ACE
Engineering Academy

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ESE- 2018 (Prelims) - Offline Test Series **Test - 3**

MECHANICAL ENGINEERING

SUBJECT: MECHANISMS AND MACHINES + DESIGN OF MACHINE ELEMENTS - SOLUTIONS

01. Ans: (d)

Sol: Amplitude = 1cm

Time period = 8 sec

$$\Rightarrow \omega = \frac{2\pi}{T} = \frac{2\pi}{8} = \frac{\pi}{4}\text{rad/s}$$

$$x = A\sin\omega t = 1(\text{cm})\sin\left(\frac{\pi}{4}t\right)$$

$$\text{Acceleration} = \frac{d^2x}{dt^2} = -\left(\frac{\pi}{4}\right)^2 \times 1\sin\left(\frac{\pi}{4}t\right)$$

$$\text{Acceleration}\left(\text{at } t = \frac{4}{3}\text{s}\right) = -\frac{\pi^2}{16}\times 1\times\frac{\sqrt{3}}{2}$$

$$= \frac{-\sqrt{3}\pi^2}{32}\text{cm/s}^2$$

02. Ans: (a)

Sol: A shaft supported by long bearings is assumed to have both ends fixed but when supported in short bearings it is considered to be simply supported. The natural frequency in transverse vibration is called critical speed.

$$\omega_n = \sqrt{\frac{g}{\delta_{st}}}$$

$$(\delta_{st})_{\text{long bearing}} = \frac{mgL^3}{192EI}$$

$$(\delta_{st})_{\text{short bearing}} = \frac{mgL^3}{48EI}$$

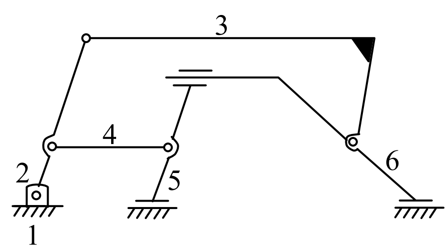
$$N = \frac{60}{2\pi}\sqrt{\frac{g}{\delta_{st}}} = \frac{60}{2\pi}\times\sqrt{\frac{192EI}{mL^3}}$$

$$N_1 = \frac{60}{2\pi}\sqrt{\frac{48EI}{\left(\frac{m}{4}\right)\times L^3}} = \sqrt{\frac{192EI}{mL^3}}$$

$$\therefore N_1 = N$$

03. Ans: (c)

Sol:



The prismatic pair connecting links 5 & 6 should not be taken into consideration because the relative motion between these two links is determined by the prismatic pairs between 1 & 6 and 1 & 5, the prismatic pair connecting links 5 & 6 is not imposing any independent constraint.

$N = 6$, $J_1 = 8$, $J_r = 1$ (where J_1 = number of lower pairs, J_r = number of redundant lower pairs)

Kinematic pair between links 5 & 6 is redundant (prismatic pair)

Effective DOF $= 3(N-1) - 2(J_1 - J_r)$

$= 3(6-1) - 2\times(8-1) = 1$

04. Ans: (a)

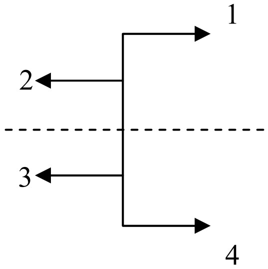
Sol: The firing order for in-line four cylinder four stroke engine is 1-3-4-2.

$$\beta = \frac{4\pi}{\text{no. of cylinders}} = \frac{4\pi}{4} = \pi$$

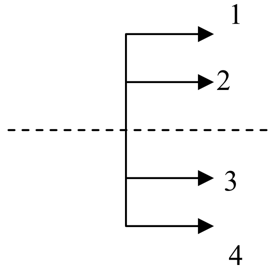
Magnitude of primary force = $mr\omega^2 \cos\theta$

Magnitude of secondary force = $\dfrac{mr\omega^2 \cos 2\theta}{n}$

Primary Force diagram for firing order 1-3-4-2



Secondary Force diagram for firing order 1-3-4-2



∴ From above diagram we can see that primary forces, primary couples and secondary couples are balanced whereas secondary forces are unbalanced.

