modulus of rigidity
(d) young's modulus
(e) modulus of elasticity.

11.112. A tapered bar of length l with diameter D at base and having specific weight ρ is suspended freely under its own weight. The elongation of bar will be
(a) $\frac{\rho l^3}{6E}$ (b) $\frac{\rho l^3}{E}$
(c) $\frac{\rho l^3}{4E}$ (d) $\frac{\rho l^3}{3E}$
(e) $\frac{\rho l^3}{2E}$.

11.113. A tapered bar of length l has diameters D and d at ends. If it is subjected to axial load F, then the elongation produced will be
(a) $\frac{\rho l}{\pi D d}$ (b) $\frac{\rho l}{Dd}$
(c) $\frac{\pi \rho l}{Dd}$ (d) $\frac{16\pi \rho l}{Dd}$
(e) $\frac{\rho l}{16\pi D d}$.

11.114. There are two bars, one prismatic and one conical. Length and weight of both are

same. The ratio of their elongation due to their own weight will be
(*a*) 0.5 : 1 (*b*) 1 : 1
(*c*) 3 : 1 (*d*) 1 : 2
(*e*) 1 : 3.

11.115. Fig. 11.3 shows the relationship of various scales of hardness w.r.t. Brinell Numbers
Curve *A* is for

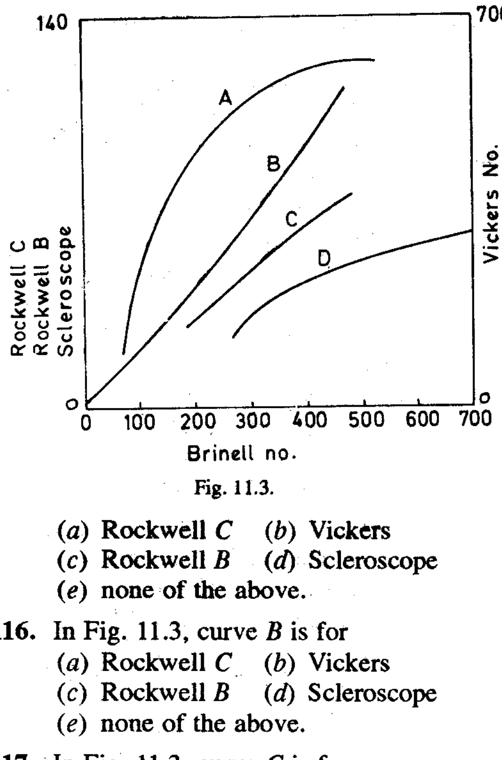

Fig. 11.3.

(*a*) Rockwell *C* (*b*) Vickers
(*c*) Rockwell *B* (*d*) Scleroscope
(*e*) none of the above.

11.116. In Fig. 11.3, curve *B* is for
(*a*) Rockwell *C* (*b*) Vickers
(*c*) Rockwell *B* (*d*) Scleroscope
(*e*) none of the above.

11.117. In Fig. 11.3, curve *C* is for
(*a*) Rockwell *C* (*b*) Vickers
(*c*) Rockwell *B* (*d*) Scleroscope
(*e*) none of the above.

11.118. In Fig. 11.3, curve *D* is for
(*a*) Rockwell *C* (*b*) Vickers
(*c*) Rockwell *B* (*d*) Scleroscope
(*e*) none of the above.

11.119. A structure, made up of sereval bars, joined together is known as
(*a*) beam (*b*) column
(*c*) strut (*d*) tie
(*e*) frame.

11.120. A frame in which the number of members is just sufficient to keep it in equilibrium, is known as
(*a*) theoretical frame
(*b*) perfect frame
(*c*) ideal frame
(*d*) deficient frame
(*e*) redundant frame.

11.121. The number of members in a perfect frame having J number of joints is equal to
(*a*) $2J - 1$ (*b*) $3J - 2$
(*c*) $2J - 3$ (*d*) $3J - 1$
(*e*) $2J + 3$.

11.122. The force acting along the circumference will cause stress in the walls in a direction normal to the longitudinal axis of cylinder; this stress is called
(*a*) longitudinal stress
(*b*) hoop stress
(*c*) yeiled stress
(*d*) ultimate stress
(*e*) none of the above.

11.123. A boiler shell 200 cm diameter and plate thickness 1.5 cm is subjected to internal pressure of 1.5 MN/m^2, then the hoop stress will be
(*a*) 30 MN/m^2 (*b*) 50 MN/m^2
(*c*) 100 MN/m^2 (*d*) 200 MN/m^2
(*e*) 300 MN/m^2.

11.124. A cylindrical section having no joint is known as
(*a*) jointless section
(*b*) homogeneous section
(*c*) perfect section
(*d*) manufactured section
(*e*) seamless section.

11.125. Longitudinal stress in a thin cylinder is
(*a*) equal to the hoop stress
(*b*) twice the hoop stress
(*c*) half of the hoop stress
(*d*) one-fourth of hoop stress
(*e*) four times the hoop stress.

11.126. The radius taken into consideration in calculating the stress in a hollow shaft subjected to torsion is
(*a*) inner radius (*b*) outer radius
(*c*) mean radius
(*d*) both inner and outer radii
(*e*) geometric mean of inner and outer radii.

11.127. Thin cylinders are used to store

(*a*) water (*b*) oil
(*c*) gas (*d*) steam
(*e*) any one of the above.

11.128. The safe twisting moment for a compound shaft is equal to the
(*a*) maximum calculated value
(*b*) minimum calculated value
(*c*) mean value
(*d*) extreme value
(*e*) none of the above.

11.129. The ratio of maximum shear stress to maximum normal stress at any point in a solid circular shaft is
(*a*) 1 (*b*) $\frac{1}{2}$
(*c*) 2 (*d*) $\frac{2}{3}$
(*e*) $\frac{3}{2}$.

11.130. The torsional rigidity of a shaft is expressed by the
(*a*) maximum torque it can transmit
(*b*) number of cycles it undergoes before failure
(*c*) elastic limit upto which it resists torsion, shear and bending stresses
(*d*) torque required to produce a twist of one radian per unit length of shaft
(*e*) maximum power it can transmit at highest possible speed.

11.131. Strain energy stored in a solid circular shaft is proportional to
(*a*) GJ (torsional rigidity of shaft)
(*b*) $1/GJ$ (*c*) $(GJ)^2$
(*d*) $\frac{1}{(GJ)^2}$ (*e*) $\frac{1}{\sqrt{GJ}}$.

11.132. In the case of thick cylinder, the ratio of cylinder diameter to wall thickness is less than or equal to
(*a*) 5 (*b*) 10
(*c*) 20 (*d*) 40
(*e*) 50.

11.133. If r_i and r_0 be the inner and outer radii of a cylinder then theory of thick cylinder can be applied when r_i/r_0 is equal to
(*a*) 1 (*b*) less than 1
(*c*) 1.2 (*d*) less than 1.2
(*e*) greater than 1.2.

11.134. The value of shear stress which is induced in the shaft due to the applied couple varies
(*a*) from maximum at the centre to zero at the circumference
(*b*) from zero at the centre to maximum at the circumference
(*c*) from maximum at the centre to minimum at the cricumference
(*d*) from minimum at the centre to maximum at the circumference
(*e*) none of the above.

11.135. A key is subjected to side pressure as well at shearing forces. These pressures are called
(*a*) bearing stresses
(*b*) fatigue stresses
(*c*) crushing stresses
(*d*) resultant stresses
(*e*) none of the above.

11.136. In a belt drive, the pulley diameter is doubled, the belt tension and pulley width remaining same. The changes required in key will be
(*a*) increase key length
(*b*) increase key depth
(*c*) increase key width
(*d*) double all the dimensions
(*e*) none of the above.

11.137. Shear stress induced in a shaft subjected to tension will be
(*a*) maximum at periphery and zero at centre
(*b*) maximum at centre
(*c*) uniform throughout
(*d*) average value in centre
(*e*) none of the above.

11.138. In the design of pulley, key and shaft
(*a*) all three are designed for same strength
(*b*) key is made weaker link
(*c*) pulley is made weaker
(*d*) shaft is made weaker
(*e*) key is made strongest link.

11.139. A column is fabricated by inserting one tube into another for a distance l and then brazing the two to make a homogeneous joint. The outside diameter of small tube is 5 cm and whole column is subjected to compressive load of 1000π kg. The allow-

able compressive stress is 250 kg/cm^2 and shear stress 100 kg/cm^2. The inside diameter of small tube is
(*a*) 3 cm (*b*) 2 cm
(*c*) 4 cm (*d*) 1 cm
(*e*) 6 cm.

11.140. In above example 11.139, the outside diameter of bigger tube is
(*a*) 6.0 cm (*b*) 7.0 cm
(*c*) 7.5 cm (*d*) 6.4 cm
(*e*) 12.8.

11.141. In above example 11.139, the length l is
(*a*) 5 cm (*b*) 3 cm
(*c*) 2 cm (*d*) 1 cm
(*e*) 4 cm.

11.142. A member is subjected to tensile force F and its normal cross-section perpendicular to line of force is A. The resulting normal stress in an oblique plane inclined at angle θ to traverse plane will be
(*a*) $\frac{P}{A}\cos^2\theta$ (*b*) $\frac{P}{2A}\sin 2\theta$
(*c*) $\frac{P}{A}\sin^2\theta$ (*d*) $\frac{P}{2A}\cos 2\theta$
(*e*) $\frac{P}{A}\cos 2\theta$.

11.143. In above example 11.142, the resulting shear stress in oblique plane inclined at angle θ to transverse plane will be
(*a*) $\frac{P}{A}\cos^2 2\theta$ (*b*) $\frac{P}{2A}\sin 2\theta$
(*c*) $\frac{P}{A}\sin^2\theta$ (*d*) $\frac{P}{2A}\cos 2\theta$
(*e*) $\frac{P}{2A}\sin^2\theta$.

11.144. Mohr's circle can be used to determine following stress on inclined surface
(*a*) principal stresses
(*b*) normal stress
(*c*) tangential stress
(*d*) maximum shear stress
(*e*) all of the above.

11.145. Which of the following constitutes statically determinate beams
(*a*) simply supported cantilevers and overhung beams
(*b*) cantilevers and fixed beams
(*c*) continuous beams and beams carrying uniformly distributed loads
(*d*) fixed beams and simply supported beams
(*e*) none of the above.

11.146. At the principal planes
(*a*) the normal stress is maximum or minimum and the shear stress is zero
(*b*) the tensile and compressive stresses are zero
(*c*) the tensile stress is zero and the shear stress is maximum
(*d*) no stress acts
(*e*) all the stresses are maximum.

11.147. If a material is subjected to a tensile load, then to avoid the shear failure of a material along a plane inclined at 45° to the direction of the tensile stress, the material should have its shear strength at least equal to
(*a*) its tensile strength
(*b*) half the tensile strength
(*c*) its compressive strength
(*d*) principal stress
(*e*) half the difference of tensile and compressive stresses.

11.148. The longitudinal stress induced in a thin walled cylindrical vessel is
(*a*) $\frac{pD}{2t}$ (*b*) $\frac{pD}{4t}$
(*c*) $\frac{pD}{t}$ (*d*) $\frac{pD}{3t}$
(*e*) $\frac{pD}{6t}$.

11.149. A flat plate with fillets of radius r shown in Fig. 11.4 is subjected to tensile loading. As the value $\frac{h}{r}$ increases, the stress concentration factor will

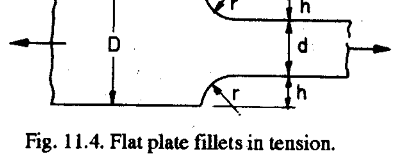

Fig. 11.4. Flat plate fillets in tension.

(*a*) decrease (*b*) increase
(*c*) remain same (*d*) unpredictable
(*e*) none of the above.

11.150. In Fig. 11.4 as the value $\frac{h}{d}$ increases, the stress concentration factor will

(*a*) decrease (*b*) increase
(*c*) remain same (*d*) unpredictable
(*e*) none of the above.

11.151. The elongation produced in a tapered shaft with end diameters d_1 and d_2 due to tensile or compressive axial load is proportional to
(*a*) $d_1 + d_2$ (*b*) $\frac{1}{d_1 + d_2}$
(*c*) $d_1 d_2$ (*d*) $\frac{1}{d_1 d_2}$
(*e*) $\sqrt{\frac{1}{d_1 d_2}}$.

11.152. Fig. 11.5 shows the effect of mean stress on the variable stress for failure. Which is correct curve

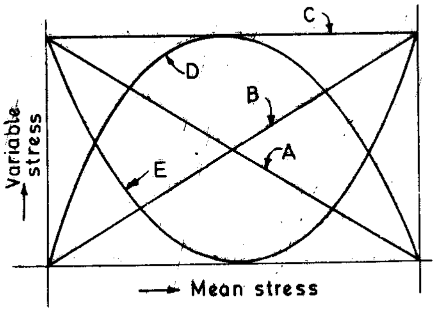

Fig. 11.5.

(*a*) curve *A* (*b*) curve *B*
(*c*) curve *C* (*d*) curve *D*
(*e*) curve *E*.

11.153. The circumferential stress induced in a thin-walled cylindrical vessel is
(*a*) $\frac{pD}{2t}$ (*b*) $\frac{pD}{4t}$
(*c*) $\frac{pD}{t}$ (*d*) $\frac{pD}{3t}$
(*e*) $\frac{pD}{6t}$.

11.154. The forces in the various members of a perfect frame can be found out from
(*a*) method of joints
(*b*) method of section
(*c*) graphical method
(*d*) any one of the above
(*e*) none of the above.

11.155. The maximum load to which fillet joint of length *L* can be subjected is equal to
(*a*) 0.7 × permissible shear stress × fillet size × *L*
(*b*) 2 × permissible shear stress × fillet size × *L*
(*c*) permissible shear stress × fillet size × *L*
(*d*) $\frac{\text{permissible shear stress} \times \text{fillet size} \times L}{3}$
(*e*) none of the above.

11.156. A large cylindrical vessel was sealed in summer. What is likely to happen to it in winter
(*a*) nothing
(*b*) explode
(*c*) buckle and collapse
(*d*) become lighter
(*e*) seal will get loosened.

11.157. Determine the smallest hole that can be punched in a plate of thickness *t* if maximum crushing stress in punch is 4 times the shear stress of plate.
(*a*) *t* (*b*) *t*/2
(*c*) 2*t* (*d*) *t*/4
(*e*) 4*t*.

11.158. Pressure vessels are not made of rectangular shape, because
(*a*) these are difficult to fabricate
(*b*) they are not economical
(*c*) they do not give pleasing appearance
(*d*) it has been practice to use cylindrical vessels
(*e*) none of the above.

11.159. A spherical vessel with an inside diameter of 2 m is made of material having an allowable stress in tension of 500 kg/cm^2. The thickness of a shell to withstand a pressure of 25 kg/cm^2 should be
(*a*) 5 cm (*b*) 10 cm
(*c*) 2.5 cm (*d*) 1.25 cm
(*e*) none of the above.

11.160. Units of strain are
(*a*) dimensionless (*b*) cm/cm
(*c*) kg/cm^2/cm (*d*) kg/cm
(*e*) none of the above.

11.161. A cylindrical bar of *L* metres deforms by *l* cm. The strain in bar is
(*a*) *l*/*L* (*b*) 0.1 *l*/*L*
(*c*) 0.01 *l*/*L* (*d*) 100 *l*/*L*
(*e*) none of the above.

11.162. A cylindrical steel bar having length of 0.25 m is subjected to a tensile force of 2000 kg. If stress and total elongation are not to exceed 1000 kg/cm^2 and 0.01 cm respectively, and $E = 2 \times 10^6$ kg/cm^2 then its cross-sectional area should be
(*a*) 2 cm^2 (*b*) 2.5 cm^2
(*c*) 2.25 cm^2 (*d*) 5 cm^2
(*e*) unpredictable.

11.163. A rigid beam of negligible weight is supported in a horizontal position by two rods of steel and aluminium, 2 m and 1 m long having values of cross-sectional areas 1 cm^2 and 2 cm^2 and E of 2×10^6 kg/cm^2 and 1×10^6 kg/cm^2 respectively. What is the guide line for placing a load P on beam such that it remains horizontal
(*a*) forces on both rods should be equal
(*b*) force on aluminium rod should be twice the force on steel
(*c*) force on steel rod should be twice the force on aluminium
(*d*) unpredictable
(*e*) none of the above.

11.164. In Prob. 11.163, determine the position of P with respect to steel rod, if beam length is 2 m
(*a*) in the middle of beam
(*b*) 1.33 m from steel rod
(*c*) 0.67 m from steel rod
(*d*) 1.66 m from steel rod
(*e*) none of the above.

11.165. A load of 20,000 kg applied to a brass cylinder 40 cm long and 10 cm in diameter caused the length to increase 0.8 cm and the diameter to decrease 0.005 cm. Poisson's ratio of brass is
(*a*) 0.25 (*b*) 0.925
(*c*) 0.4 (*d*) 2.5
(*e*) 4.

11.166. A load of 10,000 kg is supported on three columns, two steel and one bronze, all of equal length and equal cross-sectional area of 10 cm^2 initially. Determine temperature change necessary to just relieve the bronze column of all stresses

$\alpha_s = 10 \times 10^{-6}$ cm/cm/°C

$\alpha_b = 20 \times 10^{-6}$ cm/cm/°C

$E_s = 2 \times 10^6$ kg/cm^2

$E_b = 1 \times 10^6$ kg/cm^2

(*a*) 25°C decrease
(*b*) 12.5°C increase
(*c*) 1.25°C decrease
(*d*) 2.5°C decrease
(*e*) 1.25°C increase.

11.167. A composite bar of copper and steel is heated. The ratio of tensile force in steel and the compressive stress in copper will be
(*a*) 1.0 (*b*) 0.5
(*c*) 2.0
(*d*) in proportion of values E of copper and steel
(*e*) inversely proportional to values of E of copper and steel.

11.168. A steel bar and a brass bar each of 1 metre length and cross-sectional areas 10 cm^2 and 20 cm^2, each secured to a rigid support, are rastened at their free ends by a pin. If temperature drops by 25°C, the pin will be loaded by
(*a*) 2500 kg (*b*) 25,000 kg
(*c*) 250 kg (*d*) 25 kg
(*e*) none of the above.

Take $\alpha_s = 10 \times 10^{-6}$ cm/cm/°C

$\alpha_b = 20 \times 10^{-6}$ cm/cm/°C

$E_s = 2 \times 10^6$ kg/cm^2

$E_b = 1 \times 10^6$ kg/cm^2.

11.169. A composite bar made of steel and copper is heated up. The stresses developed in steel and copper will be
(*a*) compressive and tensile
(*b*) compressive and bending
(*c*) bending and tensile
(*d*) tensile and compressive
(*e*) tensile and torsional.

11.170. The relationship between modulus of elasticity E, modulus of rigidity G is
(*a*) $E = G(1 + \mu)$ (*b*) $G = E(2 - \mu)$
(*c*) $G = \dfrac{E}{2(1+\mu)}$ (*d*) $G = \dfrac{E}{1 + 2\mu}$
(*e*) $G = \dfrac{2E}{1 + 2\mu}$.
where μ = Poisson's ratio.

11.171. Volumetric strain of a rectangular body subjected to an axial force, in terms of

linear strain e and Poisson's ratio μ, is equal to
(a) $e(1-2\mu)$ (b) $e(1-\mu)$
(c) $e(1-3\mu)$ (d) $e(1+\mu)$
(e) $e(1+2\mu)$.

11.172. A cube is subjected to three mutually perpendicular tensile stresses s, the volumetric strain will be
(a) $\frac{3s}{E}(1-2\mu)$ (b) $\frac{E}{3s}(1-2\mu)$
(c) $\frac{E}{3s}(2\mu-1)$ (d) $\frac{3s}{E}(2\mu-1)$
(e) $\frac{s}{E}(1-2\mu)$.

11.173. The Poisson's ratio determined by taking readings when load is applied gradually compared to that taken with load applied at a faster rate would be
(a) same (b) different
(c) more or less same
(d) depends on other factors
(e) none of the above.

11.174. Bulk modulus K in terms of modulus of elasticity E and Poisson's ratio μ is given as equal to
(a) $\frac{E}{3(1-2\mu)}$ (b) $E(1-2\mu)$
(c) $3E(1-2\mu)$ (d) $\frac{E}{3}(1+2\mu)$
(e) $\frac{E}{3}(1-3\mu)$.

11.175. A solid cube is subjected to equal normal forces of the similar nature on all the faces. The ratio of volumetric strain and linear strain in any of the three axes will be
(a) 1 (b) 2
(c) 3 (d) $\sqrt{3}$
(e) a value dependent on Poisson's ratio.

11.176. A structural member subjected to an axial compressive force is called
(a) beam (b) column
(c) frame (d) strut
(e) structure.

11.177. The ratio of shear modulus to the modulus of elasticity if Poisson's ratio is 0.25 will be
(a) 0.4 (b) 0.25
(c) 4 (d) 0.5
(e) 2.

11.178. Two solid shafts are made of same material and have their diameters D and $D/2$. The ratio of strength of bigger shaft to smaller one in torsion is
(a) 4 (b) 2
(c) 8 (d) 16
(e) 32.

11.179. The strain energy stored in a hollow shaft of outer and inner diameters D and d subjected to shear stress s_s and having modulus of rigidity C is equal to
(a) $\frac{s_s^2}{4C}\left(\frac{D^2-d^2}{D}\right)\times$ volume
(b) $\frac{s_s^2}{2C}\left(\frac{D^2-d^2}{D}\right)\times$ volume
(c) $\frac{s_s}{4C}\left(\frac{D^2-d^2}{D}\right)\times$ volume
(d) $\frac{s_s^2}{2C}\left(\frac{D^2-d^2}{D^2}\right)\times$ volume
(e) $\frac{s_s^2}{4C}\left(\frac{D^2+d^2}{D}\right)\times$ volume.

11.180. Compare the strengths of solid and hollow shafts both having outside diameter D and hollow shaft having inside diameter of $D/2$ in torsion. The ratio of strength of solid to hollow shafts in torsion will be
(a) 0.5 (b) 0.75
(c) 15/16 (d) 0.25
(e) 1/16.

11.181. Torsion bars are in series
(a) if same torque acts in each
(b) if they have equal angles of twist and an applied torque apportioned between them
(c) are not possible
(d) if their ends are welded together
(e) none of the above.

11.182. 100 kW is to be transmitted by each of two separate shafts. A is turning at 250 rpm and B at 300 rpm. Which shaft must have greater diameter
(a) A (b) B
(c) both will have same diameter
(d) unpredictable
(e) none of the above.

11.183. Torsional rigidity of a solid circular shaft of diameter 'd' is proportional to
(*a*) d (*b*) d^2
(*c*) $\frac{1}{d^2}$ (*d*) d^4
(*e*) $\frac{1}{d^4}$.

11.184. The ratio of maximum shear stress to the average shear stress in case of a circular beam transmitting power is equal to
(*a*) 3/2 (*b*) 4/3
(*c*) 7/4 (*d*) 2
(*e*) 5/2.

11.185. The ratio of maximum shear stress to the average shear stress in a rectangular beam subjected to torsion is
(*a*) 3/2 (*b*) 4/3
(*c*) 7/4 (*d*) 2
(*e*) 5/2.

11.186. The elongation of a close coiled helical spring subjected to tensile load is proportional to
(*a*) mean diameter of spring
(*b*) reciprocal of length of spring
(*c*) diameter of wire of coil
(*d*) shear modulus of the material or spring
(*e*) reciprocal of mean diameter of spring.

11.187. The minimum thickness of a flange forged at the end of shaft is determined by the
(*a*) compression between two flanges
(*b*) tightening of bolts
(*c*) fact that it must be sufficient to prevent the shaft from shearing out of the flange on the cylindrical surface
(*d*) any one of the above
(*e*) maximum of the above.

11.188. Compression members tend to buckle in the direction of
(*a*) axis of load
(*b*) perpendicular to axis of load
(*c*) minimum cross-section
(*d*) least radius of gyration
(*e*) unpredictable.

11.189. Moment of inertia of an area is always least with respect to
(*a*) bottom-most axis
(*b*) radius of gyration
(*c*) central axis
(*d*) centroidal axis
(*e*) none of the above.

11.190. Torsion bars are in parallel
(*a*) if same torque acts on each
(*b*) if they have equal angles of twist and applied torque apportioned between them
(*c*) are not possible
(*d*) if their ends are connected together
(*e*) none of the above.

11.191. When the external forces and moments that support an object can be found by the equations of statics alone, the object is said to be
(*a*) free body
(*b*) statically determinate
(*c*) statically indeterminate
(*d*) homogeneous (*e*) none of the above.

11.192. The reactions of each support of beam can be determined from following condition of equilibrium
(*a*) algebraic sum of all vertical forces is zero
(*b*) algebraic sum of all horizontal forces is zero
(*c*) algebraic sum of moments about any point is zero
(*d*) all of the above
(*e*) none of the above.

11.193. Section modulus Z is expressed as
(*a*) I/y (*b*) E/I
(*c*) M/I (*d*) EI
(*e*) I_y.

11.194. The section modulus of a circular section about an axis through its c.g. is
(*a*) $\frac{\pi d^3}{16}$ (*b*) $\frac{\pi d^3}{32}$
(*c*) $\frac{\pi d^3}{64}$ (*d*) $\frac{\pi d^2}{32}$
(*e*) $\frac{\pi d^3}{8}$.

11.195. If the section modulus of a beam decreases, then bending stress will
(*a*) decrease (*b*) increase
(*c*) remain same
(*d*) there is no such correlation
(*e*) none of the above.

11.196. For a given stress the ratio of moment of resistance of a square beam with its sides horizontal, compared to when the diagonal is horizontal, is
(*a*) 2 (*b*) $\sqrt{2}$
(*c*) 1.2 (*d*) 1.5
(*e*) $2\sqrt{2}$.

11.197. The moment diagram for a cantilever beam subjected to bending moment at end of beam will be
(*a*) rectangle (*b*) triangle
(*c*) parabola (*d*) cubic parabola
(*e*) elliptical.

11.198. If the load at free end on a cantilever is increased so as to cause rupture, same will occur
(*a*) below the load
(*b*) at fixed end
(*c*) between fixed end and centre
(*d*) at centre
(*e*) between centre and free end.

11.199. Fig. 11.6 (*a*) shows a central load on a beam supported firmly at the ends. The bending moment diagram for it will be as shown in Fig. 11.6
(*a*) (*b*) (*b*) (*c*)
(*c*) (*d*) (*d*) (*e*)
(*e*) none of the above.

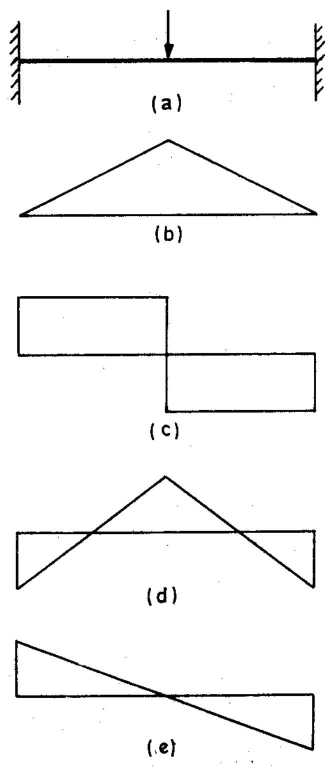

Fig. 11.6.

11.200. Shear force at any point on the beam is the algebraic sum of
(*a*) all vertical forces
(*b*) all horizontal forces
(*c*) forces on either side of the point
(*d*) moments of forces on either side of the point
(*e*) all of the above.

11.201. Bending moment at any point is equal to the algebraic sum of
(*a*) all vertical forces
(*b*) all horizontal forces
(*c*) forces on either side of the point
(*d*) moments of forces on either side of the point
(*e*) all of the above.

11.202. The rate of change of shearing force at any section is equal to the rate of
(*a*) loading at that section
(*b*) change of deflection at that section
(*c*) change of bending moment at that section
(*d*) integration of bending moment at that section
(*e*) none of the above.

11.203. Two equal length beams are fixed at their ends. One carries a distributed load and other carries same load but concentrated in the middle. The ratio of maximum deflections will be
(*a*) 2 (*b*) 3
(*c*) 4 (*d*) 6
(*e*) 8.

11.204. Two cantilever beams are of equal length. One carries a uniformly distributed load and other carries same load but concentrated at the free end. The ratio of maximum deflections is
(*a*) 5/6 (*b*) 2/3
(*c*) 1/2 (*d*) 1/3
(*e*) 5/12.

11.205. Two simply supported beams are of equal length. One carries a central load of W and other carries the uniformly distributed load such that total load is W. The ratio of maximum deflection in two cases is

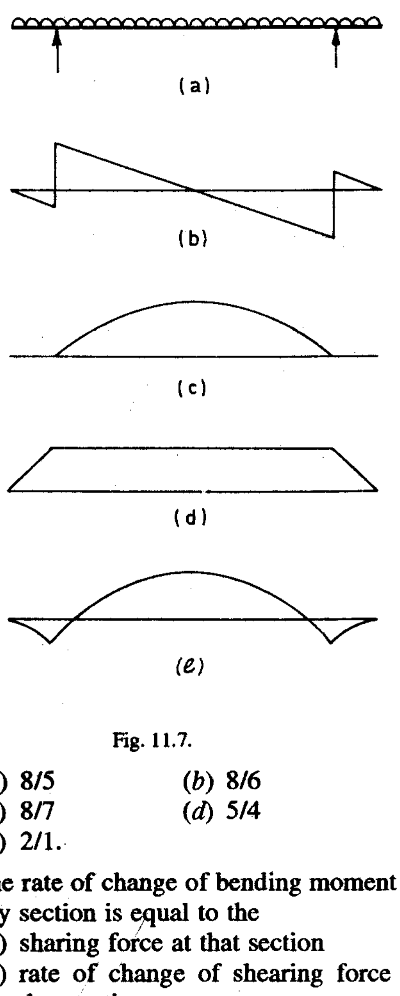

Fig. 11.7.

(*a*) 8/5 (*b*) 8/6
(*c*) 8/7 (*d*) 5/4
(*e*) 2/1.

11.206. The rate of change of bending moment at any section is equal to the
(*a*) sharing force at that section
(*b*) rate of change of shearing force at that section
(*c*) deflection at that section
(*d*) rate of change of deflection at that section
(*e*) rate of loading at that section.

11.207. Fig. 11.7 (*a*) shows a beam supported at two ends with some overhanging portions and carrying uniformly distributed load. The bending moment diagram for it will be as shown in Fig. 11.7
(*a*) (*b*) (*b*) (*c*)
(*c*) (*d*) (*d*) (*e*)
(*e*) none of the above.

11.208. The moment diagram for a cantilever beam carrying uniformly distributed load will be
(*a*) rectangle (*b*) triangle
(*c*) parabola (*d*) cubic
(*e*) elliptical.

11.209. The reaction in the case of hinged support
(*a*) acts perpendicular to beam
(*b*) perpendicular to surface of hinge
(*c*) along the surface of beam
(*d*) opposite to the direction of load
(*e*) in any direction depending upon the loads.

11.210. If the shear force is zero along a section, the M.B. at that section will be
(*a*) minimum (*b*) maximum
(*c*) zero
(*d*) could have any value
(*e*) either minimum or maximum.

11.211. Zero bending moment in a fixed beam of length 'l' carrying uniformly distributed load will occur at
(*a*) $l/2$ (*b*) $l/3$
(*c*) $l/4$ (*d*) $l/6$
(*e*) none of the above.

11.212. If a beam is cut in halves horizontally and the two halves laid side by side then the later in comparison to the original beam will carry
(*a*) same load (*b*) double load
(*c*) half load (*d*) one fourth load
(*e*) four-times load.

11.213. The bulk modulus of a material is defined as the ratio of
(*a*) volume change to modulus of elasticity
(*b*) stress intensity to volumetric strain
(*c*) volume change to original volume
(*d*) pressure applied to the change in volume
(*e*) volumetric strain to the stress intensity.

11.214. A beam is fixed at one end and freely supported at the other end. A load acts in the centre. The maximum bending moment will occur at
(*a*) under the load
(*b*) fixed end (*c*) supported end
(*d*) between centre and fixed support
(*e*) between centre and free end.

11.215. Strength of a beam is directly proportional to its

(*a*) length (*b*) depth
(*c*) width
(*d*) moment of inertia
(*e*) all of the above.

11.216. Strength of a beam is proportional to the square of its
(*a*) length (*b*) depth
(*c*) width
(*d*) moment of inertia
(*e*) none of the above.

11.217. Section modulus of a rectangular beam is equal to
(*a*) $\frac{\text{Width }(b) \times (\text{depth})\,(h)^2}{6}$
(*b*) $\frac{hb^2}{6}$ (*c*) $\frac{h^2b^2}{6}$
(*d*) $\frac{bh^2}{12}$ (*e*) $\frac{hb^2}{12}$.

11.218. Section modulus of solid circular rod of diameter *d* is equal to
(*a*) $\frac{d^2}{10}$ (*b*) $\frac{d^3}{10}$
(*c*) $\frac{d^4}{10}$ (*d*) $\frac{d^3}{20}$
(*e*) $\frac{d^3}{5}$

11.219. Section modulus of hollow circle with average diameter '*d*' and wall thickness '*t*' is equal to
(*a*) $\frac{4}{5}td^2$ (*b*) $\frac{4}{5}t^2d^2$
(*c*) $\frac{4}{5}dt^2$ (*d*) $\frac{5}{4}td^2$
(*e*) $\frac{5}{4}d^2t$.

11.220. Moment of a beam is defined as its section modulus multiplied by
(*a*) moment of inertia
(*b*) stress (*c*) strain
(*d*) coefficient of elasticity
(*e*) half the depth.

11.221. If a freely supported beam at its ends is loaded by a central concentrated load, then maximum moment is *M*. If the same weight be equally distributed over the beam, then its maximum moment will be
(*a*) *M* (*b*) *M*/2
(*c*) *M*/3 (*d*) *M*/4
(*e*) 2 *M*.

11.222. The point of contra-flexure occurs only in
(*a*) cantilever beams
(*b*) overhanging beams
(*c*) simply supported beams
(*d*) continuous beams
(*e*) uniform beams.

11.223. Maximum shearing stress planes are inclined at the following angle to the principal planes
(*a*) at 45° (*b*) at 90°
(*c*) at $22\frac{1}{2}$°
(*d*) depends on the magnitude of the loads
(*e*) none of the above.

11.224. The point of inflexion or contra-flexure is the point where
(*a*) bending moment diagram changes sign
(*b*) stress is minimum
(*c*) deflection changes sign
(*d*) shear force and bending moment cross each other
(*e*) bending moment is maximum.

11.225. The bending moment on a section is maximum where shearing force
(*a*) is maximum
(*b*) is minimum (*c*) is equal
(*d*) changes sign (*e*) is zero.

11.226. In a fixed beam carrying a central load in the middle, the bending moment will be zero at a distance of
(*a*) length of beam (*l*)
(*b*) 0 (*c*) *l*/2
(*d*) *l*/4 (*e*) *l*/3.

11.227. Fig. 11.8 shows a simply supported beam of length which carries a concentrated load *W* at *C*, distance '*a*' from support *A* and '*b*' from support *B*. The maximum deflection occurs at point

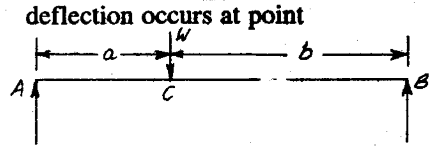

Fig. 11.8.

(*a*) *A* (*b*) *C*
(*c*) *B*

(*d*) between A and C
(*e*) between C and B.

