11

SCHEME OF VALUATION

(Scoring Indicators)

Revision : 2015		Course Code: 5021		
Course Title : Design of Machine Elements				
Qst. No.	**Scoring Indicator**	**Split up score**	**Sub Total**	**Total**
I. 1.	**PART. A** It is defined as the maximum stress a material can stand without failure.	2		2
I. 2.	It is the distamce moved by the thread parallel to screw axis in one complete rotation.	2		2
I. 3.	Shear stress and Bending stress.	2		2
I. 4.	The function of governor is to maintain as closely as possible the speed of an engine within specified limits.	2		2
I. 5.	Slip in belt is defined as the relative motion between pulleys and belt.	2		2
	PART B			
II.1.	1 Are convenient to assemble and disassemble. 2 Are highly reliable joints. 3 Can be placed in any position. 4 Compact construction. 5 Adopted to various operating conditions. 6 Are economical to manufacture due to standardisation.	1	1 x 6	6
II.2.	If helix angle α is greater than friction angle ϕ, the torque required to lower the load will be negative. Which means that the load will lower itself without the application of any effort. Such a condition is known as overhauling of screws. If helix angle α is less than friction angle ϕ the torque required to lower the load will be positive. Which means that an effort is required to lower the load. Such a condition is known as self locking.	3 3		6
II.3.	$T = 20$ kN-m $= 20 \times 10^6$ N-mm $\tau = 55$ MPa $T = \frac{\pi}{16} \tau d^3$ $d = \sqrt[3]{\frac{16T}{\pi \tau}}$ $= \sqrt[3]{\frac{16 \times 20 \times 10^6}{\pi \times 55}} = \underline{\underline{122.8 \text{ mm}}}$ std $d = \underline{\underline{125 \text{ mm}}}$.	2 3 1		6

II.4	1. Design of shaft. Find the mean torque from power equation compute maximum torque as per given condition from strngth equation find diameter of shaft. $P = \frac{2\pi N T}{60}$ $T = \frac{60P}{2\pi N}$ $T = \frac{\pi}{16} \tau d^3$ $d = \sqrt[3]{\frac{16T}{\pi \tau}}$	2		
	2.. Design of sleeve or muff. Outer diameter of sleeve D = 2d + 13 mm Length of muff L = 3.5 d check torsional shear stress induced $\tau = \frac{16T}{\pi D^3 (1-k^4)}$, $k = \frac{d}{D}$	2		6
	3. Design of key. compare crushing and shear stresses then select square or rectangular key width of key w = d/4 and thickness t = d/6 length of key l = L/2 check torsional shear stress and crushing stress induced. $T = l \cdot w \tau \frac{d}{2}$, $\tau = ?$ $T = l \cdot \frac{t}{2} \cdot \sigma_c \cdot \frac{d}{2}$, $\sigma_c = ?$	2		
II.5.	**Sensitiveness** - A governor is said to be sensitive when it readily responds to a small change of speed. The movement of sleeve is a small fraction mean equilibrium speed if its change of speed no load to ful load is a measure of sensitivity. Sensitiveness is defined as the of the mean equilibrium speed to the difference between the and minimum **Hunting** - This is cause dby a governor whch is too sensitive. If a governor is too sensitive , it may fluctuate continuously above and below the nean speed because when load on engine falls or increase. When load falls, the sleeve rises rapidly to a maximum position. This shut off the supply of working fluid and affect a sudden fall of speed. As the speed falls below the mean value, the sleeve again moves rapidly and falls to a mnimum position to increase the supply of working fluid. The speed subsequently rises and becomes more than the average, the sleeve again falls. This process continues and is known as huntng.	3 3		6
II.6.	V- BELT / FLAT BELT 1. Suitable for shorter distances. / 1. Longer distances. 2. Trapezoidal in cross section / 2. Rectangular in cross section. 3. High frictional grip. / 3. Less grip. 4. No possibility of belt coming out of pulley / 4. Possibility of belt coming out. 5. Low percentage of slip. / 5. High percentage of slip. 6. Velocity ratio high. / 6. Velocity ratio low.	1	1 x 6	6