05. Ans: (d)

Sol: According to total strain energy theory,

$$\sigma_1^2 + \sigma_2^2 - 2\mu\sigma_1\sigma_2 = (S_{yt})^2$$

Under pure torsion,

$$\tau^2 + (-\tau)^2 + 2\mu\tau^2 = (S_{yt})^2$$

$$S_{ys}^2 + S_{ys}^2 + 2\mu S_{ys}^2 = (S_{yt})^2$$

$$\frac{S_{yt}}{S_{ys}} = \sqrt{2(1+\mu)}$$

Where, S_{yt}, S_{ys} → Tensile and shear strengths

06. Ans : (b)

Sol: $\sigma_1 = 120$ MPa, $\sigma_2 = 70$ MPa ,

$S_{yt} = 240$ MPa.

According to Maximum Normal Stress Theory

$$\frac{S_{yt}}{FS} = 120$$

$$\frac{240}{FS} = 120 \Rightarrow FS = 2$$

07. Ans (c)

Sol: $T_{eq} = \sqrt{M^2 + T^2}$

$\Rightarrow 5 = \sqrt{M^2 + 4^2}$

$\Rightarrow M = 3$ kN-m

08. Ans: (a)

Sol: In spur gear the Tangential component of force transmits the torque and radial component of force causes bending.

09. Ans: (c)

Sol: Scoring is due to Excessive surface pressure, high surface speed and inadequate supply of Lubricant results in breakdown of the Lubricant film which is called as Scuffing.

Pitting is a fatigue failure which occurs when the load on the bearing part exceeds the surface endurance strength of the material.

10. Ans: (a)

Sol: Initial deflection of the spring,

$$\delta_1 = \frac{(m_1 + m_2)g}{k}$$

When mass m_1 is removed only, mass m_2 undergoes free vibrations. At this instant, velocity of mass m_2 is zero. So it is in extreme position.

At mean position, $\delta_2 = \dfrac{m_2 g}{k}$

The amplitude of the oscillation of mass m_2

$$= \frac{(m_1 + m_2)g}{k} - \frac{m_2 g}{k} = \frac{m_1 g}{k}$$

11. Ans: (c)

Sol: $\omega_2 = 10$ rad/s, $V_s = 1$ m/sec

When the crank is perpendicular to line of stroke,

$V_s = r\omega_2$

$\Rightarrow 1 = r \times 10$

$\Rightarrow r = 0.1$ m

$\Rightarrow l = 4r = 0.4$ m

12. Ans: (a)

Sol: By energy method

Total Mechanical energy = $\dfrac{1}{2}kA^2$

($\because$ At extreme position kinetic energy = 0)

$K = m\omega_n^2$

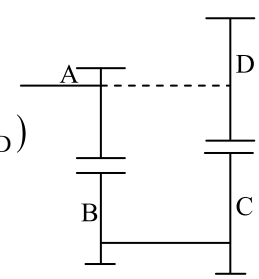
$\omega_n = 2\pi f = 2 \times \pi \times \dfrac{25}{\pi} = 50$ rad/s

$\dfrac{1}{2} \times m \times \omega_n^2 \times A^2 = 0.9$

Solving, we get, A = 0.06 m

13. Ans: (b)

Sol: $N_A = 600$ rpm (CCW)



$$\frac{m}{2}(T_A + T_B) = \frac{m}{2}(T_C + T_D)$$

$\Rightarrow 24 + 48 = 24 + T_D$

$\Rightarrow T_D = 48$

$$\frac{N_D}{N_A} = \frac{-T_A}{T_B} \times \frac{-T_C}{T_D} = \frac{24}{48} \times \frac{24}{48} = \frac{1}{4}$$

$$N_D = \frac{N_A}{4} = \frac{60}{4} = 150 \text{ rpm (CCW)}$$

14. Ans: (b)

Sol:

- The flywheel limits the fluctuations of speed during each cycle which arise from fluctuations of turning moment on the crank shaft.
- In non Grashof's kinematic chain, all inversions result in double rocker mechanism.
- When $\frac{\omega}{\omega_n} > \sqrt{2}$, the lesser amount of damping gives lower transmissibility which is always less than 1.