11.228. In above case, the maximum deflection occurs at following distance from B
(*a*) $\frac{l}{3}$ (*b*) $\frac{l^2 - 2a^2}{3}$
(*c*) $\frac{l^2 - b^2}{3}$ (*d*) $\sqrt{\frac{l^2 - a^2}{3}}$
(*e*) $\sqrt{\frac{l^2 - b^2}{3}}$

11.229. In above case, the deflection at C is
(*a*) $\frac{Wl^3}{48EI}$ (*b*) $\frac{Wa^2b^2}{3EIl}$
(*c*) $\frac{Wa}{b\sqrt{3EIl}}(l^2 - a^2)^{3/2}$
(*d*) $\frac{5Wl^3}{384EI}$ (*e*) $\frac{W\sqrt{a^3b^3}}{48\,EI}$

11.230. The rate of change of bending moment is equal to
(*a*) shear force at that section
(*b*) deflection at that section
(*c*) loading at that section
(*d*) slope at that section
(*e*) intensity of loading at that section.

11.231. For a beam of uniform strength, if its depth is kept constant, then its width will vary in proportion to
(*a*) bending moment (M)
(*b*) $\sqrt{M}$ (*c*) M^2
(*d*) $1/M$ (*e*) none of the above.

11.232. A beam is said to be of uniform strength, if
(*a*) B.M. is same throughout the beam
(*b*) shear stress is same throughout the beam
(*c*) deflection is same throughout the beam
(*d*) bending stress is same at every section along its longitudinal axis
(*e*) none of the above.

11.233. In a continuous curve of bending moment, the point of zero bending moment, where it changes sign is called
(*a*) the point of inflexion
(*b*) the point of contra-flexure
(*c*) the point of a virtual hinge
(*d*) all of the above
(*e*) none of the above.

11.234. A reinforced cement concrete beam is said to be made of
(*a*) homogeneous material
(*b*) heterogeneous material
(*c*) isotropic material
(*d*) all of the above
(*e*) none of the above.

11.235. The steel bars in reinforced cement concrete beam are located at
(*a*) top (*b*) bottom
(*c*) centre (*d*) neutral axis
(*e*) uniformly distributed.

11.236. When a rectangular beam is loaded longitudinally, shear develops on
(*a*) every horizontal plane
(*b*) every vertical plane
(*c*) top fibre
(*d*) bottom fibre
(*e*) middle fibre.

11.237. If a beam of constant section is subjected throughout its length to a uniform bending moment, it will bend to
(*a*) a circular arc (*b*) a parabolic arc
(*c*) a catenary (*d*) elliptical shape
(*e*) none of the above.

11.238. The shear force diagram for a cantilever beam carrying a uniformly distributed load over its length is a
(*a*) triangle (*b*) rectangle
(*c*) hyperbola (*d*) parabola
(*e*) cubic parabola.

11.239. The rate of change of shear force at any section is equal to
(*a*) bending moment
(*b*) loading (*c*) deflection
(*d*) intensity of loading
(*e*) slope.

11.240. The moment diagram for a cantilever beam whose free end is subjected to a bending moment will be
(*a*) triangle (*b*) rectangle
(*c*) parabola (*d*) cubic parabola
(*e*) circular arc.

11.241. Two beams have same width but one beam has double the depth of the other. The elastic strength of double depth beam compared to other beam will be
(*a*) double (*b*) four times

(c) six times (d) eight times
(e) none of the above.

11.242. The ratio of deflection of two identical beams subjected to a central load W, but one fixed at both ends and other supported freely will be
(a) 1/2 (b) 1/3
(c) 1/4 (d) 1/8
(e) 1/16.

11.243. Two beams have same depth but one beam has double the width of the other. The elastic strength of double width beam compared to other beam will be
(a) same (b) half
(c) double (d) one-fourth
(e) none of the above.

11.244. Two beams have same width and depth. The span of one is twice the span of other. The elastic strength of longer span beam compared to other beam will be
(a) same (b) half
(c) double (d) one-fourth
(e) none of the above.

11.245. The moment diagram for a cantilever beam carrying concentrated load at end of the beam will be
(a) rectangle (b) triangle
(c) parabola (d) cubic parabola
(e) elliptical.

11.246. Beams of uniform strength so vary in section that the
(a) bending moment remains constant
(b) deflection remains constant
(c) unit stress remains constant
(d) all of the above remain constant
(e) any of the above remains constant.

11.247. For a cantilever beam of uniform width in plan and loaded by a concentrated load at the end, the profile of the shape of the beam in elevation, in order that beam is of uniform strength, should be
(a) uniform depth
(b) triangular
(c) parabola
(d) cubic parabola
(e) none of the above.

11.248. For a cantilever beam of uniform depth in elevation and loaded by concentrated load at the end, the profile of the shape of the beam in plan in order that beam is of uniform strength, should be
(a) uniform width
(b) triangular (c) parabola
(d) cubic parabola
(e) none of the above.

11.249. Point of contraflexure occurs in
(a) simply supported beam
(b) beams carrying load varying from zero at one end to maximum at other
(c) cantilevers
(d) overhanging beams
(e) any type of beam.

11.250. The bending moment diagram will be a cubic parabola in the case of a cantilever loaded as follows
(a) bending moment applied at free end
(b) concentrated load at the end
(c) uniformly distributed load
(d) varying load, zero at free end and maximum at fixed end
(e) uniformly distributed load and concentrated load at free end.

11.251. The moment diagram for a cantilever beam carrying linearly varying load from zero at free end to maximum at supported end will be
(a) rectangle (b) triangle
(c) parabola (d) cubic parabola
(e) elliptical.

11.252. In the case of cantilever, irrespective of the type of loading, the maximum bending moment and maximum shear force occur at
(a) free end (b) under the load
(c) fixed end (d) middle
(e) any point.

11.253. When a rectangular beam is loaded longitudinally, shear develops on
(a) top fibre (b) middle fibre
(c) bottom fibre
(d) every horizontal plane
(e) vertical fibre.

11.254. When shear force along a section is zero
(a) B.M. is maximum or minimum
(b) B.M. is zero
(c) B.M. is in between maximum and minimum value
(d) B.M. is infinity

(e) B.M. is unpredictable.

11.255. When a rectangular beam is loaded transversely, the maximum compressive stress develops on
(a) bottom fibre (b) top fibre
(c) neutral axis
(d) every cross-section
(e) middle fibre.

11.256. Coefficient of cubical expansion is
(a) equal to the coefficient of linear expansion
(b) twice the coefficient of linear expansion
(c) thrice the coefficient of superficial expansion
(d) 1.5 times the coefficient of superficial expansion
(e) 0.75 times the coefficient of superficial expansion.

11.257. Hoop stress in thin walled cylinder is
(a) compressive stress
(b) radial stress
(c) circumferential tensile stress
(d) longitudinal stress
(e) shear stress.

11.258. When a strip made of iron and copper is heated
(a) it does not bend
(b) it gets twisted
(c) it bends with iron on concave side
(d) it bends with iron on convex side
(e) none of the above.

11.259. Modulus of resilience is
(a) property to resist shocks
(b) the property to store energy without undergoing permanent deformation
(c) an index of elasticity
(d) an index of compressibility
(e) property to withstand heavy pressure.

11.260. During the tensile test of a glass rod the nature of stress-strain curve is
(a) straight and drooping
(b) sudden break (c) straight line
(d) parabolic (e) none of the above.

11.261. During impact loading the stress developed as compared to gradually applied load is
(a) 1.5 times (b) 2 times
(c) 2.5 times (d) 3 times
(e) 4 times.

11.262. In a laminated spring the strips are provided in different lengths for
(a) equal distribution of stress
(b) better look
(c) light in weight
(d) ease in installing
(e) none of the above.

11.263. Ratio of thickness to diameter for thin cylinder is
(a) 1/10 (b) 1/15
(c) 1/20 (d) 1/40
(e) 1/30.

11.264. It is be assumed that the longitudinal strain is constant at any point in the thickness of the cylinder, then radial stress s_r and hoop stress s_h are related as follows
(a) $s_r - s_h = \text{constant}$
(b) $s_r + s_h' = \text{constant}$
(c) $\frac{s_r}{s_h} = \text{constant}$ (d) $\frac{s_r - s_h}{s_r} = \text{constant}$
(e) $\frac{s_r - s_h}{s_h} = \text{constant}$.

11.265. The hoop stress of a thick cylinder is considered at *i.e.* it is maximum at
(a) near centre (b) outer radius
(c) inner radius
(d) depends on the type of loading
(e) none of the above.

11.266. Hoop stress in a thin cylinder of diameter d and thickness t subjected to pressure p will be
(a) $\frac{pd}{4t}$ (b) $\frac{pd}{t}$
(c) $\frac{pd}{2t}$ (d) $\frac{2pd}{t}$
(e) $\frac{4pd}{t}$

11.267. In a thin cylinder, all along the thickness of the cylinder
(a) hoop stress and longitudinal stress are almost constant
(b) hoop stress and longitudinal stress are equal
(c) hoop stress is constant and longitudinal stress varies considerably
(d) longitudinal stress is constant and hoop stress varies considerably

(*e*) both hoop stress and longitudinal stress vary considerably.

11.268. In a thick cylinder along the thickness of the cylinder
(*a*) hoop stress and longitudinal stress are almost constant
(*b*) hoop stress and longitudinal stress are equal
(*c*) hoop stress is constant and longitudinal stress varies considerably
(*d*) longitudinal stress is constant and hoop stress varies considerably
(*e*) both hoop and longitudinal stresses vary considerably.

11.269. The stresses at any point in the thick cylinder are
(*a*) tensile (*b*) compressive
(*c*) shear (*d*) compound
(*e*) principal.

11.270. According to Lame's equation, hoop stress for a thick cylinder at any point at a radius *r* from centre is equal to
(*a*) $\frac{b}{r^2}+a$ (*b*) $\frac{b}{r^2}-a$
(*c*) $\frac{b}{r}+a$ (*d*) $\frac{b}{r}-a$
(*e*) $\frac{b}{r^3}+a$.

11.271. In thick cylinders, the stress can be uniformly distributed over the thickness by the method of pre-stressing as follows
(*a*) self-hooping
(*b*) constructing laminated cylinders
(*c*) shrinking hollow cylinder over main cylinder
(*d*) any one of the above
(*e*) none of the above.

11.272. The longitudinal stress in a thick cylinder of diameter *d*, thickness *t* and subjected to pressure *p* will be
(*a*) $\frac{pd}{2t}$ (*b*) $\frac{pd}{4t}$
(*c*) $\frac{pd}{t}$ (*d*) $\frac{2pd}{t}$
(*e*) none of the above.

11.273. The radial pressure and hoop tension in case of thick cylinder is
(*a*) maximum at inner surface and decreases towards outer surface
(*b*) minimum at inner surface and increases towards outer surface
(*c*) minimum at inner and outer surfaces and maximum in middle
(*d*) maximum at inner and outer surfaces and minimum in middle
(*e*) none of the above.

11.274. Thin cylindrical shell of diameter *d* and thickness *t* is subjected to internal pressure *p*. Poisson's ratio of material is μ. The circumferential or hoop strain is
(*a*) $\frac{pd}{2tE}(1-2\mu)$ (*b*) $\frac{pd}{4tE}(1-2\mu)$
(*c*) $\frac{pd}{2tE}\left(\frac{1}{2}-\mu\right)$ (*d*) $\frac{pd}{4tE}\left(\frac{1}{2}-\mu\right)$
(*e*) $\frac{pd}{4tE}\left(1-\frac{1}{2}\mu\right)$

11.275. In above case the longitudinal strain is
(*a*) $\frac{pd}{2tE}(1-2\mu)$ (*b*) $\frac{pd}{4tE}(1-2\mu)$
(*c*) $\frac{pd}{2tE}\left(\frac{1}{2}-\mu\right)$ (*d*) $\frac{pd}{4tE}\left(\frac{1}{2}-\mu\right)$
(*e*) $\frac{pd}{4tE}\left(1-\frac{1}{2}\mu\right)$

11.276. If the hoop strain and longitudinal strain in case of a thin cylindrical shell are e_n and e_l, then volumetric strain is equal to
(*a*) e_n+e_l (*b*) $2e_n+2e_l$
(*c*) e_n+2e_l (*d*) $2e_n-e_l$
(*e*) $2e_l-e_n$.

11.277. The design of thin cylindrical shells is based on
(*a*) hoop stress
(*b*) longitudinal stress
(*c*) volumetric stress
(*d*) average of hoop and longitudinal stress
(*e*) all of the above.

11.278. The volumetric strain in case of cylindrical shell of diameter '*d*' and thickness '*t*', subjected to internal pressure *p*, having coefficient of elasticity *E* and Poisson's ratio μ is equal to
(*a*) $\frac{pd}{tE}(2-\mu)$ (*b*) $\frac{pd}{2tE}(3-2\mu)$
(*c*) $\frac{pd}{3tE}(4-3\mu)$ (*d*) $\frac{pd}{4tE}(5-4\mu)$

(e) $\frac{pd}{4tE}(4-5\mu)$.

11.279. In above case, the ratio of longitudinal to volumetric strain is
(a) $\frac{1-2\mu}{5-4\mu}$ (b) $\frac{2-\mu}{5-4\mu}$
(c) $\frac{1-2\mu}{4-5\mu}$ (d) $\frac{1-2\mu}{3-4\mu}$
(e) $\frac{3-4\mu}{1-2\mu}$

11.280. In above case the ratio of longitudinal strain to hoop strain is
(a) $\frac{1-2\mu}{2-\mu}$ (b) $\frac{2-\mu}{1-2\mu}$
(c) $\frac{1-2\mu}{1-\mu}$ (d) $\frac{2\mu+1}{1-2\mu}$
(e) $\frac{1+2\mu}{2+\mu}$

11.281. Auto frettage is the method of
(a) calculating stresses in thick cylinders
(b) relieving thick cylinders
(c) prestressing thick cylinders
(d) increasing life of thick cylinders
(e) joining thick cylinders.

11.282. Proof load for springs is the maximum load that it can undertake
(a) without producing permanent deformation in spring material
(b) upto elastic limit
(c) upto yield point
(d) to straighten fully the leafs of a carriage spring
(e) to satisfy fatigue requirements.

11.283. If all the strips of a leaf spring are made of the same length then it would result in
(a) uniform distribution of shearing force
(b) uniform deflection
(c) maximum capability of taking maximum bending moment
(d) uneconomical use of spring material
(e) spring of uniform strength.

11.284. A composite shaft, consisting of two stepped portions having spring constants of k_1 and k_2, is held between two rigid supports at ends. Its equivalent spring constant is
(a) $\frac{k_1+k_2}{2}$ (b) $\frac{k_1k_2}{k_1+k_2}$
(c) k_1+k_2 (d) $\frac{k_1+k_2}{k_1k_2}$
(e) none of the above.

11.285. A composite shaft, consisting of two stepped portions having spring constants of k_1 and k_2, is held firmly at one end and other end is free and subjected to a torque T. Its equivalent spring constant is
(a) $\frac{k_1+k_2}{2}$ (b) $\frac{k_1k_2}{k_1+k_2}$
(c) k_1+k_2 (d) $\frac{k_1+k_2}{k_1k_2}$
(e) none of the above.

11.286. If a shaft of radius r and polar moment of inertia J be subjected to bending moment M and torque T, then maximum combined shear and bending stress is equal to
(a) $\frac{r}{J}\sqrt{M^2+T^2}$ (b) $\frac{J}{r}\sqrt{M^2+T^2}$
(c) $\frac{2r}{J}\sqrt{M^2+T^2}$ (d) $\frac{r}{J}\left[\frac{M}{2}+\sqrt{M^2+T^2}\right]$
(e) $\frac{r}{J}\frac{\sqrt{M^2+T^2}}{2}$.

11.287. A torsion bar with a spring constant k is cut into n equal lengths. What is the spring constant of each portion
(a) k/n (b) $\sqrt[n]{k}$
(c) k^n (d) nk
(e) n^k.

11.288. Two identical springs of spring constant k in series are attached in series with a parallel combination of two identical springs of spring constant k. The overall equivalent spring constant is
(a) 2.5 k (b) 1.25 k
(c) 0.4 k (d) 0.8 k
(e) 0.2 k.

11.289. Two identical leaf springs of spring constant k are arranged like cantilevers in parallel and attached at free end by a spring of spring constant k. The equivalent spring constant of combination is
(a) 2.5 k (b) 1.5 k
(c) 0.4 k (d) 0.75 k
(e) 3 k.

11.290. An overhanging beam is

(*a*) same as cantilever
(*b*) not same as cantilever
(*c*) one which extends beyonds its support at either end
(*d*) one which extends beyond its support at both ends
(*e*) none of the above.

11.291. Neutral plane of a beam
(*a*) is in the middle
(*b*) passes through the c.g.
(*c*) is one whose length remains unchanged during the deformation
(*d*) lies at bottom most fibre
(*e*) none of the above.

11.292. Section modulus is defined as
(*a*) $\frac{W}{gy}k^2$ (*b*) $\frac{p}{\delta V/V}$
(*c*) $\frac{\text{M.I}}{y}$ (*d*) $\Sigma y \Delta A$
(*e*) $\frac{\text{M.I}}{y^2}$

11.293. If the thickness and width of each plate of a laminated spring be *t* and *w* respectively, then its moment of inertia is equal to
(*a*) $\frac{wt^3}{12}$ (*b*) $\frac{tw^3}{12}$
(*c*) $\frac{wt^2}{12}$ (*d*) $\frac{tw^2}{12}$
(*e*) $\frac{wt^4}{12}$.

11.294. The load taken by a laminated spring to make it flat is called
(*a*) ultimate load
(*b*) proof load
(*c*) bending load
(*d*) maximum safe load
(*e*) yielding load.

11.295. Beam of uniform strength can be obtained by
(*a*) keeping depth constant throughout the length and varying the width suitably
(*b*) keeping the width constant throughout the length and varying the depth suitably
(*c*) varying both the depth and width suitably
(*d*) any one of the above
(*e*) none of the above.

11.296. When a beam is loaded, the horizontal or longitudinal shear should be accounted for materials like
(*a*) mild steel (*b*) concrete
(*c*) cast iron (*d*) wood
(*e*) lead.

11.297. According to Hooke's law, stress and strain
(*a*) are directly proportional
(*b*) are inversely proportional
(*c*) are curvilinearly related
(*d*) have unpredictable relationship
(*e*) none of the above.

11.298. Radius of curvature of a stressed beam and modulus of elasticity
(*a*) are directly proportional
(*b*) are inversely proportional
(*c*) are curvilinearly related
(*d*) have unpredictable relationship
(*e*) none of the above.

11.299. Stress in a beam and the second modulus
(*a*) are directly proportional
(*b*) are inversely proportional
(*c*) are curvilinearly related
(*d*) have unpredictable relationship
(*e*) none of the above.

11.300. Stress in a beam and moment
(*a*) are directly proportional
(*b*) are inversely proportional
(*c*) have curvilinearly related
(*d*) have unpredictable relationship
(*e*) none of the above.

11.301. In an unsymmetrical beam, the maximum compressive stress at top was measured as 1200 kg/cm^2 and the maximum tensile stress at bottom was 300 kg/cm^2. If the beam is 3 cm deep, the neutral axis from top will be at
(*a*) 6 cm (*b*) 4 cm
(*c*) 2 cm (*d*) 6.4 cm
(*e*) 3.2 cm.

11.302. The steel bars in a concrete beam are embedded
(*a*) uniformly (*b*) near bottom section
(*c*) in the centre (*d*) near top section
(*e*) anywhere.

11.303. The point on a beam where shearing force changes sign is known as the point of
(*a*) zero shear (*b*) uniform strength

(c) no load
(d) neutrality
(e) none of the above.

11.304. The point of contraflexure is a point where
(a) shear force in zero
(b) shear force changes sign
(c) bending moment changes sign
(d) bending moment is maximum
(e) beam is liable to break.

11.305. A cantilever beam is deflected by d due to load P. If load is doubled then deflection compared to earlier case will be changed by a factor of
(a) 2 (b) $\frac{1}{2}$
(c) $\frac{1}{8}$ (d) 8
(e) 4.

11.306. The maximum shear stress in Mohr's circle will act at following angle to the principal plane
(a) 0° (b) 30°
(c) 45° (d) 60°
(e) none of the above.

11.307. A cantilever beam is deflected by d due to load P. If beam depth is doubled, then deflection compared to earlier case will be changed by a factor of
(a) 2 (b) $\frac{1}{2}$
(c) $\frac{1}{8}$ (d) 8
(e) 4.

11.308. A cantilever beam is deflected by d due to load P. If beam width is doubled, then deflection compared to earlier case will be changed by a factor of
(a) 2 (b) $\frac{1}{2}$
(c) $\frac{1}{8}$ (d) 8
(e) 4.

11.309. A cantilever beam is deflected by d due to load P. If length of beam is doubled, then deflection compared to earlier case will be changed by a factor of
(a) 2 (b) $\frac{1}{2}$
(c) $\frac{1}{8}$ (d) 8
(e) 4.

11.310. Moment of intertia of a square of side d about the diagonal is
(a) $d^4/12$ (b) $d^4/24$
(c) $d^4/18$ (d) $d^4/8$
(e) $d^4/16$.

11.311. Principal plane is one which carries
(a) no shear stress
(b) maximum shear stress
(c) no normal stress
(d) maximum resultant of stresses
(e) no resultant of stresses.

11.312. For biaxial stress the maximum shear stress occurs on plane inclined at following angle to the principal normal plane
(a) 90° (b) 45°
(c) 145° (d) 135°
(e) none of the above.

11.313. On the planes having maximum or minimum principal stresses, the tangential stress will be
(a) minimum (b) maximum
(c) zero (d) infinity
(e) some value depending on magnitude of two stresses.

11.314. Maximum shear stress in a Mohr's circle is
(a) equal to radius of Mohr's circle
(b) greater than radius of Mohr's circle
(c) less than the radius of Mohr's circle
(d) could be any of the above
(e) none of the above.

11.315. The normal stress on an oblique section of a body subjected to a direct stress σ in one plane, if θ be the inclination of the oblique section with normal to the stress, is equal to
(a) $\sigma \cos\theta$ (b) $\sigma^2 \cos\theta$
(c) $\frac{\sigma}{2}\cos\theta$ (d) $\sigma \cos 2\theta$
(e) $\sigma \cos^2\theta$.

11.316. If a principal plane be subjected to maximum principal stress, then shear stress on this plane will be
(a) zero (b) maximum
(c) minimum (d) infinity

(*e*) any value.

11.317. In Mohr's circle radius is taken as

(*a*) $\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2}$

(*b*) $\frac{\sigma_x - \sigma_y}{2}$ (*c*) $\sqrt{(\sigma_x - \sigma_y)^2 + (\tau_{xy})^2}$

(*d*) $\sqrt{\frac{(\sigma_x - \sigma_y)^2}{2} + (\tau_{xy})^2}$

(*e*) $\sqrt{\frac{(\sigma_x - \sigma_y^2)}{2} - (\tau_{xy})^2}$.

11.318. If a prismatic bar is subjected to direct tensile stresses σ_x and σ_y on two perpendicular faces and shear stress τ_{xy} then stress normal to plane inclined at θ to vertical is

(*a*) $\frac{\sigma_x + \sigma_y}{2} + \frac{\sigma_x - \sigma_y}{2}\cos 2\theta - \tau_{xy}\sin 2\theta$

(*b*) $\frac{\sigma_x - \sigma_y}{2}\sin 2\theta + \tau_{xy}\cos 2\theta$

(*c*) $\frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

(*d*) $\frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

(*e*) none of the above.

11.319. The stress along the plane in above case is

(*a*) $\frac{\sigma_x + \sigma_y}{2} + \frac{\sigma_x - \sigma_y}{2}\cos 2\theta - \tau_{xy}\sin 2\theta$

(*b*) $\frac{\sigma_x - \sigma_y}{2}\sin 2\theta + \tau_{xy}\cos 2\theta$

(*c*) $\frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

(*d*) $\frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

(*e*) none of the above.

11.320. The maximum principal stress in above case is

(*a*) $\frac{\sigma_x + \sigma_y}{2} + \frac{\sigma_x - \sigma_y}{2}\cos 2\theta - \tau_{xy}\sin 2\theta$

(*b*) $\frac{\sigma_x - \sigma_y}{2}\sin 2\theta + \tau_{xy}\cos 2\theta$

(*c*) $\frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

(*d*) $\frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

(*e*) none of the above.

11.321. The minimum principal stress in above case is

(*a*) $\frac{\sigma_x + \sigma_y}{2} + \frac{\sigma_x - \sigma_y}{2}\sin 2\theta - \tau_{xy}\cos 2\theta$

(*b*) $\frac{\sigma_x - \sigma_y}{2}\sin 2\theta + \tau_{xy}\cos 2\theta$

(*c*) $\frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

(*d*) $\frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

(*e*) none of the above.

11.322. The principal stresses in above case occur at angle of

(*a*) $\tan^{-1}\left(-\frac{2\tau_{xy}}{\sigma_x - \sigma_y}\right)$

(*b*) $\frac{1}{2}\tan^{-1}\left(-\frac{2\tau_{xy}}{\sigma_x - \sigma_y}\right)$

(*c*) $\frac{1}{2}\tan^{-1}\left(\frac{\sigma_x - \sigma_y}{2\tau_{xy}}\right)$

(*d*) $\frac{1}{2}\tan^{-1}\left(-\frac{2\tau_{xy}}{\sigma_x + \sigma_y}\right)$

(*e*) $\frac{1}{2}\tan^{-1}\left(-\frac{\tau_{xy}}{\sigma_x - \sigma_y}\right)$.

11.323. In above case, planes of maximum shear are located at following angle to each other

(*a*) 90° (*b*) 30°
(*c*) 45° (*d*) 60°
(*e*) 0°.

11.324. In above case, the planes of maximum shear are inclined at following angle to the principal planes

(*a*) 0° (*b*) 30°
(*c*) 45° (*d*) 60°
(*e*) 90°.

11.325. Maximum shear stress in Mohr's circle is equal to

(*a*) radius of circle
(*b*) diameter of circle
(*c*) centre of circle from *y*-axis
(*d*) chord of circle

(*e*) none of the above.

11.326. Maximum shear stress in a body subjected to two perpendicular stresses σ_x, σ_y and shear stress τ_{xy} is equal to

(*a*) $\sqrt{\left(\frac{\sigma_x + \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

(*b*) $\frac{\sigma_x - \sigma_y}{2}$

(*c*) $\sqrt{(\sigma_x - \sigma_y)^2 + (\tau_{xx})^2}$

(*d*) $\sqrt{\frac{(\sigma_x - \sigma_y)^2}{2} + (\tau_{xy})^2}$

(*e*) $\sqrt{\frac{(\sigma_x - \sigma_y)^2}{2} - (\tau_{xy})^2}$

11.327. If σ_1 and σ_2 be the major and minor tensile stresses, then maximum value of tangential stress is equal to

(*a*) σ_1 (*b*) σ_2

(*c*) $\sigma_1 - \sigma_2$ (*d*) $\sigma_1 + \sigma_2$

(*e*) $\frac{\sigma_1 - \sigma_2}{2}$.

11.328. If a prismatic bar be subjected to an axial tensile stress σ, then the shear stress induced on a plane inclined at angle θ with the axis will be

(*a*) $\frac{\sigma}{2}\sin^2\theta$ (*b*) $\frac{\sigma}{2}\sin 2\theta$

(*c*) $\sigma \sin^2\theta$ (*d*) $\sigma \cos^2\theta$

(*e*) $\frac{\sigma}{2}\cos^2\theta$.

11.329. In Mohr's circle, centre of circle from y-axis is taken at

(*a*) $\sigma_x + \sigma_y$ (*b*) $\frac{\sigma_x + \sigma_y}{2} + \tau_{xy}$

(*c*) $\frac{\sigma_x + \sigma_y}{2}$ (*d*) $\frac{\sigma_x - \sigma_y}{2}$

(*e*) $\frac{\sigma_x + \sigma_y}{4}$.

11.330. Shear stresses on mutually perpendicular planes are

(*a*) zero (*b*) minimum

(*c*) maximum (*d*) equal

(*e*) none of the above.

11.331. If one principal stress be zero at a point, then other principal stress compared to the maximum shear stress at that point will be

(*a*) same (*b*) twice

(*c*) $1\frac{1}{2}$ (*d*) $2\frac{1}{2}$

(*e*) half.

11.332. The shear stress along the principal plane subjected to maximum principal stress is

(*a*) minimum (*b*) maximum

(*c*) zero

(*d*) any value depending on loading conditions

(*e*) negative.

11.333. Mohr's circle is used to determine the stresses on an oblique section of a body subjected to

(*a*) direct stresses in two mutually perpendicular directions

(*b*) a direct stress in one plane alongwith simple shear stress

(*c*) direct stresses in two mutually perpendicular directions accompanied by a simple shear stress

(*d*) all of the above

(*e*) none of the above.

11.334. The planes of minimum shear stress with reference to principal planes are located at

(*a*) 0° (*b*) $22\frac{1}{2}°$

(*c*) 45° (*d*) 90°

(*e*) 135°.

11.335. The normal stress on the planes of minimum stress in the cases of plane stress problems will be

(*a*) zero (*b*) maximum

(*c*) half the difference of two mutually perpendicular stresses

(*d*) principal stress

(*e*) none of the above.

11.336. The extremities of any diameter on Mohr's circle represent

(*a*) principal stresses

(*b*) normal stresses on planes at 45°

(*c*) shear stresses on planes at 45°

(*d*) normal and shear stresses on a plane

(*e*) none of the above.

11.337. The normal stresses on planes of maximum shear stresses in the case of plane stress problems will be

(a) $\frac{\sigma_1+\sigma_2}{2}, \frac{\sigma_1-\sigma_2}{2}$

(b) $\frac{\sigma_1}{2}, \frac{\sigma_2}{2}$ (c) zero

(d) σ_1, σ_2 (e) principal stresses.

11.338. In case of pure shear at a point, the sum of normal stresses on two orthogonal planes is equal to
(a) maximum shear stress
(b) twice the maximum shear stress
(c) half the maximum shear stress
(d) zero
(e) none of the above.

11.339. The principal stress in the case of pure shear will be equal in magnitude
(a) and similar in nature
(b) but opposite in nature
(c) to half the maximum shear stress and similar in nature
(d) to half the maximum shear stress but opposite in nature
(e) zero.

11.340. Ten cubical blocks of size 25 cm × 25 cm × 25 cm are held together by a horizontal force of 50 kg to form a beam. What is the greatest force P that could act vertically at the mid span. Weight of bricks may be neglected.
(a) 50 kg (b) 10 kg
(c) 5 kg (d) 3.34 kg
(e) 6.68 kg.

11.341. Two loads P act at right angles to one another at the free end of a cantilever beam having square cross-section $d \times d$ and length l on the vertical and horizontal faces. Maximum bending stress in the beam will be equal to
(a) $12\, Pl/d^3$ (b) $24\, Pl/d^3$
(c) $6\, Pl/d^3$ (d) $18\, Pl/d^3$
(e) $3\, Pl/d^3$.

11.342. In a fillet weld, the maximum load that can be applied is equal to
(a) permissible shearing stress $(s_s) \times$ fillet size $\times$ fillet length (L)
(b) $0.707\, s_s \times$ fillet size $\times L$
(c) $\frac{s_s \times \text{fillet size} \times L}{2}$
(d) $2 \times s_s \times$ fillet size $\times L$
(e) none of the above.

11.343. For eccentrically loaded struts
(a) solid members are preferred
(b) hollow members are preferred
(c) both are equally good
(d) reinforced members are preferred
(e) none of the above.

11.344. In eccentrically loaded members, a neutral axis can be drawn by joining the points which represent the greatest permissible eccentricity in each coordinate direction. By symmetry, four neutral axes exist in the section : these lines form the boundaries of a core known as kern. Compressive forces acting within the kern of a section
(a) cannot produce tensile stress
(b) will sometimes produce tensile stresses
(c) will always produce tensile stresses
(d) are zero
(e) none of the above.

11.345. The ratio of maximum shear stress and average stress in a circular section is
(a) 1 (b) 1.25
(c) 1.5 (d) 4/3
(e) 2.

11.346. The stress at the boundary of the kern is
(a) maximum (b) minimum
(c) zero (d) any value
(e) none of the above.

11.347. Kern of a circular section of diameter D is a concentric circular area having a diameter of
(a) $D/4$ (b) $D/2$
(c) $D/8$ (d) $3\,D/8$
(e) none of the above.

11.348. The minimum limiting load at which the column tends to have lateral displacement is known as
(a) critical load (b) crippling load
(c) buckling load (d) any of the above
(e) none of the above.

11.349. A tower is to be designed to nave same compressive stress at all sections under a load and its own weight; its cross section should be of shape
(a) uniform (b) taper
(c) parabolic (d) hyperbolic

(*e*) none of the above.

11.350. The slenderness ratio for long columns is
(*a*) less than 32 (*b*) 50–60
(*c*) 80–100 (*d*) less than 120
(*e*) more than 120.

11.351. For keeping the stress wholly compressive, the load may be applied on a circular column of diameter '*d*' anywhere within a concentric circle of following diameter
(*a*) $d/2$ (*b*) $d/3$
(*c*) $d/4$ (*d*) $d/8$
(*e*) $d/10$.

11.352. The slenderness ratio of a vertical column of square cross-section of 2.5 cm on edge and 3 m long is
(*a*) 120 (*b*) 240
(*c*) 416 (*d*) 550
(*e*) none of the above.

11.353. If the slenderness ratio of a column is less than 32, it is known as
(*a*) short column (*b*) medium column
(*c*) long column
(*d*) extra long column
(*e*) none of the above.

11.354. The equivalent length of a column supported firmly at both ends is
(*a*) $2l$ (*b*) $0.7l$
(*c*) l (*d*) $0.5l$
(*e*) none of the above.

11.355. The equivalent length of a column supported firmly at one-end and free at other end is
(*a*) $2l$ (*b*) $0.7l$
(*c*) l (*d*) $0.5l$
(*e*) none of the above.

11.356. A cantilever carrying a uniformly distributed load '*w*' kg per unit length is propped at its free end. The reaction at the prop will be
(*a*) wl (*b*) $\frac{wl}{2}$
(*c*) $\frac{wl}{8}$ (*d*) $\frac{3wl}{8}$
(*e*) $\frac{5wl}{8}$.

11.357. In the above problem if prop is installed in the middle then reaction at the prop will be
(*a*) wl (*b*) $\frac{wl}{2}$
(*c*) $\frac{wl}{8}$ (*d*) $\frac{3wl}{8}$
(*e*) $\frac{5wl}{8}$.

11.358. A certain high tensile strength steel has a modulus of elasticity of 2×10^6 kg/cm^2 and a yield point stress of 6,000 kg/cm^2. Find the minimum limiting value of the slenderness ratio for which Euler's equation is valid
(*a*) 99 (*b*) 80
(*c*) 75 (*d*) 57
(*e*) none of the above.

11.359. The extension of a uniform bar produced by its own weight as compared to that produced by the load equal to self weight of bar, applied at free end is
(*a*) same (*b*) half
(*c*) one-fourth (*d*) double
(*e*) four times.

11.360. Three beams have the same length, the same allowable stress and the same B.M. The cross-section of the beams are a circle, a square, and a rectangle with depth twice the width. Weightwise best section in order of merit will be
(*a*) circle, square, rectangle
(*b*) rectangle, square, circle
(*c*) square, circle, rectangle
(*d*) rectangle, circle, square
(*e*) square, rectangle, circle.

11.361. A beam of uniform strength is one in which
(*a*) B.M. is same throughout the beam
(*b*) deflection is same throughout the length
(*c*) the bending stress is same in every section along the longitudinal axis
(*d*) shear stress is uniform throughout the beam
(*e*) none of the above.

