Subject Code

- Any -

Chapter Name

- Any -

Apply

Examination:

Que.No	Marks	
Q 4a)(iii)	4	Question: **State the effect of keyway on the strength of the shaft.** Answer: The keyway is a slot machined either on the shaft or in hub to accommodate the key. It is cut by vertical or horizontal milling cutter. A little consideration will show that the keyway cut into the shaft reduces the load carrying capacity of the shaft. This is due to the stress concentration near the corners of the keyway and reduction in the crosssectional area of the shaft. It other words, the torsional strength of the shaft is reduced. The following relation for the weakening effect of the keyway is based on the experimental results by H.F. Moore. $e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$ where e = Shaft strength factor. w = Width of keyway, d = Diameter of shaft, and h = Depth of keyway =Thickness of key (t)/2 It is usually assumed that the strength of the keyed shaft is 75% of the solid shaft, which is somewhat higher than the value obtained by the above relation. In case the keyway is too long and the key is of sliding type, then the angle of twist is increased in the ratio K_θ as given by the following relation $K_\theta = 1 + 0.4\left(\frac{w}{d}\right) - 0.7\left(\frac{h}{d}\right)$ where k_θ= Reduction factor for angular twist. --

Examination: 2017 SUMMER

Que.No	Marks	
Q 1a)(iii)	8	Question: **A hollow shaft for a rotary compressor is to be designed to transmit maximum torque of 4750 N-m. The shear stress in the shaft is limited to 50 MPa. Determine the inside outside diameter of the shaft if the ratio of inside to outside diameter of the shaft is 0.4.** Answer: Design of Hollow shaft: Given Data: T =4750 N-m = 4750 X 103 N-mm , Ʈ = 50 N/mm2 , K=Di/Do =0.4 The hollow shaft is designed on the basis of strength from the derived torsion equation. 4750 X 103 N-mm ---- Thus Do= 79.18 mm 80 mm (Say) Di = 0.4 x Do = 0.4 x 80 = 32 mm --

Que.No	Marks	
Q 1b)(ii)	6	Question: **Design a bushed pin type flexible coupling for connecting a motor shaft to a pump shaft for the following service conditions. Power to be transmitted = 40 KW. Speed of the motor shaft = 1000 RPM. Diameter of the motor shaft = 50 mm Diameter of the pump shaft = 45 mm The bearing pressure in the rubber bush and allowable stress in the pins are to be limited to 0.45 N/mm2 and 25 MPa respectively** Answer: (see below)

Given Data: P= 40 KW = 40 X 10^3 W , N= 1000 rpm , d= 50 mm dp= 45 mm, τ_{ci}=15 N/mm^2

P_b=0.45 N/mm^2, τ=25 N/mm^2

1) Power Transmitted $P = \frac{2\pi NT}{60}$

$T = \frac{P \, X \, 60}{2\pi N} = \frac{40 \, X \, 103; X \, 60}{2\pi \, X \, 1000}$ = 381.97 N.m =381.97 x 10^3 N.mm

Let Number of Pins = 6

2) Diameter of pin: d_1 =0.5d/$\sqrt{n}$ =0.5 x 50/$\sqrt{6}$ =10.20

In order to permit the bending stress induced in the pin due to compressibility of brass bush .let us modify diameter of pin d1=20 mm .This diameter is threaded and secured Right hand half coupling .

Let us take, diameter of the enlarged portion in the left half coupling d1 =24 mm. A brass bush of 2mm is fitted over the enlarged portion of pin. also brass bush carries rubber bush of 6 mm.

Diameter of rubber bush =d_2 = d_1+2x2+2 x 6 =24+4+12 = 40 mm.

Diameter of pitch circle of pin =D1 = 2 x d + d2 + 2 x6 =100+40+12 = 152 mm.

3)Bearing load acting on each pin W = Pb x d_2 x l =0.45 x 40 x l = 18 x l

Total bearing load on all pins = n x W

Torque transmitted by coupling = T =n x W x D1/2

381.97 x 10^3 = 6 x 18 x l x 152/2

.l=46.54 mm

W=18 x l = 18 x 46.54 = 837.72 N-

4.Direct shear stress in coupling halves

$\tau = \frac{W}{\frac{\pi}{4}d1^2} = \frac{837.72}{\frac{\pi}{4}20^2}$ =2.67 N/mm^2

σ_b = (M / Z) , M= W X (l/2 +5)= 837.72 x(46.54/2 +5) Z = (π/32) 20^3

σ_b = [837.72 x(46.54/2 +5) / (π/32) 20^3] = 30.15 N/mm^2

Checking of maximum stress

According to Maximum shear stress theory

Do= 100 mm , W =300 KN = 300 X 10^3 N, P=12 mm , μ= μ1= 0.15

Since,Screw is double start, Lead of screw = 2 p =2 x12 =24 mm

dc= do-P =100-12 =88

Mean diameter d =(do+dc)/2 =(100+88)/2 =94 mm

Tan $\alpha = \frac{Lead}{\pi d} = \frac{2p}{\pi d}$, $\alpha = tan^{-1}\left(\frac{2p}{\pi d}\right)$

$\alpha = tan^{-1}\frac{24}{\pi x 94}$ = 4.64°

$\emptyset = tan^{-1}\mu$ $= tan^{-1} x \, 0.15$ = 8.53°

Torque Required to lift the load , T1= W.tan ($\alpha + \emptyset$) $\frac{d}{2}$

T1= 300 x 10^3x tan (4.64° + 8.53°) $\frac{94}{2}$ = 3301.15 x 10^3 N.mm

Total Torque =Tt=T1+T2

=3301.15 x 10^3 + 0 = 3301.15 x 10^3 N.mm ………….

Efficiency of screw:

n = $\frac{tan \propto}{tan(\propto + \emptyset)} = \frac{tan 4.64}{tan(4.64+8.53)}$ **= 0.347 i.e 34.71 %**

<table>
<tr><th>Que.No</th><th>Marks</th><th></th></tr>
<tr><td>Q 2b)(ii)</td><td>8</td><td>Question:
Draw neat sketch of a protected type flanged coupling showing all details.

