DETAILED SOLUTIONS

# MADE EASY

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**Test Centres:** Delhi, Hyderabad, Bhopal, Jaipur, Bhubaneswar, Pune, Kolkata

## ESE 2024 : Prelims Exam
## CLASSROOM TEST SERIES

## MECHANICAL ENGINEERING

## Test 10

**Section A :** Strength of Materials & Engineering Mechanics
**Section B :** Heat Transfer-1 + IC Engines-1
**Section C :** Fluid Mechanics and Turbo Machinery-2

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| **1.** | (c) | **16.** | (d) | **31.** | (a) | **46.** | (a) | **61.** | (a) |
| **2.** | (b) | **17.** | (d) | **32.** | (d) | **47.** | (d) | **62.** | (b) |
| **3.** | (a) | **18.** | (c) | **33.** | (a) | **48.** | (a) | **63.** | (b) |
| **4.** | (d) | **19.** | (b) | **34.** | (a) | **49.** | (c) | **64.** | (c) |
| **5.** | (d) | **20.** | (d) | **35.** | (c) | **50.** | (a) | **65.** | (b) |
| **6.** | (d) | **21.** | (c) | **36.** | (a) | **51.** | (b) | **66.** | (c) |
| **7.** | (c) | **22.** | (c) | **37.** | (a) | **52.** | (d) | **67.** | (c) |
| **8.** | (d) | **23.** | (b) | **38.** | (a) | **53.** | (d) | **68.** | (c) |
| **9.** | (c) | **24.** | (b) | **39.** | (c) | **54.** | (c) | **69.** | (b) |
| **10.** | (c) | **25.** | (b) | **40.** | (c) | **55.** | (c) | **70.** | (b) |
| **11.** | (b) | **26.** | (c) | **41.** | (b) | **56.** | (d) | **71.** | (d) |
| **12.** | (c) | **27.** | (d) | **42.** | (c) | **57.** | (d) | **72.** | (b) |
| **13.** | (d) | **28.** | (c) | **43.** | (c) | **58.** | (b) | **73.** | (d) |
| **14.** | (b) | **29.** | (d) | **44.** | (c) | **59.** | (d) | **74.** | (b) |
| **15.** | (c) | **30.** | (c) | **45.** | (a) | **60.** | (c) | **75.** | (c) |

## DETAILED EXPLANATIONS

**1. (c)**

$\sigma_x$ = 180 MPa; $\sigma_y$ = –140 MPa; $\sigma_1$ = 300 MPa

$$\text{Largest principal stress, } \sigma_1 = \frac{1}{2}(\sigma_x + \sigma_y) + \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}$$

$$300 = \frac{180 - 140}{2} + \frac{1}{2}\sqrt{(180+140)^2 + 4\times\tau_{xy}^2}$$

$$300 = 20 + \frac{1}{2}\sqrt{(320)^2 + 4\times\tau_{xy}^2}$$

$$280 \times 2 = \sqrt{(320)^2 + 4\tau_{xy}^2}$$

$$560 = \sqrt{(320)^2 + 4\tau_{xy}^2}$$

$$560^2 = (320)^2 + 4\tau_{xy}^2$$

$$\therefore \quad \tau_{xy} = 229.78 \simeq 230 \text{ MPa}$$

**2. (b)**

$b$ = 3 mm, $t$ = 0.5 mm, $R$ = 0.5 m, $E$ = 220 GPa

$$\sigma = \frac{Ey}{R} = \frac{220\times10^9 \times 0.25\times10^{-3}}{0.5}$$

$$\sigma = 110 \text{ MPa}$$

$\therefore$ Strain energy stored per metre length.

$$U = \frac{M^2L}{2EI} = \frac{\left(\frac{\sigma I}{y}\right)^2 \times L}{2EI} = \frac{\sigma^2 \cdot I \cdot L}{2E\,y^2} \qquad \left[\because \frac{M}{I} = \frac{\sigma}{y}\right]$$

$$= \frac{110^2 \times \left(\frac{1}{32}\right)\times 1000}{2\times220\times10^3\times(0.25)^2} \qquad \left[\because I = \frac{bt^3}{12}\right]$$

$$= 13.75 \text{ N-mm}$$

**4. (d)**

$$\text{Shear force, } F = \frac{wl}{2} = \frac{w\times2}{2}$$

$$F = w \text{ N}$$

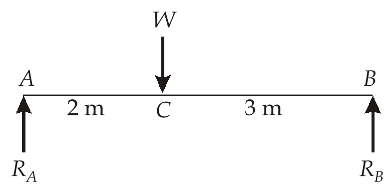
Maximum shear stress in rectangular beam,

$$\tau_{\max} = \frac{3}{2}\tau_{avg} = \frac{3}{2}\times\frac{F}{A}$$

$$5 = \frac{3}{2} \times \frac{w}{bh} = \frac{3}{2} \times \frac{w}{100 \times 200}$$

$$\therefore \quad w = \frac{5 \times 200^2}{3} = 66.67 \text{ kN/m}$$

**5. (d)**




$l$ = 5 m; $W$ = 30 kN; $a$ = 2 m; $b$ = 3 m; $I = 60 \times 10^{-6}$ m$^4$; $E$ = 200 GPa