11.362. In a beam of uniform strength, if depth is kept constant, then its width varies in proportion to the
(*a*) Bending moment (M)
(*b*) M^2 (*c*) $\sqrt{M}$
(*d*) $1/M$ (*e*) none of the above.

11.363. Strain rosettes are used to
(*a*) produce strains for testing purpose
(*b*) relieve strain in heavily loaded components
(*c*) measure strain
(*d*) analyse property of materials
(*e*) none of the above.

11.364. Damping capacity of a material is its ability to
(*a*) absorb shock
(*b*) absorb vibrations
(*c*) withstand compression
(*d*) absorb impact loads
(*e*) withstand creep failures.

11.365. Each section of a close coiled helical spring is subjected to
(*a*) tensile stress and shear stress due to load
(*b*) compressive stress and shear stress due to torque
(*c*) tensile and compressive stresses
(*d*) tensile stress and shear stress due to torque
(*e*) shear stress due to torque and direct shear due to load.

11.366. Disruptive strength is the maximum strength of a metal
(*a*) when subjected to 3 principal tensile stresses at righ angles to one another and all of equal magnitude
(*b*) when loaded in tension
(*c*) when loaded in compression
(*d*) when loaded in shear
(*e*) all of the above.

11.367. Dilatometer is an instrument used for measuring
(*a*) strain
(*b*) expansion contraction due to change in temperature
(*c*) stress
(*d*) damping capacity
(*e*) vibrations.

11.368. Flow stress is
(*a*) the shear stress required to cause plastic deformation of solid metal
(*b*) concerned with liquid
(*c*) corresponding to breaking point
(*d*) there is nothing like flow stress
(*e*) used in connection with acceptance tests of materials.

11.369. Proof stress
(*a*) is the safest stress
(*b*) is that which will cause a specified permanent deformation in a material, usually 0.01% or less
(*c*) is used in connection with materials like mild steel
(*d*) does not exist
(*e*) is used in connection with acceptance tests for materials.

11.370. Rupture stress is
(*a*) breaking stress
(*b*) the stress obtained by dividing the load at the moment of incipient fracture, by the area supporting that load
(*c*) never allowed to reach in members
(*d*) highest stress in a test
(*e*) none of the above.

11.371. Maximum deflection in a cantilever due to pure bending moment M at its end is
(*a*) $\frac{Ml^2}{2EI}$ (*b*) $\frac{Ml^2}{3EI}$
(*c*) $\frac{Ml^2}{4EI}$ (*d*) $\frac{Ml^2}{6EI}$
(*e*) $\frac{Ml^2}{8EI}$.

11.372. If A_1 and A_2 be areas of shear force diagram and bending moment diagram respectively over the portion of the beam, then the change of slope over any portion of a loaded beam is equal to
(*a*) $\frac{A_1}{EI}$ (*b*) $\frac{A_2}{EI}$
(*c*) $\frac{A_1 - A_2}{EI}$ (*d*) $\frac{A_1 + A_2}{EI}$
(*e*) $\frac{A_1 \times A_2}{EI}$.

11.373. Maximum deflection in a beam supported freely at both ends due to a central load P at middle is
(*a*) $\frac{Pl^3}{48EI}$ (*b*) $\frac{Pl^3}{32EI}$
(*c*) $\frac{Pl^3}{96EI}$ (*d*) $\frac{Pl^3}{64EI}$
(*e*) $\frac{Pl^3}{128EI}$.

11.374. Maximum deflection in a cantilever beam of length l carrying a load P at its end will be

(a) $\frac{Pl^3}{3EI}$ (b) $\frac{Pl^2}{8EI}$

(c) $\frac{Pl^3}{32EI}$ (d) $\frac{Pl^3}{64EI}$

(e) $\frac{Pl^3}{128EI}$.

11.375. Maximum slope in case of a cantilever of length l carrying a load P at its end will be

(a) $\frac{Pl^2}{EI}$ (b) $\frac{Pl^2}{2EI}$

(c) $\frac{Pl^2}{3EI}$ (d) $\frac{Pl^2}{4EI}$

(e) $\frac{Pl^2}{6EI}$.

11.376. Maximum deflection in case of a cantilever beam carrying uniformly distributed load w per unit length will be

(a) $\frac{wl^4}{EI}$ (b) $\frac{wl^4}{8EI}$

(c) $\frac{wl^4}{4EI}$ (d) $\frac{5}{64}\frac{wl^4}{EI}$

(e) $\frac{5}{384}\frac{wl^4}{EI}$.

11.377. Ties are load carrying members which carry
(a) torsional loads
(b) axial compressive loads
(c) axial tension loads
(d) transverse loads
(e) vertical compression loads.

11.378. For a given material, Young's modulus E = 200 GN/m^2 and modulus of rigidity G = 80 GN/m^2. Its bulk modulus K will be
(a) 100 GN/m^2 (b) 133.33 GN/m^2
(c) 200 GN/m^2 (d) 250 GN/m^2
(e) none of the above.

11.379. For above question, the value of Poission's ratio will be
(a) 0.15 (b) 0.20
(c) 0.25 (d) 0.30
(e) 0.35.

11.380. The extension of a circular tapering rod having diameters d_1 and d_2 at its two ends and subjected to axial pull P, as compared to a circular bar of diameter $\sqrt{d_1 d_2}$ and subjected to pull P will be
(a) more (b) less
(c) equal
(d) depends on other parameters
(e) none of the above.

11.381. The extension of uniform bar by its own weight as compared to a bar loaded by same weight at the free end will be
(a) same (b) half
(c) one-fourth (d) one-third
(e) three-fourth.

11.382. If the compressive stress in a tower due to a load and its own weight is to be constant at all sections, then its cross-section at various sections should be of
(a) uniform shape
(b) tapering shape
(c) parabolic shape
(d) hyperbolic shape
(e) some other shape.

11.383. If D be the diameter of coil of a close coiled helical spring and total angle of twist in full length be θ, then deflection of spring is equal to

(a) $D\theta$ (b) $\frac{D}{2}\theta$

(c) $2D\theta$ (d) $D\theta^2$

(e) $\frac{D^2\theta}{2}$.

11.384. Instantaneous stress produced by falling weight on a bar as compared to gradually applied load is
(a) two times
(b) between one and two times depending on height of fall
(c) more than two times
(d) more than ten times
(e) none of the above.

11.385. If the areas of cross-sections of square and circular beams are same and both are subjected to equal bending moment, then which of the following is correct
(a) the circular beam is more economical
(b) the square beam is more economical
(c) both the beams are equally strong
(d) both the beams are equally economical

(*e*) none of the above.

11.386. The ratio of central deflection due to a central load in the case of a beam freely supported at both ends to the beam fixed at both ends will be
(*a*) 1/2 (*b*) 2
(*c*) 1/4 (*d*) 4
(*e*) none of the above.

11.387. There are two bars of equal length and equal volume and same material, one having stepped diameters and other having uniform diameter. If maximum stress produced in both bars is same then stored energy will be
(*a*) more in stepped diameter shaft
(*b*) more in uniform diameter shaft
(*c*) equal in both
(*d*) would depend on other factors
(*e*) none of the above.

11.388. In above bars, if gradually applied load is same then the stored energy will be
(*a*) more in stepped diameter shaft
(*b*) more in uniform diameter shaft
(*c*) equal in both
(*d*) would depend on other factors
(*e*) none of the above.

11.389. A direct tensile stress on a specimen will induce the following shear stress on a plane inclined at 45° to its own plane
(*a*) same as tensile stress
(*b*) half of tensile stress
(*c*) zero
(*d*) two times the tensile stress
(*e*) depends on other parameters.

11.390. Sum of normal stresses on any two mutually perpendicular planes as compared to sum of stresses s_x and s_y is
(*a*) same
(*b*) more
(*c*) less
(*d*) depends on other conditions
(*e*) could be anything.

11.391. Three beams of circular, square, rectangular (depth = twice the width) sections and of same length are subjected to same bending moment. If the allowable stress is same then least weight of same material will be required for
(*a*) circular section
(*b*) square section
(*c*) rectangular section
(*d*) more data are required to determine same
(*e*) none of the above.

11.392. When a beam is subjected to a transverse shearing force, the shear stress in the upper fibres will be
(*a*) maximum (*b*) minimum
(*c*) zero
(*d*) depends on other data
(*e*) none of the above.

11.393. The central deflection in a fixed beam *i.e.,* supported firmly at both ends and loaded in the centre compared to a freely supported beam will be
(*a*) same (*b*) double
(*c*) one-half (*d*) one-fourth
(*e*) one-eight.

11.394. In a fixed beam, *i.e.* firmly supported at both the ends with uniformly distributed load, the location of zero bending moment from either end will be at
(*a*) zero (*b*) $l/8$
(*c*) $l/4$ (*d*) $l/3$
(*e*) none of the above.

11.395. A continuous beam is one which is
(*a*) infinitely long
(*b*) supported at two places
(*c*) supported at one point
(*d*) supported at more than two supports
(*e*) none of the above.

11.396. For a simply supported beam having a load at the centre the bending moment will be
(*a*) minimum at support
(*b*) minimum at the centre
(*c*) maximum at the supports
(*d*) minimum and maximum could be any where, along the length
(*e*) none of the above.

11.397. Bending moment in the centre of a beam of length l firmly supported at both ends and having a central load of W is
(*a*) Wl (*b*) $\frac{Wl}{2}$
(*c*) $\frac{Wl}{4}$ (*d*) $\frac{Wl}{8}$
(*e*) none of the above.

11.398. When a number of loads rest upon a beam, the deflection at any point is equal to the sum of the deflections at this point due to each of the loads taken separately. This is according to
(*a*) Castigliano's theorem
(*b*) principle of least work
(*c*) Maxwell's theorem
(*d*) theory of flexure
(*e*) moment shear relation.

11.399. The ratio of hoop stress to longitudinal stress in thin walled cylinders is
(*a*) 1 (*b*) $\frac{1}{2}$
(*c*) 2 (*d*) $\frac{1}{4}$
(*e*) 4.

11.400. The deflection at the point of application of an external force acting on a beam is equal to the partial derivative of the work of deformation w.r.t. this force. This is according to
(*a*) Castigliano's theorem
(*b*) principle of least work
(*c*) Maxwell's theorem
(*d*) theory of flexure
(*e*) moment shear relation.

11.401. The ratio of longitudinal stress to shear stress in thin walled cylinders is
(*a*) 1 (*b*) $\frac{1}{2}$
(*c*) 2 (*d*) $\frac{1}{4}$
(*e*) 4.

11.402. Rankine's constant for a M.S. column with both ends hinged is
(*a*) $\frac{1}{1500}$ (*b*) $\frac{1}{3500}$
(*c*) $\frac{1}{5500}$ (*d*) $\frac{1}{7500}$
(*e*) $\frac{1}{9500}$.

11.403. In a fixed beam *i.e.* firmly supported at both the ends, with a central load in the middle, the bending moment will be zero at
(*a*) one place (*b*) two places
(*c*) three places (*d*) four places
(*e*) none of the places.

11.404. The deformation of any structure takes place in such a manner that the work of deformation is a minimum. This is according to
(*a*) Castigliano's theorem
(*b*) principle of least work
(*c*) Maxwell's theorem
(*d*) theory of flexure
(*e*) moment shear relation.

11.405. In above question, 11.403, the location of point where bending moment is zero from either end is at
(*a*) zero (*b*) $l/8$
(*c*) $l/4$ (*d*) $l/2$
(*e*) none of the above.

11.406. In order to determine whether a column is long or short, one should know
(*a*) length (*b*) cross-section
(*c*) M.I.
(*d*) slenderness ratio
(*e*) conditions of end fixations.

11.407. As the slenderness ratio of a column increases, its compressive strength
(*a*) increases
(*b*) decreases
(*c*) remains unchanged
(*d*) may increase or decrease depending on length
(*e*) unpredictable.

11.408. Euler's formula is applicable for determining the buckling load for
(*a*) long columns
(*b*) intermediate columns
(*c*) medium size columns
(*d*) intermediate columns, but with certain amendments it can be used both for short as well as long columns
(*e*) short columns.

11.409. The load at which the column first buckles is known as
(*a*) buckling load
(*b*) crippling load
(*c*) critical load
(*d*) any one of the above
(*e*) none of the above.

11.410. A long column fails by
(*a*) crushing (*b*) tension
(*c*) shearing (*d*) buckling
(*e*) buckling and crushing.

11.411. The Rankine Gordon formula is applicable for determining the buckling load for
(*a*) long columns
(*b*) intermediate columns
(*c*) medium size columns
(*d*) intermediate columns, but with certain amendments it can be used for short as well as long columns
(*e*) short columns.

11.412. For eccentrically loaded strut, following section is preferred
(*a*) solid (*b*) hollow
(*c*) reinforced (*d*) composite
(*e*) any one of the above.

11.413. The ratio of length of column to the minimum radius of gyration of the cross sectional area of the column is known as
(*a*) slenderness ratio
(*b*) buckling factor
(*c*) crippling factor
(*d*) compressive factor
(*e*) column factor.

11.414. A column is defined as short column if
(*a*) its length is less than 10 m
(*b*) the ratio of its effective length to the least lateral dimension is less than 15
(*c*) the ratio of its effective length to least radius of gyration is less than 50
(*d*) both (*b*) and (*c*) above
(*e*) none of the above.

11.415. According to Euler's formula, the buckling load P, for a column of length l with both ends hinged and having I = least moment of inertia of the section of the column, and E = modulus of the elasticity of the material of the column
(*a*) $P = \dfrac{2\pi^2 EI}{l}$ (*b*) $P = \dfrac{\pi^2 EI}{4l^2}$
(*c*) $P = \dfrac{\pi^2 EI}{l^2}$ (*d*) $P = \dfrac{4\pi^2 EI}{l^2}$
(*e*) none of the above.

11.416. Slenderness ratio of a column is the ratio of its
(*a*) length of least lateral dimension
(*b*) length to radius of gyration
(*c*) both (*a*) and (*b*) above
(*d*) lateral dimension to radius of gyration
(*e*) none of the above.

11.417. The ratio of the equivalent length of the column to the minimum radius of gyration of the cross-sectional area of the column is called
(*a*) slenderness ratio
(*b*) buckling factor
(*c*) column factor
(*d*) crippling factor
(*e*) compressive factor.

11.418. A short column of external diameter D and internal diameter d carries an external load W. The greatest eccentricity which the load can have without producing tension on the cross section of the column is
(*a*) $\dfrac{D+d}{8}$ (*b*) $\dfrac{D^2+d^2}{8}$
(*c*) $\dfrac{D^2+d^2}{8D}$ (*d*) $\dfrac{D^2+d^2}{8d}$
(*e*) $\dfrac{D^2+d^2}{8D^2}$

11.419. The effective length of a column with one end fixed and other end free is
(*a*) its own length
(*b*) twice its length
(*c*) half its length
(*d*) $1/\sqrt{2} \times$ its length
(*e*) none of the above.

11.420. Who enunciated the following theorem

If unit loads rest upon a beam at the two points R and S, the deflection at R due to unit load at S equals the deflection at S due to the load at R
(*a*) Castigliano (*b*) Rankine
(*c*) Mohr (*d*) Maxwell
(*e*) Gordon.

11.421. A column with maximum equivalent length has
(*a*) both ends fixed
(*b*) both ends hinged
(*c*) one end fixed and other hinged
(*d*) one end fixed and other free
(*e*) both ends free.

11.422. A column has its equivalent length equal to its length in case of
(*a*) both ends fixed
(*b*) both ends hinged
(*c*) one end fixed and other hinged

(*d*) one end fixed and other free
(*e*) both ends free.

11.423. The slope and deflection at a point in a loaded cantilever or beam carrying several loads can be found out by the
(*a*) principle of least work
(*b*) moment area method
(*c*) double integration method
(*d*) Macaulay's method
(*e*) any one of the above.

11.424. Deflection in a beam is maximum where the slope is
(*a*) minimum (*b*) maximum
(*c*) zero (*d*) changes sign
(*e*) any value.

11.425. Who enunciated the following theorem

Deflection of the point of application of an external force acting on a beam is equal to the partial derivative of the work of deformation w.r.t. this force
(*a*) Castigliano (*b*) Rankine
(*c*) Mohr (*d*) Maxwell
(*e*) Gordon.

11.426. In case of both ends fixed of a column, the effective length is
(*a*) its own length (*b*) twice its length
(*c*) half its length
(*d*) $1/\sqrt{2} \times$ its length
(*e*) none of the above.

11.427. Longitudinal strain for a specimen is 0.01 and it is found to undergo 1 mm change in its thickness. Its thickness will be
(*a*) 100 mm (*b*) 200 mm
(*c*) 400 mm (*d*) 800 mm
(*e*) none of the above.

11.428. The distribution of stress in a beam of rectangular cross-section will be
(*a*) triangular (*b*) rectangular
(*c*) parabolic (*d*) elliptical
(*e*) part of a circle.

11.429. For a beam shown in Fig. 11.9 what will be the shear force at the centre

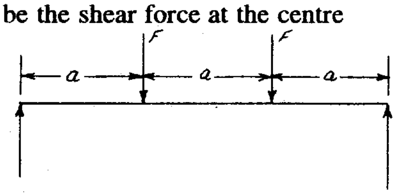

Fig. 11.9.

(*a*) F (*b*) $2F$
(*c*) $-F$ (*d*) 0
(*e*) $F/2$.

11.430. The shape of a cantilever for uniformly distributed load will be
(*a*) straight line (*b*) parabolic
(*c*) parabolic (*d*) elliptical
(*e*) none of the above.

11.431. The deflection of the centre of a simply supported beam of length l carrying a uniformly distributed load w kg/m will be
(*a*) $\frac{wl^4}{8\,EI}$ (*b*) $\frac{5wl^3}{384\,EI}$
(*c*) $\frac{5}{8}\frac{wl^4}{EI}$ (*d*) $\frac{5}{384}\frac{wl^4}{EI}$
(*e*) $\frac{wl}{48EI}$.

11.432. In a beam at a place where the shear force is maximum, the bending moment will be
(*a*) maximum (*b*) minimum
(*c*) zero
(*d*) neither minimum nor maximum
(*e*) none of the above.

11.433. In a tensile testing experiment on a specimen of 1 cm^2 area, the maximum load observed was 5 tonnes and neck area 0.5 cm^2. The ultimate tensile strength of specimen is
(*a*) 5 tonnes/cm^2
(*b*) 10 tonnes/cm^2
(*c*) 2.5 tonnes/cm^2
(*d*) 20 tonnes/cm^2
(*e*) none of the above.

11.434. Elasticity of a M.S. specimen is defined by
(*a*) Hooke's law (*b*) yield point
(*c*) when plastic flow starts
(*d*) proof stress
(*e*) permanent set.

11.435. A coil is cut into two halves, the stiffness of cut coils will be
(*a*) double (*b*) half
(*c*) same (*d*) something else
(*e*) none of the above.

11.436. A hollow shaft of same cross-section area as solid shaft transmits
(*a*) same torque (*b*) less torque
(*c*) more torque

(d) more or less depending on external diametei
(e) none of the above.

11.437. The value of J in equation $\frac{T}{J} = \frac{s_s}{y} = \frac{G\theta}{l}$ for a circular shaft of diameter d is

(a) $\frac{\pi d^4}{32}$ (b) $\frac{\pi d^4}{64}$
(c) $\frac{\pi d^4}{16}$ (d) $\frac{\pi d^3}{32}$
(e) $\frac{\pi d^3}{16}$.

11.438. The value of M.I. for a solid shaft of diameter 'd 'is equal to

(a) $\frac{\pi d^4}{32}$ (b) $\frac{\pi d^4}{64}$
(c) $\frac{\pi d^4}{16}$ (d) $\frac{\pi d^3}{32}$
(e) $\frac{\pi d^3}{16}$.

11.439. Torque in a solid shaft of diameter d and shear strength of s_s is given by

(a) $\frac{\pi}{8} s_s d^3$ (b) $\frac{\pi}{16} s_s d^3$
(c) $\frac{\pi}{32} s_s d^3$ (d) $\frac{\pi}{64} s_s d^3$
(e) $\frac{\pi}{16} s_s d^4$.

11.440. Two shafts, one hollow and other solid have same material and mass. The inner diameter of hollow shaft is half the outside diameter. The ratio of torque that can be transmitted by these two shafts is

(a) $\frac{15}{16}$ (b) $\frac{1}{16}$
(c) $\frac{1}{4}$ (d) $\frac{1}{32}$
(e) $\frac{3}{4}$.

11.441. The value of shear stress induced in a shaft due to applied torque is

(a) uniform throughout
(b) zero at circumference and maximum at centre
(c) zero at cnetre and maximum at circumference
(d) zero in middle and maximum at centre and circumference
(e) none of the above.

11.442. Value of longitudinal stress for thin cylinder is

(a) $\frac{pd}{2t}$ (b) $\frac{pd}{4t}$
(c) $\frac{2pd}{t}$ (d) $\frac{pd}{2.5t}$
(e) none of the above.

11.443. Which one of the following statements for a shaft is false. Shear stress intensity at a point is

(a) directly proportional to the distance from the axis
(b) inversely proportional to the distance from the axis
(c) inversely proportional to the polar moment of inertia
(d) directly proportional to the applied torque
(e) none of the above.

12

Theory of Machines

12.1. Which of the following disciplines provides study of inertia forces arising from the combined effect of the mass and the motion of the parts
(*a*) theory of machines
(*b*) applied mechanics
(*c*) mechanisms (*d*) kinetics
(*e*) kinematics.

12.2. Which of the following disciplines provides study of relative motion between the parts of a machine
(*a*) theory of machines
(*b*) applied mechanics
(*c*) mechanisms (*d*) kinetics
(*e*) kinematics.

12.3. Which of the following disciplines provides study of the relative motion between the parts of a machine and the forces acting on the parts
(*a*) theory of machines
(*b*) applied mechanics
(*c*) mechanisms
(*d*) kinetics
(*e*) kinematics.

12.4. The type of pair formed by two elements which are so connected that one is constrained to turn or revolve about a fixed axis of another element is known as
(*a*) turning pair (*b*) rolling pair
(*c*) sliding pair (*d*) spherical pair
(*e*) lower pair.

12.5. Which of the following is a lower pair
(*a*) ball and socket
(*b*) piston and cylinder
(*c*) cam and follower
(*d*) (*a*) and (*b*) above
(*e*) belt drive.

12.6. If two moving elements have surface contact in motion, such pair is known as
(*a*) sliding pair (*b*) rolling pair
(*c*) surface pair (*d*) lower pair
(*e*) higher pair.

12.7. The example of lower pair is
(*a*) shaft revolving in a bearing
(*b*) straight line motion mechanisms
(*c*) automobile steering gear
(*d*) all of the above
(*e*) none of the above.

12.8. Pulley in a belt drive acts as
(*a*) cylindrical pair
(*b*) turning pair
(*c*) rolling pair
(*d*) sliding pair
(*e*) surface pair.

12.9. The example of rolling pair is
(*a*) bolt and nut
(*b*) lead screw of a lathe
(*c*) ball and socket joint
(*d*) ball bearing and roller bearing
(*e*) all of the above.

12.10. Any point on a link connecting double slider crank chain will trace a
(*a*) straight line (*b*) circle
(*c*) ellipse (*d*) parabola
(*e*) hyperbola.

12.11. The purpose of a link is to
(*a*) transmit motion
(*b*) guide other links
(*c*) act as a support

(*d*) all of the above
(*e*) none of the above.

12.12. A universal joint is an example of
(*a*) higher pair (*b*) lower pair
(*c*) rolling pair (*d*) sliding pair
(*e*) turning pair.

12.13. Rectilinear motion of piston is converted into rotary by
(*a*) cross head (*b*) slider crank
(*c*) connecting rod
(*d*) gudgeon pin
(*e*) four bar chain mechanism.

12.14. A heavy ball is suspended from a fixed point by a string of length 1 m and is rotating about a vertical axis through this point with uniform angular velocity of 10 rad/sec. Angle between cord and vertical axis will be
(*a*) $\cos^{-1} 9.81$ (*b*) $\cos^{-1} 0.981$
(*c*) $\cos^{-1} 0.0981$ (*d*) $\cos^{-1} 0.00981$
(*e*) more data is required to determine same.

12.15. The values of velocity and acceleration of piston at near dead centre for a slider-crank mechanism will be
(*a*) 0, and more than $\omega^2 r$
(*b*) 0, and less than $\omega^2 r$
(*c*) 0, 0 (*d*) ωr, 0
(*e*) none of the above.

12.16. The example of spherical pair is
(*a*) bolt and nut
(*b*) lead screw of a lathe
(*c*) ball and socket joint
(*d*) ball bearing and roller bearing
(*e*) none of the above.

12.17. Cross head and guides form a
(*a*) lower pair (*b*) higher pair
(*c*) turning pair (*d*) rolling pair
(*e*) sliding pair.

12.18. The number of instantaneous centres for a four-bar chain mechanism and in general for *n* links are
(*a*) 6, $\frac{n(n-1)}{2}$ (*b*) 4, *n*
(*c*) 12, $n(n-1)$ (*d*) 3, $n-1$
(*e*) none of the above.

12.19. A circular bar moving in a round hole is an example of
(*a*) incompletely constrained motion
(*b*) partially constrained motion
(*c*) completely constrained motion
(*d*) successfully constrained motion
(*e*) none of the above.

12.20. If some links are connected such that motion between them can take place in more than one direction, it is called
(*a*) incompletely constrained motion
(*b*) partially constrained motion
(*c*) completely constrained motion
(*d*) successfully constrained motion
(*e*) none of the above.

12.21. If there are *L* number of links in a mechanism then number of possible inversions is equal to
(*a*) $L+1$ (*b*) $L-1$
(*c*) L (*d*) $L+2$
(*e*) $L-2$.

12.22. Kinematic pairs are those which have two elements that
(*a*) have line contact
(*b*) have surface contact
(*c*) permit relative motion
(*d*) are held together
(*e*) have dynamic forces.

12.23. A simple mechanism has
(*a*) 1 link (*b*) 2 links
(*c*) 3 links (*d*) 4 links
(*e*) 5 links.

12.24. The lower pair is a
(*a*) open pair (*b*) closed pair
(*c*) sliding pair (*d*) point contact pair
(*e*) does not exist.

12.25. Automobile steering gear is an example of
(*a*) higher pair (*b*) sliding pair
(*c*) turning pair (*d*) rotary pair
(*e*) lower pair.

12.26. In higher pair, the relative motion is
(*a*) purely turning (*b*) purely sliding
(*c*) purely rotary
(*d*) purely surface contact
(*e*) combination of sliding and turning.

12.27. Which of the following has sliding motion
(*a*) crank (*b*) connecting rod
(*c*) crank pin (*d*) cross-head
(*e*) cross head guide.

12.28. The example of higher pair is

(a) belt, rope and chain drives
(b) gears, cams
(c) ball and roller bearings
(d) all of the above
(e) none of the above.

12.29. Which of the following mechanism is obtained from lower pair
(a) gyroscope
(b) pantograph
(c) valve and valve gears
(d) generated straight line motions
(e) all of the above.

12.30. Which of the following would constitute a link
(a) piston, piston rings and gudgeon pin
(b) piston, and piston rod
(c) piston rod and cross head
(d) piston, crank pin and crank shaft
(e) piston, piston-rod and cross head.

12.31. The Scott-Russell mechanism consists of
(a) sliding and turning pairs
(b) sliding and rotary pairs
(c) turning and rotary pairs
(d) sliding pairs only
(e) turning pairs only.

12.32. Davis steering gear consists of
(a) sliding pairs (b) turning pairs
(c) rolling pairs (d) higher pairs
(e) lower pairs.

12.33. Ackermann steering gear consists of
(a) sliding pairs (b) turning pairs
(c) rolling pairs (d) higher pairs
(e) lower pairs.

12.34. A completely constrained motion can be transmitted with
(a) 1 link with pin joints
(b) 2 links with pin joints
(c) 3 links with pin joints
(d) 4 links with pin joints
(e) all of the above.

12.35. The motion transmitted between the teeth of gears in mesh is
(a) sliding
(b) rolling
(c) rotary
(d) could be either sliding or rolling depending upon shape of teeth
(e) partly sliding and partly rolling.

12.36. Oldham's coupling is the
(a) second inversion of double slider crank chain
(b) third inversion of double slider crank chain
(c) second inversion of single slider crank chain
(d) third inversion of slider crank chain
(e) fourth inversion of double slider crank chain.

12.37. Sense of tangential acceleration of a link
(a) is same as that of velocity
(b) is opposite to that of velocity
(c) could be either same or opposite to velocity
(d) is perpendicular to that of velocity
(e) none of the above.

12.38. A mechanism is an assemblage of
(a) two links (b) three links
(c) four links or more than four links
(d) all of the above
(e) none of the above.

12.39. The number of links in pantograph mechanism is equal to
(a) 2 (b) 3
(c) 4 (d) 5
(e) 6.

12.40. Elements of pairs held together mechanically is known as
(a) closed pair (b) open pair
(c) mechanical pair
(d) rolling pair (e) none of the above.

12.41. Shaft revolving in a bearing is the following type of pair
(a) lower pair (b) higher pair
(c) spherical pair (d) cylindrical pair
(e) bearing pair.

12.42. Rectangular bar in a rectangular hole is the following type of pair
(a) completely constrained motion
(b) partially constrained motion
(c) incompletely constrained motion
(d) freely constrained motion
(e) none of the above.

12.43. A foot step bearing and rotor of a vertical turbine form examples of
(a) incompletely constrained motion
(b) partially constrained motion
(c) completely constrained motion

(*d*) successfully constrained motion
(*e*) none of the above.

12.44. A slider crank chain consists of following numbers of turning and sliding pairs
(*a*) 1, 3 (*b*) 2, 2
(*c*) 3, 1 (*d*) 4, 0
(*e*) 0, 4.

12.45. There is a relation between number of joints and number of links, where L = number of links and J = number of joints which constitute a kinematic chain and this is given by the expression
(*a*) $L = 3/2\,(J + 2)$
(*b*) $L = \frac{1}{3}(J + 2)$ (*c*) $L = \frac{2}{3}(J + 2)$
(*d*) $L = \frac{2}{3}(J + 1)$ (*e*) $L = \frac{2}{3}(J + 3)$.

12.46. Relationship between the number of links (L) and number of pairs (P) is
(*a*) $P = 2L - 4$ (*b*) $P = 2L + 4$
(*c*) $P = 2L + 2$ (*d*) $P = 2L - 2$
(*e*) $P = L - 4$.

12.47. The criterion of constraint of a chain as enunciated by A.W. Klein, connecting the number of binary joints (J), number of higher pairs (H), and number of links (L) is
(*a*) $J + \frac{1}{2}H = \frac{3}{2}L - 2$
(*b*) $J + H = 3L - 2$
(*c*) $J + \frac{1}{2}H = 3L - 2$
(*d*) $J + \frac{1}{2}H = \frac{3L - 2}{2}$
(*e*) $J + \frac{3}{2}H = L - 2$.

12.48. In problem 12.47, the chain is locked when
(*a*) L.H.S. = R.H.S.
(*b*) L.H.S. > R.H.S.
(*c*) L.H.S. < R.H.S.
(*d*) there is no such criterion for checking above requirement
(*e*) none of the above.

12.49. In problem 12.47, the chain is unconstrained when
(*a*) L.H.S. = R.H.S.
(*b*) L.H.S. > R.H.S.
(*c*) L.H.S. < R.H.S.
(*d*) there is no such criterion for checking above requirement
(*e*) none of the above.

12.50. In problem 12.47, the chain is constrained when
(*a*) L.H.S. = R.H.S.
(*b*) L.H.S. < R.H.S.
(*c*) L.H.S. > R.H.S.
(*d*) there is no such criterion for checking above requirement
(*e*) none of the above.

12.51. The tendency of a body to resist change from rest or motion is known as
(*a*) mass (*b*) friction
(*c*) inertia (*d*) resisting force
(*e*) resisting torque.

12.52. A flywheel weighs $\frac{981}{\pi}$ kg and has a radius of gyration of 100 cm. It is given a spin of 100 r.p.m. about its horizontal axis. The whole assembly is rotating about a vertical axis at 6 rad/sec. The gyroscopic couple experienced will be
(*a*) 2000 kg m (*b*) 19,620 kg m
(*c*) 20,000 kg m (*d*) 1962 kg m
(*e*) none of the above.

12.53. The type of coupling used to join two shafts whose axes are neither in same straight line nor parallel, but intersect is
(*a*) flexible coupling
(*b*) universal coupling
(*c*) chain coupling
(*d*) Oldham's coupling
(*e*) American coupling.

12.54. The advantage of the piston valve over D-slide valve is that in the former case
(*a*) wear is less
(*b*) power absorbed is less
(*c*) both wear and power absorbed are low
(*d*) the pressure developed being high provides tight sealing
(*e*) there is overall economy of initial cost, maintenance and operation.

12.55. Flexible coupling is used because
(*a*) it is easy to disassemble
(*b*) it is easy to engage and disengage
(*c*) it transmits shocks gradually
(*d*) it prevents shock transmission and eliminates stress reversals
(*e*) it increases shaft life.

12.56. With single Hooke's joint it is possible to connect two shafts, the axes of which have an angular misalignment upto
(*a*) 10° (*b*) 20°
(*c*) 30° (*d*) 40°
(*e*) 60°.

12.57. The Hooke's joint consists of :
(*a*) two forks (*b*) one fork
(*c*) three forks (*d*) four forks
(*e*) five forks.

12.58. The Klein's method of construction for reciprocating engine mechanism
(*a*) is based on acceleration diagram
(*b*) is a simplified form of instantaneous centre method
(*c*) utilises a quadrilateral similar to the diagram of mechanism for reciprocating engine
(*d*) enables determination of Corioli's component
(*e*) none of the above.

12.59. It is required to connect two parallel shafts, the distance between whose axes is small and variable. The shafts are coupled by
(*a*) universal joint
(*b*) knuckle joint (*c*) Oldham's coupling
(*d*) flexible coupling
(*e*) electromagnetic coupling.

12.60. The c.g. of a link in any mechanism would experience
(*a*) no acceleration
(*b*) linear acceleration
(*c*) angular acceleration
(*d*) both angular and linear accelerations
(*e*) none of the above.

12.61. In elliptical trammels
(*a*) all four pairs are turning
(*b*) three pairs turning and one pair sliding
(*c*) two pairs turning and two pairs sliding
(*d*) one pair turning and three pairs sliding
(*e*) all four pairs sliding.

12.62. In automobiles the power is transmitted from gear box to differential through
(*a*) bevel gear (*b*) universal joint
(*c*) Hooke's joint (*d*) Knuckle joint
(*e*) Oldham's coupling.

12.63. The indicator using Watt mechanism is known as
(*a*) Thompson indicator
(*b*) Richard indicator
(*c*) Simplex indicator
(*d*) Thomson indicator
(*e*) none of the above.

12.64. The Ackermann steering mechanism is preferred to the Davis type in automobiles because
(*a*) the former is mathematically accurate
(*b*) the former is having turning pair
(*c*) the former is most economical
(*d*) the former is most rigid
(*e*) none of the above.

12.65. Transmission of power from the engine to the rear axle of an automobile is by means of
(*a*) compound gears
(*b*) worm and wheel method
(*c*) Hooke's joint (*d*) crown gear
(*e*) bevel gears.

12.66. When a ship travels in a sea, which of the effect is more dangerous
(*a*) steering (*b*) pitching
(*c*) rolling (*d*) all of the above
(*e*) none of the above.

12.67. In an ideal machine, the output as compared to input is
(*a*) less (*b*) more
(*c*) equal
(*d*) may be less or more depending on efficiency
(*e*) always less.