15. Ans: (c)

Sol: $\xi = \frac{C}{2\sqrt{km}}$,

$$\xi_1 = \frac{C}{2\sqrt{2mk}} = \frac{C}{\sqrt{2} \times 2\sqrt{mk}}$$

In a slider crank mechanism if the radius of crank (r) is equal to length of the connecting rod then stroke length = 4r

16. Ans: (a)

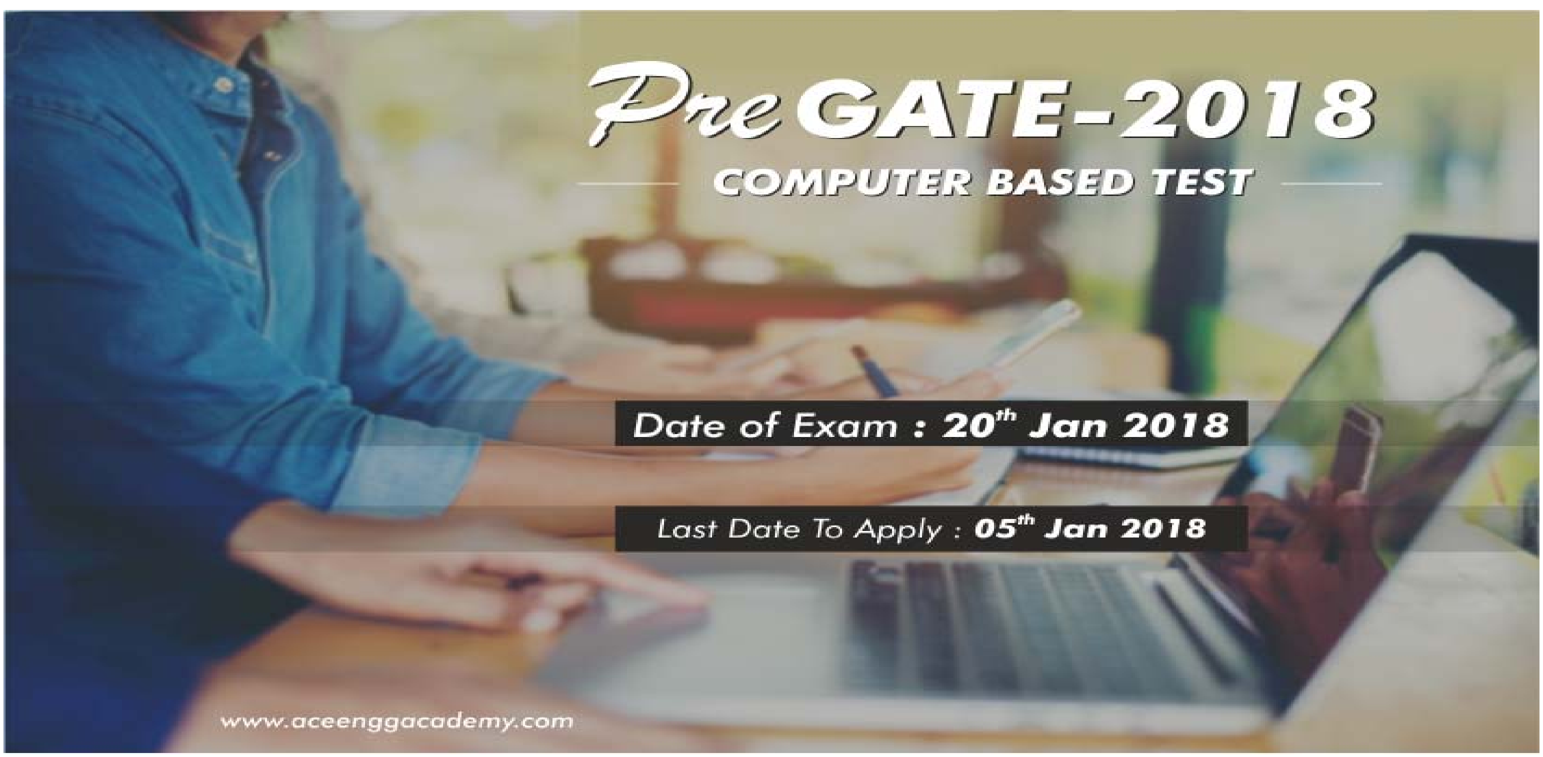
Sol: 20° stub teeth has following advantages and disadvantages,

(i) Stub teeth are stronger than full depth involute.

(ii) Interference is reduced due to shorter addendum.

(iii) Number of teeth requirement is less due to shorter addendum and hence, production cost is low.

(iv) Due to insufficient overlap between the mating parts, vibration is likely to occur.



17. Ans: (a)

Sol: Preloading is done to

1. To prevent leakage of fluid.
2. Increase the fatigue strength of bolt.
3. Increase the locking effect.

18. Ans: (b)

Sol: Shear load on weld

$P = 0.707 \times s \times \ell \times \tau_{max}$.

