# CBCS SCHEME



USN | | | | | | | | | | |

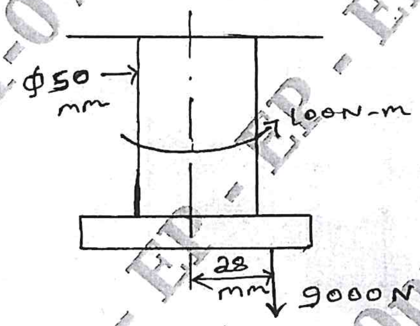
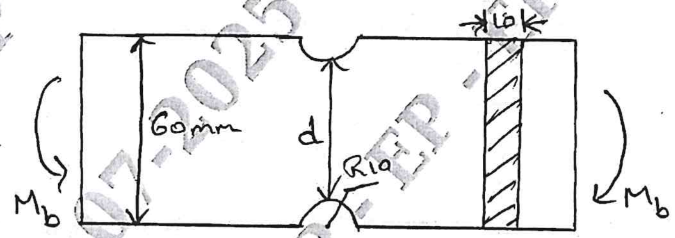
**BME602**

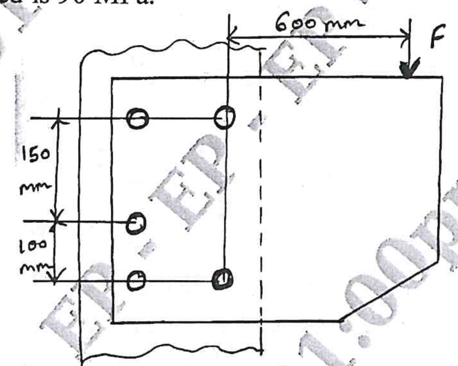
**Sixth Semester B.E./B.Tech. Degree Examination, June/July 2025**

## Machine Design

Time: 3 hrs. Max. Marks: 100

*Note: 1. Answer any FIVE full questions, choosing ONE full question from each module.*
*2. Use of Design data hand book is permitted.*
*3. Missing data can be assumed.*
*3. M : Marks , L: Bloom's level , C: Course outcomes.*

| | | Module – 1 | M | L | C |
|---|---|---|---|---|---|
| Q.1 | a. | List and explain theories of material failure. | 10 | L2 | CO2 |
| | b. | A 50 mm diameter steel rod supports 9 KN load and in addition it is subjected to a torsion moment of 100 N-m as shown in Fig. Q1 (b). Identify maximum tensile and maximum shear stress.   Fig. Q1 (b) | 10 | L2 | CO2 |
| | | **OR** | | | |
| Q.2 | a. | Derive Soderberg's and Goodman equation for designing member subjected to fatigue loading. | 10 | L2 | CO2 |
| | b. | A notched flat plate shown in Fig. Q2 (b) is subjected to bending moment of 10 N-m. Identify the maximum stress induced in the member by taking the stress concentration into account.   Fig. Q2 (b) | 10 | L3 | CO1 |
| | | **Module – 2** | | | |
| Q.3 | | Design the shaft of armature of a motor. If the magnetic poll on the shaft is equivalent to a uniformly distributed load of 10 N per mm length over the middle one third of 600 mm length of shaft between bearings. The motor transmits a power of 15 kW @ 1200 rpm. The allowable shear stress is 50 MPa. Take $C_m = 1.5$ and $C_t = 1.25$. | 20 | L3 | CO3 |

| | | | | | |
|---|---|---|---|---|---|
| | | **OR** | | | |
| **Q.4** | **a.** | Show that the squeeze key is equally strong in shear and compression. | **4** | **L4** | **CO3** |
| | **b.** | A rectangular C/S key 8*7*36 is used to transmit 6 kW @ 1200 rpm. The shaft diameter is 30 mm. If the allowable shear and crushing stress for key material are 60 MPa and 135 MPa respectively and find whether key is safe or not. | **6** | **L4** | **CO3** |
| | **c.** | Design a rigid flange coupling to transmit 18 kW @ 1440 rpm. The allowable shear stress for CI flange is 4 MPa. The shafts, keys and bolts are made of annealed steel having allowable shear stress of 93 MPa. Allowable crushing stress for key is 186 MPa. Assume key way factor as 0.75. | **10** | **L4** | **CO3** |
| | | **Module – 3** | | | |
| **Q.5** | **a.** | A double riveted butt joint with two cover plates for the longitudinal seam of a boiler shell 1.5 m in diameter subjected to a pressure of 0.95 MPa. Assume an efficiency of 75%, allowable tensile strength in the plate 90 MPa, allowable crushing stress 140 MPa and allowable shear stress 56 MPa. Design and interpret efficiency of the joint. | **10** | **L4** | **CO3** |
| | **b.** | Calculate the safe load that can be applied to an eccentrically loaded riveted bracket as shown in Fig. Q5 (b). The allowable shear stress for 25 mm diameter rivets used is 90 MPa.   Fig. Q5 (b) | **10** | **L3** | **CO3** |
| | | **OR** | | | |
| **Q.6** | | Investigate the design requirements for a pair of spur gears to transmit a power of 18 kW from a shaft running @ 1000 rpm to a parallel shaft running @ 250 rpm, maintaining a centre distance of 160 mm between the shaft centres. Suggest suitable surface hardness for the gear pair. | **20** | **L4** | **CO4** |
| | | **Module – 4** | | | |
| **Q.7** | | Analyze the requirements for designing a pair of helical gears to transmit power of 15 kW @ 3200 rpm with a speed ratio of 4 : 1. Given that the pinion is made of cast steel with 0.4 % carbon content untreated and gear is made of high grade cast iron, with a helix angle of 26° and a minimum of 20 teeth on each gear. Suggest suitable surface hardness for gear and pinion. | **20** | **L4** | **CO4** |
| | | **OR** | | | |
| **Q.8** | | A pair of 20° FDI gear are to be designed to connect two shafts @ right angles having a velocity ratio 4:1, the gear is made of cast steel 0.2% untreated and the pinion is made up of C30 steel heat treated, the pinion has 20 teeth and transmit power of 40 kW @ 720 rpm. Design the bevel gears completely. | **20** | **L4** | **CO4** |

| Module – 5 | | | | | |
|---|---|---|---|---|---|
| Q.9 | a. | Derive Petrofit equation for Journal bearing. | 10 | L3 | CO5 |
| | b. | A simple band brake of drum diameter 600 mm has a band passing over it with an angle of contact of 225°, while one end is connected to fulcrum, the other end is connected to the brake lever at a distance of 400 mm from the fulcrum. The brake lever is 1 m long. The brake is to absorb a power of 15 kW @ 720 rpm. Design the brake lever of rectangular C/S, assuming depth to be thrice the width. Take allowable stress 80 MPa. | 10 | L4 | CO3 |
| OR | | | | | |
| Q.10 | a. | Write a short note on Hydrodynamic theory of lubrication, showing pressure distribution and a graph of friction (Vs) speed. | 10 | L3 | CO5 |
| | b. | A 75 mm long full journal bearing of diameter 75 mm supports a radial load of 12 kN at the shaft speed of 1800 rpm. Assume the ratio of diameter to the radial clearance as 1000. The viscosity of oil 0.01 PaS at the operating temperature. Determine the following :<br>(i) Sommerfeld number.<br>(ii) The coefficient of friction based on Mckee's equation.<br>(iii) Amount of heat generated. | 10 | L4 | CO5 |