II.7.	1.Positive drives and constant speed ratio. 2.More compact 3.Operated at higher speeds. 4.High efficiency. 5.lighter loads on shafts and bearings. 6.Used where prcise timing is desired. 7.Wide range of power transmitted. 8.Less maintenance. 9.Used for nonintersecting and nonparallel shafts. (any six)	1	1 x 6	6
	PART C			
III.a.	Cylinder dia $D = 300\ mm$ $p = 1.5\ N/mm^2$ $n = 16$ $\sigma_t = 30\ MPa$ Force on cylinder cover $P = \frac{\pi D^2}{4} \cdot p$ $P = \frac{\pi \times 300^2}{4} \cdot 1.5 = 105975\ N$ — ① Resistance on studs $P = \frac{\pi d_c^2}{4} \sigma_t \cdot n$ — ② equating ① & ② $105975 = \frac{\pi d_c^2}{4} \times 30 \times 16$ $d_c = \sqrt{\frac{4 \times 105975}{\pi \times 30 \times 16}} = 16.77\ mm$ $d = \frac{d_c}{0.84} = \frac{16.77}{0.84} = 19.96 = 20\ mm$	2 2 2 2		8
III.b.	$d = 40\ mm$ $F = 20\ kN = 20 \times 10^3\ N$ $\tau = 60\ MPa$ width of key $w = d/4 = \frac{40}{4} = 10\ mm$ Thickness of key $t = d/6 = \frac{40}{6} = 6.67\ mm = 7\ mm$ Tangential force $F = l \cdot w \cdot \tau$ Length of key $l = \frac{F}{w \cdot \tau} = \frac{20 \times 10^3}{10 \times 60}$ $= 33.33\ mm$ $= 34\ mm$ Size of key is $10\ mm \times 7\ mm \times 34\ mm$.	2 2 3		7

IV.a.	n = 2 d_m = 50 mm p = 10 mm μ = 0.1 W = 25 kN $= 25 \times 10^3$ N Helix angle $\alpha = \tan^{-1}\left(\frac{n.p}{\pi d_m}\right)$ $= \tan^{-1}\left(\frac{2 \times 10}{\pi \times 50}\right) = \underline{7.26^\circ}$ Friction angle $\phi = \tan^{-1}\mu$ $= \tan^{-1}(0.1) = \underline{5.71^\circ}$ Effort required $P = W \tan(\alpha + \phi)$ $= 25 \times 10^3 \times \tan(7.26 + 5.71)$ $= \underline{5757.92\ N}$ Torque required $T = P . \frac{d_m}{2} = 5757.92 \times \frac{50}{2}$ $= \underline{143948 N}$	2 2 4		8
IV.b.	n = 2 P = 9 kN $= 9 \times 10^3$ N σ_t = 40 Mpa $P = \frac{\pi d_c^2}{4} . \sigma_t . n$ $d_c = \sqrt{\frac{4.P}{\pi \sigma_t n}} = \sqrt{\frac{4 \times 9 \times 10^3}{\pi \times 40 \times 2}}$ $= \underline{11.97\ mm}$ $d = \frac{d_c}{0.84} = \frac{11.97}{0.84} = 14.25\ mm$ standard $d = \underline{16\ mm}$	Eqn. 2 2 3		7
V.a.	T = 250 N-m $= 250 \times 10^3$ N-mm n = 4 τ = 50 Mpa for shaft, bolt and key. σ_c = 100 Mpa τ = 16 Mpa for flange and hub. $T = \frac{\pi}{16} \tau d^3$ Diameter of shaft, $d = \sqrt[3]{\frac{16T}{\pi \tau}} = \sqrt[3]{\frac{16 \times 250 \times 10^3}{\pi \times 50}}$ $= 29.42 = \underline{30\ mm}$ 1. Design of hub. Inside dia: of hub = 30 mm	2		

Outside dia: of hub $D = 2d$

$= 2 \times 30 = \underline{60\ mm}$

Length of hub $L = 1.5d$

$= 1.5 \times 30 = \underline{45\ mm}$

$$\tau = \frac{16T}{\pi D^3 (1-k^4)} = \frac{16 \times 250 \times 10^3}{\pi \times 60^3 \left[1 - \left(\frac{30}{60}\right)^4\right]}$$ 3

$= \underline{6.29\ N/mm^2} < 16\ MPa$

8

2. Design of flange

Outside dia: $D_o = 4d = 4 \times 30 = \underline{120\ mm}$

Thickness $t_f = 0.5d = 0.5 \times 30 = \underline{15\ mm}$

$$T = \pi \cdot D\, t_f\, \tau\, D/2$$

$$\tau = \frac{2T}{\pi D^2 t_f} = \frac{2 \times 250 \times 10^3}{\pi \times 60^2 \times 15} = \underline{2.95\ N/mm^2}$$ 3

$< 16\ MPa$

V.b.