12.68. Governor is used in automobile to
(*a*) decrease the variation of speed
(*b*) to control $\frac{\delta N}{\delta t}$
(*c*) to control δN
(*d*) all of the above
(*e*) none of the above.

12.69. In gramophones for adjusting the speed of the turntable, the following type of governor is commonly employed
(*a*) Hartung governor
(*b*) Wilson Hartnell governor
(*c*) Pickering governor
(*d*) Inertia governor
(*e*) none of the above.

12.70. For fluctuating loads, well suited bearing is
(*a*) ball bearing (*b*) roller bearing
(*c*) needle roller bearing

(*d*) thrust bearing (*e*) sleeve bearing.

12.71. Crowning on pulleys helps
(*a*) in increasing velocity ratio
(*b*) in decreasing the slip of the belt
(*c*) for automatic adjustment of belt position so that belt runs centrally
(*d*) increase belt and pulley life
(*e*) none of the above.

12.72. Idler pulley is used
(*a*) for changing the direction of motion of the belt
(*b*) for applying tension
(*c*) for increasing velocity ratio
(*d*) all of the above
(*e*) none of the above.

12.73. In multi-V-belt transmission, if one of the belt is broken, we have to change the
(*a*) broken belt
(*b*) broken belt and its adjacent belts
(*c*) all the belts
(*d*) there is no need of changing any one as remaining belts can take care of transmission of load
(*e*) all the weak belts.

12.74. The moment on the pulley which produces rotation is called
(*a*) inertia (*b*) momentum
(*c*) moment of momentum
(*d*) work (*e*) torque.

12.75. Creep in belt drive is due to
(*a*) material of the pulley
(*b*) material of the belt
(*c*) larger size of the driver pulley
(*d*) uneven extensions and contractions due to varying tension
(*e*) expansion of belt.

12.76. The horse power transmitted by a belt is dependent upon
(*a*) tension on tight side of belt
(*b*) tension on slack side of belt
(*c*) radius of pulley
(*d*) speed of pulley
(*e*) all of the above.

12.77. The locus of a point on a thread unwound from a cylinder will be
(*a*) a straight line (*b*) a circle
(*c*) involute (*d*) cycloidal
(*e*) helix.

12.78. To transmit power from one rotating shaft to another whose axes are neither parallel nor intersecting, use
(*a*) spur gear (*b*) spiral gear
(*c*) bevel gear (*d*) worm gear
(*e*) crown gear.

12.79. In a gear drive, module is equal to
(*a*) $\dfrac{1}{\text{diametral pitch}}$
(*b*) $\dfrac{1}{\text{circular pitch}}$
(*c*) $\dfrac{\text{circular pitch}}{\pi}$
(*d*) diametral pitch/π
(*e*) π/diametral pitch.

12.80. Addendum is given by
(*a*) $\dfrac{\pi}{\text{circular pitch}}$ (*b*) diametral pitch
(*c*) one module (*d*) 1.25 module
(*e*) none of the above.

12.81. To obviate axial thrust, following gear drive is used
(*a*) double helical gears having opposite teeth
(*b*) double helical gears having identical teeth
(*c*) single helical gear in which one of the teeth of helix angle α is more
(*d*) mutter gears
(*e*) none of the above.

12.82. Which of the following is false statement in respect of differences between machine and structure
(*a*) Machines transmit mechanical work, whereas structures transmit forces
(*b*) In machines, relative motion exists between its members, whereas same does not exist in case of structures
(*c*) Machines modify movement and work, whereas structures modify forces
(*d*) Efficiency of machines as well as structures is below 100%
(*e*) Machines are run by electric motors, but structures are not.

12.83. If D_1 and D_2 be the diameters of driver and driven pulleys, then belt speed is proportional to
(*a*) D_1/D_2 (*b*) D_2/D_1

(c) $D_1 - D_2$. (d) D_1
(e) $D_1 + D_2$.

12.84. Typewriter constitutes
(a) machine (b) structure
(c) mechanism (d) inversion
(e) none of the above.

12.85. Lower pairs are those which have
(a) point or line contact between the two elements when in motion
(b) surface contact between the two elements when in motion
(c) elements of pairs not held together mechanically
(d) two elements that permit relative motion
(e) none of the above.

12.86. A point on a link connecting double slider crank chain traces a
(a) straight line (b) circle
(c) parabola (d) hyperbola
(e) ellipse.

12.87. A pantograph is a mechanism with
(a) lower pairs (b) higher pairs
(c) rolling pairs (d) turning pairs
(e) spherical pairs.

12.88. Kinematic pairs are those which have
(a) point or line contact between the two elements when in motion
(b) surface contact between the two elements when in motion
(c) elements of pairs not held together mechanically
(d) two elements that permit relative motion
(e) none of the above.

12.89. If the opposite links of a four bar linkage are equal, the links will always form a
(a) triangle (b) rectangle
(c) parallelogram (d) pentagon
(e) trapezoid.

12.90. Higher pairs are those which have
(a) point or line contact between the two elements when in motion
(b) surface contact between the two elements when in motion
(c) elements of pairs not held together mechanically
(d) two elements that permit relative motion
(e) none of the above.

12.91. A cam mechanism imparts following motion
(a) rotating (b) oscillating
(c) reciprocating (d) all of the above
(e) none of the above.

12.92. A cam with a roller follower would constitute following type of pair
(a) lower pair (b) higher pair
(c) open pair (d) close pair
(e) cam pair.

12.93. The approximate straight line mechanism is a
(a) four bar linkage
(b) 6 bar linkage (c) 8 bar linkage
(d) 3 bar linkage (e) 5 bar linkage.

12.94. Open pairs are those which have
(a) point or line contact between the two elements when in motion
(b) surface contact between the two elements when in motion
(c) elements of pairs not held together mechanically
(d) two elements that permit relative motion
(e) none of the above.

12.95. Peaucellier mechanism has
(a) eight links (b) six links
(c) four links (d) twelve links
(e) five links.

12.96. Hart mechanism has
(a) eight links (b) six links
(c) four links (d) twelve links
(e) five links.

12.97. A chain comprises of 5 links having 5 joints. Is it kinematic chain ?
(a) yes (b) no
(c) it is a marginal case
(d) data are insufficient to determine it
(e) unpredictable.

12.98. In the following equation $[L = \frac{2}{3}(J + 2)]$ to determine whether or not the given chain in kinematic, higher pair is treated equivalent to
(a) two lower pairs and two additional links
(b) two lower pairs and two additional links

(c) one lower pair and two additional links
(d) any one of the above
(e) none of the above.

12.99. The main disadvantage of the sliding pair is that it is
(a) bulky (b) wears rapidly
(c) difficult to manufacture
(d) (a) and (b) above
(e) (a) and (c) above.

12.100. For a kinematic chain to be considered as mechanism
(a) two links should be fixed
(b) one link should be fixed
(c) none of the links should be fixed
(d) there is no such criterion
(e) none of the above.

12.101. An eccentric sheave pivoted at one point rotates and transmits oscillatory motion to a link whose one end is pivoted and other end is connected to it. This mechanism has
(a) 2 links (b) 3 links
(c) 4 links (d) 5 links
(e) none of the above.

12.102. Whitworth quick return mechanism is obtained by inversion of
(a) slider crank mechanism
(b) kinematic chain
(c) five link mechanism
(d) roller cam mechanism
(e) none of the above.

12.103. In its simplest form, a cam mechanism consists of following number of links
(a) 1 (b) 2
(c) 3 (d) 4
(e) none.

12.104. Which of the following mechanisms produces mathematically an exact straight line motion
(a) Grasshopper mechanism
(b) Watt mechanism
(c) Peaucellier's mechanism
(d) Tchabichiff mechanism
(e) Ackermann mechanism.

12.105. In a mechanism, usually one link is fixed. If the fixed link is changed in a kinematic chain, then relative motion of other links
(a) will remain same
(b) will change
(c) could change or remain unaltered depending on which link is fixed
(d) will not occur
(e) none of the above.

12.106. A kinematic chain requires at least
(a) 2 links and 3 turning pairs
(b) 3 links and 4 turning pairs
(c) 4 links and 4 turning pairs
(d) 5 links and 4 turning pairs
(e) none of the above.

12.107. In a darg link quick return mechanism, the shortest link is always fixed. The sum of the shortest and longest link is
(a) equal to sum of other two
(b) greater than sum of other two
(c) less than sum of other two
(d) there is no such relationship
(e) none of the above.

12.108. The following is the inversion of slider crank chain mechanism
(a) Whitworth quick return mechanism
(b) hand pump
(c) oscillating cylinder engine
(d) all of the above
(e) none of the above.

12.109. Kinematic pairs are those which have
(a) two elements held together mechanically
(b) two elements having relative motion
(c) two elements having Coroili's component
(d) minimum of two instantaneous centres
(e) all of the above.

12.110. According to criterion of constraint by A.W. Klein
(a) $J+\frac{1}{2}H=3/2L-2$
(b) $H+\frac{1}{2}J=\frac{2}{3}L-2$
(c) $J+\frac{1}{2}H=3/2L-1$
(d) $J+3/2H=\frac{1}{2}L-2$
(e) none of the above.
where J = number of binary joints,
H = number of higher pairs,
and L = number of links.

12.111. A quarternary joint is equivalent to

(*a*) one binary joint
(*b*) two binary joints
(*c*) three binary joints
(*d*) four binary joints
(*e*) none of the above.

12.112. A typewriter mechanism has 7 number of binary joints, six links and none of higher pairs. The mechanism is
(*a*) kinematically sound
(*b*) not sound
(*c*) soundness would depend upon which link is kept fixed
(*d*) data is not sufficient to determine same
(*e*) none of the above.

12.113. In a four-bar chain it is required to give an oscillatory motion to the follower for a continuous rotation of the crank. For the lengths of 50 mm of crank and 70 mm of the follower, determine theoretical maximum length of coupler. The distance between fixed pivots of crank and followers is
(*a*) 95 mm
(*b*) slightly less than 95 mm
(*c*) slightly more than 95 mm
(*d*) 45 mm
(*e*) none of the above.

12.114. In above example, the minimum length of the coupler will be
(*a*) 45 mm
(*b*) slightly less than 45 mm
(*c*) slightly more than 45 mm
(*d*) 95 mm
(*e*) none of the above.

12.115. In S.H.M., acceleration is proportional to
(*a*) velocity
(*b*) displacement
(*c*) rate of change of velocity
(*d*) all of the above
(*e*) none of the above.

12.116. In S.H.M., the ratio of acceleration and displacement is proportional to
(*a*) frequency (ω) (*b*) ω
(*c*) ω^2 (*d*) $1/\omega^2$
(*e*) $\sqrt{\omega}$.

12.117. In S.H.M., the velocity vector w.r.t. displacement vector
(*a*) leads by 90° (*b*) lags by 90°
(*c*) leads by 180° (*d*) are in phase
(*e*) could be anywhere.

12.118. A body having moment of inertia of 30 kg m^2 is rotating at 210 RPM and meshes with another body at rest having M.I. of 40 kg m^2. The resultant speed after meshing will be
(*a*) 90 RPM (*b*) 100 RPM
(*c*) 80 RPM
(*d*) data are insufficient
(*e*) none of the above.

12.119. Inertia force acts
(*a*) perpendicular to the accelerating force
(*b*) along the direction of accelerating force
(*c*) opposite to the direction of accelerating force
(*d*) in any direction w.r.t. accelerating force depending on the magnitude of two
(*e*) none of the above.

12.120. The frequency of oscillation at moon compared to earth will be
(*a*) 6 times more
(*b*) 6 times less
(*c*) 2.44 times more
(*d*) 2.44 times less
(*e*) 36 times less.

12.121. Polar moment of inertia (I_P) of a circular disc is to be determined by suspending it by a wire and noting the frequency of oscillations (f)
(*a*) $I_P \propto f$ (*b*) $I_P \propto f^2$
(*c*) $I_P \propto \frac{1}{f^2}$ (*d*) $I_P \propto \frac{1}{f}$
(*e*) none of the above.

12.122. The frequency of oscillation of a bigger diameter cylinder compared to a small cylinder inside a cylindrical concave surface will be
(*a*) less (*b*) more
(*c*) same
(*d*) data are insufficient to determine same
(*e*) none of the above.

12.123. The frequency of oscillation of a cylinder inside a cylindrical concave surface of

bigger radius compared to a small radius will be
(*a*) less (*b*) more
(*c*) same
(*d*) data are insufficient to determine same
(*e*) none of the above.

12.124. If the radius of gyration of a compound pendulum about an axis through c.g. is more, then its frequency of oscillation will be
(*a*) less (*b*) more
(*c*) same
(*d*) data are insufficient to determine same
(*e*) none of the above.

12.125. A body of mass m and radius of gyration 'k' is to be replaced by two masses m_1 and m_2 located at distances h_1 and h_2 from the c.g. of original body. These will be kinetically equivalent to original body if
(*a*) $h_1 + h_2 = k$ (*b*) $h_1^2 + h_2^2 = k^2$
(*c*) $h_1 h_2 = k^2$ (*d*) $\frac{h_1 + h_2}{2} = k$
(*e*) $k = h_1 h_2$.

12.126. The Bifilar suspension method is used to determine
(*a*) natural frequency of vibration
(*b*) position of balancing weights
(*c*) moment of inertia
(*d*) centripetal acceleration
(*e*) angular acceleration of a body.

12.127. Which is the false statement about the properties of instantaneous centre
(*a*) at the instantaneous centre of rotation, one rigid link rotates instantaneously relative to another for the configuration of mechanism considered
(*b*) the two rigid links have no linear velocities relative to each other at the instantaneous centre
(*c*) the two rigid links which have no linear velocity relative to each other at this centre have the same linear velocity to the third rigid link
(*d*) the double centre can be denoted either by O_{21} or O_{12}, but proper selection should be made
(*e*) none of the above.

12.128. Instantaneous center of rotation of a link in a four bar mechanism lies on
(*a*) right side pivot of this link
(*b*) lift side pivot of this link
(*c*) a point obtained by intersection on extending adjoining links
(*d*) can't occur
(*e*) none of the above.

12.129. The total number of instantaneous centres for a mechanism of n links is
(*a*) $\frac{n(n-1)}{2}$ (*b*) n
(*c*) $n - 1$ (*d*) $\frac{n}{2}$
(*e*) $n(n-1)$.

12.130. The number of links and instantaneous centres in a reciprocating engine mechanism are
(*a*) 4, 4 (*b*) 4, 5
(*c*) 5, 4 (*c*) 6, 4
(*e*) 4, 6.

12.131. According to Kennedy's theorem, if three bodies have plane motions, their instantaneous centres lie on
(*a*) a triangle (*b*) a point
(*c*) two lines (*d*) a straight line
(*e*) a curve.

12.132. In a rigid link OA, velocity of A w.r.t. O will be
(*a*) parallel to OA
(*b*) perpendicular to OA
(*c*) at 45° to OA
(*d*) along AO
(*e*) along OA.

12.133. Two systems shall be dynamically equivalent when
(*a*) the mass of two are same
(*b*) c.g. of two coincides
(*c*) M.I. of two about an axis through c.g. is equal
(*d*) all of the above
(*e*) none of the above.

12.134. A link is rotating about O. Velocity of point P on link w.r.t. point Q on link will be perpendicular to
(*a*) OP (*b*) OQ
(*c*) PQ
(*d*) line in between OP and OQ

(*e*) none of the above.

12.135. The velocity of any point in mechanism relative to any other point on the mechanism on velocity polygon is represented by the line
(*a*) joining the corresponding points
(*b*) perpendicular to line as per (*a*)
(*c*) not possible to determine with these data
(*d*) at 45° to line as per (*a*)
(*e*) none of the above.

12.136. The absolute acceleration of any point P in a link about centre of rotation O is
(*a*) along PO
(*b*) perpendicular to PO
(*c*) at 45° to PO (*d*) along OP
(*e*) none of the above.

12.137. Angular acceleration of a link can be determined by dividing the
(*a*) centripetal component of acceleration with length of link
(*b*) tangential component of acceleration with length of link
(*c*) resultant acceleration with length of link
(*d*) all of the above
(*e*) none of the above.

12.138. Corioli's component of acceleration exists whenever a point moves along a path that has
(*a*) linear displacement
(*b*) rotational motion
(*c*) tangential acceleration
(*d*) centripetal acceleration
(*e*) none of the above.

12.139. The direction of Corioli's component of acceleration is the direction
(*a*) of relative velocity vector for the two coincident points rotated by 90° in the direction of the angular velocity of the rotation of the link
(*b*) along the centripetal acceleration
(*c*) along tangential acceleration
(*d*) along perpendicular to angular velocity
(*e*) none of the above.

12.140. In a shaper mechanism, the Corioli's component of acceleration will
(*a*) not exist (*b*) exist
(*c*) depend on position of crank
(*d*) unpredictable
(*e*) none of the above.

12.141. The magnitude of tangential acceleration is equal to
(*a*) velocity^2 × crank radius
(*b*) velocity^2/crank radius
(*c*) (velocity/crank radius)^2
(*d*) velocity × crank radius^2
(*e*) none of the above.

12.142. Tangential acceleration direction is
(*a*) along the angular velocity
(*b*) opposite to angular velocity
(*c*) may be any one of these
(*d*) perpendicular to angular velocity
(*e*) none of the above.

12.143. The magnitude of the Corioli's component of acceleration of a slider moving at velocity V on a link rotating at angular speed ω is
(*a*) $V\omega$ (*b*) $2V\omega$
(*c*) $\dfrac{V\omega}{2}$ (*d*) $\dfrac{2V}{\omega}$
(*e*) none of the above.

12.144. In a rotary engine the angular velocity of the cylinder centre line is 25 rad/sec and the relative velocity of a point on the cylinder centre line w.r.t. cylinder is 10 m/sec. Corioli's acceleration will be
(*a*) 500 m/sec^2 (*b*) 250 m/sec^2
(*c*) 1000 m/sec^2 (*d*) 2000 m/sec^2
(*e*) unpredictable.

12.145. Corioli's component is encountered in
(*a*) quick return mechanism of shaper
(*b*) four bar chain mechanism
(*c*) slider crank mechanism
(*d*) (*a*) and (*c*) above
(*e*) all of the above.

12.146. Klein's construction gives a graphical construction for
(*a*) slider-crank mechanism
(*b*) velocity polygon
(*c*) acceleration polygon
(*d*) four bar chain mechanism
(*e*) angular acceleration.

12.147. The velocity of a slider with reference to a fixed point about which a bar is rotating and slider sliding on the bar will be

(a) parallel to bar
(b) perpendicular to bar
(c) somewhere in between above two
(d) unpredictable
(e) none of the above.

12.148. Klein's construction can be used to determine acceleration of various parts when the crank is at
(a) inner dead centre
(b) outer dead centre
(c) right angles to the link of the stroke
(d) at 45° to the line of the stroke
(e) all of the above.

12.149. The number of dead centres in a crank driven slider crank mechanism are
(a) 0 (b) 2
(c) 4 (d) 6
(e) may be any number depending upon position of mechanism.

12.150. Corioli's component acts
(a) perpendicular to sliding surfaces
(b) along sliding surfaces
(c) somewhere in between above two
(d) unpredictable
(e) none of the above.

12.151. The sense of Corioli's component is such that it
(a) leads the sliding velocity vector by 90°
(b) lags the sliding velocity vector by 90°
(c) is along the sliding velocity vector
(d) leads the sliding velocity vector by 180°
(e) none of the above.

12.152. Klein's construction can be used when
(a) crank has a uniform angular velocity
(b) crank has non-uniform velocity
(c) crank has uniform angular acceleration
(d) crank has uniform angular velocity and angular acceleration
(e) there is no such criterion.

12.153. Klein's construction is useful to determine
(a) velocity of various parts
(b) acceleration of various parts
(c) displacement of various parts
(d) angular acceleration of various parts
(e) all of the above.

12.154. A circle passing through the pitch point with its centre at the centre of cam axis is known as
(a) pitch circle (b) base circle
(c) prime circle (d) outer circle
(e) cam circle.

12.155. The pressure angle of a cam depends upon
(a) offset between centre lines of cam and follower
(b) lift of follower
(c) angle of ascent
(d) sum of radii of base circle and roller follower
(e) all of the above.

12.156. Cam size depends upon
(a) base circle (b) pitch circle
(c) prime circle (d) outer circle
(e) none of the above.

12.157. Cylindrical cams can be classified as
(a) circular (b) tangent
(c) reciprocating
(d) all of the above
(e) none of the above.

12.158. The maximum value of the pressure angle in case of cam is kept as
(a) 10° (b) 14°
(c) 20° (d) 30°
(e) 25°.

12.159. For the same lift and same angle of ascent, a smaller base circle will give
(a) a small value of pressure angle
(b) a large value of pressure angle
(c) there is no such relation with pressure angle
(d) something else
(e) none of the above is true.

12.160. Cam angle is defined as the angle
(a) during which the follower returns to its initial position
(b) of rotation of the cam for a definite displacement of the follower
(c) through which the cam rotates during the period in which the follower remains in the highest position
(d) moved by the cam from the instant the follower begins to rise, till it reaches its highest position
(e) moved by the cam from beginning of ascent to the termination of descent.

12.161. Angle of descent of cam is defined as the angle
- (*a*) during which the follower returns to its initial position
- (*b*) of rotation of the cam for a definite displacement of the follower
- (*c*) through which the cam rotates during the period in which the follower remains in the highest position
- (*d*) moved by the cam from the instant the follower begins to rise, till it reaches its highest position
- (*e*) moved by the cam from beginning of ascent to the termination of descent.

12.162. Angle of action of cam is defined as the angle
- (*a*) during which the follower returns to its initial position
- (*b*) of rotation of the cam for a definite displacement of the follower
- (*c*) through which the cam rotates during the period in which the follower remains in the highest position
- (*d*) moved by the cam from the instant the follower begins to rise, till it reaches its highest position
- (*e*) moved by the cam from beginning of ascent to the termination of descent.

12.163. Angle of dwell of cam is defined as the angle
- (*a*) during which the follower returns to its initial position
- (*b*) of rotation of the cam for definite displacement of the follower
- (*c*) through which the cam rotates during the period in which the follower remains in the highest position
- (*d*) moved by the cam from the instant the follower begins to rise, till it reaches its highest position
- (*e*) moved by the cam from a beginning of ascent to the termination of descent.

12.164. Angle of ascent of cam is defined as the angle
- (*a*) during which the follower returns to its initial position
- (*b*) of rotation of the cam for a definite displacement of the follower
- (*c*) through which the cam rotates during the period in which the follower remains in highest position
- (*d*) moved by the cam from the instant the follower begins to rise, till it reaches its highest position
- (*e*) moved by the cam from beginning of ascent to the termination of descent.

12.165. The angle at any point on the pitch curve of the cam included between the normal to that point on the curve and line of motion of the follower at that instant is known as
- (*a*) cam angle (*b*) profile angle
- (*c*) pressure angle (*d*) dwell angle
- (*e*) prime angle.

12.166. For S.H.M. cam, the acceleration of the follower at the ends of the stroke and at mid-stroke respectively, is
- (*a*) maximum and zero
- (*b*) zero and maximum
- (*c*) minimum and maximum
- (*d*) zero and minimum
- (*e*) maximum and minimum.

12.167. Throw of a cam is the maximum distance of the follower from
- (*a*) base circle (*b*) pitch circle
- (*c*) root circle (*d*) prime circle
- (*e*) inner circle.

12.168. For simple harmonic motion of the cam follower, a cosine curve represents
- (*a*) displacement diagram
- (*b*) velocity diagram
- (*c*) acceleration diagram
- (*d*) all of the above
- (*e*) none of the above.

12.169. Pitch point on a cam is
- (*a*) any point on pitch curve
- (*b*) the point on cam pitch curve having the maximum pressure angle
- (*c*) any point on pitch circle
- (*d*) the point on cam pitch curve having the minimum pressure angle
- (*e*) none of the above.

12.170. In the scotch yoke mechanism, Corioli's component is
- (*a*) involved
- (*b*) not involved
- (*c*) possible in some position

(*d*) a rare possibility
(*e*) unpredictable.

12.171. In a cam follower motion, jerk is expressed by

(*a*) $\omega \dfrac{d^2y}{d\theta^2}$ (*b*) $\omega \dfrac{d^3y}{d\theta^3}$

(*c*) $\omega^2 \dfrac{d^2y}{d^2\theta}$ (*d*) $\omega^2 \dfrac{d^3y}{d\theta^3}$

(*e*) $\left(\omega \dfrac{dy}{d\theta}\right)^3$.

12.172. A rotating mass having moment of inertia of 30 kgm^2 rotates at 800 rpm and is travelling in a curve of 170 metres radius at a speed of 240 km/hr. It will experience a gyroscopic reaction of
(*a*) 10 m kg_f (*b*) 100 m kg_f
(*c*) 1,000 m kg_f (*d*) 10,000 m kg_f
(*e*) none of the above.

12.173. A cam in which the follower reciprocates or oscillates in a plane parallel to the axis of the cam is known as
(*a*) cylindrical cam
(*b*) circular cam (*c*) reciprocating cam
(*d*) tangent cam (*e*) none of the above.

12.174. In the case of flat pivot bearing, the frictional force in case of uniform pressure can be assumed to be acting at

(*a*) $\dfrac{2}{3}r$ (*b*) $\dfrac{1}{2}r$

(*c*) $\dfrac{3}{4}r$ (*d*) $\dfrac{3}{8}r$

(*e*) $\dfrac{5}{8}r$.

12.175. In the case of flat pivot bearing, the frictional force in case of uniform wear can be assumed to be acting at

(*a*) $\dfrac{2}{3}r$ (*b*) $\dfrac{1}{2}r$

(*c*) $\dfrac{3}{4}r$ (*d*) $\dfrac{3}{8}r$

(*e*) $\dfrac{5}{8}r$.

12.176. In the case of flat collar bearing, the frictional force in case of uniform pressure can be assumed to be acting at

(*a*) $\dfrac{2}{3}\dfrac{r_1^2 - r_2^2}{r_1 - r_2}$ (*b*) $\dfrac{1}{2} \times \dfrac{r_1 + r_2}{2}$

(*c*) $\dfrac{2}{3}\left(\dfrac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right)$ (*d*) $\dfrac{1}{2}(r_1 + r_2)$

(*e*) none of the above.

12.177. In the case of flat collar pivot bearing, the frictional force in case of uniform wear can be assumed to be acting at

(*a*) $\dfrac{2}{3}\dfrac{r_1^2 - r_2^2}{r_1 - r_2}$ (*b*) $\dfrac{1}{2} \times \dfrac{r_1 + r_2}{2}$

(*c*) $\dfrac{2}{3}\left(\dfrac{r_1^2 - r_2^3}{r_1^2 - r_2^2}\right)$ (*d*) $\dfrac{1}{2}(r_1 + r_2)$

(*e*) none of the above.

12.178. Rope brake dynamometer uses
(*a*) oil as lubricant
(*b*) water as lubricant
(*c*) grease as lubricant
(*d*) no lubricant
(*e*) special lubricant.

12.179. The most commonly used dynamometer for tests in the laboratory is
(*a*) rope brake dynamometer
(*b*) prony brake dynamometer
(*c*) froude water vortex dynamometer
(*d*) amsler dynamometer
(*e*) electrical load.

12.180. The following dynamometer is used for power measurement when the speed is high and the viscous force is small
(*a*) tesla fluid friction dynamometer
(*b*) froude water vortex dynamometer
(*c*) rope brake dynamometer
(*d*) amsler dynamometer
(*e*) belt transmission dynamometer.

12.181. For large ranges of power and speed and for accurate measurement of the power, the following dynamometer is used
(*a*) tesla fluid friction dynamometer
(*b*) froude water vortex dynamometer
(*c*) rope brake dynamometer
(*d*) amsler dynamometer
(*e*) belt transmission dynamometer.

12.182. The following dynamometer is widely used for absorption of wide range of powers at wide range of speeds
(*a*) hydraulic (*b*) belt transmission
(*c*) rope brake (*d*) electric generator
(*e*) torsion.

12.183. For measuring powers of machines having high speed and comparatively low outputs, following dynamometer is used
(*a*) tesla fluid friction dynamometer
(*b*) electric generator dynamometer
(*c*) belt transmission dynamometer
(*d*) rope brake dynamometer
(*e*) froude water vortex dynamometer.

12.184. Which of the following is transmission dynamometer
(*a*) rope brake (*b*) electric generator
(*c*) prony brake
(*d*) hydraulic dynamometer
(*e*) none of the above.

12.185. The following type of dynamometer is used when it is desired to measure the large powers of the steam turbine propelling the naval ship
(*a*) epicyclic train dynamometer
(*b*) torsion dynamometer
(*c*) electrical dynamometer
(*d*) belt transmission dynamometer
(*e*) amsler dynamometer.

12.186. The brake commonly used on train boggies is
(*a*) internal expanding
(*b*) band brake
(*c*) band and block brake
(*d*) shoe brake
(*e*) electric brake.

12.187. Pick up the wrong statement. A flywheel
(*a*) is used to limit the inevitable fluctuation of speed during each cycle
(*b*) controls the mean speed of rotation
(*c*) stores up energy and gives up whenever required
(*d*) regulates the speed during one cycle of a prime mover
(*e*) absorbs energy when turning moment is greater than the resisting moment.

12.188. The radius of gyration of a disc type flywheel of diameter D is
(*a*) D (*b*) $D/2$
(*c*) $D/4$ (*d*) $\frac{3}{\sqrt{2}}D$
(*e*) $\frac{\sqrt{2}}{3}D$.

12.189. A flywheel absorbs energy during those periods of crank rotation when the turning moment is greater than the resisting moment. The absorption is
(*a*) at constant speed
(*b*) accompanied by increase in speed
(*c*) accompanied by decrease in speed
(*d*) possible at all speeds
(*e*) not concerned with increase/decrease in speed.

12.190. The contact angle in tapered roller bearings varies between
(*a*) 2 – 5° (*b*) 12 – 16°
(*c*) 5 – 10° (*d*) 28 – 30°
(*e*) 30 – 40°.

12.191. For heavy thrust, the contact angle in tapered roller bearings varies between
(*a*) 28 – 30° (*b*) 5 – 10°
(*c*) 2 – 5° (*d*) 15 – 20°
(*e*) 30 – 40°.

12.192. In a well greased ball bearing, the coefficient of friction may be
(*a*) 0.01 to 0.1 (*b*) 0.1 to 0.25
(*c*) 0.25 to 0.35 (*d*) 0.35 to 0.5
(*e*) 0.5 to 0.65.

12.193. Fluctuation of energy of an engine is the
(*a*) variation of energy above and below the mean resisting torque line
(*b*) ratio of maximum and minimum energies
(*c*) difference between the maximum and minimum energies
(*d*) ratio of the maximum fluctuation of energy to the work done per cycle
(*e*) ratio of maximum fluctuation of speed to the mean speed.

12.194. Maximum fluctuation of energy is the
(*a*) variation of energy above and below the mean resisting torque line
(*b*) ratio of maximum and minimum energies
(*c*) difference between the maximum and minimum energies
(*d*) ratio of the maximum fluctuation of energy to the work done per cycle
(*e*) ratio of maximum fluctuation of speed to the mean speed.

12.195. Coefficient of fluctuation of energy is the
(*a*) variation of energy above and below the mean resisting torque line

(*b*) ratio of maximum and minimum energies
(*c*) difference between the maximum and minimum energies
(*d*) ratio of the maximum fluctuation of energy to the work done per cycle
(*e*) ratio of maximum fluctuation of speed to the mean speed.

12.196. Coefficient of fluctuation of speed is the
(*a*) variation of energy above and below the mean resisting torque line
(*b*) ratio of maximum and minimum energies
(*c*) difference between the maximum and minimum energies
(*d*) ratio of the maximum fluctuation of energy to the work done per cycle
(*e*) ratio of maximum fluctuation of speed to the mean speed.

12.197. Maximum fluctuation of energy in a flywheel is
(*a*) mass M.I. of flywheel × (mean angular speed × difference of maximum and minimum speed)
(*b*) mass M.I. of flywheel × (mean angular speed) × co-efficient of fluctuation of speed
(*c*) 2 × maximum fluctuation of energy × co-efficient of fluctuation of speed
(*d*) all of the above
(*e*) none of the above.

12.198. In the rim type of flywheel, the major mass is
(*a*) concentrated around the periphery
(*b*) concentrated at the centre
(*c*) contributed due to arms
(*d*) ineffective
(*e*) balanced by centripetal forces.

12.199. The supply of working fluid to the engine to suit the load conditions is controlled by
(*a*) Meyer's expansion valve
(*b*) D-slide valve
(*c*) flywheel
(*d*) governor
(*e*) throttle valve.

12.200. The speed variations of the engine caused by the fluctuation of engine turning moment are controlled by
(*a*) Meyer's expansion valve
(*b*) D-slide valve (*c*) flywheel
(*d*) governor (*e*) throttle valve.

12.201. For same lift of sleeve, range of speed of Proell governor as compared to Porter governor is
(*a*) less (*b*) more
(*c*) equal (*d*) half
(*e*) double.

12.202. In a Hartnell governor, if the stiffness of spring is increased, governor will become
(*a*) more sensitive (*b*) less sensitive
(*c*) hunting (*d*) isochronous
(*e*) sensitivity remains unchanged.

12.203. The governor used in gramophone is of the following type
(*a*) Pickening (*b*) Porter
(*c*) hartnell (*d*) watt
(*e*) hartung.

12.204. For a governor running at constant speed, the force acting on the sleeve is
(*a*) constant (*b*) minimum
(*c*) maximum (*d*) zero
(*e*) variable depending on the load.

12.205. Hartnell governor could be classified under the head of
(*a*) inertia type governors
(*b*) pendulum type governors
(*c*) centrifugal type governors
(*d*) dead weight type governors
(*e*) none of the above.

12.206. A porter governor could be classified as
(*a*) inertia type governor
(*b*) pendulum type governor
(*c*) centrifugal type governor
(*d*) dead weight type governor
(*e*) none of the above.

12.207. A watt governor could be classified as
(*a*) inertia type governor
(*b*) pendulum type governor
(*c*) centrifugal type governor
(*d*) dead weight type governor
(*e*) none of the above.

12.208. The quality of a governor can be judged by its
(*a*) stability
(*b*) sensitivity
(*c*) effort and power
(*d*) all of the above
(*e*) none of the above.

12.209. Sensitiveness of governor is defined as

(*a*) $\frac{\text{range of speed}}{\text{mean speed}}$

(*b*) $\frac{\text{mean speed}}{\text{range of speed}}$

(*c*) mean speed × range of speed

(*d*) $\frac{2 \times \text{mean speed}}{\text{range of speed}}$

(*e*) $\frac{\text{range of speed}}{2 \times \text{mean speed}}$.

12.210. Which of the following is spring controlled governor

(*a*) hartnell (*b*) hartung
(*c*) pickering (*d*) wilson-hartnell
(*e*) all of the above.

12.211. The function of a governor is to

(*a*) store energy and give up whenever required
(*b*) regulate the speed during one cycle of a prine mover
(*c*) decrease variation of speed
(*d*) increase variation of speed
(*e*) adjust variatión of speed by varying the input to the engine.

12.212. The height of watt's governor is proportional to

(*a*) speed (N) (*b*) N^2
(*c*) $1/N$ (*d*) $1/N^2$
(*e*) $1/\sqrt{N}$.

12.213. Centrifugal type governor is preferred to the inertia type governor because

(*a*) former has low initial cost
(*b*) former consumes no power
(*c*) latter results in difficulties of balancing inertia forces
(*d*) latter has less controlling force
(*e*) latter is highly sensitive type.

12.214. If the controlling force of a governor increases with increase in speed, the governor is said to be

(*a*) sensitive (*b*) insensitive
(*c*) isochronous (*d*) powerful
(*e*) unstable.