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# June/July 2025
## Machine Design BME602

1) a) Theories of Material Failure

i) Maximum normal stress theory or Rankine's theory
ii) Maximum shear stress theory or Guest's theory
iii) Maximum distortion energy theory or Hencky-Mises theory
iv) Maximum principal or Normal strain theory or Saint Venant's theory
v) Maximum strain energy theory or Haigh's theory
vi) Mohr's theory.

1.) Maximum normal stress theory or Rankine's theory

According to this theory, failure of a component takes place if the maximum principal stress at any point exceeds the ultimate or yield point stress. If the principal stresses acting in the three directions are $\sigma_1$, $\sigma_2$ and $\sigma_3$ and if $\sigma_1 > \sigma_2 > \sigma_3$, then design equ is

$$\sigma_1 \leq \frac{\sigma_y}{n} \quad \text{where } n = \text{FOS and } \sigma_y = \text{Yield point stress}$$

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_1 = \frac{\sigma_y}{n} \quad \text{where } \frac{\sigma_y}{n} = \sigma_e = \text{Allowable stress}$$

$$\therefore \sigma_e = \frac{1}{2}\left[(\sigma_x + \sigma_y) + \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$

a) If the external static loads produce direct tensile or compressive stresses $\sigma_x$, $\sigma_y$ and $\sigma_z$ acting normal to one another and shear stresses are zero, then the normal stresses itself are the principal stresses.

b) If the external loads produce both direct and shear stress, then principal stresses can be determined. If the numerical maximum principal stress is positive, The design equ is

$$\sigma_{max} \leq \frac{\sigma_{yt}}{n} \quad \text{where } n = \text{FOS}$$

c) This theory is based on failure in tension or compression and ignores shearing. Therefore this theory is good for brittle material.

2) Maximum shear stress theory or Guest's theory

According to this theory, failure occurs when the maximum induced shear stress exceeds the shear stress at yield point. Therefore the design equation is

$$\tau_{max} \leq \frac{\tau_y}{n}$$ where $n = FOS$ & $\tau_y$ = shear stress at yield point

$$\tau_{max} = \frac{\tau_y}{FOS}$$

Since shear stress at the yield point is equal to one half the yield stress in tension, the equation may be written as.

$$\tau_{max} = \frac{\sigma_{yt}}{2 \times FOS}$$ where $\frac{\sigma_{yt}}{FOS} = \sigma_e$ = Allowable normal stress

a) for three dimensional stress state $\tau_{max}$ is the largest among the three values of $\frac{\sigma_1 - \sigma_2}{2}$, $\frac{\sigma_2 - \sigma_3}{2}$ and $\frac{\sigma_1 - \sigma_3}{2}$ where $\sigma_1$, $\sigma_2$ and $\sigma_3$ are the principal stresses

b) For two dimensional stress state $\tau_{max}$ is the largest among the three values of $\frac{\sigma_1 - \sigma_2}{2}$, $\frac{\sigma_1}{2}$, $\frac{\sigma_2}{2}$

c) For the two dimensional stress state

1) If the principal stresses $\sigma_1$ and $\sigma_2$ are of opposite sign the maximum shear stress will be equal to $\frac{\sigma_1 - \sigma_2}{2}$

2) If the principal stresses $\sigma_1$ and $\sigma_2$ are of same sign then maximum shear stress will be equal to $\frac{\sigma_1}{2}$ or $\frac{\sigma_2}{2}$.

3) $\tau_{max} = \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}$, if $\sigma_y = 0$ then

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_x)^2 + 4\tau_{xy}^2}$$ in case when $\sigma_1$ and $\sigma_2$ are of opposite sign. This theory is mostly suitable for ductile materials.

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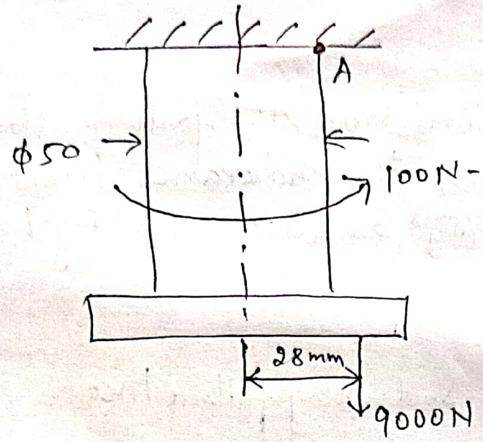
1.) b.)




Solution:- The critical stress is at point A.
The given load 9000N is an eccentric load
Due to this load, the steel rod is subjected to
i) Direct stress ii) Bending stress due to bending moment

$\therefore$ Combined stress at $A = \frac{F}{A} + \frac{M_b}{I} \cdot c$ where $M_b = F \times e$.

$$\sigma = \frac{9000}{\frac{\pi}{4}(50)^2} + \frac{(9000 \times 28)}{\frac{\pi}{64}(50)^4} \cdot \left(\frac{50}{2}\right) = 25.12 \text{ N/mm}^2 \text{ (Tensile)}$$

Shear stress at A due to torque

$$\tau = \frac{M_t}{J} \cdot r = \frac{(100 \times 10^3)}{\frac{\pi}{32}(50)^4} \cdot \left(\frac{50}{2}\right) = 4.074 \text{ N/mm}^2$$

$\therefore$ Max normal stress at A

$$\sigma_{max} = \left(\frac{\sigma}{2}\right) + \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$= \frac{25.12}{2} + \sqrt{\left(\frac{25.12}{2}\right)^2 + (4.074)^2} = 25.764 \text{ N/mm}^2 \text{ (Tensile)}$$