Deflection at C is given by,

$$y_c = \frac{Wa^2b^2}{3EIl} = \frac{30 \times 10^3 \times 2^2 \times 3^2}{3 \times 200 \times 10^9 \times 60 \times 10^{-6} \times 5}$$

$$\therefore \quad y_c = 6 \times 10^{-3} \text{ m} = 6 \text{ mm}$$

**7. (c)**

$D$ = 500 mm; $P$ = 10 MPa; $\sigma_y$ = 250 MPa; FOS = 4

$$\text{Hoop stress, } \sigma_h = \frac{Pd}{2t} = \frac{10 \times 500}{2 \times t} = \frac{2500}{t} \text{MPa}$$

$$\text{Longitudinal stress, } \sigma_L = \frac{Pd}{4t} = \frac{10 \times 500}{4 \times t} = \frac{1250}{t} \text{MPa}$$

According to MPST, we have,

$$\sigma_h = \sigma_{max} = \frac{\sigma_y}{FOS}$$

$$\therefore \quad \frac{2500}{t} = \frac{250}{4}$$

$$\therefore \quad t = \frac{2500 \times 4}{250} = 40 \text{ mm}$$

**8. (d)**

In thick cylinders hoop stress for any radius $R$

$$\sigma_h = \frac{B}{R^2} + A$$

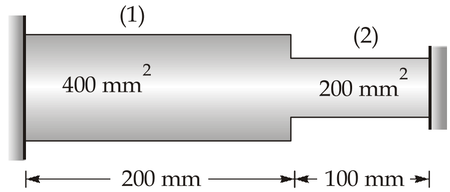
So, the variation of the hoop stress across the thickness of cylinder will be hyperbolic.

**9. (c)**

For bending without twisting, plane of loading must contain one of the principal central axis of the section. In case the section is having a plane of symmetry, the symmetrical plane contain the principal central axis. Thus statement I is correct. But the bending axis will be perpendicular to the plane of loading, hence statement II is wrong.

**10. (c)**

Refer figure,




Total expansion of bar due to temperature = Total contraction of bar due to force $P$

$$\therefore \quad (L_1 + L_2)\,\alpha\Delta t = \frac{PL_1}{A_1E} + \frac{PL_2}{A_2E}$$

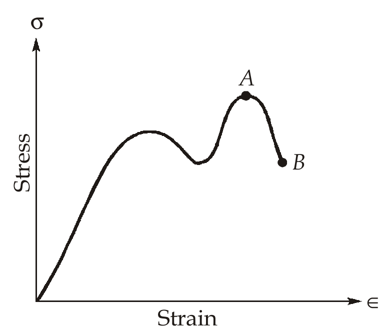
$$300 \times 12.5 \times 10^{-6} \times 50 = \frac{P}{200 \times 10^3}\left[\frac{200}{400} + \frac{100}{200}\right]$$

$$37500 = P\left[\frac{200}{400} + \frac{100}{200}\right]$$

$$\therefore \quad P = 37500 \text{ N}$$

$$\therefore \quad \sigma_{max} = \frac{P}{A_{min}} = \frac{37500}{200} = 187.5 \text{ MPa}$$

**11. (b)**




$A \rightarrow$ Ultimate strength

$B \rightarrow$ Rupture strength

**12. (c)**

$$\text{Power, } P = T_{mean} \cdot \omega$$

$$\therefore \quad T_{mean} = \frac{P \times 60}{2\pi N}$$

$$T_{mean} = \frac{150 \times 10^3 \times 60}{2\pi \times 300} = \frac{15000}{\pi} \text{Nm}$$

$$\text{Maximum torque, } T_{max} = 1.2T_{mean} = \frac{1.2 \times 15000}{\pi} \text{Nm}$$

Now,
$$\tau = \frac{16T_{max}}{\pi D^3} = \frac{16 \times 1.2 \times 15000}{\pi^2 \times 0.1^3}$$

$$\tau = 29.18 \text{ MPa}$$

**13. (d)**

Let '$P$' be the radial pressure at the common surface.