Answer:
distance to some degree. c) For effective conjugate action i.e for maintaining a constant velocity ration, in case of involute gearing system, the center distance can be changed without affecting angular velocity ratio. d)In involute gearing as the path of contact is a straight line and the pressure angle is constant . Sketch of Protected type flanged coupling with details :

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<tr><td>Q 3 e)</td><td>4</td><td>Question:
A shaft 30 mm. diameter is transmitting power at a maximum shear stress of 80 MPa. If a pulley is connected to the shaft by means of a key, find the dimension of the key so that stress in the key is not to exceed 50 MPa and length of the key is 4 times the width.

Answer:
Comparison of welded joints with screwed joint. 1) Welded Joint is rigid & permanent. Screwed joint is temporary. 2) Cost of welded assembly is lower than that of screwed joints. 3) Strength of welded structure is more than screwed joints. 4) For welding joints, highly skilled worker are required 5) Welded joints are tight & leak proof as compared to Screwed joints. 6) Welded joint is very difficult to inspect
Design of Key
Data: d =30 mm , τ_{Max}=80 Mpa , τ_{Key}=50 Mpa , l_K =4 W_k
Torque transmitted by shaft is given by
$T = \frac{\pi}{16}\ x\ d^3\ x\ \tau$, $T = \frac{\pi}{16}\ x\ 30^3\ x\ 80$ = 424.115 x10 ^{3} N/mm^2
Considering shear failure of key,
$T = W_k \times l_k \times \tau \times \frac{d}{2}$
$424.115 \times 10^3 \frac{N}{mm2} = W_k \times 4\ W_k \times 50 \times \frac{30}{2}$
$\boldsymbol{W_k = 11.89\ mm}$
l_K =4 W_k , l_K= 4 x 11.89 = 47.56 mm
if Key is Square then $t_k = w_k$ = 11.89 mm
if Key is Rectangular thent_k = 2/3 w_k = 2/3 x 11.89 mm= 7.92 mm
compared to other joints.

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<tr><td>Q 4a)(iii)</td><td>4</td><td>Question:
Explain effect of keyways on strength of shaft. Name one type of key which does not affect strength of shaft

Answer:
Effect of keyways – when the keyways are cut on the shafts, material is removed at the skin, there by weakening the cross section of the shaft. Stress concentration effect is also serious at the corner of the keyways. Thus the shaft become weak. Type of key- Hollow saddle key or Tangent key
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</table>

Que.No	Marks	
Q 4b)(i)	6	Question: **(i) Explain different causes of gear tooth failure and suggest possible remedies to avoid such failures** Answer: Causes-1) Bending failure-every gear tooth acts as a cantilever. If the total repetitive dynamic load acting on the gear tooth is greater than the beam strength of the gear tooth then the gear tooth will fail in bending remedies-module and face width of the gear is adjusted so that the beam strength is greater than the dynamic load 2)Pitting-surface fatigue failure which occurs due to many repetition of Hertz contact stresses , failure occurs when the surface contact stresses are higher than the endurance limit of the material. It starts with the formation of pits which continue to grow resulting in the rupture of the tooth surface. Remedies- dynamic gear tooth load the of gear tooth between the gear tooth should be less than the wear strength 3)Scoring-the excessive heat is generated when there is a excessive surface pressure, high speed or supply of lubricant fails. it is stick-slip phenomenon in which alternate shearing and welding takes place rapidly at high spots. Remedies- by proper designing of the parameters such as speed, pressure and proper flow of lubricant, so that the temperature at the rubbing faces is within the permissible limits. 4)Abrasive wear- the foreign particles of lubricants such as dirt, dust or burr enter between the tooth and damage the form of tooth Remedies- by providing filters for the lubricating oil or using high viscosity lubricant oil which unable the formation of thicker oil film and hence permits easy passage of such particles with ought damage of gear tooth surface. 5)Corrosive wear- due to presence of corrosive elements such as additives present in the lubricating oils. Remedies- proper anti corrosive additives should be used. --

Examination: 2017 WINTER

Que.No	Marks	
Q 1 j)	2	Question: **State four types of keys.** Answer: (i) Sunk key: Rectangular sunk key, Square sunk key and Parallel sunk key (ii) Gib-head key (iii) Feather key Any four (iv) Woodruff key types of keys (v) Saddle key: Flat saddle key, Hollow saddle key (vi) Tangent key (vii) Round key --
Q 1 m)	2	Question: **What are the requirement of a good coupling?** Answer: A good coupling should have the following requirements: (i) It should be easy to connect and disconnect. (ii) It should transmit the full power from one shaft to another shaft without losses. (iii) It should hold the shafts in perfect alignment. (iv) It should reduce the transmission of shock loads from one shaft to another shaft. (v) It should have no projecting parts --

Que.No	Marks	
Q 3 b)	8	Question: **In a rigid flaninged complended to transmit 20K.W at 700.r.p.m**

Answer:

Solution :- Given data ⇒ $P = 20\ kW = 20 \times 10^3\ W$

$N = 700\ r.p.m$

$\tau_s = 40\ MPa = 40\ N/mm^2 = \tau_k = \tau_b$

$\sigma_{ckt} = 110\ N/mm^2$

$\tau = 10\ N/mm^2$, $\sigma_t = \sigma_{ck} = 100\ N/mm^2$, $n = 6$

The power transmitted by shaft,

$$P = 2\pi NT/60$$

$$\therefore \text{Torque},\ T = \frac{P \times 60}{2\pi N} = \frac{20 \times 10^3 \times 60}{2\pi \times 700} = 272.84\ N\text{-}m$$

$$\therefore T = 272.84 \times 10^3\ N\text{-}mm$$

We know that, torque transmitted by shaft is given by,

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

$$\therefore 272.84 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$$

$$\therefore d = 32.6230\ mm \approx 33\ mm\ (say)$$

$$\therefore \text{Diameter of shaft},\ \underline{d = 33\ mm}$$

Step I :- Design of Hub

Usual proportions are, D = Outer diameter of hub

$$\therefore \underline{D = 2d = 2 \times 33 = 66\ mm}$$

$$L = \text{Length of hub} = 1.5d = 1.5 \times 33 = \underline{49.5\ mm}$$

$$k = \frac{d}{D} = \frac{33}{66} = 0.5$$

Considering hub as a hollow shaft transmitting the same torque as that of the shaft, we have,

$$T = \frac{\pi}{16} \times \tau_{ci} \times D^3 (1 - k^4)$$

$$\therefore 272.84 \times 10^3 = \frac{\pi}{16} \times \tau_{ci} \times (66)^3 \times [1 - (0.5)^4]$$

$$\therefore \tau_{ci} = 5.15\ N/mm^2 < 10\ N/mm^2$$

Thus, the induced shear stress in the cast iron hub is less than the given permissible shear stress. Hence, the design is safe.