12.215. If the controlling force of a spring controlled governor decreases with increase in radius of rotation then governor is said to be

(*a*) sensitive (*b*) insensitive
(*c*) isochronous (*d*) powerful
(*e*) unstable.

12.216. Practically the sensitiveness of Watt and Porter governor are

(*a*) same (*b*) half
(*c*) double (*d*) 4 times
(*e*) none of the above.

12.217. The term "effort of governor" refers to

(*a*) centrifugal force of balls
(*b*) useful power developed
(*c*) force acting on sleeve for given % change of speed
(*d*) minimum force required on sleeve for % change of speed
(*e*) none of the above.

12.218. The speed range suitable for Watt's governor is

(*a*) 20—50 rpm (*b*) 60—80 rpm
(*c*) 80—125 rpm (*d*) 125—250 rpm
(*e*) 250—500 rpm.

12.219. For spring controlled governors the controlling force curve would be

(*a*) straight line (*b*) circle
(*c*) parabola (*d*) hyperbola
(*e*) unpredictable.

12.220. If the controlling force line for a spring controlled governor when produced intersects the y-axis at the origin, then governor is said to be

(*a*) stable (*b*) unstable
(*c*) isochronous (*d*) sensitive
(*e*) powerful.

12.221. For isochronous, spring controlled governor, the controlling force with increase in radius of rotation

(*a*) increase
(*b*) decreases
(*c*) remains constant
(*d*) behaves in unpredictable way
(*e*) may increase or decrease depending on size.

12.222. If ϕ be the angle of friction for a square threaded screw, then maximum efficiency of the screw-jack will be

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

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

(*e*) $\frac{\sin\phi}{1-\sin\phi}$.

12.223. In above problem, maximum efficiency of screw jack for square threads will occur when the angle of threads is equal to

(*a*) $\phi/2$ (*b*) $\frac{\pi}{2}-\frac{\phi}{2}$

(*c*) $\frac{\pi}{4}-\frac{\phi}{2}$ (*d*) $\frac{\pi}{2}-\phi$

(*e*) $\frac{\pi}{4}+\frac{\phi}{2}$.

12.? The maximum efficiency of a screw jack having square threads and friction angle of 30° will be

(*a*) 11% (*b*) 22%
(*c*) 30% (*d*) 33%
(*e*) 50%.

12.225. For a machine to be self-sustaining

(*a*) $\alpha = \phi$ (*b*) $\alpha > \phi$
(*c*) $\alpha < \phi$ (*d*) unpredic table
(*e*) none of the above.

where α = slope of threads
ϕ = angle of friction.

12.226. Which of the following clutches is positive type

(*a*) cone (*b*) disc
(*c*) jaw (*d*) centrifugal
(*e*) hydraulic.

12.227. Which of the following is not a flexible coupling

(*a*) universal (*b*) bushed pin
(*c*) muff (*d*) Oldham's
(*e*) none of the above.

12.228. The moment of friction with assumption of uniform pressure compared to uniform wear is

(*a*) same
(*b*) greater
(*c*) lower
(*d*) could be anything
(*e*) none of the above.

12.229. Length of cross belt, in addition to centre length, depends

(*a*) only on the sum of the radii of pulleys
(*b*) on the sum and difference of the radii of the pulleys
(*c*) square of difference of radii of the pulleys
(*d*) square of sum of radii of pulleys
(*e*) none of the above.

12.230. The coefficient of friction between pulley and belt is reduced by 50%. If initial ratio of tension in belt was 5 then new value will be

(*a*) 5 (*b*) 2.5
(*c*) 10 (*d*) 1.25
(*e*) none of the above.

12.231. Length of open belt, in addition to centre length, depends

(*a*) only on the sum of the radii of pulleys
(*b*) on the sum and difference of the radii of the pulleys
(*c*) square of difference of radii of pulleys
(*d*) square of sum of radii of pulleys
(*e*) none of the above.

12.232. Abbreviation P.I.V. drive stands for

(*a*) positive, infinitely variable drive
(*b*) positive, independently variable drive
(*c*) purely, incremental variable drive
(*d*) purely, integral variable drive
(*e*) all of the above.

12.233. The centrifugal tension in belts

(*a*) reduces power transmission
(*b*) increases power transmission
(*c*) does not affect power transmission
(*d*) increases power transmission upto certain speed and then decreases
(*e*) none of the above.

12.234. The belting can transmit maximum power when maximum total tension in belt equals

(*a*) twice the centrifugal tension
(*b*) thrice the centrifugal tension
(*c*) four times the centrifugal tension
(*d*) centrifugal tension
(*e*) half the centrifugal tension.

12.235. Pitching of a ship produces forces on the bearings

(*a*) in the direction of motion of ship
(*b*) which act horizontally perpendicular to the motion of ship
(*c*) in the plane of the pitching
(*d*) which act along the axis of the bearings
(*e*) which can be resolved into all the three components.

12.236. In the case of gyroscopic effect, the planes of spin, gyroscopic couple and precession are

(a) in same plane
(b) in different planes
(c) any two in perpendicular planes and third in different plane
(d) mutually perpendicular
(e) unpredictable.

12.237. It is possible to obtain unity velocity ratio at every instant in double Hooke's joint, if
(a) axes of driving and driven shafts are in same plane
(b) axes of driving and driven shafts are in different planes
(c) intermediate shaft makes equal angles with driving and driven shafts
(d) (a) and (c)
(e) (b) and (c).

12.238. The contact surfaces in a single plate clutch are usually lined with leather, cork etc. in order to
(a) enable quick replacement of worn parts
(b) increase friction force
(c) increase power transmitted
(d) reduce slip
(e) none of the above.

12.239. The maximum permissible velocity of the belt is given by
(a) $\sqrt{Tg/3w}$ (b) $\sqrt{2Tg/3w}$
(c) $\sqrt{3Tg/2w}$ (d) $\sqrt{3Tg/w}$
(e) $\sqrt{T/3wg}$.
where T = maximum tension
and w = weight per metre length of belt.

12.240. The power transmitted by a belt is maximum when the maximum tension in the belt compared to centrifugal tension is
(a) 2 times (b) 3 times
(c) 4 times (d) 2.5 times
(e) 3.5 times.

12.241. Initial tension in belts, when stationary, is
(a) T_1 (b) T_2
(c) $T_1 + T_2$ (d) $\frac{T_1 + T_2}{2}$
(e) $T_1 - T_2$.

12.242. Can simple band brake be made self-energising type
(a) yes (b) no
(c) with lot of sophistication
(d) it may not be economical
(e) none of the above.

12.243. Which is false statement about flywheel
(a) flywheel smoothens the cyclic fluctuation of speed when delivering constant output h.p.
(b) it has no influence on the mean speed of the prime mover
(c) it takes care of output fluctuations and controls input accordingly
(d) it has no influence over the varying load demand on prime mover
(e) it several times acts as pulley.

12.244. Which is false statement above govenor
(a) it has no influence on mean speed of the prime mover
(b) it has no influence over the cyclic speed fluctuation
(c) it adjusts supply energy of prime mover with varying output
(d) it controls mean speed over a period for output load variations by manipulating input energy
(e) it usually employs centrifugal force type speed sensors.

12.245. The frictional torque transmitted in a conical or flat pivot bearing assuming uniform pressure in comparison to assumption of uniform wear is
(a) same (b) more
(c) less
(d) more or less depending on load to be transmitted
(e) unpredictable.

12.246. If T_1 and T_2 be the tensions in kg on tight and slack sides of a belt and v be its velocity in m/sec, then h.p. transmitted is equal to
(a) $\frac{(T_1 - T_2)v}{4500}$ (b) $\frac{T_1 v}{75}$
(c) $\frac{(T_1 - T_2)v}{75}$ (d) $\frac{(T_1 - T_2)v}{5500}$
(e) $\frac{(T_1 - T_2)v}{3300}$.

12.247. The ratio of number of teeth and pitch circle diameter is called
(a) pitch (b) circular pitch
(c) diametral pitch
(d) module (e) addendum.

12.248. Which of the following does not change with the conditions of the mating gears
(*a*) pitch circle diameter
(*b*) base circle
(*c*) pressure angle
(*d*) all of the above
(*e*) none of the above.

12.249. The circle passing through the bottom of the teeth of gear is known as
(*a*) inner circle (*b*) prime circle
(*c*) base circle (*d*) addendum circle
(*e*) dedendum circle.

12.250. Pitch circle diameter of an involute gear is
(*a*) independent of any other element
(*b*) dependent of pressure angle
(*c*) constant for a set of meshing gears
(*d*) proportional to base diameter
(*e*) most important element when manufacturing a gear.

12.251. Intermediate gears are used for
(*a*) obtaining rotation in desired direction
(*b*) reducing the size of the individual gear
(*c*) bridging the gap between the first and last wheels of the train
(*d*) driving auxiliaries incidental to the main drive
(*e*) any one of the above.

12.252. In an involute gear, the normal to the involute is tangent to the
(*a*) pitch circle (*b*) base circle
(*c*) addendum circle
(*d*) dedendum circle
(*e*) average of addendum and dedendum circles.

12.253. Gears are considered to be medium velocity type if their peripheral velocity lies in the range of
(*a*) 1—3 m/sec (*b*) 3—15 m/sec
(*c*) 15—25 m/sec (*d*) 25—50 m/sec
(*e*) none of the above.

12.254. The surface of the gear tooth below the pitch surface is called
(*a*) bottom tooth (*b*) face
(*c*) flank (*d*) dedendum portion
(*e*) tooth depth.

12.255. Which is correct
(*a*) $\text{inv}(\phi) = \tan\phi - \phi$
(*b*) $\text{inv}(\phi) = \tan(\phi - 1)$
(*c*) $\text{inv}(\phi) = \tan\phi - 1$
(*d*) $\text{inv}(\phi) = \phi - \tan\phi$
(*e*) $\text{inv}(\phi) = 1 - \tan\phi$

12.256. The transverse section of a helical gear is identical to
(*a*) bevel gear (*b*) spur gear
(*c*) worm gear (*d*) all of the above
(*e*) none of the above.

12.257. According to law of gearing
(*a*) teeth should be of involute type
(*b*) clearance between mating teeth should be provided
(*c*) dedendum should be equal to 1.157 m
(*d*) teeth should be of cycloidal type
(*e*) none of these.

12.258. Bevel gears are used to transmit rotary motion between two shafts whose axes are
(*a*) parallel (*b*) non-intersecting
(*c*) non-coplaner (*d*) any of the above
(*e*) none of the above.

12.259. Which is incorrect statement about gears
(*a*) pitch circle is always bigger than base circle
(*b*) angular velocity ratio is inversely proportional to radius of base circle from which the involute is generated
(*c*) addendum is bigger than dedendum
(*d*) PCD = module × No. of teeth.
(*e*) dedendum is bigger than addendum.

12.260. If D and T be the pitch circle diameter and no. of teeth of a gear, then its circular pitch $p =$
(*a*) D/T (*b*) T/D
(*c*) $\pi \frac{T}{D}$ (*d*) $\pi \frac{D}{T}$
(*e*) $\frac{\pi D}{2T}$.

12.261. If ϕ = friction angle and α is shaft angle then maximum efficiency of spiral gears is
(*a*) $\frac{\cos(\alpha + \phi) + 1}{\cos(\alpha - \phi) + 1}$
(*b*) $\frac{\cos(\alpha - \phi) - 1}{\cos(\alpha - \phi) - 1}$
(*c*) $\frac{\cos(\alpha + \phi)}{\cos(\alpha - \phi)}$

(d) $\dfrac{\cos(\alpha-\phi)+1}{\cos(\alpha+\phi)+1}$

(e) $\dfrac{\sin(\alpha+\phi)+1}{\sin(\alpha-\phi)+1}$.

12.262. Law of gearing is satisfied if
(a) two surfaces slide smoothly
(b) common normal at the point of contact passes through pitch point on the line joining the centres of rotation
(c) number of teeth $= \dfrac{PCD}{\text{module}}$
(d) addendum is greater than dedendum
(e) none of the above.

12.263. If D_1 and T_1 be the diameter and no. of teeth of gear 1 and D_2 and T_2 the corresponding values of other gear in mesh, then speed ratio N_1/N_2 will be equal to
(a) D_1/D_2 (b) T_1/T_2
(c) $\dfrac{D_1}{D_2}\times\dfrac{T_2}{T_1}$ (d) $\dfrac{D_2}{D_1}$
(e) $\dfrac{D_1}{D_2}\times\dfrac{T_1}{T_2}$.

12.264. The path of contact in involute gears is
(a) a straight line (b) involute path
(c) curved line (d) circle
(e) cycloidal.

12.265. The distance measured parallel to the axis to represent the distance advanced by each tooth per revolution is known as
(a) pitch (b) axial pitch
(c) normal pitch (d) base pitch
(e) lead.

12.266. The pressure angle for involute gears is
(a) variable (b) always constant
(c) dependent on type of meshing
(d) dependent on size of teeth
(e) never constant.

12.267. The centre distance between two meshing involute gears is
(a) $\dfrac{\text{sum of base circle radii}}{\cos(\text{pressure angle }\phi)}$
(b) $\dfrac{\text{sum of base circle diameters}}{\cos\phi}$
(c) $\dfrac{\text{sum of base circle radii}}{\sin\phi}$
(d) $\dfrac{\text{sum of pitch circle radii}}{\cos\phi}$
(e) $\dfrac{\text{sum of outer circle radii}}{\cos\phi}$

12.268. Contact ratio for gears is the ratio of length of arc of contact and the
(a) circular pitch
(b) cos (pressure angle ϕ)
(c) $\sin\phi$ (d) $\tan\phi$
(e) $1-\cos\phi$.

12.269. Dedendum circle diameter is the product of $\cos\phi$ and
(a) pitch circle diameter
(b) base circle diameter
(c) addendum circle diameter
(d) length of arc of contact
(e) contact ratio.

12.270. For a cycloidal tooth profile, pressure angle at (i) commencement of engagement, (ii) pitch point and at (iii) end of engagement will be
(a) constant
(b) zero, maximum, zero
(c) max., zero, max.
(d) max., zero, zero
(e) zero, zero, max.

12.271. Which is false statement about the properties of involute profile
(a) the shape of involute profile is dependent only on the dimensions of base circle
(b) the angular velocity ratio when two involutes are in mesh, is directly proportional to the size of the base circles
(c) involute is the only tooth from that is not sensitive to centre distance of their base circles
(d) basic rack for involute tooth profile has straight line form
(e) involute profile is generated by the locus of a point on a thread unwound from a cylinder.

12.272. The gear train in which the first and last gear are on the same axis, is known as
(a) uniaxial gear train
(b) simple gear train
(c) compound gear train
(d) epicyclic gear train
(e) reverted gear train.

12.273. In involute teeth, normal to the involute is tangent to

(a) the pitch circle
(b) the base circle
(c) the pitch circle diameter
(d) tooth profile
(e) pitch point.

12.274. The minimum number of teeth on a gear with $14\frac{1}{2}°$ pressure angle will be
(a) 11 (b) 17
(c) 25 (d) 32
(e) 42.

12.275. Mittre gears are used for
(a) great speed reduction
(b) transmitting motion between two intersecting shafts
(c) equal speed
(d) minimum axial thrust
(e) minimum backlash.

12.276. The difference between dedendum ana addendum is known as
(a) backless (b) clearance
(c) flank (d) tooth space
(e) module.

12.277. Larger pressure angle results in
(a) wider base and stronger teeth
(b) weaker teeth
(c) little pulsating motion
(d) bigger size of gear
(e) smaller size of gear.

12.278. The contact ratio or engagement factor in case of gears should be
(a) less than 1 (b) 1
(c) 1.3 to 1.5 (d) 1.8 to 2.0
(e) 2.0 to 2.5.

12.279. Which of the following is not the gear for non-parallel, non-intersecting shafts
(a) cross (b) helical
(c) bevel (d) worm
(e) hypoid.

12.280. The product of circular pitch and diameteral pitch is equal to
(a) module (b) unity
(c) π (d) $\frac{1}{\pi}$
(e) $\pi \times$ module.

12.281. Which is false statement about cycloidal gears
(a) interference exists
(b) complex profile and less flexible
(c) have spreading flanks
(d) less wear
(e) ease in transmitting pulsating loads.

12.282. Bevel gears have their teeth
(a) straight over the wheel rim
(b) inclined to wheel rim
(c) curved over the wheel rim
(d) cut on the surfaces of the frusta of cones
(e) none of the above.

12.283. Helical gears have their teeth
(a) straight over the wheel rim
(b) inclined to wheel rim
(c) curved over the wheel rim
(d) cut on the surfaces of the frusta of cones
(e) none of the above.

12.284. Which is false statement about involute gears
(a) pressure angle varies from zero at pitch to maximum at commencement and end of engagement
(b) slight variation in centre distance can be tolerated
(c) simple tools required for manufacture
(d) smooth running
(e) are most commonly used.

12.285. The centre distance between involute gears is a function of the base circle radii of the meshing gears and
(a) the pressure angle
(b) No. of teeth
(c) pitch circle diameter
(d) speed
(e) nothing else.

12.286. Best profile to obtain resistance against wear is
(a) $14\frac{1}{2}°$ rack
(b) $14\frac{1}{2}°$ full depth involute
(c) $14\frac{1}{2}°$ involute stub
(d) 20° full depth involute
(e) 20° involute stub.

12.287. For a given total arc of action, the work wasted in friction will be least when arc of approach is

(*a*) greater than arc of recess
(*b*) equal to arc of recess
(*c*) less than arc of recess
(*d*) arc of recess has no relation with arc of approach
(*e*) none of the above.

12.288. The maximum efficiency in case of worm and worm wheel is

(*a*) $\frac{1-\sin\phi}{1+\sin\phi}$ (*b*) $\frac{1-\sin\phi}{\sin\phi}$
(*c*) $\frac{1+\sin\phi}{1-\sin\phi}$ (*d*) $\frac{\sin\phi}{1-\sin\phi}$
(*e*) none of the above
where ϕ is friction angle.

12.289. The maximum efficiency of spiral gears with shaft angle α and pressure angle ϕ is equal to

(*a*) $\frac{1+\cos(\alpha+\phi)}{1+\cos(\alpha-\phi)}$
(*b*) $\frac{1-\cos(\alpha+\phi)}{1-\cos(\alpha-\phi)}$
(*c*) $\frac{\cos\alpha}{1+\cos\phi}$ (*d*) $\frac{\cos\alpha}{\cos\phi}$
(*e*) $\frac{\sin\alpha}{1+\sin\phi}$.

12.290. A gear having 100 teeth is fixed and another gear having 25 teeth revolves around it, the centre lines of both gears being joined by an arm. How many revolutions will be made by gear of 25 teeth for one revolution of arm

(*a*) 4 (*b*) 3
(*c*) 5 (*d*) 6
(*e*) 10.

12.291. If the force transmitted between two meshing gears is *F* and if the pitch circle speed is v m/mt, then h.p. transmitted will be

(*a*) *F*v/75 (*b*) *F*v/330
(*c*) *F*v/4500 (*d*) *F*v/33,000
(*e*) *F*v/102.

12.292. The interference or undercutting in involute gears can be avoided by

(*a*) varying the centre distance by changing pressure angle
(*b*) using modified involute or composite system
(*c*) increasing the addendum of small wheel and reducing it for the larger wheel
(*d*) any one of the above
(*e*) none of the above.

12.293. The cranks of locomotives with two cylinders, in order to facilitate starting of locomotive in any position, are invariably placed

(*a*) in same plane
(*b*) at 90° to each other
(*c*) at 180° to each other
(*d*) at 45° to each other
(*e*) may be placed anywhere.

12.294. Secondary forces in reciprocating mass on engine frame are

(*a*) of same frequency as of primary forces
(*b*) twice the frequency as of primary forces
(*c*) four times the frequency as of primary forces
(*d*) half the frequency as of primary forces
(*e*) none of the above.

12.295. In reciprocating engines, primary forces are

(*a*) completely balanced
(*b*) partially balanced
(*c*) can't be balanced
(*d*) balanced by secondary forces
(*e*) none of the above.

12.296. In locomotives, the ratio of length of connecting rod to crank radius is kept very large in order to

(*a*) facilitate quick starting
(*b*) minimise primary forces
(*c*) minimise the effect of secondary forces
(*d*) achieve perfect balancing
(*e*) minimise swaying couple.

12.297. Partial balancing in locomotives results in

(*a*) hammer blow, variation of tractive effort, swaying couple
(*b*) least wear
(*c*) most smooth operation
(*d*) better performance of engine
(*e*) none of the above.

12.298. A disc oscillates freely at the end of a shaft, the other end of which is fixed. With increase in shaft stiffness, the natural frequency of vibration will

(*a*) increase (*b*) decrease
(*c*) remain same (*d*) unpredictable
(*e*) none of the above.

12.299. Pick up the wrong statement. Viscous force required to cause a plate to slide over a parallel plate is proportional to
(*a*) coefficient of viscosity of lubricant separating two plates
(*b*) area of plate
(*c*) velocity of sliding
(*d*) lubricant thickness
(*e*) all of the above.

12.300. In hydrostatic bearing, pressure to lubricant is supplied by
(*a*) external source
(*b*) partially external and partially from rotation of journal
(*c*) not supplied by external source
(*d*) shaft driven pump
(*e*) none of the above

12.301. Swaying couple results due to
(*a*) primary disturbing force
(*b*) secodary disturbing force
(*c*) partial balancing
(*d*) use of two cylinders
(*e*) hammer blow.

12.302. The effect of swaying couple is resisted by
(*a*) the side pressure between the flanges of the tyres of the wheel and the inside of the rails
(*b*) cylinders
(*c*) dead weight on the wheels
(*d*) balancing weight in the wheels
(*e*) is not resisted but acts freely.

12.303. Purpose of using differential gear in automobile is to
(*a*) control speed
(*b*) avoid jerks
(*c*) help in turning
(*d*) obtain rear movement
(*e*) none of the above.

12.304. In order to balance the reciprocating masses
(*a*) primary and secondary forces must be balanced
(*b*) primary couple must be balanced
(*c*) secondary couple must be balanced
(*d*) all of the above
(*e*) none of the above.

12.305. Partial balancing means
(*a*) partially balance the revolving masses
(*b*) partially balance the reciprocating masses
(*c*) best balancing of engines
(*d*) all of the above
(*e*) none of the above.

12.306. Gear box of a car utilises
(*a*) compound train
(*b*) simple train (*c*) epicyclic gears
(*d*) complex train (*e*) none of the above.

12.307. If a more stiff spring is used in Hartnell governor, then the governor will be
(*a*) more sensitive
(*b*) less sensitive
(*c*) sensitivity remains unaffected
(*d*) isochronous
(*e*) none of the above.

12.308. A spring controlled governor will be stable if the straight control line force curve when produced will intersect y-axis
(*a*) at origin (*b*) below x-axis
(*c*) above x-axis
(*d*) there is no such criterion
(*e*) none of the above.

12.309. The motion of a pendulum is S.H.M. only when its amplitude is
(*a*) small
(*b*) large
(*c*) equal to length of pendulum
(*d*) may have any value
(*e*) a particular value.

12.310. The periodic time of a simple pendulum depends on
(*a*) size of bob
(*b*) mass of bob
(*c*) material of bob
(*d*) amplitude of swing
(*e*) length of pendulum.

12.311. The period of a simple pendulum can be doubled by increasing the length
(*a*) 2 times (*b*) 1/2 times
(*c*) 4 times (*d*) 1/4 times
(*e*) keeping it constant and doubling the mass of bob.

12.312. The acceleration in S.H.M. is proportional to
(*a*) length of pendulum
(*b*) time period
(*c*) angular velocity
(*d*) displacement
(*e*) all of the above.

12.313. Frequency of oscillation of a compound pendulum having same distance between point of suspension and c.g. of mass as a simple pendulum will be
(*a*) same
(*b*) more
(*c*) less
(*d*) depends on other factors
(*e*) none of the above.

12.314. The period of oscillation in case of S.H.M. in terms of angular velocity ω is equal to
(*a*) $\frac{2\pi}{\omega}$ (*b*) $\frac{\omega}{\pi}$
(*c*) $\frac{2\omega}{\pi}$ (*d*) $\frac{\omega}{2\pi}$
(*e*) $\sqrt{2\frac{\omega}{\pi}}$.

12.315. The maximum efficiency of worm and gear wheel is
(*a*) dependent on whether worm is the driver
(*b*) dependent on whether gear wheel is the driver
(*c*) independent of whether the worm or gear wheel is the driver
(*d*) there is no such criterion
(*e*) none of the above.

12.316. The natural frequency for the beam shown in Fig. 12.1 will be

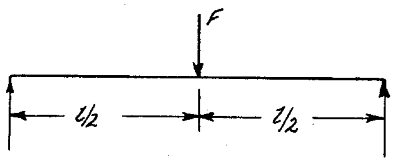

Fig. 12.1.

(*a*) $\frac{1}{2\pi}\sqrt{\frac{48EIG}{Fl^3}}$ (*b*) $2\pi\sqrt{\frac{48EI}{l^4}}$
(*c*) $\frac{1}{2\pi}\sqrt{\frac{l^4}{48EI}}$ (*d*) $2\pi\sqrt{\frac{l^4}{48EI}}$
(*e*) $\frac{1}{2\pi}\sqrt{\frac{48EIg}{Fl^4}}$

12.317. The natural frequency for the beam shown in Fig. 12.2 will be

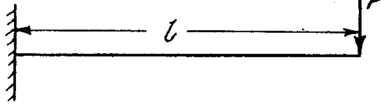

Fig. 12.2.

(*a*) $\frac{1}{2\pi}\sqrt{\frac{l^3}{3EI}}$ (*b*) $\frac{1}{2\pi}\sqrt{\frac{l^3}{8EI}}$
(*c*) $\frac{1}{2\pi}\sqrt{\frac{8EI}{Fl^3}}$ (*d*) $\frac{1}{2\pi}\sqrt{\frac{3EIg}{Fl^3}}$
(*e*) $\frac{1}{2\pi}\sqrt{\frac{3EI}{Fl^6}}$

12.318. The motion of a particle from one extremity to other executes following number of oscillations
(*a*) one full oscillation
(*b*) 2 oscillations (*c*) half oscillation
(*d*) 4 oscillation
(*e*) one-fourth oscillation.

12.319. The critical speed depends on
(*a*) mass (*b*) stiffness
(*c*) mass and stiffness
(*d*) mass, stiffness and eccentricity
(*e*) none of the above.

12.320. The factor which affects critical speed of shaft is
(*a*) eccentricity (*b*) span of shaft
(*c*) diameter of disc
(*d*) all of the above
(*e*) none of the above.

12.321. The primary disturbing force due to inertia of reciprocating parts in an engine is equal to
(*a*) $\frac{W}{g}\omega^2 r\cos\theta$ (*b*) $\frac{W}{g}\omega^2 r\frac{\cos 2\theta}{n}$
(*c*) $\frac{W}{g}\omega^2 r\cos 2\theta$ (*d*) $\frac{W}{g}\omega r\frac{\cos 2\theta}{n}$
(*e*) $\frac{W}{g}\omega^2 r\sin 2\theta$.

12.322. The maximum magnitude of the unbalanced force in a line perpendicular to the line of stroke is known as
(*a*) swaying couple
(*b*) hammer blow

(*c*) variation in tractive effort
(*d*) unbalanced force
(*e*) resultant force.

12.323. The secondary disturbing force due to inertia of reciprocating parts in an engine is equal to
(*a*) $\frac{W}{g}\omega^2 r\cos\theta$ (*b*) $\frac{W}{g}\omega^2 r\frac{\cos 2\theta}{n}$
(*c*) $\frac{W}{g}\omega^2 r\cos 2\theta$ (*d*) $\frac{W}{g}\omega r\frac{\cos 2\theta}{n}$
(*e*) $\frac{W}{g}\omega^2 r\sin 2\theta$.

2.324. In locomotives, the effect of secondary forces is proportional to weight of reciprocating parts transferred to rotational mass at a radius of
(*a*) crank radius (r)
(*b*) connecting rod length (l)
(*c*) r/l (*d*) $r^2/4l$
(*e*) r^2/l.

12.325. The acceleration of the piston of a reciprocating engine is equal to
(*a*) $\omega^2 r\sin\left(\sin\theta+\frac{\sin 2\theta}{n}\right)$
(*b*) $\omega^2 r\left(\sin\theta+\frac{\cos 2\theta}{n}\right)$
(*c*) $\omega^2 r\left(\cos\theta+\frac{\cos 2\theta}{n}\right)$
(*d*) $\omega^2 r\left(\cos\theta+\cos\frac{2\theta}{n}\right)$
(*e*) $\omega^2 r\left(\cos\theta+\frac{\cos^2 2\theta}{n}\right)$

12.326. The time period of a simple pendulum when oscillating in water in comparison to when oscillating in air will be
(*a*) more (*b*) less
(*c*) same
(*d*) could be more or less depending upon its length
(*e*) unpredictable.

12.327. The time for complete oscillation of a simple pendulum when its amplitude is increased will
(*a*) increase (*b*) decrease
(*c*) remain same
(*d*) may increase or decrease depending on length
(*e*) unpredictable.

12.328. If the primary direct crank of a reciprocating engine is located at θ° anticlockwise, then secondary direct crank will be located
(*a*) θ° clockwise (*b*) 2 θ° clockwise
(*c*) 2θ° anticlockwise
(*d*) θ° anticlockwise
(*e*) none of the above.

12.329. In balancing of reciprocating cylinders, the resultant unbalanced force will be minimum when .... of the reciprocating masses are balanced by rotating masses (Fill in the gap by correct answer from following)
(*a*) full (*b*) half
(*c*) one-quarter (*d*) two-thirds
(*e*) one-third.

12.330. Usually following fraction of reciprocating masses, is balanced in case of reciprocating engines
(*a*) full (*b*) half
(*c*) one-quarter (*d*) two-thirds
(*e*) one-third.

12.331. In order to facilitate the starting of locomotive in any position, the cranks of a locomotive with two cylinders are placed at following angle to each other
(*a*) 45° (*b*) 90°
(*c*) 135° (*d*) 180°
(*e*) 225°.

12.332. Hammer blow occur when the c.g. of the balance weight is
(*a*) directly above the wheel centre
(*b*) directly below the wheel centre
(*c*) directly above or below the wheel centre
(*d*) at angle θ
(*e*) perpendicular to wheel vertical plane.

12.333. Partial balancing in locomotive results in
(*a*) hammer blow
(*b*) variation in tractive effort
(*c*) swaying couple
(*d*) all of the above
(*e*) none of the above.

12.334. Primary forces are usually balanced
(*a*) fully
(*b*) partially, half
(*c*) partially, two-thirds

(*d*) partially, one-third
(*e*) partially, three-fourths.

12.335. The principle of direct and reverse cranks is readily applicable to
(*a*) primary balance
(*b*) secondary balance
(*c*) balancing of in-line engines
(*d*) balancing of radial engines
(*e*) partial primary balance.

12.336. The effect of hammer blow can be reduced by
(*a*) fully balancing the reciprocating masses
(*b*) decreasing the speed
(*c*) using two or three pairs of wheels coupled together
(*d*) all of the above
(*e*) (*b*) and (*c*) above.

12.337. The value of swaying couple is maximum or minimum when the crank is inclined at following angle with the line of stroke
(*a*) 135° and 45° (*b*) 135° and 90°
(*c*) 225° and 135° (*d*) 225° and 90°
(*e*) 225° and 45°.

12.338. The velocity of the piston of a reciprocating engine is equal to
(*a*) $\omega r \left(\cos\theta + \frac{\cos 2\theta}{n} \right)$
(*b*) $\omega r \left(\sin\theta + \frac{\sin 2\theta}{n} \right)$
(*c*) $\omega r \left(\sin\theta + \sin\frac{2\theta}{n} \right)$
(*d*) $\omega r \left(\frac{\sin\theta + \sin 2\theta}{n} \right)$
(*e*) $\omega r \left(\sin\theta + \cos\frac{2\theta}{n} \right)$.

12.339. In balancing of coupled locomotive one has to consider following planes
(*a*) two planes of cylinders
(*b*) two planes of coupling rods
(*c*) two planes of driving wheels containing balance weights
(*d*) all of the above
(*e*) none of the above.

12.340. If damping factor in a vibration system is unity, then the system will be
(*a*) having no vibrations
(*b*) highly damped
(*c*) under damped
(*d*) critically damped
(*e*) none of the above.

12.341. The secondary critical speed of a shaft occurs at
(*a*) twice the speed of primary critical speed
(*b*) half the speed of primary critical speed
(*c*) tour times the speed of primary critical speed
(*d*) one-fourth the speed of primary critical speed
(*e*) unpredictable.

12.342. The shaft of a steam turbine is usually rotated at
(*a*) natural frequency of vibration
(*b*) much blow the natural frequency of vibration
(*c*) much above the natural frequency of vibration
(*d*) there is no such criterion
(*e*) none of the above.

12.343. For 20° pressure angle, minimum number of teeth on gear will be
(*a*) 6 (*b*) 12
(*c*) 17 (*d*) 20
(*e*) 24.

12.344. Maximum fluctuation of energy of a flywheel is proportional to
(*a*) difference of maximum and minimum speed of flywheel
(*b*) sum of maximum and minimum speeds of flywheel
(*c*) difference of square of maximum and minimum speeds of flywheel
(*d*) sum of square of maximum and minimum speeds of flywheel
(*e*) square of mean speed of flywheel.

12.345. In the case of coupled wheels locomotives, the magnitude of hammer blow is
(*a*) reduced
(*b*) increased
(*c*) same
(*d*) may increase/decrease depending on speed
(*e*) unpredictable.

12.346. Coefficient of fluctuation of energy is

(a) the variation of energy above and below the mean resisting torque
(b) difference between the maximum and the minimum energies divided by M.I. of flywheel
(c) the ratio of the maximum fluctuation of energy of the work done per cycle
(d) M.I. of flywheel multiplied by difference between square of maximum and minimum angular speeds
(e) none of the above.

12.347. The fluctuation of energy in a turning moment diagram is
(a) the variation of energy above and below the mean resisting torque
(b) difference between the maximum and the minimum energies divided by M.I. of flywheel
(c) the ratio of the maximum fluctuation of energy of the work done per cycle
(d) M.I. of flywheel multiplied by difference between square of maximum and minimum angular speeds
(e) none of the above.

12.348. The following valve is used to vary the cut off, while engine is in motion without reversing facility
(a) Meyer expansion valve
(b) Allen link motion valve
(c) Joy valve gear
(d) Hockworth valve gear
(e) Walscharet valve.

12.349. The valve gear which does not call for eccentric is known as :
(a) Meyer expansion valve gear
(b) Hockworth valve gear
(c) Walscharet valve gear
(d) Joy valve gear
(e) all of the above.

12.350. Governor sensitivity is the ratio of
(a) range of speed to the mean speed
(b) maximum speed to the minimum speed
(c) mean speed to the range of speed
(d) effort of governor to the range of speed
(e) governor lift to the range of speed.

12.351. The weight of a porter governor is
(a) proportional to angular speed (ω)
(b) inversely proportional to ω
(c) proportional to ω^2
(d) inversely proportional to ω^2
(e) independent of ω.

12.352. The height of a simple watt governor is proportional to
(a) speed N (b) $1/N$
(c) N^2 (d) $\frac{1}{N^2}$
(e) none of the above.

12.353. If governor balls have one particular fixed radius for each given speed in the equilibrium position, such a governor is said to be
(a) sensitive (b) insensitive
(c) stable (d) unstable
(e) isochronous.

12.354. The effort of a porter governor is
(a) proportional to percentage increase in speed (s)
(b) inversely proportional to s
(c) proportional to s^2
(d) inversely proportional to s^2
(e) independent of s.

12.355. The power of a porter governor is
(a) proportional to percentage increase in speed (s)
(b) inversely proportional to s
(c) proportional to s^2
(d) inversely proportional to s^2
(e) independent of s.