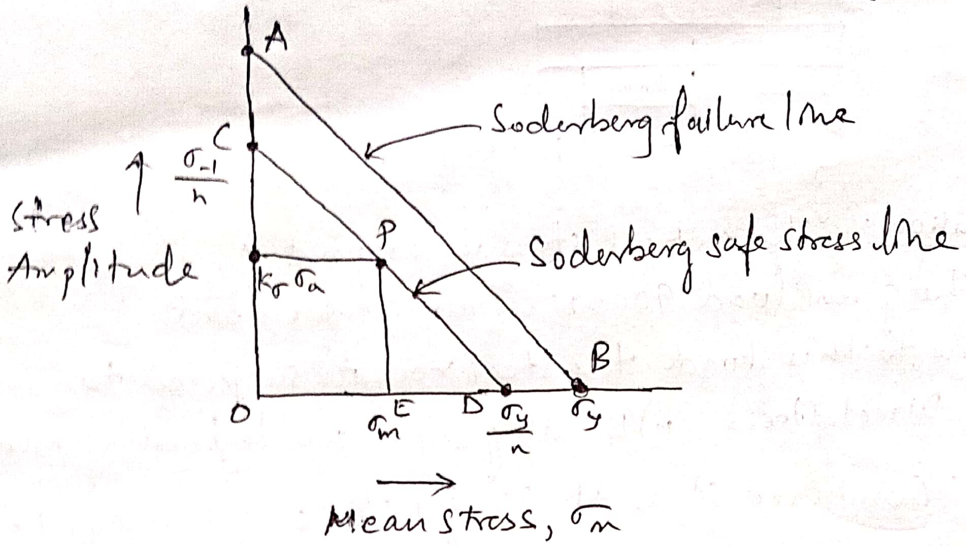
Max shear stress at A

$$\tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$= \sqrt{\left(\frac{25.12}{2}\right)^2 + (4.074)^2} = 13.2 \text{ N/mm}^2$$

2) a) i) Soderberg Criterion

According to Soderberg criterion, the failure line is taken as the straight line joining the Endurance limit ($\sigma_{-1}$) and the yield strength ($\sigma_y$) as shown in Fig below.




Here the effect of strain hardening is neglected.
In fig. the line AB is termed as Soderberg failure line.
Considering suitable factor of safety the line CD can be drawn parallel to the line AB.
This line is termed as Soderberg safe stress line.

Consider a point P on the Soderberg safe stress line as shown in fig.
Let $\sigma_m$ and $\sigma_a$ be the mean and amplitude stress at this point respectively. Now draw PE perpendicular to OB.
From similar triangles PED and COD

$$\frac{PE}{OC} = \frac{ED}{OD}$$

$$= \frac{OD - OE}{OD}$$

$$\therefore \frac{PE}{OC} = 1 - \frac{OE}{OD}$$

$$\therefore \frac{\sigma_a}{\frac{\sigma_{-1}}{n}} = 1 - \frac{\sigma_m}{\frac{\sigma_y}{n}}$$

Neglecting stress concentration $K_{\sigma_{-1}}$, $PE = \sigma_a$

$$\therefore \frac{n \cdot \sigma_a}{\sigma_{-1}} + \frac{n \cdot \sigma_m}{\sigma_y} = 1$$

$$\therefore \frac{\sigma_a}{\sigma_{-1}} + \frac{\sigma_m}{\sigma_y} = \frac{1}{n} \quad \text{where } n = \text{Factor of safety} \quad \text{—(i)}$$

Considering the stress concentration factor for the reversed component of stress, the equation (i) becomes

$$\frac{K_\sigma \sigma_a}{\sigma_{-1}} + \frac{\sigma_m}{\sigma_y} = \frac{1}{n} \quad \text{—(ii)}$$

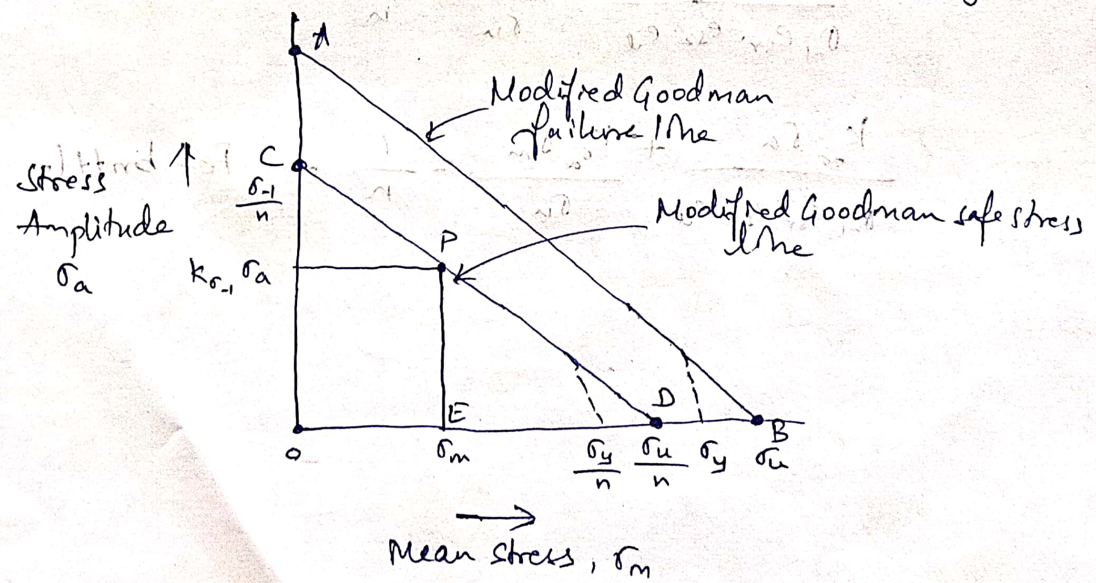
Considering the important three modifying factors for the endurance limit, the equation (ii)

$$\frac{K_{\sigma} \sigma_a}{\sigma_{-1} \cdot e_{sr} \cdot e_{sz} \cdot e_l} + \frac{\sigma_m}{\sigma_y} = \frac{1}{n} \quad \text{—(iii)}$$

Hence Soderberg equation suitable for ductile material.

ii) <u>Goodman Criterion</u>

According to Goodman criterion, the failure line is taken as the straight line joining the endurance limit ($\sigma_{-1}$) and the ultimate strength ($\sigma_u$) as shown in fig.