Now, increase in inner diameter of the collar,

$$\delta d = \frac{\sigma_c + \mu\sigma_r}{E} \times d$$

or $$0.2 = \frac{1}{E}\left[\frac{250^2 + 200^2}{250^2 - 200^2} \cdot P + \mu P\right] \times d$$

or $$0.2 = \frac{Pd}{E}[4.55 + 0.3]$$

$\therefore$ $$P = \frac{0.2 \times 200 \times 10^3}{4.85 \times 200} = 41.2 \text{ MPa}$$

**14. (b)**

Reduction in diameter of shaft, $$\frac{\delta d}{d} = \frac{\sigma_c + \mu\sigma_r}{E} = \frac{(-P + \mu P)}{E}$$

$$\delta d = \frac{P \cdot d}{E}(-1 + 0.3) = \frac{41.23 \times 200}{200 \times 10^3} \times (-0.7)$$

$\therefore$ $$\delta d = -0.028 \text{ mm}$$

or reduction in diameter $= 0.028$ mm

**15. (c)**

Hoop stress, $$\sigma_c = \frac{250^2 + 200^2}{250^2 - 200^2} \times 41.2$$

$$\sigma_c = 187.8 \text{ MPa}$$

**17. (d)**

Rate of change of shear force (or slope of the shear force curve) is equal to intensity of loading.

**18. (c)**

$d = 50$ mm; $L = 1$ m; $P = 200$ kN; $\Delta = 0.25$ mm; $T = 3$ kN-m; $\theta = 1.8°$

Now, deformation, $$\Delta = \frac{PL}{AE}$$

$\Rightarrow$ $$E = \frac{PL}{A\Delta}$$

$\therefore$ $$E = \frac{200 \times 10^3 \times 1000}{\frac{\pi}{4} \times 50^2 \times 0.25}$$

$$E = 4.07 \times 10^5 \text{ N/mm}^2$$

**19. (b)**

Shear modulus, $$G = \frac{TL}{J\theta} = \frac{32TL}{\pi d^4 \times \theta}$$

$$\therefore \qquad G = \frac{32 \times 3 \times 10^6 \times 1000}{\pi \times 50^4 \times \frac{\pi}{180} \times 1.8} = 1.55 \times 10^5 \text{ N/mm}^2$$

**20. (d)**

Poisson's ratio is obtained from the relation,

$$E = 2G(1 + \mu)$$

$$\therefore \qquad \mu = \frac{E}{2G} - 1$$

$$= \frac{4.07 \times 10^5}{1.55 \times 10^5 \times 2} - 1 = 0.31$$

**21. (c)**

The deformation of a tapering conical bar is given by,

$$\Delta = \frac{4PL}{\pi E d_1 d_2} = \frac{4 \times 100 \times 10^3 \times 500}{\pi \times 2 \times 10^5 \times 20 \times 40}$$

$$= 0.397 \simeq 0.4 \text{ mm}$$

**22. (c)**

MSST or Guest's theory is not applicable to materials subjected to hydrostatic pressure, since it will predict the shear stress to be almost zero, meaning thereby that the material will never fail, however larger the applied stresses be. This is physically impossible.

**23. (b)**

According to maximum shear strain energy theory, we have

$$\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 = \sigma_y^2$$

$$\left(\frac{16T}{\pi d^3}\right)^2 + \left(\frac{16T}{\pi d^3}\right)^2 + \left(\frac{16T}{\pi d^3}\right)^2 = \left(\frac{32M}{\pi d^3}\right)^2$$

or

$$3\left(\frac{16T}{\pi d^3}\right)^2 = \left(\frac{32M}{\pi d^3}\right)^2$$

$$\therefore \qquad \frac{M}{T} = \frac{\sqrt{3}}{2} = 0.867$$

**25. (b)**

Strain energy in bending is,

$$U = \frac{1}{2EI}\int_0^L {M_x}^2 dx$$

$$U = \frac{1}{2EI}\int_0^L M^2 dx \qquad \because M_x = M$$

$$\therefore \qquad U = \frac{M^2 L}{2EI}$$

**26. (c)**

$$\text{Kinetic energy of load} = \text{Energy absorbed by rope}$$

$$\therefore \quad \frac{1}{2}MV^2 = \frac{\sigma^2}{2E} \cdot A \cdot L$$

or
$$\sigma^2 = \frac{MV^2 \cdot E}{A \cdot L}$$

$$\sigma^2 = \frac{1000 \times 1^2 \times 200 \times 10^9}{\frac{\pi}{4} \times 0.05^2 \times 10}$$

$$\sigma = 100.92 \text{ MPa} \simeq 101 \text{ MPa}$$

**27. (d)**

Maximum instantaneous elongation is,

$$\Delta = \frac{\sigma \cdot L}{E} = \frac{101 \times 10^4}{200 \times 10^3} = 5.05 \text{ mm}$$