Step - II :- Design of flange

$$\text{Take},\ t_f = \frac{d}{2} = \frac{33}{2} = 16.5\ mm$$

While transmitting the torque, the flange is under shear.

$\therefore T$ = Circumference of hub × Thickness of flange × Shear stress × Radius of hub

$$\therefore T = (\pi \times D) \times t_f \times \tau_f \times \frac{D}{2}$$

$$\therefore 272.84 \times 10^3 = \pi \times 66 \times 16.5 \times \tau_f \times \frac{66}{2}$$

$$\therefore \tau_f = 2.42\ N/mm^2 < 10\ N/mm^2$$

Thus, induced shear stress is less than given permissible shear stress for flange material. Hence, design is safe.

The other proportions are, $\underline{D_2 = 4d = 4 \times 33 = 132\ mm}$

& thickness of protective circumferential flange,

$$\underline{t_p = 0.25\,d = 0.25 \times 33 = 8.25\ mm}$$

Step - III :- Design of bolts

The bolts are subjected to shear stress due to torque transmitted.

$$\therefore \text{Load on each bolt} = \frac{\pi}{4} \times (d_c)^2 \times \tau_b$$

$$\therefore \text{Total load on all bolts} = n \times \frac{\pi}{4} \times (d_c)^2 \times \tau_b$$

$$\therefore \text{Torque transmitted} = \text{Load} \times \text{radius} = n \times \frac{\pi}{4} \times (d_c)^2 \times \tau_b \times \frac{D_1}{2}$$

Taking, $\underline{D_1 = 3d = 3(33) = 99\ mm}$

& $n = 6$ (given), the above equation becomes,

$$\therefore 272.84 \times 10^3 = 6 \times \frac{\pi}{4} \times (d_c)^2 \times 40 \times \frac{99}{2}$$

$$\therefore d_c = 5.407\ mm$$

We have, $d_c = 0.84 \times d_o$

$$\therefore d_o = \frac{d_c}{0.84} = \frac{5.407}{0.84} = 6.428\ mm \simeq 8\ mm\ (say)$$

∴ We can use M8 bolts. For safety reasons, we can increase the size of bolts upto M16.

Que.No	Marks	
Q 3 c)	4	Question: **What are the considerations in design of dimensions of formed and parallel key having rectangular cross section?** Answer: Bending moment on lever, $R_b = d_2/2 = 64/2 = 32$ mm $M = P \times [L - R_b] = 800 \times [1000 - 32] = 774.4 \times 10^3$ N.mm $\sigma_b = (M / Z)$, $Z = 1/6\, t\, B^2 = 1.5 \times t^3$ $73 = (774.4 \times 10^3 / 1.5 \times t^3)$, $\mathbf{t = 19.9\ mm \cong 20\ mm}$ **& B = 3t = 3 x 20 = 60 mm** --

Que.No	Marks	
Q 6 a)	8	Question: Effect of Keyway on strength of shaft

Question:
Effect of Keyway on strength of shaft

Answer:

Effect of Keyway on strength of shaft:

The keyway is a slot machined either on the shaft or in hub to accommodate the key.It is cut by vertical or horizontal milling cutter.

A little consideration will show that the keyway cut into the **shaft reduces the load carrying capacity of the shaft.**

This is due to the stress concentration near the corners of the keyway and **reduction in the cross-sectional area of the shaft**. It other words, the **torsional strength of the shaft is reduced.**

The following relation for the weakening effect of the keyway is based on the experimental results by H.F. Moore. $e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$

where e = Shaft strength factor.

w = Width of keyway,

d = Diameter of shaft, and

h = Depth of keyway =Thickness of key (t)/2

It is usually assumed that the **strength of the keyed shaft is 75% of the solid shaft**, which is somewhat higher than the value obtained by the above relation.

In case the keyway is too long and the key is of sliding type, then the **angle of twist is increasedin the ratio K_θ** as given by the following relation

$$K_\theta = 1 + 0.4\left(\frac{w}{d}\right) - 0.7\left(\frac{h}{d}\right)$$

where k_θ= Reduction factor for angular twist.

The different CAUSES of gear teeth failure:

1. *Bending failure.*
2. *Pitting.*
3. *Scoring.*
4. *Abrasive wear.*
5. *Corrosive wear*

1. *Bending failure.*

Gear tooth behave like a cantilever beam subjected to repetitive bending stress. The tooth may crack due to repetitive bending stress

In order to avoid such failure, the module and face width of the gear is adjusted so that the beam strength is greater than the dynamic load.

2. *Pitting*

It is a surface fatigue failure due to repetitive contact stresses. Pitting starts when total load acting on the gear tooth exceeds the wear strength of the gear.

In order to avoid the pitting, the dynamic load between the gear tooth should be less than the wear strength of the gear tooth.

3. *Scoring.*

It is lubrication failure. Inadequate lubrication along with high tooth load & poor surface finish results in breakdown of oil film and causes metal to metal contact.

This type of failure can be avoided by properly designing the parameters such as speed, pressure and proper flow of the lubricant, so that the temperature at the rubbing faces is within the permissible limits.

4. *Abrasive wear.*

It is a surface damage caused by particles trapped in between the matting teeth surfaces.

This type of failure can be avoided by providing filters for the lubricating oil or by using high viscosity lubricant oil which enables the formation of thicker oil film and hence permits easy passage of such particles without damaging the gear surface.