$S = 10mm$, $\ell = 1cm = 10$ mm

$\tau = 80$ MPa $= 8000$ N /cm^2

$P = 0.707 \times 10 \times 10 \times 80$

$= 5656$ N $= 5.65$ kN

19. Ans: (c)

Sol: Let $P = F_1 + F_2$ = Total load on weld

F_1, F_2 are load carried by upper weld & lower weld respectively.

$a = 30$ mm, $b = 20$ mm

$\Sigma M_{C.G} = 0$

$$\Rightarrow F_1 \times a = F_2 \times b = \frac{F_1 + F_2}{a + b}.$$

$$\Rightarrow F_1 = \left(\frac{b}{a+b}\right).P, F_2 = \left(\frac{a}{a+b}\right).P.$$

$F_1 = 0.707 \times s \times \ell_a \times \tau$.

$F_2 = 0.707 \times s \times \ell_b \times \tau$.

$P = 0.707 s(\ell_a + \ell_b)\tau$

$$\therefore 0.707 \times s \times l_a \times \tau = \left(\frac{b}{b+a}\right) \times 0.707 \times s \times (\ell_a + \ell_b) \times \tau$$

$$\ell_a = \frac{b}{a+b} \times \ell = \frac{20}{50} \times 100$$

$\Rightarrow \ell_a = 40$mm, $\ell_b = 60$mm.

20. Ans: (b)

Sol: Attitude, $\varepsilon = 1 - \frac{h_0}{c}$

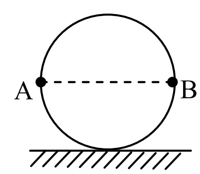
c = radial clearance $= \frac{0.1}{2} = 0.05$.

$$0.5 = 1 - \frac{h_0}{0.05} \Rightarrow h_0 = 0.025.$$

21. Ans: (c)

22. Ans: (b)

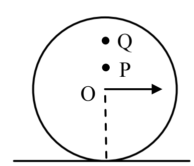
Sol:



- The points A and B have same speed
- The centre of the disc has zero acceleration
- The contact point has centripetal acceleration

23. Ans: (c)

Sol: $V_p = 8$ m/s



$V_q = 14$ m/s

$V_q - V_P = PQ.\omega = 6$

$PQ = 0.3$ m

$\therefore \omega = 20$ rad/s

Now, $(OQ)\omega = 14$

$\Rightarrow$ OQ = 700 mm

24. Ans: (d)

Sol: On increasing centre distance:

- Arc of contact changes ,
- Contact ratio changes $[\because AOC = \pi m \times n]$ where n = contact ratio; m = module
- Pressure angle increases and interference will decrease

25. Ans: (c)

Sol: In Involute gear profile pressure angle is constant at each point of contact. In Cycloidal gear variation of centre distance is not permitted otherwise it will affect the velocity ratio.

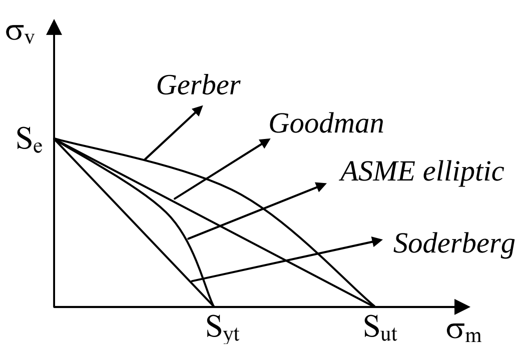
The relative velocity of sliding when two gears are in mesh = $(\omega_1 + \omega_2) \times PC$

At pitch point path of contact = 0

∴ The velocity of sliding is zero at the pitch point.

26. Ans: (c)

Sol:



So from figure, Gerber is the most non-conservative criterion.

27. Ans: (b)

Sol: $F_b = F_{ini} + k_b \dfrac{F_{ext}}{k_b + k_j}$

$k_j = 3\ k_b$

$F_b = 3 + k_b \times \dfrac{8}{k_b + 3k_b}$

$F_b = 3 + \dfrac{8}{4} = 5\,\text{kN}$

28. Ans: (a)

Sol: Due to preloading in bolts the stresses induced are:

1. A tensile stress in bolt due to initial tension $\sigma_i = \dfrac{P_i}{\frac{\pi}{4} d^2}$

2. Compressive stress in threads

$$\sigma_c = \frac{P_i}{\frac{\pi}{4}\left(d_{max}^{\ 2} - d_{min}^2\right)}$$

3. Torsional shear stress due to frictional resistance of thread during tightening

$$\tau = \frac{16T}{\pi d_c^3}$$

4. Bending stress occur when head or not perfectly normal to the bolt axis

5. Transverse shear stress across the thread

$$\tau = \frac{P}{\pi d_c h}$$

where, h = height of thread

29. Ans: (a)

Sol: We know that,

P.r = constant

$P_{max}.r_2 = C$

$C = 0.1\times100 = 10$ N/mm

Load (W) = $2\pi C(r_1 - r_2)$

$W = 2\pi\times10(200-100)$

$$W = \frac{2\pi\times10\times100}{1000} = 2\pi \text{ kN}$$

30. Ans: (c)

Sol: $$\mu = \frac{33}{10^3}\left\{\frac{ZN}{P}\right\}\left\{\frac{R}{C}\right\} + 0.002$$

$$\mu = \frac{33}{10^3}\times\frac{28\times10^{-3}\times2400}{1.4}\times100 + 0.002$$

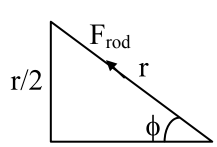
$\mu = 3.58\times10^{-3}$

31. Ans: (c)

Sol: If the relative motion between two links is both rolling and sliding, the relative instantaneous centre lies on the common normal to the surfaces of these links passing through the point of contact.

32. Ans: (a)

Sol: $\sin\phi = \frac{1}{2}$



$\phi = 30°$

$$F_{rod} = \frac{F_P}{\cos\phi}$$

$$F_{rod} = \frac{100}{\cos 30} = 115.4\,\text{N}$$

33. Ans: (c)

Sol: $I = 10$ kg-m^2, $\omega = 50$ rad/sec

$\Delta E = 250 = I\omega^2 C_s$

$$\Rightarrow 250 = 10\times50^2\times\left(\frac{\omega_1-\omega_2}{\omega}\right)$$

$$\Rightarrow 250 = 10\times50^2\times\frac{\Delta\omega}{50}$$

$\Rightarrow \Delta\omega = 0.5$ rad/sec

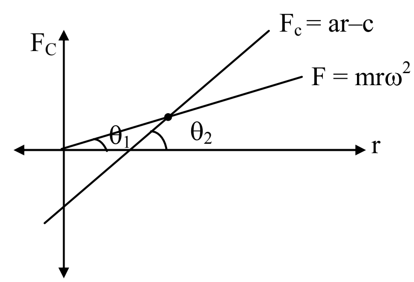
34. Ans: (b)

Sol: A governor is said to be stable if it brings the speed of the engine to the required value and there is not much hunting. The ball masses occupy a definite position for each speed. So stability and sensitivity are two opposite characteristics.

For Porter governor,

$$h_1 = \frac{g}{\omega_1^2}\left(1+\frac{M}{m}\right),\ h_2 = \frac{g}{\omega_2^2}\left(1+\frac{M}{m}\right),$$

For isochronism, $\omega_1 = \omega_2$ thus $h_1 = h_2$. However, from the configuration of Porter governor, it can be judged that it is impossible to have two positions of the balls at the same speed. Thus, a pendulum type of governor cannot possibly be isochronous.



For a stable governor, slope of the controlling force > slope of centrifugal force line ($\theta_2 > \theta_1$)

When load drops suddenly, throttle valve opens and the angular velocity of governor increases, therefore sleeve reaches top-most position.

35. Ans: (a)

Sol: The dedendum for stub teeth

= 1 module = 5 mm

Addendum for stub teeth = 0.8 module

36. Ans: (a)

Sol: $$\frac{x}{x_{static}} = \frac{1}{2\xi} \quad (\text{at resonance})$$

$$\xi = \frac{1}{5} = 0.2$$

37. Ans: (c)

Sol: Due to vibrations in actual running condition, the center distance between the gears changes and under such condition only involute profile satisfies law of gearing.