$d_i = 0.60\, d_o$

$P = 200\ kW = 200 \times 10^3\ W$

$N = 80\ rpm$

$\tau = 60\ MPa$

$k = \frac{d_i}{d_o} = 0.60$

$$P = \frac{2\pi N T}{60}$$

$$T = \frac{60P}{2\pi N} = \frac{60 \times 200 \times 10^3}{2 \times \pi \times 80} = \underline{23873.2\ N\cdot m}$$ 2

7

$= \underline{23873.2 \times 10^3\ N\text{-}mm}$

$$T = \frac{\pi}{16} \tau\, d_o^3 (1-k^4)$$

$$d_o = \sqrt[3]{\frac{16T}{\pi \tau (1-k^4)}} = \sqrt[3]{\frac{16 \times 23878.2 \times 10^3}{\pi \times 60 (1-0.6^4)}}$$

$= 132.5 = \underline{140\ mm}$ 3

$d_i = 0.60\, d_o$

$= 0.60 \times 140 = \underline{84\ mm}$ 2

VI.a

$P = 40\ kW = 40 \times 10^3\ W$

$N = 140\ rpm$

$\tau = 42\ MPa$

$\sigma_c = 84\ MPa$

for muff $\tau = 14\ MPa$

1. Design of shaft.

$$T = \frac{60P}{2\pi N} = \frac{60 \times 40 \times 10^3}{2 \times \pi \times 140} = 2729.75\ N\cdot m = 2729.75 \times 10^3\ N\cdot mm$$

8

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

$$d = \sqrt[3]{\frac{16T}{\pi \cdot \tau}} = \sqrt[3]{\frac{16 \times 2729.75 \times 10^3}{\pi \times 42}}$$

$$= 69.18 = \underline{\underline{70\ mm}}$$

2

2. Design of muff.

Inside dia $= d = 70\ mm$

Outside dia $= D = 2d + 13$

$$= 2 \times 70 + 13 = \underline{\underline{153\ mm}}$$

Length $L = 3.5d$

$$= 3.5 \times 70 = \underline{\underline{245\ mm}}$$

3

Induced stress $\tau = \dfrac{16T}{\pi D^3 (1 - k^4)}$ $\quad k = \dfrac{d}{D} = 0.46$

$$= \frac{16 \times 2729.75 \times 10^3}{\pi \times 153^3 (1 - 0.46^4)}$$

$$= \underline{\underline{4.065\ N/mm^2}} < 14\ MPa.$$

3. Design of key

$$w = d/4 = \frac{70}{4} = \underline{\underline{17.5\ mm}}$$

Square key $t = d/4 = \dfrac{70}{6} = \cancel{11.67\ mm}\ \underline{\underline{17.5\ mm}}$

$$l = \frac{L}{2} = \frac{245}{2} = \underline{\underline{122.5\ mm}}$$

3

Induced shear stress $\tau = \dfrac{2T}{l \cdot w \cdot d}$

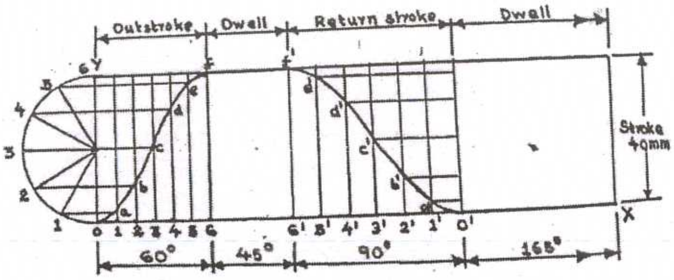
$$= \frac{2 \times 2729.75 \times 10^3}{122.5 \times 17.5 \times 70}$$