5. *Corrosive wear..*

It is due to chemical action by the improper lubricant or sometimes it may be due to surrounding atmosphere which may be corrosive nature .In order to avoid this type of wear, proper anti-corrosive additives should be used.

Q 6b)(ii) | 8

Question:
Design consideration while designing the spur Gear

Answer:
1) The power to be transmitted 2) The velocity ration or speed of gear drive. 3) The central distance between the two shafts 4) Input speed of the driving gear. 5) Wear characteristics of the gear tooth for a long satisfactory life. 6) The use of space & material should be economical. 7) Efficiency & speed ratio 8) Cost

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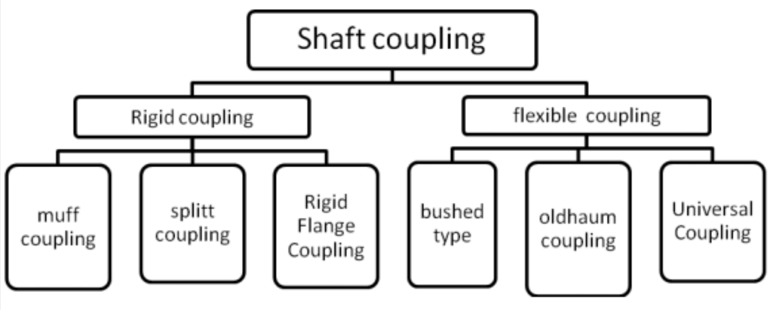
Que.No	Marks	
Q 6c)(ii)	8	Question: **State of the Classification of shaft coupling :** Answer:  ---

Examination: 2016 SUMMER

Que.No	Marks	
Q 1b)(ii)	6	

Question:

The shaft running at 125 r.p.m. transmits 440 kW. Find the diameter of shaft (d) if allowable shear stress in shaft material is 55 N/mm2 and the angle of twist must not be more than 1□ on a length of 16(d). The modulus of rigidity G = 0.80 □ 105 N/mm

Answer:

Given Data: P= 40 KW = 40 X 10^3 W , N= 1000 rpm , d= 50 mm dp= 45 mm,τ_{ci}=15 N/mm^2

P_b=0.45 N/mm^2, τ=25 N/mm^2

1) Power Transmitted $P = \frac{2\pi NT}{60}$

$T = \frac{P\ X\ 60}{2\pi N} = \frac{40\ X\ 103;X\ 60}{2\pi\ X\ 1000}$ = 381.97 N.m =381.97 x 10^3 N.mm

Let Number of Pins = 6

2) Diameter of pin: d_1 =0.5d/$\sqrt{n}$ = 0.5 x 50/$\sqrt{6}$ =10.20

In order to permit the bending stress induced in the pin due to compressibility of brass bush .let us modify diameter of pin d1=20 mm .This diameter is threaded and secured Right hand half coupling .

Let us take, diameter of the enlarged portion in the left half coupling d1 =24 mm. A brass bush of 2mm is fitted over the enlarged portion of pin. also brass bush carries rubber bush of 6 mm.

Diameter of rubber bush =d_2 = d_1+2x2+2 x 6 =24+4+12 = 40 mm.

Diameter of pitch circle of pin =D1 = 2 x d + d2 + 2 x6 =100+40+12 = 152 mm.

3)Bearing load acting on each pin W = Pb x d_2 x l =0.45 x 40 x l = 18 x l

Total bearing load on all pins = n x W

Torque transmitted by coupling = T =n x W x D1/2

381.97 x 10^3 = 6 x 18 x l x 152/2

.l=46.54 mm

W=18 x l = 18 x 46.54 = 837.72 N-

4.Direct shear stress in coupling halves

$\tau = \frac{W}{\frac{\pi}{4}d1^2} = \frac{837.72}{\frac{\pi}{4}20^2}$ =2.67 N/mm^2

σ_b = (M / Z) , M= W X (l/2 +5)= 837.72 x(46.54/2 +5) Z = (π/32) 20^3

σ_b = [837.72 x(46.54/2 +5) / (π/32) 20^3] = 30.15 N/mm^2

Checking of maximum stress

According to Maximum shear stress theory

$$\boldsymbol{d = 146mm}$$

iii. Diameter of shaft on the basis of rigidity

$\frac{T}{J} = \frac{G\theta}{L}$-------------------- **(1 Mark)**

$$\frac{33.61 \times 10^6}{J} = \frac{(0.80 \times 10^5) \times \frac{\pi}{180}}{16 \times d}$$