Consider a point P on the modified Goodman safe stress line as shown and let $\sigma_m$ and $\sigma_a$ be the mean and amplitude stress at this point respectively. Now draw PE perpendicular to OB

From similar triangles PED and COD

$$\frac{PE}{OC} = \frac{ED}{OD} = \frac{OD - OE}{OD} = 1 - \frac{OE}{OD}$$

$$\therefore \frac{\sigma_a}{\frac{\sigma_{-1}}{n}} = 1 - \frac{\sigma_m}{\frac{\sigma_u}{n}}$$ [Neglecting stress concentration $K_{\sigma_{-1}}$. $PE = \sigma_a$]

$$\therefore \frac{n \cdot \sigma_a}{\sigma_{-1}} + \frac{n \cdot \sigma_m}{\sigma_u} = 1$$

$$\therefore \frac{\sigma_a}{\sigma_{-1}} + \frac{\sigma_m}{\sigma_u} = \frac{1}{n}$$ where $n$ = Factor of safety —(i)

For ductile material, stress concentration may be ignored under steady loads, but suitable stress concentration factor must be considered under fatigue load. Therefore for ductile material equation (i) becomes

$$\frac{K_{-\sigma} \cdot \sigma_a}{\sigma_{-1}} + \frac{\sigma_m}{\sigma_u} = \frac{1}{n} \quad \text{—(ii)}$$

For brittle materials

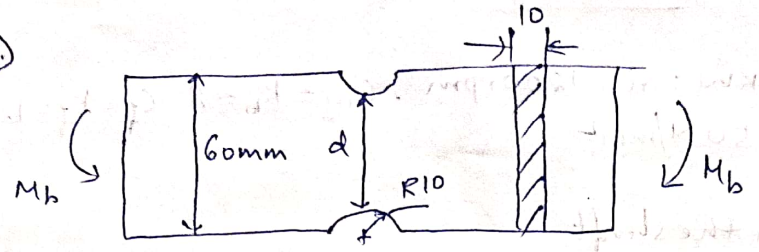
$$\frac{K_{-\sigma} \sigma_a}{\sigma_{-1}} + \frac{K_{\sigma_a} \sigma_m}{\sigma_u} = \frac{1}{n} \quad \text{—(iii)}$$

Considering three modifying factors:

$$\frac{K_{-\sigma} \sigma_a}{\sigma_{-1} \, e_{sr} \cdot e_{s2} \cdot e_d} + \frac{\sigma_m}{\sigma_u} = \frac{1}{n}$$ — for ductile

$$\frac{K_{-\sigma} \sigma_a}{\sigma_{-1} \, e_{sr} \cdot e_{s2} \cdot e_d} + \frac{K_{\sigma_a} \sigma_m}{\sigma_u} = \frac{1}{n}$$ — for brittle

2) b.)




Given :- $M_b = 10\ Nm = 10\times10^3\ N\text{-}mm$

$D = 60mm$ ; $r = 10mm$ ; $h = 10mm$

Soln

$d = D - 2R = 60 - 2\times10 = 40mm$

$r/d = 10/40 = 0.25$, $D/d = 60/40 = 1.5$

Stress Concentration factor $k_\sigma = 1.575$

For bending load

$$\sigma_{nom} = \frac{M_b}{I} \cdot c \qquad I = \frac{hd^3}{12} ; \; c = \frac{d}{2}$$

$$= \frac{10\times10^3}{\frac{10\times40^3}{12}} \cdot \left(\frac{40}{2}\right) = 3.75\ N/mm^2$$

$$K_\sigma = \frac{\sigma_{max}}{\sigma_{nom}}$$

$$1.575 = \frac{\sigma_{max}}{3.75}$$

$\therefore$ Maximum stress induced $\sigma_{max} = 5.906\ N/mm^2$

3) Given: $N = 15\,kW$ ; $n = 1200\,rpm$ . $C_m = k_b = 2$ $C_t = k_t = 1.25$

$\tau_{ed} = 50\,N/mm^2$

Soln Torque on the shaft

$$M_t = 9550 \times 1000 \times \frac{N}{n}$$

$$= 9550 \times 1000 \times \frac{15}{1200}$$

Taking moments about A

$$R_B \times 600 = 10 \times 200 \left[\frac{200}{2} + 200\right]$$

$\therefore R_B = 1000\,N$ ; $R_A = 200 \times 10 - 1000 = 1000\,N$

$(\because R_A + R_B = 10 \times 200 = 2000\,N)$

Bending moment diagram (BMD)

BM at A = 0

BM at C = $1000 \times 200 = 2 \times 10^5$ N-mm

BM at D = $1000 \times 400 - 200 \times 10 \times \frac{200}{2} = 2 \times 10^5$ N-mm

BM at B = 0

Shear Force diagram (SFD)