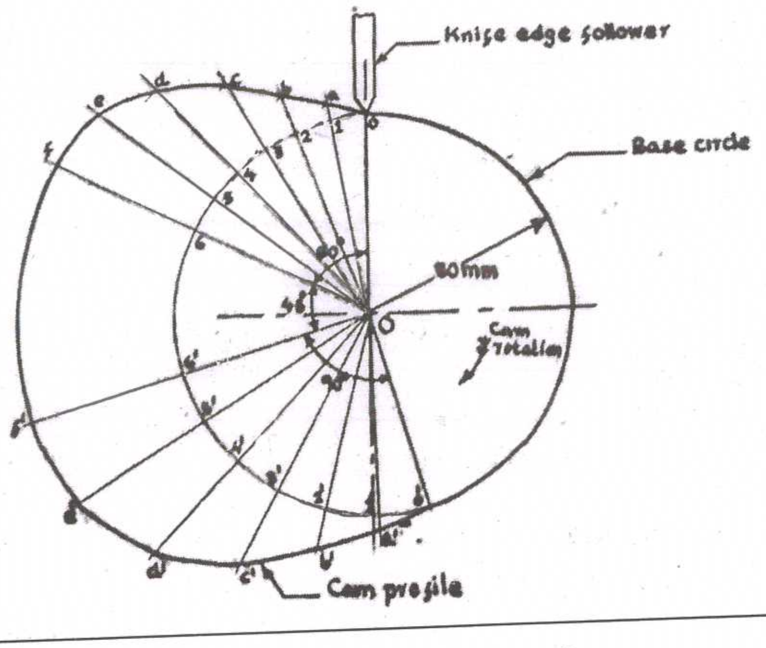
$$= 36.38\ MPa < 42\ MPa$$

" $\sigma_c = \dfrac{4T}{l \cdot t \cdot d} = \dfrac{4 \times 2729.75 \times 10^3}{122.5 \times 17.5 \times 70} = 72.76\ MPa$

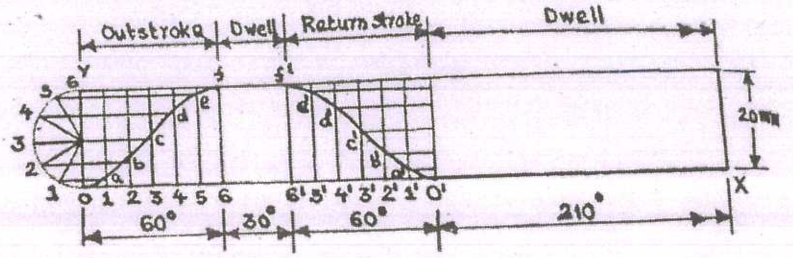
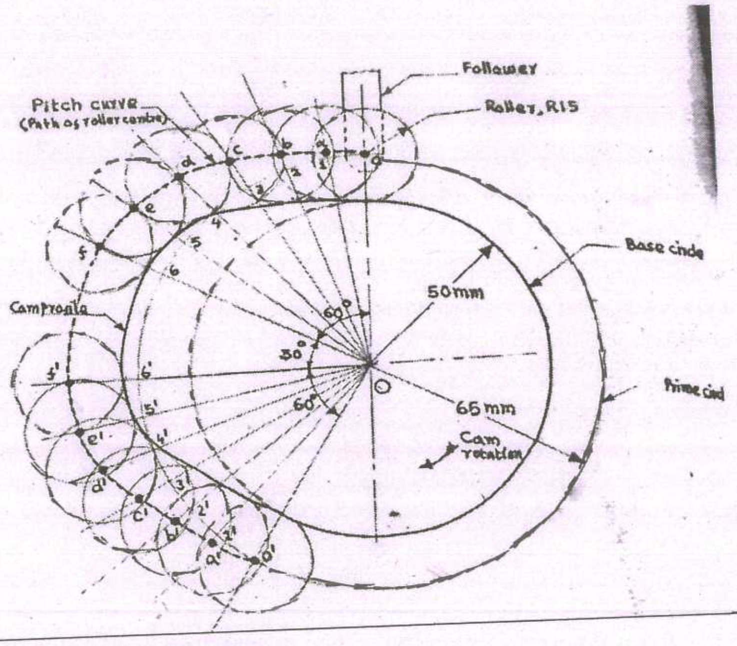
$< 84\ MPa$

6

VI.b.	$T = 2860 \text{ N-m} = 2860 \times 10^3 \text{ N-mm}$ $\theta = 1^\circ$ $l = 1400 \text{ mm}$ $G = 80 \text{ GPa} = 80 \times 10^3 \text{ MPa}$ Rigidity equation $\theta = \dfrac{584 \cdot T \cdot l}{G d^4}$ $d = \sqrt[4]{\dfrac{584 \cdot T \cdot l}{G \theta}}$ $= \sqrt[4]{\dfrac{584 \times 2860 \times 10^3 \times 1400}{80 \times 10^3 \times 1}}$ $= 73.53 \text{ mm}$ $= \underline{\underline{80 \text{ mm}}}$	Eqn. 2 5		7
VII.a.	 (a) Displacement diagram	3		