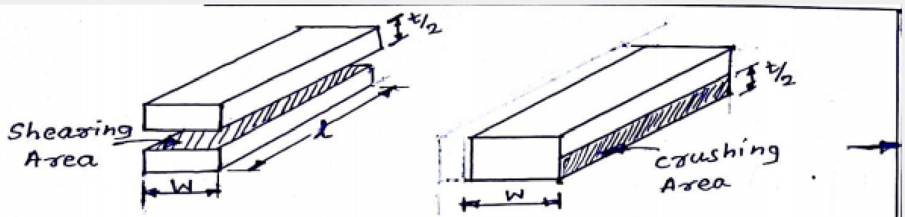
$$385.14 \times 10^3 d = J$$

$$385.14 \times 10^3 d = \frac{\pi}{32} \times d^4$$

$$\boldsymbol{d = 157.71mm}$$

Que.No	Marks	
Q 1 iii)	4	Question: **State any four factors to be considered while selecting the coupling.** Answer: Following factors should be consider while selecting coupling 1) Cyclic operation 2) Duration or life 3) Misalignment of shafts 4) Required torque and desired speed 5) Direction of rotation 6) Protection against overload 7) Operating conditions --
Q 2 c)	8	Question: **A belt pulley is fastened to a 90 mm diameter shaft running at 300 r.p.m. by means of a key 20 mm wide and 140 mm long. Allowable stress for the shaft and key material are 40 N/mm2 in shear and 100 N/mm2 in crushing. Find the power transmitted and the depth of the key required.** Answer: Diameter of shaft= 90mm N=300 w=20 mm l=140 mm $\tau = 40\ N/mm^2$ $\sigma_c = 100 N/mm^2$ **To find:** Power Transmitted & Depth of Key Required **Solution:** **To find torque T:** $T = w \times l \times \tau \times \frac{d}{2}$ $T = 20 \times 140 \times 40 \times \frac{90}{2}$ $T = 5.04 \times 10^6 N.mm$ $\boldsymbol{T = 5.04 \times 10^3 N.m}$ --------------------- **(2 Mark)** **To find Power P:** $P = \frac{2\pi NT}{60}$ $P = \frac{2\pi \times 300 \times 5.04 \times 10^3}{60}$ $\boldsymbol{P = 158.33 \times 10^3 Watt}$ $\boldsymbol{P = 158.33 kW}$ **To find depth of key h:** $T = l \times \frac{t}{2} \times \frac{d}{2} \times \sigma_c$ $5.04 \times 10^6 = 140 \times \frac{t}{2} \times \frac{90}{2} \times 100$ $\boldsymbol{t = 16mm}$ $Depthonkeywayonshaft(h) = \frac{t}{2}$ $Depthonkeywayonshaft(h) = \frac{16}{2}$ $\boldsymbol{h = 8mm}$ --

<table>
<tr><th>Que.No</th><th>Marks</th><th></th></tr>
<tr><td>Q 3 c)</td><td>4</td><td>
Question:
Prove that, for a square key, the permissible crushing stress is twice the permissible shear stress.

Answer:


(a) Shearing failure (b) Crushing failure
• Considering shearing failure of key, from fig (a),
Torque Transmitted by key $= T = F \times \frac{d}{2}$ —— (1)
Where, F = Shearing force (tangential)
F = Shearing Area × Shearing stress
$= l \times w \times \tau$, put in eq^n 1
$T = l \times w \times \tau \times \frac{d}{2}$ —— (2)
• Considering crushing failure of key from fig (b),
F = tangential crushing force
= Crushing area × crushing stress
$= l \times \frac{t}{2} \times \sigma_{ck}$, put in eq^n 1
Torque transmitted by key $= T = F \times \frac{d}{2}$
$T = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$ —— (3)
• If key is equally strong in shearing & crushing
(2) = (3)
$l \times w \times \tau \times \frac{d}{2} = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$
as, for square key $w = t$
$\boxed{\frac{\tau}{\sigma_{ck}} = \frac{1}{2} \text{ or } \sigma_{ck} = 2\tau}$

</td></tr>
<tr><td>Q 3 d)</td><td>4</td><td>
Question:
Why a coupling should be placed as close to a bearing as possible

Answer:
Coupling should be placed as close to a bearing because of following reasons i. It gives minimum vibrations ii. Bending load on the shaft can be minimized iii. It increases power transmission stability. iv. To avoid deflections of shaft.

</td></tr>
</table>

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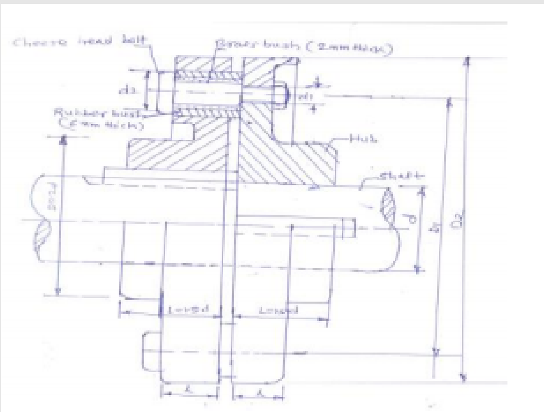
Q 4a)(ii)	4	Question: **State the effect of key-way on the strength of shaft with suitable diagram** Answer: The keyway is a slot machined either on the shaft or in hub to accommodate the key.It is cut by vertical or horizontal milling cutter. A little consideration will show that the keyway cut into the shaft reduces the load carrying capacity of the shaft. This is due to the stress concentration near the corners of the keyway and reduction in the cross-sectional area of the shaft. It other words, the torsional strength of the shaft is reduced. The following relation for the weakening effect of the keyway is based on the experimental results by H.F. Moore. $$e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$$ where$e$ = Shaft strength factor. w = Width of keyway, + --
Q 4b)(ii)	6	Question: **State any six design considerations while designing the spur gear** Answer: 1) The power to be transmitted 2) The velocity ration or speed of gear drive. 3) The central distance between the two shafts 4) Input speed of the driving gear. 5) Wear characteristics of the gear tooth for a long satisfactory life. 6) The use of space & material should be economical. 7) Efficiency & speed ratio 8) Cost --

Examination: 2016 WINTER

Que.No	Marks	
Q 1a)(iii)	4	Question: **Draw a neat sketch of flexible flange coupling and label its main components.** Answer:  --

Que.No	Marks	
Q 1b)(ii)	6	Question: **A hollow shaft is required to transmit 50 kW power at 600 rpm. Calculate its inside and outside diameters if its ratio is 0.8. Consider yield strength of material as 380N/mm2 and factor of safety as 4.** Answer: Given : P= 50 KW = 50000W Speed = 600rpm k=Di/do = 0.8 σyt= 380 N/mm2 Factor of safety= 4 Design stress σt=σyt/fos =380/4 =95 Shear stress = τ =σt/2 = 95/2 =47.5N/mm2 Torque transmitted by hollow shaft T = P x 60/2πN T = 50000 x 60/2π x600 T = 795.67 N-m T= 795670 Nmm T= π/16 Xτ X do3 (1-k 4) 795670 =π/16 X 47.5 X do3 (1-0.84) Do3=144529.313 Do = 53 mm say 55 mm Di = 0.8X 55 = 44mm --
Q 3 c)	4	Question: **State the 'Lewis equation' for spur gear design. State SI unit of each term in the equation.** Answer: Lewis equation: WT = σw.b.π.m.y, WT= Tangential load acting at the tooth in N σw= bending stress in N/mm2 b= width of the gear face in mm m= module in mm y= lewis form factor. --

<table>
<tr><th>Que.No</th><th>Marks</th><th></th></tr>
<tr><td>Q 3 e)</td><td>4</td><td>Question:
Prove that for a square key sc = 2t where sc = crushing stress t = shear stress.