**28. (c)**

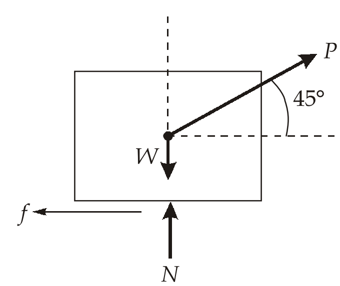
At the verge of motion, $f_{max,s} = P\cos(45°)$

$$\Rightarrow \quad \mu_s N = P\cos(45°)$$

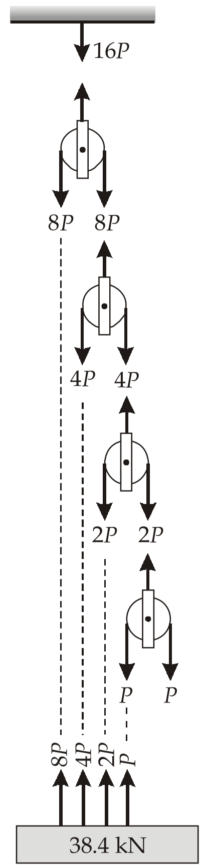
$$\Rightarrow \quad \mu_s [W - P\sin(45°)] = P\cos(45°)$$

$$\Rightarrow \quad P = \frac{\mu_s W}{\cos(45°) + \mu_s \sin(45°)}$$

$$= \frac{0.4 \times 100 \times 9.81}{\frac{1}{\sqrt{2}} + 0.4 \times \frac{1}{\sqrt{2}}} = 396.38 \text{ N}$$



**29. (d)**




At static equilibrium,

$$P + 2P + 4P + 8P = W$$

$$\Rightarrow \quad P + 2P + 4P + 8P = 38.4 \text{ kN}$$

$$\Rightarrow \quad 15P = 38.4 \text{ kN}$$

$$\Rightarrow \quad P = 2.56 \text{ kN} = 2560 \text{ N}$$

**30. (c)**

Initial velocity, $u = 10$ m/s

Acceleration, $a = 2$ m/s$^2$

Final velocity, $v = 20$ m/s

Under constant acceleration, $v^2 = u^2 + 2ax$

$$20^2 = 10^2 + 2 \times 2 \times x$$

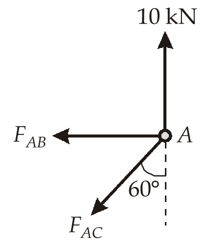
$$\Rightarrow \quad x = \frac{20^2 - 10^2}{2 \times 2} = 75 \text{ m}$$

**31. (a)**

Lami's theorem states that, "If three coplanar forces acting at a point be in equilibrium, then each force is proportional to the sine of the angle between the other two."

**32. (d)**

For static equilibrium at point *A*;




$\Sigma F_V = 0;$ $$F_{AC} \cdot \cos 60° = 10 \text{ kN}$$

$$\Rightarrow \quad F_{AC} = \frac{10}{\cos 60°} = 20 \text{ kN}$$

$\Sigma F_H = 0;$ $$F_{AB} + F_{AC} \cdot \sin 60° = 0$$

$$\Rightarrow \quad F_{AB} = -20 \times \sin 60°$$

$$= -10\sqrt{3} \text{ kN}$$

$$= 10\sqrt{3} \text{ kN (Compressive)}$$

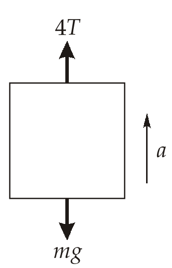
Since member *AB* and *BC* are perpendicular to each other, the horizontal reaction at support *B* will be equal to the force in the member *AB*.

$\Rightarrow$ Horizontal reaction at point $B = F_{AB} = 10\sqrt{3}$ kN

**33. (a)**

Free body diagram of the elevator:

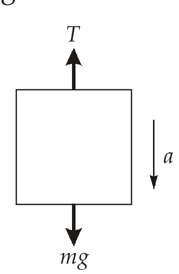
(i) While moving upwards:




Using Newton's second law of motion:

$4T - mg = ma$ ...........(i)

(ii) While moving downwards:




Using Newton's second law of motion:

$mg - T = ma$ ...........(ii)

Using equation (i) and (ii),

$$(4T - mg) - (mg - T) = ma - ma$$

$$\Rightarrow \quad 5T = 2mg$$

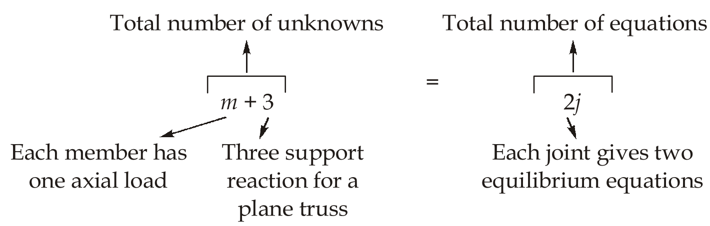
$$\Rightarrow \qquad T = \frac{2mg}{5}$$

The tension in the cable while moving upwards = $4T = \frac{8mg}{5}$

**34. (a)**

For a statically determinate plane truss, the total number of unknown should be equal to the total number of independent equilibrium equations.