12.356. Choose the correct statement

The magnitude of swaying couple due to partial balance of the primary unbalancing force in locomotive is
(a) directly proportional to the distance between the centre lines of two cylinders
(b) inversely proportional to the distance between the centre lines of the two cylinders
(c) directly proportional to the square of the distance between the centre lines of the two cylinders
(d) inversely proportional to the reciprocating mass
(e) directly proportional to the square of the crank radius.

12.357. For the same diameter, the radius of gyration of disc type flywheel as compared to rim type flywheel is

(a) $\sqrt{2}$ times (b) $\frac{1}{\sqrt{2}}$ times
(c) 2 times (d) $\frac{1}{2}$ times
(e) same.

12.358. The stress in disc type flywheel as compared to rim type flywheel is
(a) equal (b) less
(c) greater
(d) could be less or more depending on range of speed
(e) none of the above is true.

12.359. The flywheel controls the following mathematically

(a) δN (b) $\frac{\delta N}{dt}$
(c) $\frac{\delta t}{\delta N}$ (d) $\frac{\delta^2 N}{\delta t^2}$
(e) none of the above.
where N = speed, t = time.

12.360. The variation in tractive force caused by unbalanced primary force in locomotives, with increase in the fraction of the reciprocating mass to be balanced will
(a) remain unaffected
(b) increase
(c) decrease
(d) may increase or decrease depending upon the unbalance
(e) become zero.

12.361. The function of balancing a prime mover is to
(a) keep speed constant
(b) keep power output constant
(c) overcome and minimise inertia force
(d) eliminate partially or completely the effects due to resultant force and couple
(e) minimise effect of friction.

12.362. The primary unbalanced force in reciprocating masses is
(a) directly proportional to crank radius
(b) directly proportional to square of crank radius
(c) inversely proportional to crank radius
(d) inversely proportional to square of crank radius
(e) independent of crank radius.

12.363. It is safer to have the maximum unbalanced force
(a) along the line of stroke
(b) parallel to line of stroke
(c) at 45° to the line of stroke
(d) perpendicular to the line of stroke
(e) none of the above.

12.364. For complete balancing of the reciprocating parts, the following polygon must close
(a) primary force (b) primary couple
(c) secondary force
(d) secondary couple
(e) all of the above.

12.365. Most of the engines generally
(a) require balancing of secondary forces
(b) require balancing of secondary couples
(c) require balancing of both secondary forces and couples
(d) do not require balancing of secondary forces and couples
(e) none of the above.

12.366. The gear box in automobiles is placed between
(a) the clutch and differential
(b) the steering and engine
(c) the engine and clutch
(d) the differential and Hook's joint
(e) clutch and Hook's joint.

12.367. The cylinders in aero engines are arranged along
(a) parallel lines
(b) perpendicular lines
(c) radial lines
(d) any of the above arrangements
(e) none of the above.

12.368. A single row six cylinder radial engine has following number of cranks
(a) zero (b) one
(c) three (d) six
(e) twelve.

12.369. The usefulness of critical damping is that it
(a) totally eliminates vibrations

(*b*) provides basis of determining critical damping
(*c*) enables measurement of damping
(*d*) provides a measure of the relative amount of damping in a system
(*e*) predicts nature of vibrations.

12.370. The amplitude of free successive oscillations with coulomb damping will follow the following type of progression
(*a*) geometric (*b*) harmonic
(*c*) arithmetic (*d*) logarithmic
(*e*) exponential.

12.371. With viscous damping, the frequency of damped oscillations as compared to frequency of undamped vibrations is
(*a*) less (*b*) more
(*c*) same (*d*) independent
(*e*) zero.

12.372. The number of degrees of freedom in a continuous system would be
(*a*) zero
(*b*) one
(*c*) two
(*d*) dependent upon the geometry of system
(*e*) none of the above.

12.373. The frequency of vibrations with increase of damping in the case of free vibrations with coulomb damping will
(*a*) remain same
(*b*) increase (*c*) decrease
(*d*) may increase or decrease depending upon the damping coefficient
(*e*) none of the above.

12.374. The maximum displacement amplitude for an instrument having natural frequency of 20 rad/sec. and withstand capability of 4 m/sec^2 can be
(*a*) 0.001 m (*b*) 0.01 m
(*c*) 0.1 m (*d*) 0.5 m
(*e*) 0.025 m.

12.375. The type of gear used for speed reduction of 50 : 1 will be
(*a*) herringbone (*b*) spur
(*c*) bevel (*d*) worm wheel
(*e*) hypoid.

12.376. Natural period of vibration in terms of mass of the system = m and stiffness of spring = k is given as equal to
(*a*) $\frac{1}{2\pi}\sqrt{\frac{k}{m}}$ (*b*) $\frac{1}{2\pi}\sqrt{\frac{m}{k}}$
(*c*) $2\pi\sqrt{\frac{k}{m}}$ (*d*) $2\pi\sqrt{\frac{m}{k}}$
(*e*) $\sqrt{\frac{m}{k}}$.

12.377. Frequency of free torsional oscillations of the system in terms of torsional stiffness of shaft = q and mass M.I. of the disc attached at the end of shaft = I is equal to
(*a*) $2\pi\sqrt{\frac{I}{q}}$ (*b*) $2\pi\sqrt{\frac{q}{I}}$
(*c*) $\frac{1}{2\pi}\sqrt{\frac{I}{q}}$ (*d*) $\frac{1}{2\pi}\sqrt{\frac{q}{I}}$
(*e*) $\sqrt{\frac{q}{I}}$.

12.378. The type of teeth provided on a gear used in sugar crushing machinery is
(*a*) involute (*b*) paraboloid
(*c*) hyperboloid (*d*) cycloidal
(*e*) none of the above.

12.379. If two gears have moment of inertias as I_1 and I_2 respectively and mesh with a speed ratio $\omega_2/\omega_1 = n$, then equivalent moment of inertia of both gears referred to first one is
(*a*) $I_1 + I_2$ (*b*) $I_1 + nI_2$
(*c*) $I_1 + n^2I_2$ (*d*) $nI_1 + I_2$
(*e*) $n^2I_1 + I_2$.

12.380. The equation of motion of a ball of radius r rolling without slipping on a cylindrical surface of radius R is $\frac{d^2\theta}{dt^2} + \frac{2g}{3(R-r)} = 0$, then its natural frequency of oscillation about lowest point is equal to
(*a*) $\sqrt{\frac{2g}{3(R-r)}}$ (*b*) $\frac{1}{2\pi}\sqrt{\frac{2g}{3(R-r)}}$
(*c*) $\sqrt{\frac{3(R-r)}{2g}}$ (*d*) $\frac{1}{2\pi}\sqrt{\frac{3(R-r)}{2g}}$
(*e*) $\frac{1}{2\pi}\sqrt{\frac{2g}{3(R-r)}}$.

12.381. Radius of friction circle at any rotating surface is
(*a*) $r\cos\phi$ (*b*) $r\sin\phi$

(*c*) $r \sin^2 \phi$ (*d*) $r \cos^2 \phi$

(*e*) $\frac{r \sin \phi}{2}$

where r = radius of surface of contact.

12.382. Fig. 12.3 shows a crusher having several cylindrical rollers of weight W. The crushing force due to each roller will be

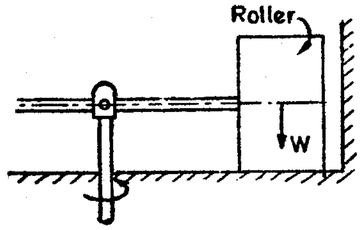

Fig. 12.3.

(*a*) W (*b*) less than W
(*c*) more than W (*d*) unpredictable
(*e*) none of the above.

12.383. The rotor of a turbine of a ship rotates clockwise when looking from the stern. If the ship travelling at certain velocity steers to the left, then the bow will
(*a*) remain in same position
(*b*) rise (*c*) fall
(*d*) unpredictable (*e*) none of the above.

12.384. The rotor of a turbine of a ship rotates clockwise when looking from the stern. If the ship when travelling starts rolling at ω rad/sec clockwise when looking from the stern, then the bow will
(*a*) remain in same position
(*d*) rise (*c*) fall
(*d*) unpredictable (*e*) none of the above.

12.385. A motor car takes a bend of certain radius in right hand direction when moving at some speed. If engine rotates in a clockwise direction when viewed from front, then due to centrifugal force
(*a*) reaction on outside wheels is increased and on inside wheels decreased
(*b*) reaction on inside wheels is increased and on outside wheels decreased
(*c*) reaction on rear wheels is increased and on front wheels decreased
(*d*) reaction on rear wheels is decreased and on front wheels increased
(*e*) unpredictable.

12.386. In Prob. 12.385, the reactions due to gyroscopic couple wheels behave as follows
(*a*) reaction on outside wheels is increased and on inside wheels decreased
(*b*) reaction on inside wheels is increased and on outside wheels decreased
(*c*) reaction on rear wheels in increased and on front wheels decreased
(*d*) reaction on rear wheels in decreased and on front wheels increased
(*e*) unpredictable.

12.387. In Prob. 12.385, the reactions due to gyroscopic couple due to engine will be
(*a*) reaction on outside wheels is increased and on inside wheels decreased
(*b*) reaction on inside wheels is increased and on outside wheels decreased
(*c*) reaction on rear wheels is increased and on front wheels decreased
(*d*) reaction on rear wheels is decreased and on front wheels increased
(*e*) unpredictable.

12.388. If δ be the static deflection of shaft under the gravity load, then natural frequency of the system in case of the longitudinal and transverse vibration is equal to
(*a*) $2\pi\sqrt{\frac{g}{\delta}}$ (*b*) $2\pi\sqrt{\frac{\delta}{g}}$
(*c*) $\frac{1}{2\pi}\sqrt{\frac{1}{\delta}}$ (*d*) $\frac{1}{2\pi}\sqrt{\frac{g}{\delta}}$
(*e*) $\frac{1}{2\pi}\sqrt{\frac{\delta}{g}}$.

12.389. Under the condition of resonance for a vibrating body, the displacement would lag behind the disturbing force, *i.e.*, phase angle would be
(*a*) 0° (*b*) 45°
(*c*) 90° (*d*) 180°
(*e*) unpredictable.

12.390. For a vibrating system, if the damping factor is unity, then maximum magnification factor will occur for ω/ω_n value of
(*a*) 0 (*b*) 1
(*c*) more than 1 (*d*) unpredictable
(*e*) none of the above.

12.391. A vibrating beam has following degrees of freedom
(*a*) 0 (*b*) 1
(*c*) 2 (*d*) 3
(*e*) multi.

12.392. For a vibrating system, as the value ω/ω_n increases above unity, the magnification factor, irrespective of value of damping factor, has tendency to move towards
(*a*) above unity (*b*) near unity
(*c*) near infinity (*d*) near zero value
(*e*) unpredictable.

12.393. An overdamped system when disturbed from equilibrium position with an initial velocity will
(*a*) vibrate about equilibrium position
(*b*) immediately return to equilibrium position
(*c*) not cross the equilibrium position
(*d*) return to equilibrium position after 1 to 2 oscillations
(*e*) none of the above.

12.394. A three rotor system has following number of natural frequencies
(*a*) one (*b*) two
(*c*) three (*d*) four
(*e*) six.

12.395. The equation $\frac{d^2x}{dt^2} + \frac{k}{m}x = 0$ represents
(*a*) free vibrations
(*b*) forced vibrations
(*c*) periodically forced vibrations
(*d*) free vibrations with viscous damping
(*e*) resonance vibrations.

12.396. The equation $m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = 0$ represents
(*a*) free vibrations
(*b*) forced vibrations
(*c*) periodically forced vibrations
(*d*) free vibrations with viscous damping
(*e*) resonance vibrations.

12.397. The equation $m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = F \sin \omega t$ represents
(*a*) free vibrations
(*b*) forced vibrations
(*c*) periodically forced vibrations
(*d*) free vibrations with viscous damping
(*e*) resonance vibrations.

12.398. In vibration isolation systems, the transmissibility *i.e.*, ratio of force transmitted to the disturbing force will be less than unity, for all values of damping factors, if ω/ω_n is
(*a*) equal to 1 (*b*) below $\sqrt{2}$
(*c*) above $\sqrt{2}$ (*d*) less than 1
(*e*) unpredictable.

12.399. In SHM, the product of periodic time and frequency is equal
(*a*) zero (*b*) unity
(*c*) π (*d*) 2π
(*e*) $\pi/2$.

12.400. Critical damping is a function of
(*a*) mass and stiffness
(*b*) mass and damping coefficient
(*c*) stiffness and natural frequency
(*d*) natural frequency and damping coefficient
(*e*) stiffness and damping coefficient.

12.401. In the case of longitudinal vibrations, effect of inertia of shaft can be considered by adding following fraction of weight of shaft at the free end
(*a*) full (*b*) half
(*c*) one-third (*d*) $\frac{33}{140}$
(*e*) one-fourth.

12.402. At node of the shaft, the vibrations are
(*a*) minimum (*b*) maximum
(*c*) some value between minimum and maximum
(*d*) zero
(*e*) corresponding to natural frequency.

12.403. A shaft carrying two rotors at ends will have following number of nodes
(*a*) 1 (*b*) 2
(*c*) 3 (*d*) 4
(*e*) zero.

12.404. A shaft carrying three rotors will have following number of nodes
(*a*) 0 (*b*) 1
(*c*) 2 (*d*) 3
(*e*) 4.

12.405. If a mass m oscilliates on a spring of mass m_s and stiffness k, then natural frequency of mass is equal to

(a) $\frac{\sqrt{k}}{\sqrt{m+m_s/3}}$ (b) $\frac{\sqrt{k}}{\sqrt{m_s+m/3}}$

(c) $\sqrt{\frac{k}{m+m_s}}$ (d) $2\pi\sqrt{\frac{k.g}{m+m_s}}$

(e) $\frac{\sqrt{k.g}}{\sqrt{m+m_s/3}}$.

12.406. The natural frequency of a spring-mass system when the mass of spring is also considered, as compared to system with negligible mass of spring will be

(a) more (b) less
(c) same
(d) depending on mass of spring
(e) unpredictable.

12.407 In vibration isolation system, if $\frac{\omega}{\omega_n} > 1$,then the phase difference between the transmitted force and the disturbing force is

(a) 0° (b) 45°
(c) 90° (d) 180°
(e) 270°.

12.408. In underdamped vibrating system, the amplitude of vibration with reference to time

(a) increases linearly
(b) increases exponentially
(c) decreases linearly
(d) decreases exponentially
(e) remains constant.

12.409. In a critically damped vibration system with single degree of freedom, the factor $\frac{\text{damping coefficient}}{\sqrt{\text{mass} \times \text{spring constant}}}$ should be equal to

(a) 0 (b) 1
(c) < 1 (d) > 1
(e) none of the above.

12.410. The rate of decay of oscillations is known as

(a) critical damping
(b) damping coefficient
(c) transmissibility
(d) logarithmic decrement
(e) damped oscillations.

12.411. In forced vibrations the magnitude of damping force at resonance equals

(a) inertia force (b) impressed force
(c) infinity (d) spring force
(e) zero.

12.412. In steady state forced vibrations, the amplitude of vibrations at resonance is

(a) directly proportional to the damping coefficient
(b) directly proportional to the resonant frequency
(c) inversely proportional to the damping coefficient
(d) inversely proportional to the resonant frequency
(e) directly proportional to the mass of the system and spring stiffness.

12.413. The ratio of the maximum displacement of the forced vibration to the deflection due to the static force is called

(a) logarithmic decrement
(b) damping coefficient
(c) critical damping coefficient
(d) magnification factor
(e) damping factor.

12.414. The ratio of the actual damping coefficient to the critical damping coefficient is called

(a) magnification factor
(b) damping factor
(c) logarithmic decrement
(d) transmissibility
(e) none of the above.

12.415. The damping force, in the forced vibrations, with reference to the spring force

(a) leads by $\pi/2$ radians
(b) lags by $\pi/2$ radians
(c) leads by π radians
(d) lags by π radians
(e) is in phase.

12.416. For steady state forced vibrations, the phase lag at resonance condition is

(a) 0° (b) 45°
(c) 90° (d) 180°.
(e) dependent on damping coefficient.

12.417. The frequency of damped oscillations in the dry friction damping, as compared to undamped vibration is

(a) equal (b) less

(c) more (d) independent
(e) may be less or more depending upon the damping coefficient.

12.418. A rotary system having damping coefficient of 50N sec m/rad, when twisted with an angular velocity of 2 rad/sec will experience the damping torque of
(a) 50 Nm (b) 100 Nm
(c) 25 Nm (d) 200 Nm
(e) 12.5 Nm.

12.419. For a vibrating body under steady state forced vibrations, if ratio ω/ω_n is very high, then phase angle would tend to approach
(a) 0° (b) 90°
(c) 180° (d) 270°.
(e) unpredictable.

12.420. The transmissibility (ratio of force transmitted to the force applied) in vibration isolation system for all values of damping factor, if

$$\left(\frac{\text{angular speed of system}}{\text{material frequency of vibration of system}}\right)$$

$\frac{\omega}{\omega_n} > 1.414$, is
(a) zero (b) unity
(c) more than unity
(d) less than unity
(e) infinity.

12.421. In vibration isolation systems, if $\frac{\omega}{\omega_n} < 1.414$ then transmissibility for all values of damping factor, is
(a) zero (b) unity
(c) more than unity
(d) less than unity
(e) infinity.

12.422. In vibration isolation systems, transmissibility is unity for all values of damping factor, if $\frac{\omega}{\omega_n}$ is equal to
(a) unity (b) $\sqrt{2}$
(c) $> \sqrt{2}$ (d) $< \sqrt{2}$
(e) zero.

12.423. In the case of transverse vibrations, the effect of inertia of the shaft can be considered by adding following fraction of mass of the shaft to that of the disc or flywheel
(a) full (b) half
(c) one-third (d) $\frac{33}{140}$
(e) $\frac{33}{280}$.

12.424. If the damping factor for a vibrating system is unity, then the system is
(a) critically damped
(b) not damped (c) under damped
(d) over damped (e) zero damped.

12.425. In a damped vibration system, the damping force is proportional to
(a) displacement (b) velocity
(c) acceleration (d) vibrations
(e) applied force.

12.426. For a vibrating body under steady state forced vibrations, if ratio ω/ω_n is very low, the phase angle would tend to approach
(a) 0° (b) 90°
(c) 180° (d) 270°
(e) unpredictable.

12.427. Which of the following effects is more dangerous for a ship
(a) rolling (b) waving
(c) pitching (d) steering
(e) disturbance caused by fishes.

12.428. A stepped shaft of diameters d_1, d_2, and d_3 of lengths l_1, l_2 and l_3 respectively is to be reduced to a torsionally equivalent shaft of uniform diameter d_1. The length of equivalent shaft is equal to

(a) $l_1 + l_2\left(\frac{d_1}{d_2}\right) + l_3\left(\frac{d_1}{d_3}\right)$

(b) $l_1 + l_2\left(\frac{d_1}{d_2}\right) + l_3\left(\frac{d_1}{d_3}\right)^2$

(c) $l_1 + l_2\left(\frac{d_1}{d_2}\right)^3 + l_3\left(\frac{d_1}{d_3}\right)^3$

(d) $l_1 + l_2\left(\frac{d_1}{d_2}\right)^4 + l_3\left(\frac{d_1}{d_3}\right)^4$

(e) $l_1 + l_2\left(\frac{d_2}{d_1}\right)^4 + l_3\left(\frac{d_3}{d_1}\right)^4$

13

Machine Design

13.1. The ultimate strength of steel in tension in comparison to shear is in the ratio of

(*a*) 1 : 1 (*b*) 2 : 1
(*c*) 3 : 2 (*d*) 2 : 3
(*e*) 1 : 2.

13.2. The permissible stress for carbon steel under static loading is generally taken as

(*a*) 2000–3000 kg/cm^2
(*b*) 3000–4000 kg/cm^2
(*c*) 4000–7500 kg/cm^2
(*d*) 7500–10,000 kg/cm^2
(*e*) 10,000–15,000 kg/cm^2.

13.3. The property of a material which enables it to resist fracture due to high impact loads is known as

(*a*) elasticity (*b*) endurance
(*c*) strength (*d*) toughness
(*e*) resilience.

13.4. For a long and narrow cross section (*i.e.*, ratio of b/t, breadth b and thickness t above 10) bar subjected to torsion T, the value of maximum shear stress will be

(*a*) $\frac{T}{bt^2}$ (*b*) $\frac{T}{2bt^2}$
(*c*) $\frac{2T}{bt^2}$ (*d*) $\frac{3T}{bt^2}$
(*e*) $\frac{T}{2bt}$.

13.5. For a rectangular cross-section beam subjected to a shearing force F, the maximum shearing stress induced will be

(*a*) $\frac{F}{bt}$ (*b*) $\frac{2F}{bt}$
(*c*) $\frac{3F}{2bt}$ (*d*) $\frac{F}{2bt}$
(*e*) none of the above.

13.6. For a circular cross-section beam subjected to a shearing force F, the maximum shear stress induced will be

(*a*) $\frac{F}{\pi d^2}$ (*b*) $\frac{4F}{\pi d^2}$
(*c*) $\frac{2F}{\pi d^2}$ (*d*) $\frac{F}{4\pi d^2}$
(*e*) $\frac{3F}{\pi d^2}$.

13.7. In bending of a beam, transverse shearing stresses are induced between the elements or fibres, if the bending moment is

(*a*) constant (*b*) varies
(*c*) increases (*d*) decreases
(*e*) depends on other conditions also.

13.8. When two cylinders of radii R and r are in axial contact with their axes parallel under force F, the area of contact being a rectangle of width b and length L, then the value of maximum pressure on the contact strip is equal to

(*a*) $\frac{F}{\pi bL}$ (*b*) $\frac{F}{2\pi bL}$
(*c*) $\frac{2F}{\pi bL}$ (*d*) $\frac{F}{bL}$
(*e*) $\frac{F}{2bL}$.

13.9. The distribution of pressure on the contact area in the above case will be

(*a*) straight line (*b*) circular
(*c*) elliptical (*d*) parabolic

(*e*) none of the above.

13.10. A hot short metal is
(*a*) brittle when cold
(*b*) brittle when hot
(*c*) brittle under all conditions
(*d*) ductile at high temperature
(*e*) hard when hot.

13.11. If the end of an elastic bar is struck axially by a hammer, a compressive wave develops that travels at the speed of sound in material equal to
(*a*) $\frac{E}{\rho}$ (*b*) $\sqrt{\frac{E}{\rho}}$
(*c*) $\left(\frac{E}{\rho}\right)^2$ (*d*) $\frac{2E}{\rho}$
(*e*) $\sqrt{\frac{2E}{\rho}}$
where E = modulus of elasticity
and ρ = mass density.

13.12. Guest's theory of failure is applicable for following type of materials
(*a*) brittle (*b*) ductile
(*c*) elastic (*d*) plastic
(*e*) tough.

13.13. Rankine's theory of failure is applicable for following type of materials
(*a*) brittle (*b*) ductile
(*c*) elastic (*d*) plastic
(*e*) tough.

13.14. If an unsupported uniform cross sectional elastic bar is subjected to a longitudinal impact from a rigid bob moving with velocity v, then a compressive wave of intensity s_c is propagated through the bar as follows
(*a*) $v\rho E$ (*b*) $v\sqrt{vE}$
(*c*) $v\sqrt{\rho E/2}$ (*d*) $2v^2\sqrt{\rho E}$
(*e*) none of the above.
where E = modulus of elasticity
and ρ = mass density.

13.15. Tensile strength of a mild steel specimen can be roughly predicted from following hardness test
(*a*) Brinell (*b*) Rockwell
(*c*) Vicker
(*d*) Shore's sceleroscope
(*e*) none of the above.

13.16. Resilience of a material is important, when it is subjected to
(*a*) combined loading
(*b*) fatigue (*c*) thermal stresses
(*d*) wear and tear (*e*) shock loading.

13.17. In the case of an elastic bar fixed at upper end and loaded by a falling weight at lower end, the shock load produced can be decreased by
(*a*) decreasing the cross-section area of bar
(*b*) increasing the cross-section area of bar
(*c*) remain unaffected with cross-section area
(*d*) would depend upon other factors
(*e*) none of the above.

13.18. Other method of reducing shock load in the above case (Q. 13.17) can be
(*a*) to decrease length
(*b*) to increase length
(*c*) unaffected by length
(*d*) other factors would decide same
(*e*) none of the above.

13.19. Yet another method to reduce shock load in the above example (Q. 13.17) can be
(*a*) to decrease E (modulus to elasticity)
(*b*) to increase E
(*c*) E has no effect on it
(*d*) other factors also require consideration in deciding same
(*e*) none of the above.

13.20. If a load W is applied instantaneously on a bar of cross section A, then the stress induced in the bar in worst case will be
(*a*) $\frac{W}{A}$ (*b*) $\frac{W}{2A}$
(*c*) $\frac{2W}{A}$
(*d*) $\frac{2W}{A}$ × (a factor greater than unity)
(*e*) none of the above.

13.21. The shear modulus of resilience of a material is proportional to
(*a*) shear stress (s_s)
(*b*) s_s^2 (*c*) s_s^3
(*d*) $\sqrt{s_s}$ (*e*) $s^{3/2}$.

13.22. If the longitudinal strain in a material is doubled in comparison to lateral strain,

then ratio of modulus of rigidity to elasticity will be
(*a*) 0.20 (*b*) 0.25
(*c*) 0.33 (*d*) 0.40
(*e*) 0.50.

13.23. If a load W is applied instantaneously on a bar; then the stress induced in bar will
(*a*) be independent of ratio of mass of load W to mass of bar (γ)
(*b*) increase with increase in γ
(*c*) decrease with decrease in γ
(*d*) depend on other considerations
(*e*) none of the above.

13.24. If a prismatic bar having an elliptical hole in centre with semi-major axis b perpendicular to direction of loading and semi-minor axis c along the direction of loading is subjected to pull F, then the maximum stress induced at edge of the hole will be
(*a*) $s\left(1+\frac{b}{c}\right)$ (*b*) $s\left(1+\frac{2b}{c}\right)$
(*c*) $s\left(1+\frac{b}{2c}\right)$ (*d*) $s\left(1+\frac{3b}{c}\right)$
(*e*) $s\left(1+\frac{b}{3c}\right)$.
where s = stress for uniform bar having no hole.

13.25. In above case, if hole is circular one, then maximum stress at edge of hole will be
(*a*) s (*b*) $2s$
(*c*) $3s$ (*d*) $4s$
(*e*) $2.5s$.

13.26. Brittle coating technique is used for
(*a*) determining brittleness
(*b*) protecting metal against corrosion
(*c*) protecting metal against wear and tear
(*d*) experimental stress analysis
(*e*) non-destructive testing of metals.

13.27. Stress concentration is caused due to
(*a*) variation in properties of material from point to point in a member
(*b*) pitting at points or areas at which loads on a member are applied
(*c*) abrupt change of section
(*d*) all of the above
(*e*) none of the above.

13.28. The endurance limit of a material with finished surface in comparison to rough surface is
(*a*) more
(*b*) less
(*c*) same
(*d*) more or less depending on quantum of load
(*e*) unpredictable.

13.29. Plastic flow in ductile materials
(*a*) increases the seriousness of static loading stress concentration
(*b*) lessens the seriousness of static loading stress concentration
(*c*) has no effect on it
(*d*) depends on other considerations
(*e*) none of the above.

13.30. The maximum stress due to stress concentration in a bar having circular transverse hole, as compared to its static stress without hole will be
(*a*) same in both cases
(*b*) 2 times more
(*c*) 3 times more
(*d*) 4 times more
(*e*) unpredictable.

13.31. The fatigue life of a part can be improved by
(*a*) electroplating (*b*) polishing
(*c*) coating (*d*) shot peening
(*e*) heat treating.

13.32. Stress concentration in static loading is more serious in
(*a*) ductile materials
(*b*) brittle materials
(*c*) equally serious in both cases
(*d*) depends on other factors
(*e*) unpredictable.

13.33. Stress concentration in cyclic loading is more serious in
(*a*) ductile materials
(*b*) brittle materials
(*c*) equally serious in both cases
(*d*) depends on other factors
(*e*) unpredictable.

13.34. Endurance limit or fatigue limit is the maximum stress that a member can withstand for an infinite number of load applications without failure when subjected to
(*a*) dynamic loading
(*b*) static loading
(*c*) combined static and dynamic loading

(*d*) completely reversed loading
(*e*) all of the above.

13.35. Pick up wrong statement. Fatigue strength can be increased by
(*a*) cold working
(*b*) shot peening
(*c*) grinding and lapping surface
(*d*) hot working
(*e*) using gradual changes of section.

13.36. Which of the following is not correct procedure to increase the fatigue limit
(*a*) cold working
(*b*) shot peening
(*c*) surface decarburisation
(*d*) under-stressing
(*e*) all of the above.

13.37. Coaxing is the procedure of increasing
(*a*) metal strength by cycling
(*b*) metal hardness by surface treatment
(*c*) metal resistance to corrosion by coating
(*d*) fatigue limit by overstressing the metal by successively increasing loadings
(*e*) none of the above.

13.38. Which is correct statement ?
Stress concentration in static loading is
(*a*) very serious in brittle materials and less serious in ductile materials
(*b*) very serious in ductile materials and less serious in brittle materials
(*c*) equally serious in both types of materials
(*d*) seriousness would depend on other factors
(*e*) none of the above.

13.39. The notch angle of the Izod impact test specimen is
(*a*) 10° (*b*) 20°
(*c*) 30° (*d*) 45°
(*e*) 60°.

13.40. In Vicker's hardness testing, the pyramid indentor apex is
(*a*) 40° (*b*) 122°
(*c*) 136° (*d*) 152°
(*e*) 161°.

13.41. Which is correct statement ?
Stress concentration in cyclic loading is
(*a*) very serious in brittle materials and less serious in ductile materials
(*b*) very serious in ductile materials and less serious in brittle materials
(*c*) equally serious in both types of materials
(*d*) seriousness would depend on other factors
(*e*) none of the above.

13.42. In testing a material for endurance strength, it is subjected to
(*a*) static load
(*b*) dynamic load
(*c*) impact load
(*d*) static as well as dynamic load
(*e*) completely reversed load.

13.43. If a material fails below its yield point, failure would be due to
(*a*) straining (*b*) fatigue
(*c*) creep (*d*) sudden loading
(*e*) impact loading.

13.44. The fatigue limit of a material
(*a*) is greatly decreased by poor surface conditions
(*b*) remains same irrespective of surface conditions
(*c*) depends mainly on core composition
(*d*) is dependent upon yield strength of material
(*e*) none of the above.

13.45. Cold working
(*a*) increases the fatigue strength
(*b*) decreases the fatigue strength
(*c*) has no influence on fatigue strength
(*d*) alone has no influence on fatigue strength
(*e*) none of the above.

13.46. Yield point in fatigue loading as compared to static loading is
(*a*) same
(*b*) higher
(*c*) lower
(*d*) depends on other factors
(*e*) none of the above.

13.47. Residual stress in materials
(*a*) acts when external load is applied
(*b*) becomes zero when external load is removed
(*c*) is independent of external loads
(*d*) is always harmful
(*e*) is always beneficial.

13.48. The building up of worn and undersized parts, subjected to repeated loads by electroplating is
(*a*) best method
(*b*) extremely hazardous
(*c*) has no effect as regards fatigue strength
(*d*) cheapest method
(*e*) all of the above.

13.49. In nitrided parts, the origins of the fatigue cracks will occur at
(*a*) surface
(*b*) just below the surface
(*c*) within the core
(*d*) could occur anywhere
(*e*) none of the above.

13.50. Which process will increase the fatigue duration of parts ?
(*a*) finishing and polishing
(*b*) shot-peening
(*c*) decarburisation
(*d*) electroplating
(*e*) all of the above.

13.51. Which is correct statement ?
(*a*) a member made of steel will generally be more rigid than a member of equal load-carrying ability made of cast iron
(*b*) a member made of cast iron will generally be more rigid than a member of equal load carrying ability made of steel
(*c*) both will be equally rigid
(*d*) which one is rigid will depend on several other factors
(*e*) none of the above.

13.52. Resistance to fatigue of a material is measured by
(*a*) Young's modulus
(*b*) coefficient of elasticity
(*c*) elastic limit
(*d*) ultimate tensile strength
(*e*) endurance limit.

13.53. In most machine members, the damping capacity of the material should be
(*a*) low
(*b*) zero
(*c*) high
(*d*) could be anything
(*e*) none of the above.

13.54. The ratio of endurance limit in shear to the endurance limit in flexure is
(*a*) 0.33 (*b*) 0.4
(*c*) 0.5 (*d*) 0.55
(*e*) 0.6.

13.55. For steel, the ultimate strength in shear as compared to ultimate strength in tension is
(*a*) same (*b*) 1/2
(*c*) 1/3 (*d*) 1/4
(*e*) 2/3.

13.56. The endurance limit in shear of carbon steel can be obtained by multiplying the endurance limit in flexure by a factor of
(*a*) 0.25 (*b*) 0.45
(*c*) 0.55 (*d*) 0.65
(*e*) 0.75.

13.57. At low temperatures (say 75°C) the notched-bar impact value of steel
(*a*) increases markedly
(*b*) decreases markedly
(*c*) remains same
(*d*) depends on heat treatment carried out
(*e*) none of the above.

13.58. A bolt
(*a*) has a head on one end and a nut fitted to the other
(*b*) has head at one end and other end fits into a tapped hole in the other part to be joined
(*c*) has both the ends threaded
(*d*) is provided with pointed threads
(*e*) requires no nut.

13.59. The crest diameter of a screw thread is same as
(*a*) major diameter
(*b*) minor diameter
(*c*) pitch diameter
(*d*) core diameter
(*e*) none of the above.

13.60. If d is the diameter of bolt hole then for a flanged pipe joint to be leak proof, the circumferential pitch of the bolts should be
(*a*) $10\sqrt{d}$ (*b*) $10\sqrt{d}$ to $15\sqrt{d}$
(*c*) $15\sqrt{d}$ to $20\sqrt{d}$ (*d*) $20\sqrt{d}$ to $30\sqrt{d}$
(*e*) $30\sqrt{d}$ to $40\sqrt{d}$.

13.61. Maximum principal stress theory is applicable for
(*a*) ductile materials
(*b*) brittle materials

(c) elastic materials
(d) all of the above
(e) none of the above.

13.62. The following type of nut is used with allen bolt
(a) allen nut
(b) hexagonal nut
(c) slotted nut
(d) castle nut
(e) any one of the above.

13.63. A stud
(a) has a head on one end and a nut fitted to the other
(b) has head at one end and other end fits into a tapped hole in the other part to be joined
(c) has both the ends threaded
(d) has pointed threads
(e) requires locking nuts.

13.64. Shear stress theory is applicable for
(a) ductile materials
(b) brittle materials
(c) elastic materials
(d) all of the above
(e) none of the above.

13.65. A tap bolt
(a) has a head on one end and a nut fitted to the other
(b) has head at one end and other end fits into a tapped hole in the other part to be joined
(c) has both the ends threaded
(d) has pointed threads
(e) requires locking devices.

13.66. For applications involving high stresses in one direction only the following type of thread would be best suited
(a) ISO metric thread
(b) acme thread
(c) square thread
(d) buttress thread
(e) British Association thread.

13.67. The included angle in unified of American National threads is
(a) 60° (b) 55°
(c) $47\frac{1}{2}°$ (d) 29°
(e) none of the above.

13.68. The included angle in Acme threads is
(a) 60° (b) 55°
(c) $47\frac{1}{2}°$ (d) 29°
(e) none of the above.

13.69. The function of a washer is to
(a) provide cushioning effect
(b) provide bearing area
(c) absorb shocks and vibrations
(d) provide smooth surface in place of rough surface
(e) act as a locking device.