38. Ans : (d)

Sol: For all bearing $L = \left(\frac{C}{W}\right)^3$

Where, W = Equivalent Load.,

C = Dynamic load rating

$\therefore L \,\alpha \frac{1}{W^3}$

$LW^3 = \left(\frac{L}{2}\right)\left(W_1^3\right)$

$\Rightarrow W_1 = 2^{\frac{1}{3}}.W = 1.26W$

39. Ans: (b)

Sol: Equating the crushing strength to shear strength, we get,

$\frac{\pi}{4}d^2f_c = \pi dtf_s$

$\therefore$ Minimum diameter $= 4t\frac{f_s}{f_c}$

$= 4 \times 15 \times \frac{3}{6} = 30$ mm

40. Ans: (b)

Sol: Safe stress in flywheel rim is

$\sigma_{cf} = \rho V^2$

$\therefore\ V < \sqrt{\frac{\sigma_{cf}}{\rho}}$

But , $V = \frac{\pi DN}{60} = \sqrt{\frac{\sigma_{cf}}{\rho}}$

$\therefore D_{max} = \frac{60}{\pi \times 1000}.\sqrt{\frac{50 \times 10^6}{7200}} = 1.5915$ m

41. Ans: (a)

Sol: W = 4 kN,

P = 1.4 MPa

$P = \frac{W}{\ell \times d}$

$1.4 = \frac{4 \times 10^3}{\ell \times 50}$, $l = 57.14$ m

$\frac{\ell}{d} = \frac{57.14}{50} = 1.14 > 1$

$\frac{\ell}{d} > 1 \Rightarrow$ long bearing

42. Ans: (c)

Sol: $\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{125}{5}} = 5$ rad/s

x = 5 cm, V = 25 cm/s

$V = \omega_n\sqrt{A^2 - x^2}$

$25 = 5 \times \sqrt{A^2 - 5^2}$

A = 7.07 cm

43. Ans: (d)

Sol: At any instant ΔT = T of engine – T of machine = 500 + 50sinθ – 500 + 50sinθ = 100sinθ

ΔT = 0⇒ 100 sinθ = 0

⇒ θ = 0, π, 2π

$\Delta E = e_{max} = \int_0^{\pi} 100\sin\theta d\theta = 200$ N-m

44. Ans: (d)

Sol: Followers with cycloidal motion

$$y = \frac{L}{2\pi}\left[\frac{2\pi\theta}{\theta_R} - \sin\left(\frac{2\pi\theta}{\theta_R}\right)\right]$$

$$V = \frac{L}{2\pi}\left[\frac{2\pi\omega}{\theta_R} - \frac{2\pi\omega}{\theta_R}\cos\left(\frac{2\pi\theta}{\theta_R}\right)\right]$$

$$a = \frac{1}{2\pi} \times \frac{4\pi^2\omega^2}{\theta_R^2}\sin\left(\frac{2\pi\theta}{\theta_R}\right)$$

$$a = \frac{2\pi.L\omega^2}{\theta_R^2}\sin\left(\frac{2\pi\theta}{\theta_R}\right)$$

∴ A pure 'sine' curve form 0 to θ_R .

45. Ans: (a)

Sol: Primary reverse crank is mirror image of primary direct crank about the line of stroke.

46. Ans: (b)

Sol: $h = \frac{g}{m\omega^2}\left[m + \frac{(M)(1+k)}{2}\right]$

k = 1 (arms are equal length)

$$\frac{0.50}{\sqrt{2}} = \frac{g}{\omega^2}(21)$$

⇒ ω = 24 rad/s

47. Ans: (c)

Sol: Theoretical stress concentration factor,

$k_t = 2.1$

Notch sensitivity (q) = 0.5

$$q = \frac{k_f - 1}{k_t - 1}$$

k_f = fatigue stress concentration factor

⇒ $k_f = 1 + q(k_t - 1)$

= 1 + 0.5 (2.1 – 1) = 1 + 0.55 = 1.55

$$\sigma_e^1 = \frac{\sigma_e}{k_f}$$

$$\Rightarrow \sigma_e^1 = \frac{\sigma_e}{1.55} = 0.6451 \times \sigma_e$$

Endurance strength is reduced by 35.5 %

48. Ans: (a)

Sol: Factor of safety does not depend upon the load and dimension of the member.

49. Ans: (a)

Sol: Power, $P = F_t V \Rightarrow F_t = \frac{P}{V}$

$$V = \frac{\pi d_A N}{60} = \frac{\pi T_A m_A N}{60} \qquad [\because d = mT]$$

$$= \frac{\pi \times 40 \times 5 \times 1440 \times 10^{-3}}{60}$$

= 15.07 m/s

$$F_t = \frac{5.6 \times 10^3}{15.07} = 371.59\,N$$

50. Ans: (b)

Sol: $Y = \frac{t^2}{6hm}$

Here, t = thickness of the tooth

h = height of the tooth

51. Ans: (c)

Sol: In actual operation the gear are subjected to dynamic loading. So, beam strength should be greater than dynamic load and wear strength should be greater than beam strength for safer design.