Answer:
F= tangential force acting at the circumference of the shaft,
d = dia. Of shaft,
l = length of key,
w = width of key
t = thickness of key
τ and σ_c= shear and crushing stress for the material of key
Consider shearing of key, the tangential shearing force acting at the circumference of the shaft ,F = Area resisting shearing X shear stress = lxw x 6s
Torque transmitted by the shaft , T = F X d/2 = lxw x 6s x d/2
Consider crushing of key, the tangential crushing force acting at the circumference of the shaft ,F = Area resisting crushing x crushing stress = lxt/2 x 6c
Torque transmitted by the shaft , T = F X d/2 = lxt/2 x 6c X d/2
The key is equally strong in shearing and crushing ,if
lxw x 6s x d/2= lxt/2 x σ_cX d/2
w/t =6c/2 τ
$\sigma_c=2\tau$
--</td></tr>
<tr><td>Q 4a)(iii)</td><td>4</td><td>Question:
State four important modes of gear failure.

Answer:
Modes of Gear Failure: ANY 4 modes 1. Bending failure. Every gear tooth acts as a cantilever. If the total repetitive dynamic load acting on the gear tooth is greater than the beam strength of the gear tooth, then the gear tooth will fail in bending, 2. Pitting. It is the surface fatigue failure which occurs due to many repetition of Hertz contactstresses. 3. Scoring. The excessive heat is generated when there is an excessive surface pressure, high speed or supply of lubricant fails. 4. Abrasive wear. The foreign particles in the lubricants such as dirt, dust or burr enter betweenthe tooth and damage the form of tooth. 5. Corrosive wear. The corrosion of the tooth surfaces is mainly caused due to the presence ofcorrosive elements such as additives present in the lubricating oils
--</td></tr>
</table>

Que.No	Marks	
Q 4b)(i)	6	Question: **Explain the design procedure of shaft on the basis of torsional rigidity. State the equation with SI units. State two applications of this approach.** Answer: **Design procedure of Shaft on the Basis of torsional rigidity.** **Torsional rigidity.** The torsional rigidity is important in the case of camshaft of an I.C.engine where the timing of the valves would be affected. The permissible amount of twist should not exceed 0.25° per metre length of such shafts. For line shafts or transmission shafts, deflections 2.5 to 3 degree per metre length may be used as limiting value. The widely used deflection for the shafts is limited to 1 degree in a length equal to twenty times the diameter of the shaft. The torsional deflection may be obtained by using the **torsion equation**, Diameter of shaft on the basis of rigidity $\frac{T}{J} = \frac{G\theta}{L}$ Where ,**θ** = Torsional deflection or angle of twist **in radians,** ***T*** = Twisting moment or torque on the shaft, N.mm ***J*** = Polar moment of inertia of the cross-sectional area about the axis of rotation, L=Length of shaft in mm G=Modulus of rigidity in N/mm^2 $J = \frac{\pi}{32} \times d^4$-----------For solid shaft $J = \frac{\pi}{32} \times (do^4 - di^4)$-----------For Hollow shaft Two Applications: Propeller shaft of automobile, marine engine shaft and Shaft of pump and motor --

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Que.No	Marks	
Q 1a)(iii)	4	Question: **Write Lewis equation for strength of gear tooth. State the meaning of each term** Answer: Torque required to lower the load $T_1 = P \times \frac{d}{2} = W.\tan(\emptyset - \alpha)\frac{d}{2}$................. **1 M** If however ,Φ > α the torque required to lower the load will be positive,indicating that an effort is applied to lower the load,such a screw is known as self-locking screw. σw = Beam strength of the tooth b = Width of the gear face Pc = Circular pitch **m** = Module Y is known as Lewis form factor or tooth form factor --

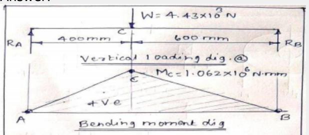
Que.No	Marks	
Q 1b)(ii)	6	(see below)
Q 2 a)	8	(see below)

Q 1b)(ii) — Marks: 6

Question:

A shaft 1000 mm long is supported between two bearings. A pulley of 250 mm diameter is keyed at 400 mm distance away from left hand bearing. The power transmitted by shaft is 10 kW at 800 r.p.m. The pulley gives power to another pulley vertically below it having an angle of contact between pulley and belt as 180°. The weight of pulley is 300 N. The coefficient of friction between belt and pulley is 0.15. Take shear stress for shaft material as 60 MPa. Find the diameter of the shaft.

Answer:



- To find R_A & R_B, from fig ⓐ

$\Sigma M_A = 0$

$4.43\times10^3 \times 400 = R_B \times 1000$

$\underline{R_B = 1773.59\ N}$

$\Sigma f_y = 0$

$R_A + R_B - 4.43\times10^3 = 0$

$\underline{R_A = 2656.41\ N}$

- To find maximum bending moment

$M_c = 400 \times 2656.4$

$\underline{M_c = 1.062\times10^6\ N\cdot mm}$

- To find diameter of shaft (d)

According to max. shear stress theory

$T_e = \frac{\pi}{16} \times \tau_{max} \times d^3$ —— ②

& $T_e = \sqrt{M^2 + T^2}$

$= \sqrt{(1.062\times10^6)^2 + (119.42\times10^3)^2}$

$\underline{T_e = 1.068\times10^6\ N\cdot mm}$

Put in eqⁿ ②

$1.068\times10^6 = \frac{\pi}{16} \times 60 \times d^3$

$\boxed{d = 44.93 \simeq 46\ mm}$ Diameter of shaft

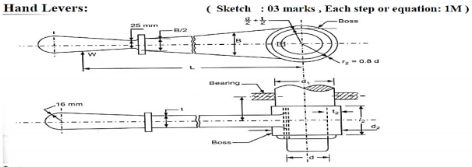
Q 2 a) — Marks: 8

Question:

Explain the design procedure of handlever with neat sketch.