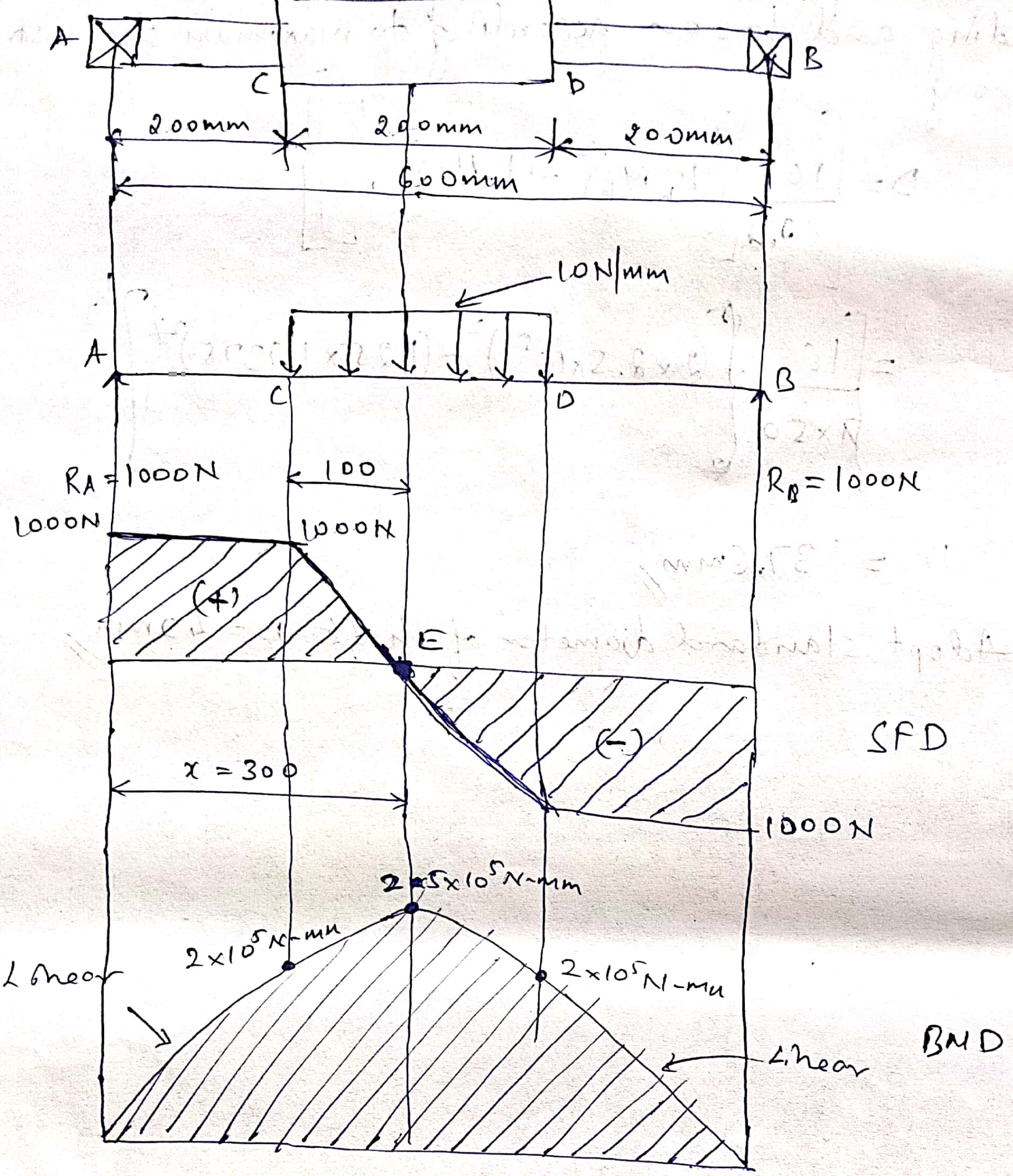
SF at A = +1000N

SF at C = +1000N

SF at D = +1000 − 200×10

= −1000N

Bending Moment Diagram and Shear Force Diagram are as below,




SF at B = −1000N

Maximum bending moment occurs at a point where the SF changes, its sign. Let E be the point at a distance 'x' from A where SF changes its sign

$\therefore$ SF at E $= +1000 - 10(x-200) = 0$

$= 1000 - 10x + 2000 = 0$

$\therefore x = 300\text{mm}$

$\therefore$ BM at E $= 1000\times300 - 10\times100\times\frac{100}{2} = 2.5\times10^5$ N-mm

Maximum torque $M_t = 119375$ N-mm

Maximum bending moment $M_b = 2.5\times10^5$ N-mm

Diameter of solid circular shaft subjected to combined bending and torsion according to maximum shear stress theory

$$D = \left[\frac{16}{\pi \tau_{d}}\left\{(k_b M_b)^2 + (k_t M_t)^2\right\}^{\frac{1}{2}}\right]^{\frac{1}{3}}$$

$$= \left[\frac{16}{\pi \times 50}\left\{(2 \times 2.5 \times 10^5)^2 + (1.25 \times 119375)^2\right\}\right]^{\frac{1}{3}}$$

$D = 37.6mm$

Adopt standard diameter of shaft $D = 40mm$

4) a) A square key (where width w equals thickness t) is equally strong in shear and compression (crushing) when the allowable crushing stress ($\sigma_c$) of the material is twice its allowable shear stress ($\tau$). This is shown by equating the torque-based shear and crushing strength formulas which simplifies $w = t$

A key fails due to either shearing or crushing (compression) based on the torque $T$ transmitted by the shaft diameter 'd'.

1. Shear Failure

- Shear Force ($F_s$) = Shear Area x Shear Stress

$$= (l \times w) \times \tau$$

- Torque ($T_s$) $= F_s \times \frac{d}{2} = l.w.\tau.\frac{d}{2}$

2. Crushing Failure

- Crushing force ($F_c$) = Crushing Area x Crushing Stress

$$= (l \times \tfrac{t}{2}) \times \sigma_c$$

- Torque ($T_c$) $= F_c \times \frac{d}{2} = l.\frac{t}{2}.\sigma_c.\frac{d}{2}$

3. Equating strengths ($T_s = T_c$)

$$l.w.\tau.\frac{d}{2} = l.\frac{t}{2}.\sigma_c.\frac{d}{2}$$

$$w.\tau = \frac{t}{2}.\sigma_c$$

4. Condition for equal strength

For standard key materials, the allowable crushing stress is roughly double the shear stress

$$\sigma_c = 2\tau$$

Substituting $\sigma_c = 2\tau$

$$w.\tau = \frac{t}{2}.(2\tau)$$

$$w.\cancel{\tau} = t.\cancel{\tau}$$

$$w = t$$

$\therefore$ when width ($w$) equals ($t$), the square key is equally strong in shear and compression.

4.) b.) The torque is

$$T = \frac{P \times 60}{2\pi N} = \frac{6000 \times 60}{2\pi \times 1200} = 47.75 \text{ N-m}$$

$$= 47746.48 \text{ N-mm}$$

The tangential force F acting at the shaft surface radius $r = \frac{d}{2} = 15\text{mm}$

$$F = \frac{T}{r} = \frac{47746.48}{15} = 3183.10\text{N}$$

Induced stresses based on the key dimensions:
$b = 8\text{mm}$, $h = 7\text{mm}$, $L = 36\text{mm}$

Shear stress ($\tau$):

$$\tau_{ind} = \frac{F}{b \times L} = \frac{3183.10}{8 \times 36} \approx 11.05 \text{ MPa}$$

Crushing stress ($\sigma_c$)

$$\sigma_{c,ind} = \frac{F}{\frac{h}{2} \times L} = \frac{3183.10}{3.5 \times 36} = 25.26 \text{ MPa}$$

The induced shear stress is 11.05MPa is less than the allowable 60MPa and the induced crushing stress 25.26 MPa is less than the allowable 135MPa. Therefore key is safe.

4) c) Given:-

$P = 18\,kW,\; n = 1440\,rpm\quad \tau_{all\,flange} = 4\,MPa$

$\tau_{all\,shaft} = \tau_{all\,key} = \tau_{all\,bolt} = 93\,MPa$

$\sigma_{c\,key} = 186\,MPa;\; \eta = 0.75$

i) Torque transmitted ($M_t$)

$$M_t = 9550\times10^3\times\left(\frac{N}{n}\right) = 9550\times10^3\times\frac{18}{1440}$$

$$M_t = 119375\,N\text{-}mm$$

ii) Diameter of shaft

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

$$119375 = \frac{\pi}{16}\times d^3 \times 0.75\times 0.93$$

$$D = 20.58 \simeq 22\,mm$$

iii) Design of bolt

No. of bolts $= i = 0.02D + 3$

$i = 0.02\times 22 + 3 = 3.44$

$\therefore i = 4$

Dia of bolt $= d = \dfrac{0.423D}{\sqrt{i}} + 7.5\,mm$

$$d = \frac{0.423\times 22}{\sqrt{4}} + 7.5$$

$$d = 12\,mm$$

Hub diameter $= D_1 = 1.8D + 20\,mm$

$D_1 = 1.8\times 22 + 20 = 59.6\,mm$

Bolt circle dia $= D_2 = D_1 + 3.2d$

$= 59.6 + 3.2\times 12$

$D_2 = 98\,mm$

Outside dia of flange $= D_3 = D_1 + 6d$

$D_3 = 59.6 + 6\times 12$

$D_3 = 131.6\,mm$

Hub length $l = 1.2D + 20\,mm$

$l = 1.2\times 22 + 20 = 46.4\,mm$

Check for stress

Torque capacity based on shear of bolts

$$T = \frac{\tau \pi . i d^2 D_2}{8}$$

$$119375 = \frac{\tau \times \pi \times 4 \times 12^2 \times 98}{8}$$

$$\tau = 5.38 < 93 MPa$$

∴ Safe

Torque capacity based on shear of flange

$$T = \frac{\tau \pi D_1^2 t}{2}$$

$$119375 \times 2 = \tau \times \pi \times 59.6^2 \times 16.7$$

$$\tau = 1.28 MPa < 4 MPa$$

∴ Safe.