52. Ans: (c)

Sol: The equation of motion about the centre of the rod by taking moment about hinge

$$\frac{ML^2\ddot{\theta}}{12} + 2\times k\times\left(\frac{L}{2}\right)^2\theta = 0$$

The frequency of oscillation,

$$f = \frac{1}{2\pi}\sqrt{\frac{\left(\frac{kL^2}{2}\right)}{\frac{ML^2}{12}}} = \frac{1}{2\pi}\sqrt{\frac{6k}{M}}$$

53. Ans: (b)

Sol: Gyroscopic couple = $I\omega\omega_p$

$$= 300\times1^2\times\frac{2\pi\times100}{60}\times6 = 6\pi \text{ kNm}$$

54. Ans: (c)

Sol: $T_1\omega_1 + T_2\omega_2 + T_3\omega_3 = 0$

$T_1 = 100$ Nm

$\omega_1 = 1000$ rpm (cw)

$\omega_2 = 50$ rpm (ccw)

$\omega_3 = 0$

$100 \times 1000 - T_2 \times 50 = 0$

$T_2 = 2000$ Nm

$T_1 + T_2 + T_3 = 0$

$T_3 = -2100$ Nm

$T_3 = 2100$ Nm is applied opposite to input torque.

55. Ans: (b)

Sol: $T_S = 48,\ T_p = 24$

$r_A = r_S + 2r_p$

$$\frac{mT_A}{2} = \frac{mT_S}{2} + \frac{2mT_p}{2}$$

$T_A = T_S + 2T_P = 48 + 2 \times 24 = 96$

$$\frac{N_s - N_a}{N_A - N_a} = -\frac{T_p}{T_s} \times \frac{T_A}{T_p}$$

$N_s = 0$

$$\frac{N_a}{N_A - N_a} = \frac{T_A}{T_s}$$

$$\frac{N_A - N_a}{N_a} = \frac{T_s}{T_A} = \frac{48}{96} = \frac{1}{2}$$

$$\frac{N_A}{N_a} - 1 = \frac{1}{2}$$

$$\frac{N_A}{N_a} = 1.5$$

56. Ans: (c)

Sol: The controlling force equation is given as

F = ar + b

1600 = a × 400 + b .... (1)

800 = a × 240 + b .... (2)

By solving these equations

a = 5, b = – 400

To make the governor isochronous, the controlling force line must pass through the origin. So, the initial tension is = 400 N

57. Ans: (b)

Sol: $$L_{90} = \frac{60NL_{90h}}{10^6} = \frac{60 \times 1400 \times 9000}{10^6}$$

= 756 million revolution

58. Ans: (a)

Sol: In uniform wear theory

Pr = constant and $r_2 > r_1$

$\Rightarrow P_1 > P_2$

59. Ans: (c)

Sol: **I:** $T \propto \frac{1}{\alpha}$

So, $\alpha > \phi$ (angle of static friction to avoid self engagement.)

II: $$\frac{T_{cone}}{T_{plate}} = \frac{1}{\sin\alpha} > 1$$

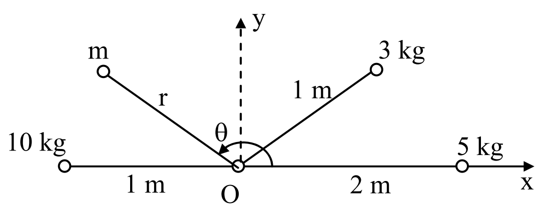
$\Rightarrow\ T_{cone} > T_{plate}$

60. Ans: (a)

Sol: When moment of force and direction of rotation of drum are different, the end of rope which is away from fulcrum is tight side and when the direction of rotation of drum and moment of force are same the end which is closer to the fulcrum is tight side.