Answer:

a) Hand Levers: (Sketch : 03 marks , Each step or equation: 1M)



Let

P = Force applied at the handle,

L = Effective length of the lever,

σt = Permissible tensile stress, and

τ = Permissible shear stress.

Step I : Considering shaft is under pure torsion, we have,

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

But, twisting moment on shaft,

$$T = P\times L$$

∴ On equating, we have,

$$P\times L = \frac{\pi}{16}\cdot\tau\cdot d^3$$

∴ Diameter of shaft (d) may be obtained.

Step II : Using the empirical relations, fix the other dimensions as,

$$d_2 = 1.6d$$
$$t_2 = 0.3d$$
$$l_2 = d \text{ to } 1.2d$$
$$l = 2\times l_2$$

Step III : Considering the shaft supported at the centre of the bearing under combined twisting and bending moment, we have,

$$M = P\times l \text{ and } T = P\times L$$

∴ Equivalent twisting moment,

$$T_e = \sqrt{M^2+T^2} = \sqrt{(P\times l)^2+(P\times L)^2} = P\sqrt{l^2+L^2}$$

Also, equivalent twisting moment,

$$T_e = \frac{\pi}{16}\cdot\tau_{max}\cdot d^3$$

$$\therefore \quad P\sqrt{l^2+L^2} = \frac{\pi}{16}\cdot\tau_{max}\cdot d^3$$

From here, value of d_1 = diameter of shaft supported at centre of bearings can be determined.

Step IV : Design of key : Let t_1 = thickness of key, w = width of key and l_1 = length of key.

After finding diameter of shaft (d), we can fix the dimensions for key as,

$$w = \frac{d}{4} \text{ and } t_1 = \frac{d}{6}$$

Considering shear failure of key,

We have, $$T = (w\cdot l_1\cdot\tau)\times\frac{d}{2}$$

∴ Length l_1 can be determined.

Also, length l_1 may be taken as length of boss i.e. l_2.

Step V : Considering bending failure of lever, we can determine the cross-section of leve near the boss.

Let t = thickness of lever near the boss and B = width of height of lever near the boss.

We have,

Bending moment on the lever = $M = P\times(L-r_b)$

and Section modulus = $Z = \frac{1}{6}\cdot t\cdot B^2$

where, r_b = Radius of boss = $\frac{d_2}{2}$

$$\therefore \quad \text{Bending stress, } \sigma_b = \frac{M}{Z} = \frac{P\cdot(L-r_b)}{\frac{1}{6}\cdot tB^2}$$

Width of lever may be taken as, B = 4t to 5t.

From this equation, values of t and B can be determined.

Que.No	Marks	
Q 2 b)	8	Question: Explain the design procedure of bush pin type flexible coupling with neat sketch

Question:

Explain the design procedure of bush pin type flexible coupling with neat sketch

Answer:

It consist of two shafts, two key, flanges, key, pin, rubber bush, brass bush, & pin ,following are the designations of various coupling dimensions which are use for the design procedure.

d = diameter of shaft , **d_1**= diameter of enlarge portion of pin ,

d_2 =diameter of rubber bush , **d_b**= nominal diameter of pin,

D = outer diameter of hub , **D_1**= diameter of bolt circle

D_2= outer diameter of flange , ***l***= length of hub

l_b= length of bush in hub , **t_f** = thickness of flange **t_p**=thickness of protection

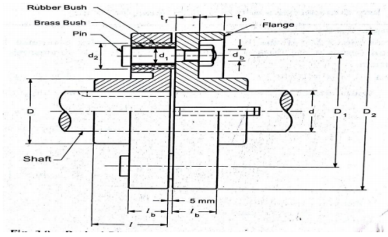
n = no of pins, and no of pins are selected from diameter of shaft

=3 for d upto 30mm

=4 for d upto 75 mm

=6 for d upto 110 mm

=8 for d upto 150 mm



Step no 1: Design of shaft (d)

Considering maximum shear stress,

$$T=\frac{\pi}{16}\times\tau_{max}\times d^3 \text{..........................(a)}$$

Step no 2: Design of key

Fix the dimension of key from the standard shaft diameter, and as per the requirement square or rectangular key can be selected,

w= width of key

t = thickness of key

l_k=length of key =**1.5d**by proportion...............(1)

1. **For Square key** , w=t=d/4...........................**..(b)**
2. **For rectangular key,** w=d/4 & t=2/3w................**(c)**
3. & length of key can be obtained **by shearing and crushing failure of key.**
   Considering shearing of key,
   $T=\tau_k\times w\times l_k\times d/2$ ....(2)
   Considering crushing failure of key,
   $T=\sigma_c\times l_k\times d/2\times t/2$.......................(3)
   - Consider maximum value of length of key from equation 1,2,3

Step no 3: Design of hub

1. D = outer diameter of hub **D=2d**
2. *l*= length of hub ***l*** **=1.5d**
3. Tortional shear stress in the hub can be calculated from below equation,
   $$T=\frac{\pi}{16}\times\tau_h\times(1-k^4)$$

   Where, k=d/D

Step no 4: Design of flange

1. t_f = thickness of flange **t_f =0.5d**
2. t_p =thickness of protection **t_p =0.25d**
3. **Torque transmitted by flange**
   T=shear area × direct shear stress× outside radius of hub

$$T=\pi D\, t_f\times\tau_f\times D/2$$

From above equation value of τ_f can be calculated and checked

Step no 5: Design of pin

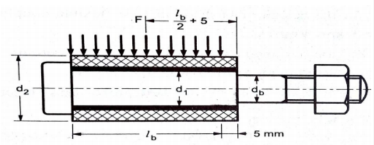


Fig .shows the pin with the rubber bush .The design of pin is as follows

- Nominal Diameter of pin (d_b)
  $d_b = \frac{0.5d}{\sqrt{n}} \longrightarrow (1)$
- Diameter of enlarge portion of Pin (d_1)
  $d_1 = d_b + 4 \longrightarrow (2)$
- Outer Diameter of Rubber bush (d_2)
  Assume that brass bush 2 mm thickness & rubber bush 6mm thickness are fitted
  $d_2 = d_1 + (2\times2) + (2\times6) \longrightarrow (3)$
- Dia. of bolt circle (D_1)
  $D_1 = D + d_2 + (2\times8) \longrightarrow (4)$
- Length of bush in the flange (l_b)
  considering bearing pressure on pin
  $F = P_b\times d_2\times l_b$ &
  Torque Transmitted,
  $T = n\times F\times D_1/2$,
  $T = n\times P_b\times d_2\times l_b\times D_1/2 \longrightarrow (5)$