Design of key

From table No 4.1 DHB - Mahadevan & Balveereddy

for $d = 22mm$

width $= b = 6mm$

Height $= h = 6mm$

Length $= l = 42mm$

width of key $= b = \frac{2M_t}{\tau d_2 \times l \times D}$

$$\tau d_2 = \frac{2M_t}{b \times l \times D} = \frac{2 \times 119375}{6 \times 42 \times 22}$$

$$\tau d_2 = 43.06 < 93 MPa$$

∴ Safe.

Thickness of key $= h = \frac{4M_t}{\sigma_b l \times D}$

$$\sigma_b = \frac{4M_t}{h l D} = \frac{4 \times 119375}{6 \times 42 \times 22}$$

$$\sigma_b = 86.13 < 186 MPa$$

∴ Safe.

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# Module 3

5a) Given:

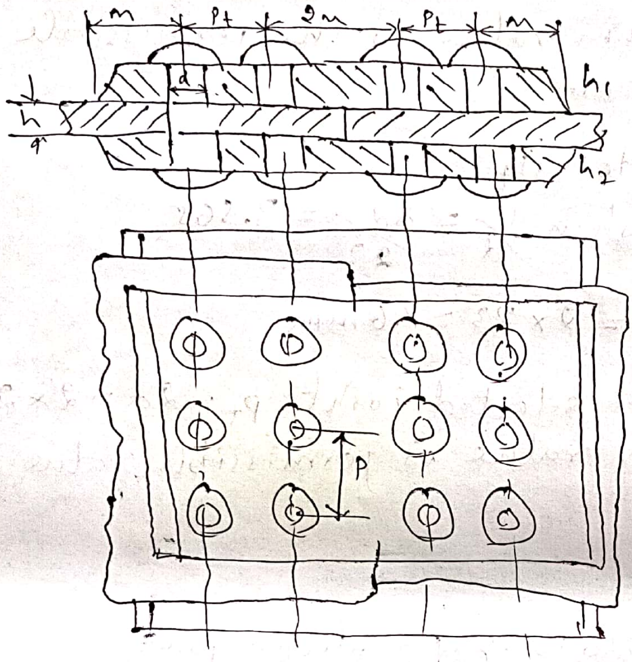
$D_i = 1.5m = 1500mm$, $P_f = 0.95 N/mm^2$, $\eta = 75\%$.

$\sigma_\theta = 90 N/mm^2$, $\sigma_c = 140 N/mm^2$, $\tau = 56 N/mm^2$

Solution

1. Type of joint

Assume equal width cover plates with chain riveting.




No. of rivets in single shear per pitch length $i_1 = 0$

No. of rivets in double shear per pitch length $i_2 = 2$

2. Thickness of Main plate

$$h = \frac{P_f D_i}{2\eta\sigma_\theta} + c \qquad c \to \text{clearance for corrosion} = 1mm$$

$$= \frac{0.95 \times 1500}{2 \times 0.75 \times 0.90} + 1 = 11.55mm \simeq 12mm$$

3. Diameter of rivet and rivet hole

Since > 8mm using Unwin's formula

$$d = 6\sqrt{h} \text{ to } 6.3\sqrt{h}$$

$$= 6\sqrt{12} \text{ to } 6.3\sqrt{12}$$

$$= 20.78 \text{ to } 21.82 \text{ mm}$$

Select standard diameter of rivet = 22mm

Standard diameter of rivet hole = 23 mm

4. Pitch(P)

$$p = \frac{(1.875 i_2 + i_1)\pi d^2 \tau}{4 h \sigma_0} + d$$

$$= \frac{(1.875\times2+0)\pi\times23^2\times56}{4\times12\times90} + 23 = 103.787\,mm$$

From table 13.14

$p = 3.5h + 40 = 3.5\times12+40 = 82\,mm$ //

Select minimum value as the permissible value

$p = 82\,mm$

5. Transverse Pitch ($P_t$)

Table 13.11 when $P/d = 82/23 = 3.565$

$P_t = 2d = 2\times23 = 46\,mm$

Table 13.14 for selected joint $P_t = 2d = 2\times23 = 46\,mm$

Select greater value as permissible value

6. Margin

From Table No 13.14 for selected joint

Margin $m = 1.5d = 1.5\times23 = 34.5\,mm$

7. Thickness of coverplates

For equal width cover plates from Tab No 13.14

$h_1 = h_2 = 0.625h = 0.625h = 0.625\times12 = 7.5\,mm$

8. Length of shank

For double cover butt joint,

Length of shank $l = h_1 + h + h_2 + 1.5d$ to $1.7d$

$l = 7.5+12+7.5+1.5\times23$

$l = 61.5\,mm$

Preferred Length $l = 63\,mm$ from Table 13.13 DHB

9. Width of cover plates

$w = 4m + 2p_t = 4\times34.5 + 2\times46 = 230\,mm$ //

10. Strength of solid plate

$F_0 = p h \sigma_0 = 82\times12\times90 = 88560\,N$ //

11. Least Strength

i) Strength of perforated plate

$$(p-d)h\sigma_t = (82-23)12\times 90$$
$$= 63720\,N$$

ii) Shear strength of rivets per pitch length

$$F_T = (1.875 i_2 + i_1)\frac{\pi}{4}d^2\tau$$
$$= (1.875\times 2 + 0)\frac{\pi}{4}\times 23^2\times 56$$
$$= 87249.88\,N$$

iii) Crushing strength of rivets in one pitch length

$$F_c = (i_2 h + i_1 h_2) d\sigma_c$$
$$= (2\times 12 + 0)(23)\times 140 = 77280\,N$$

12. Efficiency

$$\eta = \frac{\text{Least strength}}{\text{Strength of solid plate}} = \frac{63720}{88560} \approx 71.95\%$$

5) b) Given: eccentricity $e$ = 600mm.