Hence,




**35. (c)**

Given : $D$ = 200 mm = 0.2 m; $m$ = 20 kg; $N$ = 300 rpm

$$\text{Mass moment of inertia of a disc, } I = \frac{1}{2}mR^2 = \frac{1}{2}m\left(\frac{D}{2}\right)^2$$

$$= \frac{1}{2}\times 20\times\left(\frac{0.2}{2}\right)^2 = 0.1 \text{ kg.m}^2$$

$$\text{Kinetic energy, } KE = \frac{1}{2}Iw^2 = \frac{1}{2}\times I\times\left(\frac{2\pi N}{60}\right)^2$$

$$= \frac{1}{2}\times 0.1\times\left(\frac{2\pi\times 300}{60}\right)^2$$

$$= 49.348 \text{ J} \simeq 49.35 \text{ J}$$

**36. (a)**

Given : $m$ = 2 kg; $v_1$ = 20 m/s; $v_2$ = 10 m/s

Impulse is the rate of change in linear momentum with respect to time.

$$J = mv_1 - mv_2$$
$$= 2 \times 20 - 2 \times (-10)$$
$$= 2 \times 20 + 2 \times 10$$
$$= 60 \text{ N.s}$$

**37. (a)**

Given : $\eta$ = 50% = 0.5; $P$ = 200 N; $W$ = 1000 N

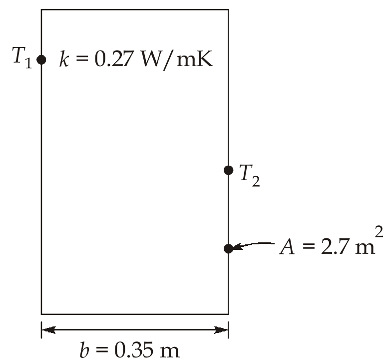
Mechanical advantage is given as:

$$\frac{W}{P} = \eta \times VR$$

$$\Rightarrow \quad VR = \frac{W}{\eta P}$$

$$= \frac{1000}{0.5 \times 200} = 10$$

**38. (a)**




The thermal conductance per unit area is given by,

$$\frac{k}{b} = \frac{0.27}{0.35} = 0.77 \text{ W/m}^2\text{K}$$

**39. (c)**

Thermal contact resistance develops when two conducting surface do not fit tightly and a thin layer of fluid is trapped between them. This resistance is primarily a function of surface roughness, the pressure holding the two surfaces in contact, the interface fluid and the interface temperature.

**40. (c)**

$D_1$ = 400 cm, $D_2 = D_1 - 2t$ = 400 – 20 = 380 cm; $k$ = 0.05 W/mK; $T_1$ = 25°C; $T_2$ = –35°C and $L$ = 20 m

The rate of heat loss from the cylindrical compartment is given by,

$$Q = \frac{2\pi k L (T_1 - T_2)}{\ln\left(\frac{r_2}{r_1}\right)} = \frac{2 \times \pi \times 0.05 \times 20 \times \{25 - (-35)\}}{\ln\left(\frac{400}{380}\right)}$$

$$= 7348.75 \text{ W}$$

**41. (b)**

Increasing the ratio of the perimeter to the cross-sectional area of the fin, P/A. The use of thin, but closely spaced, fins is preferred to thick fins.

**42. (c)**

For Biot number << 0.1, the error associated with the lumped capacitance method is small. This idealized assumption is possible if

(i) the physical size of the body is very small,

(ii) thermal conductivity of the material is very large, and

(iii) heat transfer coefficient of the surrounding fluid is very small.

**43. (c)**

For free convection heat transfer,

$$\text{Nusselt number, Nu} = \frac{h \cdot L}{k} = c(Gr \cdot \text{Pr})^m$$

where $c$ and $m$ are constant and varies from case to case.

$$\text{For laminar flow; } m = \frac{1}{4}$$

$$\text{For turbulent flow; } m = \frac{1}{3}$$

**44. (c)**

$\delta_H$ = 1.2 mm; $\mu = 20 \times 10^{-6}$ Pa-s; $c_p$ = 3 kJ/kgK and $k$ = 0.05 W/mK

$$\text{Pr} = \frac{\mu c_p}{k} = \frac{20 \times 10^{-6} \times 3 \times 10^3}{0.05} = 1.2$$

$$\frac{\delta_H}{\delta_t} = 1.026 \times \text{Pr}^{1/3}$$

$$\Rightarrow \quad \delta_t = \frac{\delta_H}{1.026 \times \text{Pr}^{1/3}}$$

$$= \frac{1.2}{1.026 \times (1.2)^{1/3}} = 1.1 \text{ mm}$$

Note : It can be easily observed for the expression, the Pr > 1, so $\delta_t < \delta_H$ (1.2 mm). So, option (c) will be correct.