13.70. Cap screws are
(a) similar to small size tap bolts except that a greater variety of shapes of heads are available
(b) slotted for a screw driver and generally used with a nut
(c) used to prevent relative motion between parts
(d) provided with detachable caps
(e) similar to stud.

13.71. An allen bolt is
(a) self locking bolt
(b) same as stud
(c) provided with hexagonal depression in head
(d) used in high speed components
(e) provided with countersunk head.

13.72. The deflection of a cantilever beam under load W is δ. If its width is halved, then the deflection under load W will be
(a) 2δ (b) $\delta/2$
(c) 4δ (d) $\delta/4$
(e) none of the above.

13.73. Ball bearing type screws are found in following application
(a) screw jack (b) aeroplane engines
(c) crane (d) steering mechanism
(e) bench vice.

13.74. Set screws are
(a) similar to small size tap bolts except that a greater variety of shapes of heads are available
(b) slotted for a screw driver and generally used with a nut
(c) used to prevent relative motion between parts
(d) similar to stud
(e) none of the above.

13.75. A self locking screw has
(*a*) fine threads
(*b*) coarse threads
(*c*) coefficient of friction ≥ tangent of load angle
(*d*) hole for inserting split pin
(*e*) two nuts for locking.

13.76. The designation M 33 × 2 of a bolt means
(*a*) metric threads of 33 nos in 2 cm.
(*b*) metric threads with cross-section of 33 mm^2
(*c*) metric threads of 33 mm outside diameter and 2 mm pitch
(*d*) bolt of 33 mm nominal diameter having 2 threads per cm
(*e*) none of the above.

13.77. Machine screws are
(*a*) similar to small size tap bolts except that a greater variety of shapes of heads are available
(*b*) slotted for a screw driver and generally used with a nut
(*c*) used to prevent relative motion between two parts
(*d*) similar to stud
(*e*) none of the above.

13.78. Rivets are generally specified by
(*a*) thickness of plates to be riveted
(*b*) length of rivet
(*c*) diameter of head
(*d*) nominal diameter
(*e*) all of the above.

13.79. The rivet head for general purpose shown in Fig. 13.1 is
(*a*) snap
(*b*) pan
(*c*) conuter sunk
(*d*) flat
(*e*) none of the above.

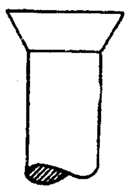

Fig. 13.1.

13.80. The rivet head for boiler applications shown in Fig. 13.2 is
(*a*) snap
(*b*) pan
(*c*) conical
(*d*) steeple
(*e*) ellipsoid.

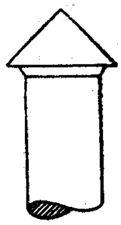

Fig. 13.2.

13.81. The edges of a boiler plate are bevelled to an angle of
(*a*) 30° (*b*) 45°
(*c*) 60° (*d*) 80°
(*e*) 85°.

13.82. Which of the following is a permanent fastening ?
(*a*) bolts (*b*) keys
(*c*) cotters (*d*) rivets
(*e*) screws.

13.83. The type of riveted joint shown in Fig. 13.3, is

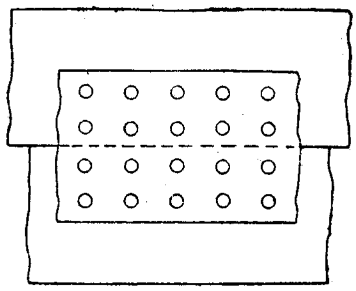

Fig. 13.3.

(*a*) double riveted lap joint
(*b*) double riveted butt joint
(*c*) quadruple riveted lap joint
(*d*) quadruple riveted butt joint
(*e*) straight riveted joint.

13.84. In order to avoid tearing of the plate at edge, the distance from the centre line of the rivet hole to the nearest edge of the plate in terms of dia. of rivet d should be equal to
(*a*) d (*b*) 1.25 d
(*c*) 1.5 d (*d*) 1.75 d
(*e*) 2 d.

13.85. If the tearing efficiency of a riveted joint is 75%, then the ratio of diameter of rivet to the pitch is equal to
(*a*) 0.2 (*b*) 0.25
(*c*) 0.50 (*d*) 0.6
(*e*) 0.75.

13.86. The drawing representation shown in Fig. 13.4, for welding is used to represent
(*a*) weld all around
(*b*) field weld
(*c*) flush contour
(*d*) chipping finish
(*e*) convex contour.

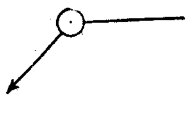

Fig. 13.4.

13.87. Transverse fillet welded joints are designed for
(a) tensile strength
(b) compressive strength
(c) shear strength
(d) bending strength
(e) torsional strength.

13.88. The drawing representation shown in Fig. 13.5, for welding is used to represent
(a) field weld
(b) weld all around
(c) flush contour
(d) chipping finish
(e) convex contour.

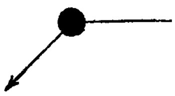

Fig. 13.5.

13.89. Jam nut is a locking device in which
(a) one smaller nut is tightened over main nut and main nut tightened against smaller one by loosening, creating friction jamming
(b) a slot is cut partly in middle of nut and then slot reduced by tightening a screw
(c) a hard fibre or nylon cotter is recessed in the nut and becomes threaded as the nut is screwed on the bolt causing a tight grip
(d) through slots are made at top and a cotter-pin is passed through these and a hole in the bolt, and cotter splitted and bent in reverse direction at other end
(e) none of the above.

13.90. The pitch of threads on a lock nut in comparison to pitch of nut is
(a) same (b) coarser
(c) finer (d) very coarse
(e) very fine.

13.91. The nut shown in Fig. 13.6, is

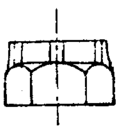

Fig. 13.6.

(a) slotted nut (b) capstan nut
(c) cap nut (d) ring nut
(e) castle nut.

13.92. The screw head shown in Fig. 13.7 is
(a) machine (b) round

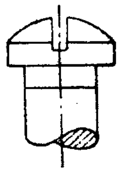

Fig. 13.6.

(c) cheese (d) fillister
(e) capstan.

13.93. Buttress threads are usually found on
(a) screw cutting lathes
(b) feed mechanisms
(c) spindles of bench vices
(d) screw jack
(e) railway carriage couplings.

13.94. The bolt shown in Fig. 13.8, is designed for

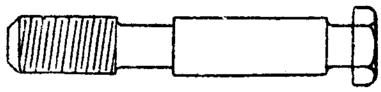

Fig. 13.8.

(a) shock loading
(b) fatigue loading
(c) avoiding stress concentration
(d) creep
(e) uniform strength.

13.95. In order to obtain bolt of uniform strength
(a) increase shank diameter
(b) increase its length
(c) drill an axial hole through head upto threaded portion so that shank area is equal to root area of thread
(d) tighten the bolt properly
(e) all of the above.

13.96. The key shown in Fig. 13.9, is

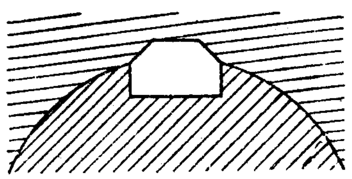

Fig. 13.9.

(a) Lewis key (b) Kennedy key
(c) Pin key (d) Barth key
(e) Woodruff key.

13.97. A key capable of tilting in a recess milled out in a shaft is known as
(*a*) wood-ruff key
(*b*) feather key
(*c*) flat saddle key
(*d*) gib head key
(*e*) hollow saddle key.

13.98. A key made from a cylindrical disc having segmental cross-section, is known as
(*a*) wood-ruff key
(*b*) feather key
(*c*) flat saddle key
(*d*) gib head key
(*e*) hollow saddle key.

13.99. A tapered key which fits in a keyway in the hub and is flat on the shaft, is known as
(*a*) wood-ruff key
(*b*) feather key
(*c*) flat saddle key
(*d*) gib head key
(*e*) hollow saddle key.

13.100. Fibrous fracture occurs in
(*a*) ductile material
(*b*) brittle material
(*c*) elastic material
(*d*) hard material
(*e*) all of the above.

13.101. Turn buckle 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
(*e*) threads in middle portion.

13.102. Eye bolts are used for
(*a*) foundation purposes
(*b*) locking devices
(*c*) absorbing shock and vibrations
(*d*) transmission of power
(*e*) lifting and transportation of machines and cubicles.

13.103. Elastic nut is a locking device in which
(*a*) one small nut is tightened over main nut and main nut tightened against smaller one by loosening, creating friction jamming
(*b*) a slot is cut partly in middle of nut and then slot reduced by tightening a screw
(*c*) hard fibre or nylon cotter is recessed in the nut and becomes threaded as the nut is srewed on the bolt causing a tight grip
(*d*) through slots are made at top and a cotter-pin is passed through these and a hole in the bolt, and cotter pin splitted and bent in reverse direction at other end
(*e*) none of the above.

13.104. If d is the normal diameter of a bolt in mm, then the initial tension in kg in a bolt used for making a fluid tight joint as for steam engine cover joint is calculated by the relation
(*a*) 102 d (*b*) 124 d
(*c*) 138 d (*d*) 151 d
(*e*) 168 d.

13.105. If threads on a bolt are left hand, threads on nut will be
(*a*) right hand with same pitch
(*b*) left hand with same pitch
(*c*) could be left or right hand
(*d*) right hand with fine pitch
(*e*) left hand with fine pitch.

13.106. Taper usually provided on cotter is
(*a*) 1 in 5 (*b*) 1 in 10
(*c*) 1 in 24 (*d*) 1 in 40
(*e*) 1 in 50.

13.107. Applications in which stresses are encountered in one direction only uses following type of threads
(*a*) metric (*b*) buttress
(*c*) acme (*d*) square
(*e*) BSW.

13.108. The draw of cotter need not exceed
(*a*) 3 mm (*b*) 5 mm
(*c*) 10 mm (*d*) 20 mm
(*e*) 25 mm.

13.109. A bench vice has following type of threads
(*a*) metric (*b*) square
(*c*) buttress (*d*) acme
(*e*) BSW.

13.110. The valve rod in a steam engine is connected to an eccentric rod by
(*a*) cotter joint
(*b*) bolted joint
(*c*) kunckle joint

(*d*) universal coupling
(*e*) gib and cotter joint.

13.111. Split nut is a locking device in which
(*a*) one smaller nut is tightened over main nut and main nut tightened against smaller one by loosening, creating friction jamming
(*b*) a slot is cut partly in middle of nut and then slot reduced by tightening screw
(*c*) a hard fibre or nylon cotter is recessed in the nut and becomes threaded as the nut is screwed on the bolt causing a tight grip
(*d*) through slots are made at top and a cotter-pin is passed through these and a hole in the bolt, and cotter pin splitted and bent in reverse direction at the other end
(*e*) none of the above.

13.112. Taper on the cotter and slot is provided
(*a*) on both the sides
(*b*) on one side only
(*c*) on none of the sides
(*d*) may be provided anywhere
(*e*) none of the above.

13.113. The function of cutting oil when threading a pipe is to
(*a*) provide cooling action
(*b*) lubricate the dies
(*c*) help remove chips
(*d*) all of the above
(*e*) none of the above.

13.114. Silver-based solder is used for
(*a*) flaring
(*b*) brazing
(*c*) soft soldering
(*d*) fusion welding
(*e*) none of the above.

13.115. For tight leakage joints, following type of thread is best suited
(*a*) metric (*b*) buttress
(*c*) square (*d*) acme
(*e*) NPT (national pipe threads).

13.116. In order to permit the thermal expansion/contraction of tubing, it should be
(*a*) crimped (*b*) honed
(*c*) flared (*b*) bent
(*e*) none of the above.

13.117. A tube has the following advantage over pipe
(*a*) lighter and easier to handle
(*b*) greater shock absorption
(*c*) smoother inside walls
(*d*) all of the above
(*e*) none of the above.

13.118. The strap end of a connecting rod of steam engine is joined by
(*a*) gib of cotter joint
(*b*) sleeve and cotter joint
(*c*) spigot socket cotter joint
(*d*) knuckle joint
(*e*) universal coupling.

13.119. A backing ring is used inside the pipe joint when making a
(*a*) butt weld (*b*) fillet weld
(*c*) sleeve weld (*d*) socket weld
(*e*) tube weld.

13.120. The shear plane in case of bolts should
(*a*) be across threaded portion of shank
(*b*) be parallel to axis of bolt
(*c*) be normal to threaded portion of shank
(*d*) never be across the threaded portion
(*e*) none of the above.

13.121. Castle nut is a locking device in which
(*a*) one smaller nut is tightened over main nut and main nut tightened against smaller one by loosening, creating friction jamming
(*b*) a slot is cut partly is middle of nut and then slot reduced by tightening a screw
(*c*) a hard fibre or nylon cotter is recessed in the nut and becomes threaded as the nut is screwed on the bolt causing a tight grip
(*d*) through slots are made at top and a cotter pin is passed through these and a hole in the bolt, and cotter pin splitted and bent in reverse direction at other end
(*e*) none of the above.

13.122. When a nut is tightened by placing a washer below it, the bolt will be subjected to following type of loads
(*a*) compression (*b*) tension
(*c*) shear (*d*) combined loads
(*e*) all of the above.

13.123. Gear box is used
(*a*) to produce torque
(*b*) for speed reduction
(*c*) to obtain variable speeds
(*d*) to increase efficiency of system
(*e*) to damp out vibrations.

13.124. The edges of the plates for cylindrical vessels are usually bevelled to an angle of 80° for
(*a*) reducing stress concentration
(*b*) ease of manufacture
(*c*) safety
(*d*) fullering and caulking
(*e*) all of the above.

13.125. The piston rod of a steam engine is usually connected to the crosshead by means of
(*a*) bolted joint (*b*) kunckle joint
(*c*) cotter joint (*d*) universal joint
(*e*) universal coupling.

13.126. Which of the following pipe joints would be suitable for pipes carrying steam
(*a*) flanged
(*b*) threaded
(*c*) bell and spigot
(*d*) expansion
(*e*) compression.

13.127. Spring index is
(*a*) ratio of coil diameter to wire diameter
(*b*) load required to produce unit deflection
(*c*) its capability of storing energy
(*d*) indication of quality of spring
(*e*) nothing.

13.128. The shearing stresses in the inner face as compared to outer face of the wire in a heavy close coiled spring is
(*a*) larger (*b*) smaller
(*c*) equal
(*d*) larger/smaller depending on diameter of spring coil
(*e*) unpredictable.

13.129. Form coefficient of spring is
(*a*) ratio of coil diameter to wire diameter
(*b*) load required to produce unit deflection
(*c*) its capability of storing energy
(*d*) concerned with strength of wire of spring
(*e*) nothing.

13.130. Spring stiffness is
(*a*) ratio of coil diameter to wire diameter
(*b*) load required to produce unit deflection
(*c*) its capability of storing energy
(*d*) its ability to absorb shocks
(*e*) none of the above.

13.131. When two springs are in series (having stiffness *K*), the equivalent stiffness will be
(*a*) K (*b*) $K/2$
(*c*) $2K$ (*d*) $K/4$
(*e*) $1/K$.

13.132. When a close coiled helical spring is compressed, its wire is subjected to
(*a*) tension
(*b*) shear
(*c*) compression
(*d*) all of the above
(*e*) none of the above.

13.133. If a spring is cut down into two springs, the stiffness of cut springs will be
(*a*) half
(*b*) same
(*c*) double
(*d*) unpredictable
(*e*) none of the above.

13.134. Belt slip may occur due to
(*a*) heavy load
(*b*) loose belt
(*c*) driving pulley too small
(*d*) all of the above
(*e*) none of the above.

13.135. Aircraft body is usually fabricated by
(*a*) welding (*b*) precasting
(*c*) rivetting (*d*) casting
(*e*) unconventional methods.

13.136. If two springs are in parallel then their overall stiffness will be
(*a*) half (*b*) same
(*c*) double (*d*) unpredictable
(*e*) none of the above.

13.137. In hydrodynamic bearings
(*a*) the oil film pressure is generated only by the rotation of the journal
(*b*) the oil film is maintained by supplying oil under pressure

(c) do not require external supply of lubricant
(d) grease is used for lubrication
(e) none of the above.

13.138. Antifriction bearings are
(a) sleeve bearings
(b) hydrodynamic bearings
(c) thin lubricated bearings
(d) ball and roller bearings
(e) none of the above.

13.139. If p = bearing pressure on projected bearing area, z = absolute viscosity of lubricant, and N = speed of journal, then the bearing characteristic number is given by
(a) ZN/p (b) p/ZN
(c) Z/pN (d) N/Zp
(e) Zp/N.

13.140. The usual clearance provided in hydrodynamic bearing per mm of diameter of shaft is
(a) 0.01 micron (b) 0.1 micron
(c) 1 micron (d) 10 microns
(e) 25 microns.

13.141. In hydrostatic bearings
(a) the oil film pressure is generated only by the rotation of the journal
(b) the oil film is maintained by supplying oil under pressure
(c) do not require external supply of lubricant
(d) grease is used for lubrication
(e) none of the above.

13.142. Oil in journal bearing should be applied at the point where load is
(a) nil or lightest (b) maximum
(c) average
(d) any one of the above
(e) unpredictable.

13.143. The rated life of a bearing varies
(a) directly as load
(b) inversely as square of load
(c) inversely as cube of load
(d) inversely as fourth power of load
(e) none of the above.

13.144. In oilness bearings
(a) the oil film pressure is generated only by the rotation of the journal
(b) the oil film is maintained by supplying oil under pressure
(c) do not require external supply of lubricant
(d) grease required to be applied after some intervals
(e) none of the above.

13.145. In V-belt drive, belt touches
(a) at bottom (b) at sides only
(c) both at bottom and sides
(d) could touch anywhere
(e) none of the above.

13.146. Three different weights fall from a certain height under vacuum. They will take
(a) same time to reach earth
(b) times proportional to weight to reach earth
(c) times inversely proportional to weight to reach earth
(d) unpredictable
(e) none of the above.

13.147. In cross or regular lay ropes
(a) direction of twist of wires in strands is opposite to the direction of twist of strands
(b) direction of twist of wires and strands are same
(c) wires in two adjacent strands are twisted in opposite direction
(d) wires are not twisted
(e) none of the above.

13.148. In standard taper roller bearings, the angle of taper of outer raceway is
(a) 5° (b) 8°
(c) 15° (d) 25°
(e) 40°.

13.149. Ball bearing type screws find application in
(a) vices
(b) screw jacks
(c) earthmoving machinery
(d) feed mechanisms
(e) steering mechanism.

13.150. In parallel lay rope
(a) direction of twist of wires in strands is opposite to direction of twist of strands
(b) direction of twist of wires and strands are same

(*c*) wires in two adjacent strands are twisted in opposite direction
(*d*) wires are not twisted
(*e*) none of the above.

13.151. In composite or reverse laid ropes
(*a*) direction of twist of wires in strands is opposite to the direction of twist of strands
(*b*) direction of twist of wires and strands are same
(*c*) wires in two adjacent strands are twisted in opposite direction
(*d*) wires are not twisted
(*e*) none of the above.

13.152. Whether any core is required in wire ropes
(*a*) yes (*b*) no
(*c*) sometimes (*d*) rarely
(*e*) very often.

13.153. If shearing stress in nut is half the tensile stress in a bolt, then nut length should be equal to
(*a*) diameter of bolt
(*b*) $0.75 \times$ diameter of bolt
(*c*) $1.25 \times$ diameter of bolt
(*d*) $1.5 \times$ diameter of bolt
(*e*) $0.5 \times$ diameter of bolt.

13.154. Basic shaft is one
(*a*) whose upper deviation is zero
(*b*) whose lower deviation is zero
(*c*) whose lower as well as upper deviations are zero
(*d*) does not exist
(*e*) none of the above.

13.155. Basic hole is one
(*a*) whose upper deviation is zero
(*b*) whose lower deviation is zero
(*c*) whose lower as well as upper deviations are zero
(*d*) does not exist
(*e*) none of the above.

13.156. Standard tolerance unit (i) =
(*a*) $0.55 \sqrt[3]{D} + 0.01\,D$
[D = diameter in mm]
(*b*) $0.45 \sqrt[3]{D} + 0.001\,D$
(*c*) $0.55 \sqrt[3]{D} + 0.001\,D$
(*d*) $0.45 \sqrt[3]{D} + 0.01\,D$
(*e*) none of the above.

13.157. I.S. specifies following total number of grades of tolerances
(*a*) 18 (*b*) 16
(*c*) 20 (*d*) 22
(*e*) 14.

13.158. In surface roughness, traverse length is the length of profile necessary for the evaluation of
(*a*) surface roughness
(*b*) surface irregularities
(*c*) surface defects
(*d*) all of the above
(*e*) none of the above.

13.159. For standard coarse threads of nut, the threads will be as strong in failure by shear as the bolt in tension, if the height of nut (h) is approximately
(*a*) 0.4 times (*b*) 0.6 times
(*c*) 0.7 times (*d*) 0.8 times
(*e*) 0.9 times
the normal diameter of bolt.

13.160. Allen bolts are
(*a*) self locking bolts
(*b*) designed for shock load
(*c*) used in aircraft application
(*d*) provided with hexagonal depression in head
(*e*) uniform strength bolts.

13.161. The resultant axial load on a bolt depends on
(*a*) initial tension
(*b*) external load applied
(*c*) relative elastic yielding (springness) of the bolt and the connected member
(*d*) (*a*) and (*b*) above
(*e*) (*a*), (*b*) and (*c*) above.

13.162. If the connected member between two members bolted together is very yielding (soft gasket) as compared to bolt, then the resultant load on the bolt will be equal to
(*a*) initial tension (*b*) external load
(*c*) higher to the two (*a*) and (*b*)
(*d*) lower of the two (*a*) and (*b*)
(*e*) sum of the two (*a*) and (*b*).

13.163. If the bolt is very yielding as compared with the connected members, then the resultant load on the bolt will be equal to
(*a*) initial tension
(*b*) external load

(*c*) higher of the two (*a*) and (*b*)
(*d*) lower of the two (*a*) and (*b*)
(*e*) sum of the two (*a*) and (*b*).

13.164. The connecting rod bolts are tightened up so that tightening stress
(*a*) is just sufficient to hold parts together
(*b*) approaches yield point
(*c*) is 50% of yield point
(*d*) is about yield point divided by safety factor
(*e*) none of the above.

13.165. The connecting rod bolts are tightened up with initial tension greater than the external load so that
(*a*) joint may not open up
(*b*) bolts are weakest elements
(*c*) the resultant load on the bolt would not be affected by the external cyclic load
(*d*) bolts will not loosen during service
(*e*) none of the above.

13.166. If an application calls for stresses on screw threads in one direction only, then the following type of thread would be best suited
(*a*) square (*b*) acme
(*c*) buttress (*d*) BSW
(*e*) metric.

13.167. When a bolt is subjected to shock loading, the resilience of the bolt should be considered in order to prevent breakage at
(*a*) shank (*b*) head
(*c*) in the middle
(*d*) at the thread
(*e*) anywhere in the bolt.

13.168. The shock absorbing capacity of a bolt can be increased by
(*a*) tightening it properly
(*b*) increasing shank diameter
(*c*) grinding the shank
(*d*) using washer
(*e*) making shank diameter equal to core diameter of thread.

13.169. Modulus of resilience is proportional to
(*a*) stress at elastic limit (s_e)
(*b*) s_e (*c*) $\sqrt{s_e}$
(*d*) $s_e^{3/2}$ (*e*) s_e^3.

13.170. Resilience of a bolt may be increased by
(*a*) increasing its length
(*b*) increasing its shank diameter
(*c*) increasing diameter of threaded portion
(*d*) increasing head size
(*e*) none of the above.

13.171. The holes in plates for riveting purposes should be made by
(*a*) punching and reaming
(*b*) drilling
(*c*) any of the two above
(*d*) depends on location and actual application
(*e*) torch cutting.

13.172. Strength of a rivet in bearing is given by
(*a*) $P = s_t (p - d)\, t$ (*b*) $P = s_b \times t \times d$
(*c*) $P = \frac{\pi}{4} d^2 \cdot s_s$ (*d*) $P = \frac{\pi}{4} d^2 s_t$
(*e*) $P = s_b (p - d)\, t$
where s_t, s_b and s_s are stresses in tension, bearing and shear, and *p*, *d* and *t* are pitch, rivet diameter and plate thickness, respectively.

13.173. In a riveted joint design, diameter of rivet '*d*' in derms of plate thickness '*t*' is equal to
(*a*) $d = 1.2\sqrt{t}$ (*b*) $d = 6\sqrt{t}$
(*c*) $d = 1.9\sqrt{t}$ (*d*) $d = \frac{1.2}{\sqrt{t}}$
(*e*) $d = \frac{1.5}{\sqrt{t}}$.

13.174. Lowest value of riveted joint efficiency is assumed in the case of
(*a*) single riveted butt joint
(*b*) single riveted lap joint
(*c*) double riveted butt joint
(*d*) double riveted lap joint
(*e*) diamond joint.

13.175. For riveted joints, the type of joint preferred is
(*a*) lap joint
(*b*) butt joint
(*c*) over lapping joint
(*d*) any of the above
(*e*) none of the above.

13.176. The distance from the centre line of the row of rivet holes nearest the edge of plate to edge of plate should be
(*a*) *d* (*b*) $1d - 1.5d$

(c) $1.5d - 2.5d$ (d) $2.5d - 2.5d$
(e) $2.5d - 3.0d$.
where d = diameter of rivet.

13.177. In the design of a riveted joint, efforts should be made to make it strong against failure due to
(a) tearing
(b) shearing
(c) bearing
(d) equal against tearing, shearing and bearing
(e) none of the above.

13.178. If the tearing efficiency of a riveted joint is 60%, then the ratio of pitch to diameter of rivet is
(a) 0.20 (b) 0.33
(c) 0.40 (d) 0.50
(e) 0.60.

13.179. In multiple-riveted (chain riveting) joints, the minimum distance between the rows of rivets should be
(a) d (b) $1.d - 1.5d$
(c) $1.5d - 2.0d$ (d) $2.0d - 2.5d$
(e) $2.5d - 3.0d$.
where d = diameter of rivet

13.180. Thickness of strap for double strap joint in terms of thickness of plate t is equal to
(a) $0.4\,t$ (b) $0.6\,t$ to t
(c) $1.2\,t$ (d) $1.75\,t$
(e) $2\,t$.

13.181. In case of a riveted joint, the maximum pitch in terms of diameter of rivet d is
(a) d (b) $d + 12$ mm
(c) $1.5 - 2.0\,d$ (d) $3d$
(e) $5d$.

13.182. The stress concentration in a riveted joint with unequal width cover plates as compared to one with equal width straps will be
(a) less (b) more
(c) equal
(d) depends on size of plate and diameter of rivet
(e) none of the above.

13.183. Failure due to tearing at an edge can be avoided by keeping the centre of the nearest rivet, from the edge of the plate at least a distance equal to
(a) d (b) $1.5d$
(c) $2.5\,d$ (d) $3d$
(e) $4d$.
where d is the diameter of the rivet.

13.184. The following type of rivet head is used for boiler plate riveting
(a) snap (b) round
(c) spherical (d) diamond
(e) counter sunk.

13.185. In order that the tearing strength of the cover plates should be equal to that of the plates to be connected, thickness of a single cover plate should be
(a) t (b) $\frac{5}{8}t$
(c) $\frac{3}{4}t$ (d) $1\frac{1}{8}t$
(e) $2t$.
where t = thickness of main plate.

13.186. With the precentage increase of carbon in steel
(a) strength of steel decreases
(b) hardness of steel decreases
(c) brittleness of steel decreases
(d) ductility of steel decreases
(e) none of the above.

13.187. Factor of safety is the ratio of
(a) yield stress/working stress
(b) tensile stress/working stress
(c) compressive stress/working stress
(d) bearing stress/working stress
(e) bearing stress/yield stress.

13.188. The rivets which are heated and then driven in the field are known as
(a) power driven shop rivets
(b) power driven filed rivets
(c) hand driven rivets
(d) cold driven rivets
(e) field rivets.

13.189. Cold driven rivets range from
(a) 6 to 10 mm in diameter
(b) 10 to 16 mm diameter
(c) 12 to 22 mm in diameter
(d) 22 to 32 mm in diameter
(e) none of the above.

13.190. In an eccentric riveted connection, the rivets have to resist
(a) linear displacement
(b) rotary displacement
(c) linear as well as rotary displacements

(d) linear or rotary displacement
(e) none of the above.

13.191. Pick up the correct statement
Diameter of the rivet hole is made larger than the diameter of the rivet by
(a) 1.00 mm for rivet diameter upto 12 mm
(b) 1.5 mm for rivet diameter exceeding 25 mm
(c) 2.0 mm for rivet diameter over 25 mm
(d) 0.5 mm for rivet diameter less than 10 mm
(e) none of the above.

13.192. The perpendicular distance between rows of rivets in chain riveting is
(a) $d+12$ (b) $1.5\,d$
(c) $2d+6$ mm (d) $3d$
(e) $3d+8$ mm.

13.193. The gross diameter of a rivet is the diameter of the
(a) cold rivet measured before driving
(b) rivet measured after driving
(c) rivet hole
(d) any one of the above
(e) none of the above.

13.194. Pick up the true statement from the following
(a) The minimum pitch should not be less than 2.5 times the gross diameter of the rivet
(b) The minimum pitch should not be less than 12 times the gross diameter of the rivet
(c) The maximum pitch should not exceed $10\,t$ or 150 mm whichever is less in compression
(d) all of the above
(e) none of the above.

13.195. According to I.B.R., safety factor of rivet joint should not be less than
(a) 2 (b) 3
(c) 4 (d) 8
(e) 12.

13.196. Efficiency of a riveted joint is the ratio between
(a) tearing strength of the joint to the strength of a pitch length of the solid plate
(b) shearing strength of the joint to the strength of a pitch length of the solid plate
(c) bearing strength of the joint to the strength of a pitch length of the solid plate
(d) the minimum of the three strengths of a joint to the strength of a pitch length of the solid plate
(e) none of the above.

13.197. According to I.B.R., the following type of joint is preferred for longitudinal joint
(a) lap
(b) butt joint
(c) welded joint
(d) any one of the above
(e) none of the above.

13.198. The edges of the boiler plate are bevelled to an angle of
(a) 45° (b) 60°
(c) 72° (d) 80°
(e) none of the above.

13.199. A riveted joint may fail due to
(a) shearing of the rivet
(b) tearing off the plate at an edge
(c) crushing of the rivet
(d) tearing off the plate across a row of rivets
(e) any or all of the above reasons.

13.200. According to I.B.R., the following type of joint is preferred for circumferential joint
(a) lap joint
(b) butt joint
(c) welded joint
(d) any one of the above
(e) none of the above.

13.201. A boiler plate thickness is 20 mm. The rivet diameter will be
(a) 20 mm (b) 10 mm
(c) 40 mm (d) 30 mm
(e) none of the above.

13.202. A thin walled cylindrical vessel consists of a central cylindrical portion having wall thickness of t_c, and two hemispherical ends of wall thickness t_h. In order to ensure uniform stress in both the portions, ratio t_c/t_h should be
(a) unity (b) half
(c) two (d) four times

(*e*) one fourth.

13.203. A riveted joint may fail due to
(*a*) failure of the rivets
(*b*) failure of the plates
(*c*) either failure of the rivets or failure of the plates
(*d*) failure of both rivets or plates
(*e*) none of the above.

13.204. Feather keys are generally
(*a*) tight in shaft and loose in hub
(*b*) loose in shaft and tight in hub
(*c*) tight in both shaft and hub
(*d*) loose in both shaft and hub
(*e*) none of the above.

13.205. The distribution of stress along the length of a key fitted in shaft
(*a*) is uniformly constant
(*b*) varies linearly
(*c*) is of exponential shape, being more at the torque input end
(*d*) is of exponential shape, being less at the torque input end
(*e*) is of parabolic shape.

13.206. Width of a key is usually taken as
(*a*) $1/10 \times$ shaft diameter (d)
(*b*) $1/8 \times d$ (*c*) $1/6 \times d$
(*d*) $1/4 \times d$ (*e*) $1/3 \times d$.

13.207. Which of the following steel key is usually strong in failure by shear and crushing
(*a*) rectangular (*b*) flat
(*c*) square (*d*) circular
(*e*) Kennedy.

13.208. For a square key of side $d/4$ (d = shaft diameter), its length should be as given below in order that it is as strong in shear as shaft is in torsion
(*a*) d (*b*) $1.5\ d$
(*c*) $2\ d$ (*d*) $2.5\ d$
(*e*) $3.0\ d$.

13.209. A saddle key
(*a*) is provided in the hub only and hollowed to fit the shaft
(*b*) has flat surface and the shaft is planned off to accommodate the key
(*c*) is fitted such that each withstands torsion in one direction only
(*d*) is designed to fit in a sunk key-way whose bed is parallel to the axis of shaft
(*e*) none of the above.

13.210. Shape of woodruff key is like
(*a*) cylinder
(*b*) semicircle (*c*) sphere
(*d*) trapezoid (*e*) tapered square.

13.211. In welded joint the throat of weld as compared to size of weld is
(*a*) about same size
(*b*) about 0.7 times
(*c*) about 0.5 times
(*d*) about 0.25 times
(*e*) about 1.25 times.

13.212. Tangent key
(*a*) is provided in the hub only and hollowed to fit the shaft
(*b*) has flat surface and the shaft is planned off to accommodate the key
(*c*) is fitted such that each withstands torsion in one direction only
(*d*) is designed to fit in a sunk key-way whose bed is parallel to the axis of shaft
(*e*) none of the above.

13.213. Kennedy keys are used for applications like
(*a*) precision duty
(*b*) light duty
(*c*) rough and heavy services
(*d*) all of the above
(*e*) none of the above.

13.214. Which key transmits power through frictional resistance only
(*a*) woodruff (*b*) kennedy
(*c*) sunk (*d*) saddle
(*e*) flat.

13.215. Sunk key
(*a*) is provided in the hub only and hollowed to fit the shaft
(*b*) has flat surface and the shaft is planned off to accommodate the key
(*c*) is fitted such that each withstands torsion in one direction only
(*d*) is designed to fit in a sunk key way whose bed is parallel to the axis
(*e*) none of the above.

13.216. Which of the following key transmits power through frictional resistance only
(*a*) square key (*b*) tapered key
(*c*) kennedy key (*d*) saddle key

(*e*) woodruff key.

13.217. Flat key
(*a*) is provided in the bulb only and hollowed to fit the shaft
(*b*) has flat surface and the shaft is planed off to accommodate the key
(*c*) is fitted such that each withstands torsion in one direction only
(*d*) is designed to fit in a sunk key way whose bed is parallel to the axis of shaft
(*e*) none of the above.

13.218. In thick cylinders, the tangential stress across the thickness of cylinder is
(*a*) zero at outside and maximum at inside
(*b*) minimum at outside and maximum at inside
(*c*) uniform throughout
(*d*) unpredictable
(*e*) none of the above.

13.219. Thick cylinders are designed by
(*a*) Lame's equation
(*b*) calculating radial stress which is uniform
(*c*) thick cylinder theory
(*d*) any one of the above
(*e*) none of the above.

13.220. For designing thick cylinders, following equation is used
(*a*) Barlow's (*b*) Birnie's
(*c*) Lame's (*d*) Clavarino
(*e*) all of the above.

13.221. In thick cylinders, the radial stress across the thickness of cylinder is
(*a*) zero at outside and maximum at inside
(*b*) minimum at outside and maximum at inside
(*c*) uniform throughout
(*d*) unpredictable
(*e*) none of the above.