VII. a.		5		8
VII.b.	$d = 200\,mm = 200 \times 10^{-3}\,m$ $W = 50\,kN = 50 \times 10^{3}\,N$ $N = 100\,rpm$ $\mu = 0.02$ Heat generated $Q_g = \mu W v$ $= \mu W \frac{\pi d N}{60}$ $= \frac{0.02 \times 50 \times 10^{3} \times \pi \times 200 \times 10^{-3} \times 100}{60}$ $= 1050\,W$ $= 1.05\,kW$	Eqn. 3 4		7

8

VIII.a.		3	8
		5	
VIII. b.	$d = 180\ mm$		
	$W = 30\ KN = 30 \times 10^3\ N$		
	$\mu = 0.04$		
	$N = 120\ rpm$		
	Mean radius $R = \frac{2}{3} r = \frac{2}{3} \times 90 = \underline{\underline{60\ mm}}$	2	
	Frictional Torque $T = \mu W R$		7
	$= 0.04 \times 30 \times 10^3 \times 60$	2	
	$= 72000\ N\text{-}mm$		
	$= \underline{\underline{72\ N\text{-}m}}$		
	Power lost $P = \frac{2\pi N T}{60}$		
	$= \frac{2 \times \pi \times 120 \times 72}{60}$	3	
	$= \underline{\underline{904.78\ W}}$		

~~VIII~~ IX. a.	$C = 5m = 5\times10^3$ mm Smaller pulley $d_1 = 500$ mm Larger pulley $d_2 = 750$ mm (i) $L = \frac{\pi}{2}(d_2+d_1)+2C+\frac{(d_2-d_1)^2}{4C}$ $= \frac{\pi}{2}(750+500)+2\times5\times10^3+\frac{(750-500)^2}{4\times5\times10^3}$ $= 11966.6$ mm $= \underline{\underline{11.97\text{m}}}$ (ii) $L = \frac{\pi}{2}(d_2+d_1)+2C+\frac{(d_2+d_1)^2}{4C}$ $= \frac{\pi}{2}(750+500)+2\times5\times10^3+\frac{(750+500)^2}{4\times5\times10^3}$ $= 12041.6$ mm $= \underline{\underline{12.04\text{m}}}$	Eqn. 2 2 Eqn. 2 2		8
IX. b.	$N_1 = 150$ rpm $d_1 = 2m = 2\times10^3$ mm $d_2 = 1m = 1\times10^3$ mm $t = 6$mm $s = 0$ $\frac{N_2}{N_1} = \frac{d_1+t}{d_2+t}$ Speed of machine shaft $N_2 = \frac{d_1+t}{d_2+t}\times N_1$ $N_2 = \left[\frac{2\times10^3+6}{1\times10^3+6}\right]\times150$ $= \underline{\underline{299.1\text{ rpm}}}$	Eqn. 2 5		7

[redacted] X.a.	Smaller pulley $d_1 = 250\ mm$			
	Larger pulley $d_2 = 500\ mm$			
	$c = 1.5\ m = 1.5 \times 10^3\ mm$			
	(i) open belt-drive			
	$\theta = \pi - 2\sin^{-1}\left(\frac{d_2 - d_1}{2c}\right)$	Eqn - 2		8
	$= 180^\circ - 2\sin^{-1}\left(\frac{500-250}{2\times1.5\times10^3}\right) = 170.44^\circ$			
	$= 170.44 \times \frac{\pi}{180} = \underline{\underline{2.97\ \text{radians}}}$.	2		
	(ii) Cross belt drive			
	$\theta = \pi + 2\sin^{-1}\left(\frac{d_2 + d_1}{2c}\right)$	Eqn - 2		
	$= 180^\circ + 2\sin^{-1}\left(\frac{500+250}{2\times1.5\times10^3}\right) = 208.96^\circ$			
[redacted]	$= 208.96 \times \frac{\pi}{180} = \underline{\underline{3.65\ \text{radians}}}$.	2		
X.b.	$d = 400\ mm = 400 \times 10^{-3}\ m$			
	$N = 750\ rpm$			
	Tension on tight side $T_1 = 300\ N$			
	Tension on slack side $T_2 = 45.35\ N$			
	Power transmitted $P = (T_1 - T_2)\frac{\pi d N}{60}$	Eqn: 3		7
	$P = (300 - 45.35)\frac{\pi \times 400 \times 10^{-3} \times 750}{60}$			
	$= 4000.03\ W$	4		
	$= \underline{\underline{4\ kW}}$.			