**45. (a)**

$$\text{Reynold's number, Re} = \frac{\rho \times U_\infty \times D}{\mu} = \frac{1.5 \times 10 \times 0.02}{2.6 \times 10^{-5}}$$

$$= 11538.46 > 2300$$

So, the flow through the tube is turbulent flow.

$$\therefore \quad \text{Nu} = \frac{\bar{h} \cdot D}{k}$$

$$\Rightarrow \quad 35.44 = \frac{\bar{h} \times 0.02}{0.04}$$

$$\Rightarrow \quad \bar{h} = 70.88 \text{ W/m}^2\text{K}$$

**46. (a)**

The heat transfer per unit length of tube is given by,

$$\frac{Q}{L} = h \times (\pi D) \times (T_w - T_b)$$

$$\Rightarrow \quad \frac{Q}{L} = 70.88 \times \pi \times 0.02 \times 20$$

$$= 89.07 \text{ W/m}$$

**49. (c)**

Let the stroke volume be $V_s$ and the clearance volume be $V_c$.

At 25% stroke,

$$V_a = V_c + 0.75V_s \quad \text{...(i)}$$

At 80% of the stroke,

$$V_b = V_c + 0.2V_s \quad \text{...(ii)}$$

From (i) and (ii),

$$\frac{V_a}{V_b} = \frac{V_c + 0.75V_s}{V_c + 0.2V_s}$$

or $$\frac{1.25}{0.5} = \frac{1 + 0.75(r-1)}{1 + 0.2(r-1)}$$

∴ $$2.5 = \frac{1 + 0.75(r-1)}{1 + 0.2(r-1)}$$

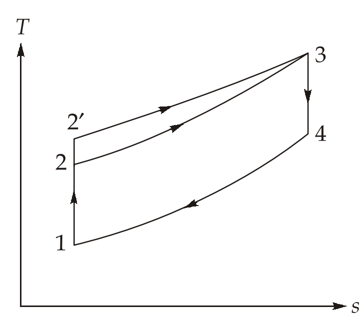
$$2.5 + 0.5(r-1) = 1 + 0.75(r-1)$$

or $$1.5 = 0.25(r-1)$$

∴ $$r = 1 + 4 \times 1.5 = 7$$

**50. (a)**

The figure below on T-s plane shows that both the cycles reject the same amount of heat energy (area under the curve 1-4) whereas the work output given by 1-2′-3-4-1 (diesel cycle is more than the work output given by 1-2-3-4-1 (Otto cycle)). Therefore the air standard efficiency is greater for diesel cycle.




**52. (d)**

| Increase in variable | Effect on delay period | Reasons |
|---|---|---|
| Fuel cetane number | Reduces | The self-ignition temperature of the fuel is reduced |
| Injection pressure | Reduces | Physical delay is reduced because of greater surface-volume ratio of fuel droplets |
| Injection advance angle | Increases | The pressure and temperature at the start of fuel injection is lowers |

**53. (d)**

In open combustion chambers, the combustion space is essentially a single cavity formed between the piston and cylinder head, the combustion chamber tends to be compact with minimum wall area per unit volume surrounding the compressed air. Statement I is true for divided combustion chambers.

**54. (c)**

In an electronic fuel injection system, there is complete freedom from icing problems because the fuel is not vapourised before the throttle. The cold starting is easy and therefore a less volatile fuel can be used and the fuel deposition at cold places can be avoided.

**55. (c)**

$$\frac{F}{A} = 0.05,\ \dot{m}_f = 6 \text{ kg/hr}$$

$\therefore$ Mass of air supplied to the engine $= \dfrac{6}{0.05 \times 60} = 2$ kg/min

**56. (d)**

Given : $D = 0.1$ m, $L = 0.12$ m

$$\text{Stroke volume} = \frac{\pi}{4}D^2L = \frac{\pi}{4} \times 0.1^2 \times 0.12$$

$$= 9.42 \times 10^{-4} \text{ m}^3$$

$$\text{Cylinder volume} = \text{Stroke volume} \times \frac{r}{r-1}$$

$$= 9.42 \times 10^{-4} \times \frac{12}{11}$$

$$= 1.027 \times 10^{-3} \text{ m}^3\text{/cycle}$$

$$\therefore \quad \text{Scavenging efficiency, } \eta_{sc} = \frac{\text{Mass of delivered air retained}}{\text{Reference mass}}$$

$$= \frac{2}{2000 \times 1.027 \times 10^{-3} \times 1.2}$$

$$= 0.8114 \text{ or } 81.14\%$$

**58. (b)**

The function of diffuser is to transform the high kinetic energy of the fluid at the impeller outlet into static pressure.