13.222. Oldham's coupling is used to connect two shafts which
(*a*) have lateral misalignment
(*b*) whose axes intersect at a small angle
(*c*) are not is exact alignment
(*d*) is the simplest type of rigid coupling
(*e*) all of the above.

13.223. For two parallel shafts, the distance between whose axes is small and variable, which coupling will you use ?
(*a*) hydraulic coupling
(*b*) universal joint
(*c*) flange coupling
(*d*) Oldham's coupling
(*e*) muff coupling.

13.224. In the flange coupling the two flanges are coupled together by means of bolts fitted in
(*a*) reamed holes (*b*) machined holes
(*c*) threaded holes (*d*) gasketed holes
(*e*) as cast holes.

13.225. The holes in the flange coupling for coupling the two flanges together by bolts are reamed because it permits
(*a*) equal sharing of load by bolts
(*b*) avoidance of stress concentration
(*c*) avoidance of any injury during dismantling
(*d*) less wear, tear and vibrations
(*e*) full utilisation of power.

13.226. Following type of pipe joint is mostly used for pipes carrying water at low pressures
(*a*) socket (*b*) nipple
(*c*) union (*d*) spigot and socket
(*e*) expansion.

13.227. The sleeve or muff coupling is designed as a
(*a*) thin vessel (*b*) thick vessel
(*c*) solid shaft (*d*) hollow shaft
(*e*) continuous shaft.

13.228. Muff coupling is used to join two shafts which
(*a*) have lateral misalignment
(*b*) whose axes intersect at a small angle
(*c*) are not in exact alignment
(*d*) is the simplest type of rigid coupling
(*e*) all of the above.

13.229. Keys are normally made from
(*a*) cold rolled mild steel bars
(*b*) forged steel
(*c*) hot rolled mild steel bars
(*d*) cold rolled carbon steel
(*e*) machined stainless steel.

13.230. The most important dimension in the design of nut is

(*a*) outside dimensions of nut
(*b*) inside diameter
(*c*) height (*d*) pitch diameter
(*e*) thread size.

13.231. Multiple threaded screws
(*a*) increase the efficiency
(*b*) increase the mechanical advantage
(*c*) increase the self locking feature
(*d*) decrease the efficiency
(*e*) increase the load lifting capacity.

13.232. Use of large thread angles in lifting machine would result in
(*a*) lower mechanical advantage
(*b*) higher mechanical advantage
(*c*) no change in mechanical advantage
(*d*) lifting the load easily
(*e*) lowering the load easily.

13.233. Bolts are designed on the basis of
(*a*) direct tensile stress with high safety factor
(*b*) direct shear stress with high safety factor
(*c*) direct compressive stress with high safety factor
(*d*) direct bearing stress with high safety factor
(*e*) all of the above.

13.234. Universal coupling is used to join two shafts which
(*a*) have lateral misalignment
(*b*) whose axes intersect at a small angle
(*c*) are not in exact alignment
(*d*) is the simplest type of rigid coupling
(*e*) all of the above.

13.235. Metal to metal joint is used for applications subjected to
(*a*) very high pressure
(*b*) very high temperature
(*c*) very high pressures and temperatures
(*d*) severe vibrations
(*e*) stress reversals.

13.236. A screw is specified by
(*a*) major diameter $\times$ pitch
(*b*) minimum diameter $\times$ length
(*c*) major diameter $\times$ length
(*d*) mean diameter $\times$ pitch
(*e*) nominal major diameter.

13.237. Screws used for power transmission should have
(*a*) high efficiency
(*b*) strong teeth
(*c*) finished threads
(*d*) high efficiency and strong teeth
(*e*) proper heat treatment.

13.238. The maximum efficiency of a screw jack having square threads and friction angle of 30° will be
(*a*) 11% (*b*) 20%
(*c*) 30% (*d*) 33%
(*e*) 50%.

13.239. If α is the helix angle of threads and ϕ is the angle of friction, then the lifting screw will be self locking when
(*a*) $\alpha = \phi$ (*b*) $\alpha > \phi$
(*c*) $\alpha < \phi$ (*d*) $\alpha = 2\phi$
(*e*) there is no such correlation.

13.240. The important criterion in case of riveted joints for material storage and other ordinary tanks is
(*a*) efficiency of joint
(*b*) economy of design
(*c*) strength and rigidly
(*d*) leakage
(*e*) environmental thermal stresses.

13.241. Bushed pin flexible coupling is used to joint two shafts which
(*a*) have lateral misalignment
(*b*) whose axes intersect at a small angle
(*c*) are not in exact alignment
(*d*) is the simplest type of rigid coupling
(*e*) all of the above.

13.242. Permanent moving coil instruments are
(*a*) A.C. type
(*b*) D.C. type
(*c*) both A.C. and D.C. type
(*d*) could be A.C. or D.C.
(*e*) none of the above.

13.243. Moving iron instruments are
(*a*) A.C. type
(*b*) D.C. type
(*c*) both A.C. and D.C. type
(*d*) do not exist
(*e*) none of the above.

13.244. Shaft coupling is used in machinery to
(*a*) alter the vibration characteristics of rotating unit
(*b*) introduce protection against overloads

(*c*) introduce mechanical flexibility
(*d*) reduce transmission of shock loads
(*e*) all of the above.

13.245. Slenderness ratio is
(*a*) $\frac{\text{shaft dia.}}{\text{shaft length}}$
(*b*) $\frac{\text{length of strut}}{\text{least radius of gyration}}$
(*c*) $\frac{\text{column width}}{\text{column depth}}$
(*d*) $\frac{\text{max. size of column}}{\text{min. size of column}}$
(*e*) none of the above.

13.246. Compression formula is valid upto the slenderness ratio of
(*a*) 10 (*b*) 20
(*c*) 30 (*d*) 40
(*e*) 60.

13.247. Rankine's formula is valid upto the slenderness ratio of
(*a*) 60 (*b*) 120
(*c*) 180 (*d*) 240
(*e*) 300.

13.248. The buckling load depends on
(*a*) cross-sectional area
(*b*) modulus of elasticity
(*c*) slenderness ratio
(*d*) all of the above
(*e*) none of the above.

13.249. Euler's buckling or crippling load corresponds to load *P* such that
(*a*) $\sqrt{\frac{P}{EI}} \times \frac{l}{2} = \frac{\pi}{2}$
(*b*) $\sqrt{\frac{P}{EI}} \times \frac{l}{2} = \pi$
(*c*) $\sqrt{\frac{P}{EI}} \times 2l = \frac{\pi}{2}$
(*d*) $\sqrt{\frac{P}{EI}} \times l = \frac{\pi}{2}$
(*e*) none of the above.

13.250. Compression members tend to buckle in the direction of
(*a*) minimum cross section
(*b*) axis of load
(*c*) perpendicular to axis of load
(*d*) least radius of gyration
(*e*) any one of the above.

13.251. If a car turns towards right, man sitting inside will move towards
(*a*) right (*b*) left
(*c*) remain erect (*d*) unpredictable
(*e*) none of the above.

13.252. Diameter of washer is generally taken
(*a*) equal to nut size
(*b*) less than nut size
(*c*) bigger than nut size
(*d*) any size irrespective of nut size
(*e*) none of the above.

13.253. Splined shaft is used in applications
(*a*) in which stress concentration due to deep keyway is to be avoided
(*b*) high torque is to be transmitted
(*c*) high r.m.p.
(*d*) calling for axial relative movcment between shaft and hub
(*e*) involving locking devices.

13.254. Which type of gear will be used for non-intersecting perpendicular shafts
(*a*) helical gears (*b*) worm gears
(*c*) hypoid gears (*d*) herringbone gears
(*e*) none of the above.

13.255. Addendum of a cycloidal gear tooth is
(*a*) cycloid (*b*) involute
(*c*) epicycloid (*d*) hypocycloid
(*e*) straight rack.

13.256. If the lead angle of a worm is $22\frac{1}{2}°$ then the helix angle will be
(*a*) $22\frac{1}{2}°$ (*b*) 45°
(*c*) $67\frac{1}{2}°$ (*d*) 90°
(*e*) $112\frac{1}{2}°$.

13.257. Which type of gear will be used to have minimum axial thrust
(*a*) helical gears (*b*) herringbone gears
(*c*) hypoid gears (*d*) worm gears
(*e*) none of the above.

13.258. Spiral gears are suitable for transmitting
(*a*) small power (*b*) huge power
(*c*) no power but motion only
(*d*) any power
(*e*) pulsating power.

13.259. Spring driven watches and clocks utilise

(*a*) involute gears
(*b*) cycloid gears
(*c*) epicycloid gears
(*d*) straight rack gears
(*e*) none of the above.

13.260. Which type of gear will be used for non-parallel and non-intersecting shafts
(*a*) helical gears
(*b*) hypoid gears
(*c*) worm gears
(*d*) herringbone gears
(*e*) none of the above.

13.261. 10 m of water column is equal to
(*a*) 10 kN/m^2 (*b*) 1 kN/m^2
(*c*) 100 kN/m^2 (*d*) 0.1 kN/m^2
(*e*) none of the above.

13.262. As pump speed increases, its NPSH (net positive suction head) requirement
(*a*) increases
(*b*) decreases
(*c*) remains same
(*d*) unpredictable
(*e*) none of the above.

13.263. If pump NPHS requirements are not satisfied
(*a*) it will not develop head
(*b*) it will be cavitated
(*c*) efficiency will be low
(*d*) it will consume excessive power
(*e*) none of the above.

13.264. The abbreviation ERW in ERW pipe stands for
(*a*) electrically resistance welded
(*b*) elastic reinforced with wire
(*c*) extra reinforcement welded
(*d*) electrically reinforced and welded
(*e*) all of the above.

13.265. One joule is equal to
(*a*) 0.23×10^{-3} kcal
(*b*) 0.102 kg m/s
(*c*) 1 kg m/s
(*d*) all the above
(*e*) not of the above.

13.266. One watt is equal to
(*a*) 0.23×10^{-3} kcal
(*b*) 0.102 kg m/s
(*c*) 1 kg m/s
(*d*) all of the above
(*e*) none of the above.

13.267. One otto is equal to
(*a*) 10^{-15} (*b*) 10^9
(*c*) 10^{-18} (*d*) 10^{12}
(*e*) 10^{15}.

13.268. One femto is equal to
(*a*) 10^{-15} (*b*) 10^{-9}
(*c*) 10^{-18} (*d*) 10^{12}
(*e*) 10^{15}

13.269. One tera is equal to
(*a*) 10^{-15} (*b*) 10^9
(*c*) 10^{-18} (*d*) 10^{12}
(*e*) 10^{15}.

13.270. One giga is equal to
(*a*) 10^{-15} (*b*) 10^9
(*c*) 10^{-18} (*d*) 10^{12}
(*e*) 10^{15}.

13.271. Units of thermal conductivity are
(*a*) Ns/m^2 (*b*) W/m°K
(*c*) J/kg°K (*d*) J/°K
(*e*) none of the above.

13.272. Units of entropy are
(*a*) Ns/m^2 (*b*) W/m°K
(*c*) J/kg°K (*d*) J/°K
(*e*) none of the above.

13.273. Units of specific heat are
(*a*) Ns/m^2 (*b*) W/m°K
(*c*) J/kg°K (*d*) J/°K
(*e*) none of the above.

13.274. Units of dynamic viscosity are
(*a*) Ns/m^2 (*b*) W/m°K
(*c*) J/kg°K (*d*) J/°K
(*e*) none of the above.

13.275. Series wound D.C. motor
(*a*) should always be started without load on
(*b*) should never be started without load on
(*c*) whether load is connected or not is immaterial
(*d*) is started with average load
(*e*) none of the above.

13.276. An alternator having 40 poles rotates at 150 r.p.m. It will generate A.C. voltage at frequency of
(*a*) 50 c/s (*b*) 60 c/s
(*c*) 100 c/s (*d*) 40 K Hz

(e) 60 K Hz.

13.277. In shunt wound D.C. motor
(a) speed is infinity at low load and very less at high load
(b) speed is maximum at no load and drops by 10 to 12% at full load
(c) speed remains constant at all the loads
(d) speed increases with increase in load
(e) none of the above.

13.278. In induction motors, power supply is connected to
(a) rotor only
(b) stator only
(c) both rotor and stator
(d) any one of (a) or (b) above
(e) none of the above.

13.279. The speed of synchronous motor having 8 poles and operating at 50 c/s power supply will be
(a) 1000 r.p.m. (b) 600 r.p.m.
(c) 750 r.p.m. (d) 800 r.p.m.
(e) 1500 r.p.m.

13.280. Which is false statement about induction motors
(a) its speed decreases with increase in load
(b) its direction of rotation can be changed by interchanging two phases
(c) its speed can't be controlled without sacrificing efficiency
(d) its efficiency is quite high
(e) its starting torque is more than d.c. shunt motor.

13.281. Rated life of a ball bearing in relation to load (P) varies as
(a) P (b) P^2
(c) P^3 (d) $1/P^2$
(e) $1/P^3$

13.282. The phenomenon of hunting is observed in
(a) induction motors
(b) d.c. shunt motors
(c) d.c. series motor
(d) synchronous motor
(e) asynchronous motor.

13.283. Helical springs are not subjected to
(a) hoop stress (b) force
(c) deflection
(d) torsional shear stress
(e) transverse shear stress.

13.284. The welding units operate at following power factor
(a) 0.3 (b) 0.6
(c) 0.8 (d) 0.9
(e) 1.0.

13.285. For traction applications, following type of motor is used
(a) induction motor
(b) synchronous motor
(c) D.C. series motor
(d) D.C. shunt motor
(e) none of the above.

13.286. The most commonly used motor in applications like elevator, machine tools is
(a) D.C. series motor
(b) D.C. shunt motor
(c) induction (squirrel cage) motor
(d) synchronous motor
(e) none of the above.

13.287. In a simply supported beam, where the shear force is zero, the bending moment will be
(a) zero (b) maximum
(c) minimum
(d) could be anything
(e) none of the above.

13.288. In stroboscopic measurement, a disc having one mark was seen to have two similar marks at 40 cps signal. Its speed is
(a) 1200 r.p.m. (b) 600 r.p.m.
(c) 2400 r.p.m. (d) 4800 r.p.m.
(e) 1800 r.p.m.

13.289. An involute gear should have minimum of
(a) 8 teeth (b) 12 teeth
(c) 16 teeth (d) 20 teeth
(e) 32 teeth.

13.290. In order to avoid interference for 20° pressure angle teeth, minimum number of teeth should be
(a) 12 (b) 18
(c) more than 11 (d) less than 18
(e) none of the above.

13.291. The shearing stress in a helical spring of wire diameter d and having mean diameter D, supporting a compressive load F is given by

(a) $\frac{2FD}{\pi d^3} \times K$ (b) $\frac{4FD}{\pi d^3} \times K$

(c) $\frac{8FD}{\pi d^3} \times K$ (d) $\frac{16FD}{\pi d^3} \times K$

(e) $\frac{32FD}{\pi d^3} \times K$.

13.292. The Wahl stress factor K for springs of spring index

$C = \frac{D}{d} = \frac{\text{Mean dia of coil}}{\text{Wire diameter}}$ is given by

(a) $\frac{4C-1}{4C-2} + \frac{0.615}{C}$

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

(c) $\frac{4C-4}{4C-1} + \frac{0.165}{C}$

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

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

13.293. Music wire is concerned with
(a) musical instruments
(b) tuning forks
(c) springs
(d) shafts
(e) measuring instruments.

13.294. Value of Wahl stress factor K for springs with increase in value of C
(a) decreases linearly
(b) increases linearly
(c) remains same
(d) decreases exponentially
(e) increases exponentially.

13.295. Which is correct statement
Fatigue cracks can spread only by
(a) tensile stress (and not by compressive or shear) and in directions perpendicular to the tensile stress
(b) tensile stress and in direction along the tensile stress
(c) compressive and shear stresses
(d) any of the three types of stresses
(e) shear stress.

13.296. The spring rate of conical and volute springs, with increase in load
(a) remains constant
(b) decreases
(c) increases
(d) increases after the largest active coil starts to "bottom"
(e) depends on other considerations.

13.297. The deflection of helical spring is directly and inversely proportional respectively to

(a) D^2, d^2 (b) D^3, d^2
(c) D^4, d^3 (d) D^3, d^4
(e) D^4, d^4.

where D = mean diameter of coil
and d = wire diameter.

13.298. Concentric helical springs should be
(a) wound in same direction
(b) wound with opposite hand helices
(c) could be wound in any direction
(d) direction of winding depends on the load to be carried
(e) none of the above.

13.299. Curvature correction factor K for helical compression spring with spring index varies as (Refer Fig. 13.10)

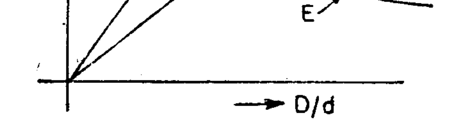

Fig. 13.10.

(a) Curve A (b) Curve B
(c) Curve C (d) Curve D
(e) Curve E.

13.300. Allowable stresses in compression springs for most of the materials with increase in size of wire will
(a) increase (b) decrease
(c) remain same (d) unpredictable
(e) none of the above.

13.301. Which is true statement about Belleville springs

(*a*) these are used for dynamic loads
(*b*) these are composed of coned discs which may be stacked up to obtain variety of load-deflection characteristics
(*c*) these are commonly used in clocks and watches
(*d*) these take up torsional loads
(*e*) these do not exist.

13.302. Angle of twist of shaft is inversely proportional to
(*a*) shaft diameter
(*b*) (shaft diameter)^2
(*c*) (shaft diameter)^3
(*d*) (shaft diameter)^4
(*e*) (shaft diameter)$^{3/2}$.

13.303. For a shaft subjected to a torque T and bending moment M, the equivalent twisting moment is
(*a*) $\sqrt{\dfrac{T^2+M^2}{2}}$ (*b*) $\sqrt{M^2-T^2}$
(*c*) $\dfrac{M}{2}+\sqrt{M^2+T^2}$ (*d*) $\sqrt{M^2+T^2}$
(*e*) $\sqrt{T^2+\left(\dfrac{M}{2}\right)^2}$.

13.304. A shaft subjected to combined bending and torsion can be designed by following stress theory
(*a*) maximum shear stress theory
(*b*) maximum normal stress theory
(*c*) maximum resultant stress theory
(*d*) all of the above
(*e*) none of the above.

13.305. The maximum shear stress induced in a shaft subjected to a shear stress s_s and bending stress s_t will be
(*a*) $\sqrt{s_s^2+s_t^2}$ (*b*) $s_s+\dfrac{1}{2}\sqrt{s_s^2+s_t^2}$
(*c*) $\sqrt{\left(s_s^2+\dfrac{s_t^2}{4}\right)}$ (*d*) $\sqrt{4s_s^2+s_t^2}$
(*e*) $\dfrac{1}{2}\sqrt{s_s^2+\left(\dfrac{s_t}{2}\right)^2}$.

13.306. Which is correct statement
(*a*) cold rolling produces a stronger shaft than hot rolling
(*b*) hot rolling produces a stronger shaft than cold rolling
(*c*) both make equally strong shafts
(*d*) shafts are not made by rolling process
(*e*) extruded shafts are most commonly used.

13.307. Stretching in a belt can be controlled by
(*a*) decreasing belt length
(*b*) increasing centre distance
(*c*) increasing pulley diameter
(*d*) reducing belt velocity
(*e*) reducing stress in the belt.

13.308. In a horizontal flat belt drive, it is customary to use
(*a*) bottom side of the belt as the slack side during the transmission of power
(*b*) top side of the belt as the slack side
(*c*) crossed-belting
(*d*) idler in between
(*e*) none of the above.

13.309. Centrifugal tension in belts
(*a*) reduces power transmission
(*b*) increases power transmission
(*c*) does not affect power transmission
(*d*) increases power transmission at high speed and decreases it at lower speed
(*e*) unpredictable.

13.310. Maximum horse power is transmitted by a belt drive when its velocity is such that the tight side driving tension in belt is equal to
(*a*) centrifugal tension
(*b*) 2 × centrifugal tension
(*c*) 3 × centrifugal tension
(*d*) 4 × centrifugal tension
(*e*) none of the above.

13.311. If F_1 and F_2 be the tight and slack side tensions in the belt, then initial tension will be equal to
(*a*) F_1-F_2 (*b*) F_1+F_2
(*c*) $\dfrac{F_1+F_2}{2}$ (*d*) $(\sqrt{F_1}+\sqrt{F_2})^2$
(*e*) $\left(\dfrac{\sqrt{F_1}+\sqrt{F_1}}{2}\right)^2$.

13.312. If F_1, F_2 and F_c be the tight side, slack side and centrifugal tensions in a belt, and μ and θ be the coefficient of friction between belt and pulley, and angle of contact, then

(a) $\frac{F_1}{F_2} = e^{\mu\theta}$ (b) $\frac{F_1 + F_c}{F_2 + F_c} = e^{\mu\theta}$

(c) $F_1 + F_2 = 4F_c$. (d) $\frac{F_1 - F_c}{F_2 - F_c} = \frac{e^{\mu\theta} - 1}{e^{\mu\theta}}$

(e) $\frac{F_1 - F_c}{F_2 - F_c} = e^{\mu\theta}$

13.313. In a V-belt drive, the belt makes contact at
(a) bottom of pulley
(b) sides of the groove of pulley
(c) sides of groove and bottom of pulley
(d) could make contact anywhere
(e) none of the above.

13.314. If F, g and w represent the tension, acceleration due to gravity and mass per unit length of belt respectively, then maximum permissible speed of belt is given by
(a) $\sqrt{\frac{Fg}{3w}}$ (b) $\sqrt{\frac{3Fg}{w}}$
(c) $\sqrt{\frac{2Fg}{3w}}$ (d) $\sqrt{\frac{Fg}{w}}$
(e) $\sqrt{\frac{3Fg}{2w}}$

13.315. In the above prob., belt drive can't transmit any power at speed of
(a) $\sqrt{\frac{Fg}{3w}}$ (b) $\sqrt{\frac{3Fg}{w}}$
(c) $\sqrt{\frac{2Fg}{3w}}$ (d) $\sqrt{\frac{Fg}{w}}$
(e) $\sqrt{\frac{3Fg}{2w}}$

13.316. The standard angle between the sides of V-belt is
(a) 25° (b) 30°
(c) 40° (d) 45°
(e) 60°.

13.317. 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 set
(b) only one belt can't be fitted with other used belts
(c) the new belt would carry more than its share and would have a short life
(d) such an arrangement would cause heavy vibration
(e) one belt can not be replaced.

13.318. A chain drive is used for
(a) short distances
(b) longer distances
(c) medium distances
(d) distance is no criterion for chain drive
(e) depends on load to be transmitted.

13.319. For a chain drive to have variation in speed of less than 1%, the minimum number of teeth in the small sprocket should be
(a) 11 (b) 17
(c) 24 (d) 33
(e) 45.

13.320. It is usually preferable in chain drive to use
(a) even number of teeth on sprocket
(b) odd number of teeth on sprocket
(c) either even or odd, but certain minimum number
(d) maximum number of teeth permissible on sprocket
(e) none of the above.

13.321. Following type of chain is used in motor cycle
(a) Bush roller (b) Silent
(c) Pintle (d) Ewast
(e) none of the above.

13.322. Silent chain is made of
(a) links and blocks
(b) links, pins, bushings and rollers
(c) 3 or more roller chains
(d) inverted tooth over-lapping links
(e) none of the above.

13.323. In designation 6 by 19 wire rope, 6 and 19 respectively stand for
(a) diameter of wire rope and number of strands
(b) diameter of wire rope and number of wires
(c) number of wires and number of strands
(d) number of strands and number of wires
(e) none of the above.

13.324. Which of the following ropes will be most flexible
(a) 6 by 7 (b) 6 by 19
(c) 8 by 19 (d) 6 by 37

(e) all are equally flexible.

13.325. Which is correct statement about flexibility and endurance of ropes
(a) lang lay rope is more flexible and endurable than regular lay rope
(b) regular lay rope is more flexible and endurable than lang lay rope
(c) both are equally good
(d) other factors decide these considerations
(e) none of the above.

13.326. Which is correct statement about drums and sheaves used in wire rope installations
(a) use largest size drum, and flat type
(b) use largest size drum with grooves, the pitch of grooves being more than the wire rope diameter
(c) use smallest size drum with grooves, the pitch of grooves being more than the wire rope diameter
(d) use largest size drum with grooves whose pitch is less than wire rope diameter
(e) none of the above.

13.327. The working load (P) for a chain for crane applications is expressed in terms of diameter of link 'd' in cm as follows
(a) $P = 1.5\,d^2$ (b) $P = 25\,d^2$
(c) $P = 50\,d^2$ (d) $P = 250\,d^2$
(e) $P = 500d^2$.

13.328. Wire ropes are used for applications experiencing
(a) low speeds and low tension
(b) low speeds and high tension
(c) high speeds and low tension
(d) high speed and high tension
(e) there is no such criterion of speed or tension.

13.329. The friction moment in clutches with assumption of uniform wear as compared to uniform pressure is
(a) more
(b) less
(c) same
(d) more/less depending on speed
(e) unpredictable.

13.330. For maximum h.p. transmission by a belt drive
(a) the centrifugal tension should be zero
(b) 50% of maximum tension should be utilised as centrifugal tension
(c) 33% of maximum tension should be utilised as centrifugal tension
(d) difference of tight side and slack side tension should be equal to centrifugal tension
(e) belt speed should be more than 100 m/sec.

13.331. If ϕ be the angle of friction, then radius of friction circle is given by
(a) r (b) $r \sin \phi$
(c) $r \cos \phi$ (d) $r \sin^2 \phi$
(e) $r \cos^2 \phi$.

13.332. Friction radius for new clutches compared to worn-out will be
(a) same (b) more
(c) less
(d) depends on overall size of clutch
(e) none of the above.

13.333. For new clutches and brakes, friction radius is equal to
(a) $\frac{D+d}{4}$ (b) $\frac{1}{3}\frac{D^3-d^3}{D^2-d^2}$
(c) $\frac{1}{2}\frac{D^3-d^3}{D^2-d^2}$ (d) $\frac{1}{4}\frac{D^3-d^3}{D^2-d^2}$
(e) none of the above

where D and d are outer and inner diameters.

13.334. For uniform wear condition of brakes and clutches : friction radius is equal to
(a) $\frac{D+d}{4}$ (b) $\frac{1}{3}\frac{D^3-d^3}{D^2-d^2}$
(c) $\frac{1}{2}\frac{D^3-d^3}{D^2-d^2}$ (d) $\frac{1}{4}\frac{D^3-d^3}{D^2-d^2}$
(e) none of the above.

13.335. The commonly used angle between the cone surface and horizontal axis for a cone clutch utilising leather to asbestos lining is about
(a) 8° (b) 12.5°
(c) 20° (d) 25°
(e) 30°.

13.336. In a cone clutch, a given torque can be transmitted by a relatively small axial force if the cone-face angle is
(a) more

(*b*) less
(*c*) any angle
(*d*) depends on power to be transmitted and speed
(*e*) none of the above.

13.337. For a flat pivot bearing of radius 'r', the moment arm of the frictional force with the assumption of uniform pressure is
(*a*) r (*b*) $r/2$
(*c*) $2r/3$ (*d*) $3r/4$
(*e*) $r/3$.

13.338. For a block brake, the equivalent coefficient of friction is equal to
(*a*) $\mu \frac{4 \sin \theta}{2\theta + \sin 2\theta}$ (*b*) $\mu \frac{2 \sin \theta}{2\theta + \sin 2\theta}$
(*c*) $\mu \frac{4 \sin 2\theta}{2\theta + \sin 2\theta}$ (*d*) $\mu \frac{\sin 2\theta}{4\theta + \sin 2\theta}$
(*e*) $\mu \frac{2 \sin \theta}{4\theta + \sin 2\theta}$.
where θ = semiblockangle
μ = coeff. of friction of material of block and wheel.

13.339. In blok brakes, the ratio of shoe width and wheel diameter is kept between
(*a*) 0.1 and 0.25 (*b*) 0.25 and 0.50
(*b*) 0.50 and 0.75 (*d*) 0.75 and 1.0
(*e*) none of the above.

13.340. The percentage of total brake effort that results from self energising action depends on
(*a*) the location of the brake arm pivot point
(*b*) the coefficient of friction
(*c*) the direction of rotation of the brake drum
(*d*) all of the above
(*e*) none of the above.

13.341. In order to prevent the brake arm from grabbing, the moment of friction force about the brake arm pivot point should be
(*a*) less than the total required braking effort
(*b*) greater than the total required braking effort
(*c*) equal to the total required braking effort
(*d*) zero (*e*) none of the above.

13.342. For a spur gear, the product of circular pitch and diametral pitch is equal to
(*a*) unity (*b*) $\frac{1}{\pi}$
(*c*) π (*d*) module
(*e*) pitch circle diameter.

13.343. In an involute gear, the base circle must be
(*a*) at root circle
(*b*) under root circle
(*c*) above root circle
(*d*) under pitch circle
(*c*) above pitch circle.

13.344. The part of the tooth between the pitch circle and dedendum circle is called
(*a*) half tooth (*b*) flank
(*c*) face (*d*) upper tooth
(*e*) lower tooth.

13.345. Stub tooth in gears
(*a*) is standard tooth
(*b*) is longer than standard tooth
(*c*) is shorter than standard tooth
(*d*) has special profile
(*e*) is used where great precision in transmission is required.

13.346. The minimum number of teeth which can be cut for standard tooth, for given pressure angle ϕ is equal to
(*a*) $\frac{\sin^2 \phi}{2}$ (*b*) $\frac{2}{\sin^2 \phi}$
(*c*) $2 \sin^2 \phi$ (*d*) $\frac{2}{\sin \phi}$
(*e*) $\frac{2}{\sin 2\phi}$.

13.347. Backlash is spur gears is the
(*a*) difference between the dedendum of one gear and the addendum of the mating gear
(*b*) difference between the tooth space of one gear and the tooth thickness of the mating gear measured on the pitch circle
(*c*) intentional extension of centre distance between two gears
(*d*) does not exist
(*e*) none of the above.

13.348. In which type of teeth, variation in centre distance within limits does not affect the velocity ratio of the mating gears
(*a*) cycloidal (*b*) involute
(*c*) hypoid (*d*) all of the above

(e) none of the above.

13.349. Which of the following tooth profiles can take very heavy load
(a) $14\frac{1}{2}°$ composite system
(b) $14\frac{1}{2}°$ full depth involute system
(c) 20° full depth involute
(d) $14\frac{1}{2}°$ stub involute
(e) 20° stub involute.

13.350. The interference in cycloidal teeth
(a) is absent completely
(b) depends on number of teeth
(c) depends on conditions of meshing
(d) depends on pressure angle
(e) is maximum.

13.351. In cycloidal gears, the work wasted in friction will be least, when for a given total arc of action, the arc of approach is
(a) greater than arc of recess
(b) less than arc of recess
(c) equal to arc of recess
(d) there is no such criterion
(e) none of the above.

13.352. The angle through which gear turns from the beginning of contact of a pair of teeth upto pitch point is called
(a) pitch angle
(b) pressure angle
(c) angle of approach
(d) angle of action
(e) angle of contact.

13.353. Lewis equation in gears is used to find the
(a) tensile stress
(b) compressive stress in bending
(c) contact stress
(d) fatigue stress
(e) endurance stress.

13.354. Involute profiles in gears are very popular because of the following advantage
(a) pressure angle is constant
(b) face and flank of a tooth form a continuous curve
(c) all gears having the same pitch and pressure angle work correctly together
(d) involute rack is a straight line
(e) all of the above.

13.355. Gear teeth are made harder to avoid
(a) greater compressive stress in bending
(b) tensile strength
(c) abrasion (d) pitting
(e) wear.

13.356. Clearance in spur gears is the
(a) difference between the dedendum of one gear and the addendum of the mating gear
(b) difference between the tooth space of one gear and the tooth thichness of the mating gear measured on the pitch circle
(c) intentional extension of centre distance between two gears
(d) does not exist
(e) none of the above.

13.357. Miter gears are
(a) right angled bevel gears having same number of teeth
(b) spur gears of equal diameter and pitch
(c) helical gears of same module
(d) gears of different module
(e) a kind of worm wheel and gear.

13.358. For proper meshing of worm and worm wheel, normal pitch of worm compared to normal pitch of worm wheel should be
(a) more
(b) less
(c) equal
(d) any one of the above
(e) none of the above.

13.359. Low pressure angle gears result in
(a) stronger teeth
(b) weaker teeth
(c) strength has nothing to do with pressure angle
(d) could be stronger or weaker depending on module adopted
(e) none of the above.

13.360. The wear on the gear teeth can be equalised (while a large and a small gear are running together) by
(a) making the gear harder than the pinion
(b) making the pinion harder than the gear
(c) making both gear and pinion of same hardness

(*d*) using non-metallic materials for both
(*e*) using non-ferrous materials for both.

13.361. The following materials give corrosion resistance to gear sets
(*a*) hard materials
(*b*) hardened steel
(*c*) non-ferrous materials
(*d*) stainless steel
(*e*) softer materials.

13.362. Larger pressure angles in comparison to smaller pressure angles make the gear
(*a*) weaker
(*b*) stronger
(*c*) have no effect as regards strength
(*d*) increase wear
(*e*) consume more power.

13.363. Fine pitch involute spur gears are those having diametral pitch
(*a*) greater than 20
(*b*) less than 20
(*c*) diametral pitch is not concerned with fineness of pitch
(*d*) equal to zero
(*e*) none of the above.

13.364. The value of form factor used in design of gear is
(*a*) independent of the size of the tooth
(*b*) depends on the number of teeth on a gear
(*c*) depends on the system of the teeth
(*d*) all of the above
(*e*) (*b*) and (*c*) above.

13.365. Larger helix angles in opposite hand helical gears result in
(*a*) smooth and quiet operation
(*b*) strong teeth
(*c*) both (*a*) and (*b*) above
(*d*) noisy operation and weaker teeth
(*e*) noisy operation and stronger teeth.

13.366. Stub teeth are cut on gears in order to
(*a*) increase capability to withstand shocks and vibrations
(*b*) reduce noise
(*c*) improve transmission efficiency
(*d*) reduce centre distance
(*e*) transmit huge power.

13.367. If the lead angle of a worm is 20°, then helix angle will be
(*a*) 20° (*b*) 70°
(*c*) 10° (*d*) 80°
(*e*) none of the above.

13.368. In zerol bevel gears, the axes
(*a*) are non-parallel and non-intersecting, and the teeth are curved
(*b*) are non-parallel and non-intersecting, and the teeth are straight
(*c*) intersect, and the teeth are curved and oblique
(*d*) intersect, and the teeth are curved and can be ground
(*e*) none of the above.

13.369. Compared to spur gears, helical gears
(*a*) run more smoothly
(*b*) run with more vibrations and noise
(*c*) run exactly alike
(*d*) consume more power
(*e*) consume less power.

13.370. Two helical gears of the same hand and a 45 degree helix angle are in mesh. The shaft of the two gears would be at following angle to each other
(*a*) 45°
(*b*) 90°
(*c*) $22\frac{1}{2}°$
(*d*) could be at any angle
(*e*) none of the above.

13.371. The gear reduction of a worm gear set with worm gear of 50 teeth and worm of double lead thread would be
(*a*) 50 : 1
(*b*) 100 : 1
(*c*) 25 : 1
(*d*) any one of the above
(*e*) none of the above.

13.372. The following material should be used for gears to run quietly at high speed
(*a*) harder steel
(*b*) softer material
(*c*) non-ferrous material
(*d*) non-metallic material
(*e*) stainless steel.

13.373. Interference is inherently absent in following type of gears
(*a*) involute (*b*) stub
(*c*) cycloidal (*d*) epicycloid
(*e*) hypocycloid.

13.374. In hypoid gears, the axes