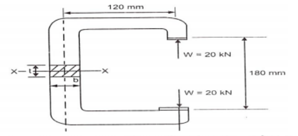
- Stress induced in pin
  Maximum bending moment on pin is ,
  $M = F\left[\frac{l_b}{2}+5\right]$
  ∴ The maximum bending stress in pin
  $\sigma_b = \frac{M}{Z} = \frac{F\times\left[\frac{l_b}{2}+5\right]}{\frac{\pi}{32}d_b^3} \longrightarrow (6)$
  where, $F = \frac{T}{n\times D_1/2}$
- Direct shear stress in pin
  $\tau_b = \frac{F}{\frac{\pi}{4}\times d_b^2} \longrightarrow (7)$
- Principal stress in pin
  $\tau_{max} = \sqrt{\left(\frac{\sigma_b}{2}\right)^2+\tau_b^2} \longrightarrow (8)$
  $\sigma_{max} = \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2+\tau_b^2} \longrightarrow (9)$

<table>
<tr><th>Que.No</th><th>Marks</th><th></th></tr>
<tr><td>Q 2c)(ii)</td><td>8</td><td>

Question:

Design ‘‘C’’ clamp frame for a total clamping force of 20 kN. The cross-section of the frame is rectangular and width to thickness ratio is 2. The distance between the load line and natural axis of rectangular cross section is 120 mm and the gap between two faces is 180 mm. The frame is made of cast steel for which maximum permissible tensile stress is 100 N/mm

Answer:



Given data : $W = 20 \text{ kN} = 20 \times 10^3 \text{ N}, \; b = 2t$

$e = 120 \text{ mm}, \; \sigma_t = 100 \text{ N/mm}^2$

At section X-X :

Step I : Direct stress : $\sigma_o = \dfrac{W}{A} = \dfrac{20 \times 10^3}{b \times t} = \dfrac{20 \times 10^3}{2t^2} = \dfrac{10 \times 10^3}{t^2}$

Step II : Bending stress : $\sigma_b = \dfrac{M}{Z} = \dfrac{W \times e}{\frac{1}{6} tb^2}$

$$\sigma_b = \frac{20 \times 10^3 \times 120 \times 6}{t \times (2t)^2} = \frac{3.6 \times 10^6}{t^3}$$

---------- **2 Marks**

Step III : Resultant stress :

$$\sigma_{tR} = \sigma_o + \sigma_b$$

$$\therefore \quad \frac{10 \times 10^3}{t^2} + \frac{3.6 \times 10^6}{t^3} = 100$$

$$\therefore \quad \frac{10 \times 10^3 \times t + 3.6 \times 10^6}{t^3} = 100$$

$$\therefore \quad 10 \times 10^3 t + 3.6 \times 10^6 = 100t^3$$

$$\therefore \quad 100t^3 - 10 \times 10^3 t = 3.6 \times 10^6$$

Divide the equation by 100,

$\therefore \quad t^3 - 100 t = 3.6 \times 10^4$; Using trial and error

---------- **1 Marks**

t= 34 mm & b= 2t =2 x 34 = 68 mm ---------- 1 Marks

</td></tr>
<tr><td>Q 3 c)</td><td>4</td><td>

Question:

Prove that for square key equally strong in shear and crushing, the permissible crushing stress is twice the permissible shear stress

Answer:

Torque required to lower the load

$T_1 = P \times \frac{d}{2} = W. \tan(\varnothing - \alpha)\frac{d}{2}$................. **1 M**

If however ,$\Phi > \alpha$ the torque required to lower the load will be positive,indicating that an effort is applied to lower the load,such a screw is known as self-locking screw.

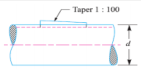
</td></tr>
<tr><td>Q 3 e)</td><td>4</td><td>

Question:

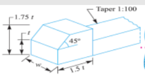
How keys are classified? Give detailed classification of keys with neat sketches; also state their applications.

Answer:

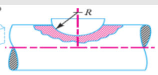
Keys are classified on the basis of shape and application of keys..................are 1) Sunk Key : a) Rectangular sunk key b) square sunk key c) Gib head key d) feather key e) Woodruff key 2) Saddle key a) Flat saddle key b) hollow saddle key 3) Round key 4) Splines ....: 1) Sunk Key : used for heavy duty application a) Rectangular sunk key : for preventing rotation of gears and pulleys on shaft b) Gib headed key: used where key to be removed frequently. c) feather key: Machine tool 2) Saddle key: for light duty or low power transmission 3) Round key : Used for low Power drive 4) Splines: Where providing axial movement between shaft and mounted member



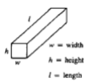
Sunk Key



Gib headed



woodruff key



(a) Flat key

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<tr><td>Q 4b)(i)</td><td>6</td><td>

Question:

State the different modes of failure of gear teeth and their possible remedies to avoid the failure.

Answer:

1. Bending failure. 2. Pitting. 3. Scoring. 4. Abrasive wear. 5. Corrosive wear Remedies to avoid failure: 1. Bending failure. In order to avoid such failure, the module and face width of the gear is adjusted so that the beam strength is greater than the dynamic load. 2. Pitting. In order to avoid the pitting, the dynamic load between the gear tooth should be less than the wear strength of the gear tooth. 3. Scoring. This type of failure can be avoided by properly designing the parameters such as speed, pressure and proper flow of the lubricant, so that the temperature at the rubbing faces is within the permissible limits. 4. Abrasive wear. This type of failure can be avoided by providing filters for the lubricating oil or by using high viscosity lubricant oil which enables the formation of thicker oil film and hence permits easy passage of such particles without damaging the gear surface. 5. Corrosive wear.. In order to avoid this type of wear, proper anti-corrosive additives should be used.

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