**59. (d)**

Reaction turbines are suitable for medium and higher power requirements, whereas impulse turbines are suitable for small power requirements.

**60. (c)**

Power developed is given by,

$$P = \dot{m}\Delta V_w \cdot u$$

$$\therefore \quad 1000 \times 10^3 = \dot{m} \times 250 \times 150$$

$$\therefore \quad \dot{m} = \frac{10^6}{250 \times 150} = 26.6 \text{ kg/s}$$

**62. (b)**

Relative velocity at inlet = 300 m/s

Relative velocity at outlet = 0.84 × 300 = 252 m/s

$$\text{Hence, blade friction loss} = \frac{V_{r_1}^2 - V_{r_2}^2}{2 \times 1000} = \frac{300^2 - 252^2}{2 \times 1000}$$

$$= 13.25 \text{ kJ/kg}$$

**63. (b)**

Part load efficiency of a closed cycle is improved as compared to the open cycle because the method of control of the output in the closed cycle does not vary the temperature and pressure ratio of the system, and also, the efficiency during the load change.

**64. (c)**

Given : $\dot{m}$ = 18 kg/s, $N$ = 12000 rpm, $\phi$ = 1.05, $\sigma$ = 0.9, $D_2$ = 0.5 m

Now,
$$U_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.5 \times 12000}{60} = 100\pi \text{ m/s}$$

$$\therefore \quad \text{Work done} = \dot{m} \times \sigma \times \phi \times U_2^2$$

$$= 18 \times 0.9 \times 1.05 \times (100\pi)^2$$

$$= 1678.8 \text{ kW}$$

**65. (b)**

Given : $\alpha_1 = 30° = \beta_2$, $\alpha_2 = 45° = \beta_1$, $V_F$ = 200 m/s, $N$ = 6000 rpm

For axial flow compressor, we have

$$u = V_f(\tan\alpha_1 + \tan\beta_1)$$

or
$$u = V_f(\tan\beta_2 + \tan\beta_1)$$

$$\therefore \quad u = 200 (\tan 45° + \tan 45°)$$

$$u = 200\left(1 + \frac{1}{\sqrt{3}}\right)$$

$$= 315.47 \text{ m/s}$$

or
$$\frac{\pi D_m N}{60} = 315.47 \text{ m/s}$$

$$\therefore \quad D_m = \frac{315.47 \times 60}{\pi \times 6000} = 1 \text{ m}$$

**66. (c)**

The minimum work required in a two stage compressor is given by

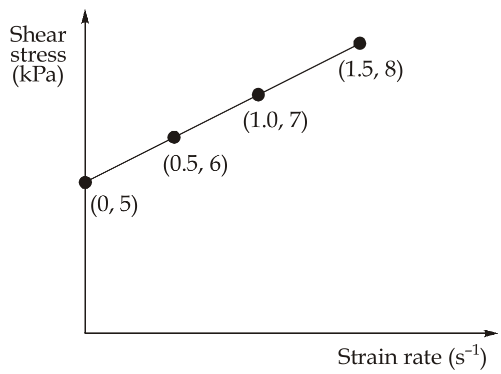
$$W_{min} = \frac{2nRT_1}{n-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

Now, $P_2 = \sqrt{P_1 P_3} = \sqrt{16 \times 1} = 4$ bar [$\because$ Perfect intercooling]

$\therefore$ $$W_{min} = \frac{2 \times 1.3 \times 0.287 \times 300}{0.3}\left[4^{0.3/1.3} - 1\right]$$

$$= 281.32 \text{ kJ/kg}$$

**67. (c)**




The fluid has an initial yield stress of 5 kPa and after which it acts as a Newtonian fluid. Hence, it is classified as Bingham plastic fluid.

**68. (c)**

- Ideal fluids are incompressible, inviscid and has zero surface tension.
- Under static condition, ideal fluids cannot withstand any shear stress. On application of shear stress, the fluids deforms continuously, however small the shear stress may be.