61. Ans: (a)

Sol: For complete balancing



$5\times2 + 3\times1\cos45 + 10\times1\cos180° + mr\cos\theta = 0$

$mr\cos\theta = 10 - 10 - \frac{3}{\sqrt{2}} = -\frac{3}{\sqrt{2}}$ ----------(1)

$\Sigma F_y = 0$

$0 + 3 \times 1\sin45 + 10 \times 1\sin180 + mr\sin\theta = 0$

$mr\sin\theta = -\frac{3}{\sqrt{2}}$ --------------(2)

$\Rightarrow \tan\theta = \frac{-1}{-1}$

($\because$ cosθ and sinθ are –ve in 3rd quadrant)

θ = 180 + 45 = 225°

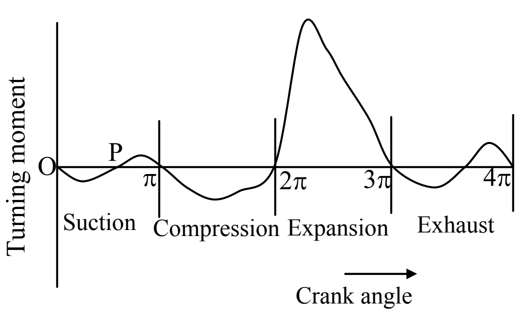
mr = 3

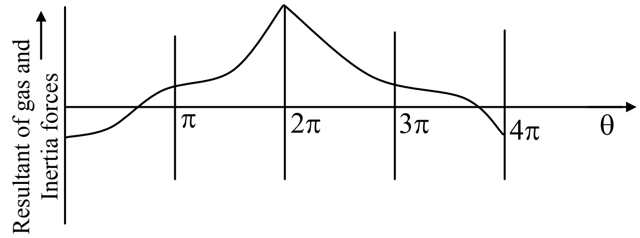
From given options we can have

m = 1.5 kg, r = 2 m, θ = 225°

62. Ans (b)

Sol:





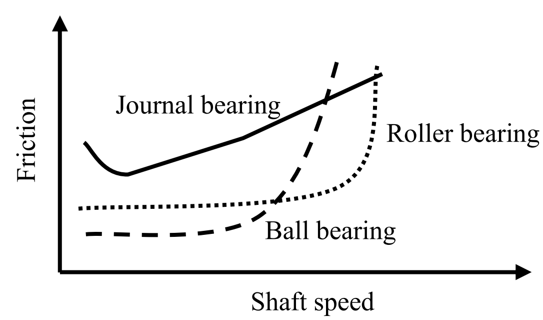
Suction (0 to π), compression (π to 2π), expansion (2π to 3π) and exhaust (3π to 4π).

63. Ans: (c)

Sol: Cam and follower is a higher pair because there is a point or line of contact between them. Surface contact takes place between two links of a lower pair.

64. Ans: (b)

Sol:

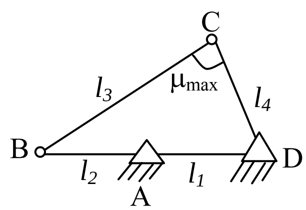


65. Ans: (c)

Sol: Torque carrying capacities of friction clutches are based on uniform wear rate condition and soft materials are used for friction linings.

66. Ans (c)

Sol:



Maximum transmission angle occurs when input crank is parallel (180°) with the fixed link.

67. Ans: (a)

Sol: Gyroscopic couple = $I(\vec{\omega} \times \vec{\omega}_p)$

There is no gyroscopic effect when angle between ω and $\omega_p = 0$.

68. Ans: (d)

Sol: Wear strength is the maximum tangential load the gear can transmit with pitting and beam strength is maximum tangential load the tooth can transmit without bending failure. Load transmitted by gear should always be less than beam and wear strength to avoid any failure.

69. Ans: (c)

Sol: Welding process is used to fabricate all kind of component not only steel, so Statement (II) is wrong.

70. Ans: (a)

Sol: In centrifugal governors balls are operated by actual change in speed but in inertial governor balls are operated by rate of change of speed.

Therefore, the response of the inertia governor is faster than centrifugal governors..

71. Ans: (a)

Sol: Since involute curve does not exist within base circle, interference is always possible if base circle radius is larger than dedendum circle radius.

72. Ans: (d)

Sol: The Lewis equation predicts the static load capacity not dynamic.

73. Ans: (a)

Sol: For a given size of a rolling contact bearing the load carrying capacity of a roller bearing is better than the ball bearing because the area of contact between the bearing surface and the race surface is greater in rolling contact bearing.

74. Ans (b)

Sol:

- To find the Coriolis acceleration, rotate the sliding velocity vector by 90° in the direction of angular velocity (ω) of the link.
- Coriolis acceleration = 2 Vω is perpendicular to the sliding velocity (V) vector.

75. Ans: (d)

Sol:

(i) For high speed cams, cycloidal motion of the follower is preferred because when follower is lifted from dwell, it is subjected to zero acceleration and less amount of jerk.

(ii) The point of pitch circle at which pressure angle is maximum is known as pitch point.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