**69. (b)**

Given : $D$ = 400 mm, $N$ = 250 rpm, $L$ = 200 mm, $c$ = 2 mm, $\mu$ = 10 poise = 1 Pa.s

Circumferential velocity, $$v = \frac{\pi DN}{60} = \frac{\pi \times (0.400) \times 250}{60} = 5.236 \text{ m/s}$$

Shear force, $$F = \mu \frac{v}{c} A = \frac{\mu v \pi DL}{c}$$

$$= \frac{1 \times 5.236 \times \pi \times 0.400 \times 0.200}{0.002}$$

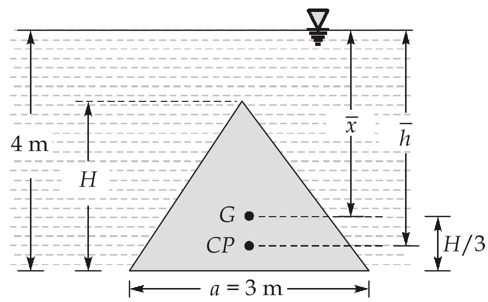
$$= 657.975 \text{ N}$$

Power loss, $$P = \frac{FD}{2} \times \frac{2\pi N}{60} = \frac{657.975 \times 0.400 \times 2\pi \times 250}{2 \times 60}$$

$$= 3445.149 \text{ W}$$

$$= 3.445149 \text{ kW} \simeq 3.45 \text{ kW}$$

**70. (b)**




Side, $a = 3$ m, Height, $H = \frac{\sqrt{3}}{2}a = \frac{\sqrt{3}}{2} \times 3 = 2.598$ m

Distance of centre of gravity from the free surface,

$$\bar{x} = 4 - \frac{H}{3} = 4 - \frac{2.598}{3} = 3.134 \text{ m}$$

**71. (d)**

$$\phi = 2xy - x$$

For a valid stream function ($\psi$);

$$\frac{\partial \phi}{\partial x} = \frac{\partial \psi}{\partial y}$$

$$\Rightarrow \quad 2y - 1 = \frac{\partial \psi}{\partial y}$$

$$\Rightarrow \quad \psi = y^2 - y + f(x) \quad \text{...(i)}$$

also,

$$\frac{\partial \phi}{\partial y} = -\frac{\partial \psi}{\partial x}$$

$$\Rightarrow \quad 2x = -\frac{\partial \psi}{\partial x}$$

$$\Rightarrow \quad \psi = -x^2 + f(y) \quad \text{...(ii)}$$

From equation (i) and (ii),

$$\psi = -x^2 + y^2 - y + \text{constant}$$

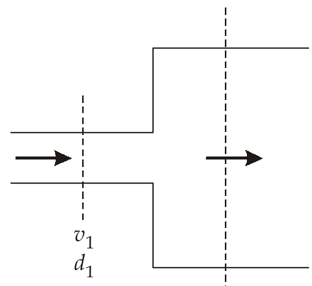
**72. (b)**

Momentum-flux correction factor is given as:

$$\beta = \frac{\text{Momentum of the flow per second based on actual velocity}}{\text{Momentum of the flow per second based on average velocity}}$$

$$\beta = \frac{1}{AV^2}\int u^2 \cdot dA$$

**73. (d)**




Given : $v_1$ = 2 m/s, $d_1$ = 10 cm, $v_2$ = ?, $d_2$ = 20 cm

$$\text{Discharge, } Q = A_1 v_1 = A_2 v_2$$

$$\Rightarrow \quad \frac{\pi}{4} d_1^2 v_1 = \frac{\pi}{4} d_2^2 v_2$$

$$\Rightarrow \quad v_2 = v_1 \times \frac{d_1^2}{d_2^2} = 2 \times \left(\frac{1}{2}\right)^2 = 0.5 \text{ m/s}$$

Head loss due to sudden expansion is given as,

$$h = \frac{(v_1 - v_2)^2}{2g} = \frac{(2 - 0.5)^2}{2 \times 9.81}$$

$$= 0.11468 \text{ m} \simeq 0.115 \text{ m}$$

**74. (b)**

Head loss in steady laminar flow through circular pipe of diameter '$D$', density '$\rho$', dynamic viscosity '$\mu$', average velocity '$v$' and length '$L$' is given as

$$h_f = \frac{32 \mu v L}{\rho g D^2}$$

Hence, at constant average velocity, $h_f \propto D^{-2}$

**75. (c)**

Momentum thickness ($\theta$) is given as:

$$\theta = \int_0^\delta \frac{u}{U}\left(1 - \frac{u}{U}\right) dy = \int_0^\delta \left(\frac{y}{\delta}\right)^{1/7} \left[1 - \left(\frac{y}{\delta}\right)^{1/7}\right] dy$$

$$= \int_0^\delta \left[\left(\frac{y}{\delta}\right)^{1/7} - \left(\frac{y}{\delta}\right)^{2/7}\right] dy = \left[\frac{7}{8} \cdot \frac{y^{8/7}}{\delta^{1/7}} - \frac{7}{9} \cdot \frac{y^{9/7}}{\delta^{2/7}}\right]_0^\delta$$

$$= \delta\left[\frac{7}{8} - \frac{7}{9}\right] = \frac{7}{72}\delta$$

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