Maximum load at bracket end, $P$ = ?

Maximum allowable shear stress for rivet, $\tau_{max}$ = 90MPa

Maximum allowable crushing stress, $\sigma_{c\,max}$ = 180 MPa

$P$ = ?

Rivet arrangement

- 2 rows × 2 columns = 4 rivets. ≈ 4 rivets.

Vertical spacing: 150mm

Horizontal spacing: 100mm

1) Area of one rivet

$$A = \frac{\pi}{4}d^2 = \frac{\pi}{4}(25)^2 = 490.87\,mm^2$$

Allowable shear force per rivet

$$F_{allow} = \tau_{allow}\times A$$
$$= 90\times 490.87 = 44178\,N = 44.18\,kN$$

Scanned with OKEN Scanner

2. Direct shear load per rivet

Total load F is shared equally

$$F_d = F/4$$

3. Distance from centroid

Distance of each rivet from centroid

$$r = \sqrt{\left(\frac{100}{2}\right)^2 + \left(\frac{150}{2}\right)^2}$$

$$r = \sqrt{(50)^2 + (75)^2} = 90.14\,mm$$

$$\Sigma r^2 = 4 \times (90.14)^2 = 4 \times 8125 = 32500$$

4. Shear due to moment

Moment

$$M = F \times 600$$
$$= 600\,F$$

Secondary shear on each rivet

$$F_m = \frac{Mr}{\Sigma r^2} = \frac{F \times 600 \times 90.14}{32500}$$

$$F_m = 1.664\,F$$

5. Resultant shear on critical rivet

Since direct shear and moment shear are perpendicular.

$$F_r = \sqrt{\left(\frac{F}{4}\right)^2 + (1.664F)^2}$$

$$= F\sqrt{0.0625 + 2.768}$$
$$= F\sqrt{2.83}$$

$$F_r = 1.683\,F$$

6. Equate to allowable shear

$$1.683F = 44178$$

$$F = \frac{44178}{1.683}$$

$$\therefore F = 26250N$$

$$F = 26.3 \, KN$$

6.) 1. Calculate the speed ratio (i)

$$i = \frac{N_1}{N_2} = \frac{1000}{250} = 4$$

2. Determine the no. of teeth on pinion and gear

Let the number of teeth on pinion be $Z_1$ and gear be $Z_2$

Pressure angle $\alpha = 20°$ Full depth involute system (assume)

Assume Class - III precision gears

Assume medium shock and 8 to 10 hours duty per day.

Service factor $C_s = 1.5$ Table 23.13 DHB

Pinion material SAE 3245

$\sigma_{o_1} = 448 - 517 MPa$

$\sigma_{o_1} = 500 MPa$

$$i = \frac{n_1}{n_2} = \frac{1000}{250} = 4 = \frac{Z_2}{Z_1} = \frac{d_2}{d_1}$$

$$a = \frac{d_1 + d_2}{2} = \frac{d_1 + id_1}{2} = \frac{d_1(1+i)}{2}$$

$$160 = \frac{(1+4)d_1}{2}$$

$\therefore d_1 = 64mm \quad d_2 = 256mm$

Lewis form factor of 20° FD involute system

$$y = 0.154 - \frac{0.912}{z}$$

To identify the weaker member temporarily assume $z_1 = 20$

$$z_2 = i z_1 = 4 \times 20 = 80$$

$$y_1 = 0.154 - \frac{0.912}{z_1} = 0.154 - \frac{0.912}{20} = 0.1084$$

$$y_2 = 0.154 - \frac{0.912}{z_2} = 0.154 - \frac{0.912}{80} = 0.1426$$

To select the gear material $\sigma_{o1} y_1$ to $\sigma_{o2} y_2$

$$500 \times 0.1084 = \sigma_{o2} \times 0.1426$$

$$\sigma_{o2} = 380.08 \ N/mm^2$$

From table 23.18, select the gear material such that its value of $\sigma_{o2}$ must be nearer to $380.08 \ N/mm^2$

Hence select SAE4640, Hardened by OQT as gear material

$$\sigma_{o2} = 379 \ N/mm^2$$

3.) Identify the weaker member

| Particulars | $\sigma_o$ $N/mm^2$ | $y$ | $\sigma_o y$ | Remarks |
|---|---|---|---|---|
| Pinion | 500 | 0.1084 | 54.21 | |
| Gear | 379 | 0.1426 | 54.045 | weaker. |

As $\sigma_{o2} y_2 < \sigma_{o1} y_1$ gear is the weaker member.

Therefore design should be based on gear

4.) Design

a) Tangential load

$$F_t = \frac{9550 \times 1000 \times N C_s}{n_r}$$

$$= \frac{9550 \times 1000 \times P C_s}{n_r}$$

Tangential tooth load of weaker member

$$F_{t_2} = \frac{9550 \times 1000 \times N C_s}{n_2 r_2} = \frac{9550 \times 1000 \times P C_s}{n_2 r_2}$$

Assume medium shock and 8-10 hrs duty per day
From table 8.33 Service factor $C_s = 1.5$
Pitch circle radius of gear $r_2 = \frac{d_2}{2} = \frac{256}{2} = 128mm$

$$F_{t_2} = \frac{9550 \times 1000 \times 18 \times 1.5}{250 \times 128} = 8057.81 N$$

b) Tangential tooth load from Lewis equation

$$F_t = \sigma_o b y p k_v$$

Tangential tooth load of weaker member

$$F_{t_2} = \sigma_{o_2} b y_2 p C_v$$

Face width $b = 3\pi m$ to $4\pi m$ or $9.5m < b < 12.5m$
Select $b = 10m$ ∴ Circular pitch $p = \pi m$

$$y_2 = 0.154 - \frac{0.912}{z_2} = 0.154 - \frac{0.912}{\frac{d_2}{m}} = 0.154 - \frac{0.912m}{256}$$

$$= 0.154 - 3.5625 \times 10^{-3} m$$

Mean pitch line velocity of weaker member

$$v_m = \frac{\pi d_2 n_2}{60,000} = \frac{\pi \times 256 \times 250}{60000} = 3.351 m/s$$

velocity factor $C_v = \frac{3}{3 + v_m} = \frac{3}{3 + 3.351} = 0.4724$
$(\because v_m < 7.5 m/s)$

Substituting all the values in Lewis equation

$$8057.81 = (379)(10)(m)(0.154 - 3.5625 \times 10^{-3} m)\pi m (0.4724)$$

$$1.4326 = 0.154 m^2 - 3.5625 \times 10^{-3} m^3$$

$$\therefore \quad m^2 = 0.02313 m^3 \geq 9.3$$

$$m^2 - 0.02313 m^3 \geq 9.3$$

Trial 1:

Select $m = 3mm$

$8.3755 < 9.3$

Not suitable

Trial 2:

Select $m = 4mm$

$14.52 > 9.3$ Hence suitable

$\therefore$ Module $m = 4mm$

c) Check for stress

Allowance stress $\sigma_{all} = (\sigma_{o2} C_v)_{all} = 379 \times 0.4724$

$= 179.03 \ N/mm^2$

Induced stress $\sigma_{ind} = (\sigma_{o2} k_v) = \frac{F_{tL}}{b y_2 P}$

$= \frac{8057.81}{(10\times4)(0.154 - 3.5625\times10^{-3}\times4)(\pi\times4)}$

$= 114.70 \ N/mm^2$

Since $(\sigma_{o2} C_v)_{ind} < (\sigma_{o2} C_v)_{allow}$

Design is safe

$\therefore$ Module $m = 4mm$

5) Dimensions

module $m = 4mm$

Facewidth $b = 10m = 10\times4 = 40mm$

No. of teeth on pinion $z_1 = \frac{d_1}{m} = \frac{64}{4} = 16$

No. of teeth on gear $z_2 = \frac{d_2}{m} = \frac{256}{4} = 64$

Centre distance $a = 160mm$

Addendum $h_a = 1m = 1\times4 = 4mm$

Dedendum $h_d = 1.25m = 1.25\times4 = 5mm$

working depth $h' = 2m = 2\times4 = 8mm$

Total depth $h = 2.25m = 2.25\times4 = 9mm$

Tooth thickness $s = \frac{\pi m}{2} = \frac{\pi \times 4}{2} = 6.2832$ mm

6.) Checking

a) Dynamic load

$$F_d = F_t + \frac{21 v_m (F_t + bc)}{21 v_m + \sqrt{F_t + bc}}$$

Tangential tooth load $F_t = F_{t_2} = 8057.81$ N

Mean pitch line velocity $V_m = 3.351$ m/s

Face width $b = 40$ mm

Dynamic load factor

$C = 145$ kN/m $= 145$ N/mm

$$\therefore F_d = 8057.1 + \frac{21 \times 3.351 (8057.81 + 40 \times 145)}{21 \times 3.351 + \sqrt{8057.81 + 40 \times 145}}$$

$= 13242.5$ N

b) Wear load

$F_w = d_1 b Q K$

Ratio Factor $Q = \frac{2 z_2}{z_1 + z_2} = \frac{2 \times 64}{16 + 64} = 1.6$

For safer design. $F_w \geq F_d$

$d_1 b Q K \geq F_d$

$64 \times 40 \times 1.6 \times K \geq 13242.5$

$K \geq 3.233$ N/mm$^2$

$\therefore$ Surface hardness for pinion $= 450$ BHN

Surface hardness for gear $= 450$ BHN

7) Given: $P = N = 15\,kW$, $n_1 = 3200\,rpm$, $\beta = 26°$

$z_1 = 20$ $i = 4$

Pinion material – 0.4% C untreated; Gear – High grade CI

velocity ratio $i = \frac{n_1}{n_2} = \frac{z_2}{z_1} = \frac{d_2}{d_1}$

Speed of gear $n_2 = \frac{n_1}{i} = \frac{3200}{4} = 800\,rpm$

No. of teeth on gear $z_2 = iz_1 = 4 \times 20 = 80$

Allowable stress for 0.4% C untreated

$\sigma_{o1} = 69.6\,MPa = 69.6\,N/mm^2$

High grade CI, $\sigma_{o2} = 31\,N/mm^2$

Formative no. of teeth

$$z_v = \frac{z}{\cos^3\beta}$$

∴ formative no. of teeth on pinion

$$z_{1v} = \frac{z_1}{\cos^3\beta} = \frac{20}{\cos^3 26} = 27.545$$

∴ Formative no. of teeth on gear

$$z_{2v} = \frac{z_2}{\cos^3\beta} = \frac{80}{\cos^3 26} = 110.18$$

Assume pressure angle in the normal plane

$\alpha_n = 20°$ Full depth

Lewis form factor for 20° full depth

$$y = 0.154 - \frac{0.912}{z_v}$$

For pinion $y_1 = 0.154 - \frac{0.912}{z_{1v}} = 0.154 - \frac{0.912}{27.54} = 0.1208$

For gear $y_2 = 0.154 - \frac{0.912}{z_{2v}} = 0.154 - \frac{0.912}{110.18} = 0.1457$

i) Identify the weaker member

| Particulars | $\sigma_o$ $N/mm^2$ | $y$ | $\sigma_o y$ | Remarks |
|---|---|---|---|---|
| Pinion | 69.6 | 0.1208 | 8.407 | |
| Gear | 31 | 0.1457 | 4.516 | weaker |

As $\sigma_{o2} y_2 < \sigma_{o1} y_1$, gear is weaker. Therefore design should be based on gear

ii) Design

a) Tangential tooth load

$$F_t = \frac{9550 \times 1000 N C_s}{n r}$$

∴ Tangential tooth load of the weaker member gear

$$F_{t_2} = \frac{9550 \times 1000 N C_s}{n_2 r_2} = \frac{9550 \times 1000 N C_s}{n_2 r_2}$$

Pitch circle radius of gear

$$r_2 = \frac{d_2}{2} = \frac{m_n z_2}{2 \cos\beta} = \frac{m_n \times 80}{2 \cos 26} = 44.5 m_n$$

Assume medium shock and 8-10 hours duty per day

From Table, service factor $C_s = 1.5$

$$F_{t_2} = \frac{9550 \times 1000 \times 15 \times 1.5}{800 \times 44.5 m_n} = \frac{6035.8}{m_n} \quad \text{--- (i)}$$

b) Lewis equation for tangential tooth load

$$F_t = \frac{\sigma_o b y p_t C_v \cos\beta}{C_w} = \frac{\sigma_o b y p_n C_v}{C_w} = \frac{\sigma_o b y p_n k_v}{C_w} \quad \left(\because p_t = \frac{p_n}{\cos\beta}\right)$$

Since gear is the weaker member

$$F_{t_2} = \frac{\sigma_{o2} b y_2 p_n C_v}{C_w}$$

Assume scant lubrication but frequent inspection

∴ From table

$$\frac{p_t}{\tan\beta} < b < \frac{20 m_t}{\tan\beta}$$

i.e. $\frac{\pi m_n}{\sin\beta} < b < \frac{20 m_n}{\sin\beta}$

$\therefore$ Select $b = 10 m_n$

$$F_{t_2} = \frac{(31)(10m_n)(0.1457)(\pi m_n) k_v}{1.25} = 113.517 m_n^2 k_v \quad -(ii)$$

Equating equ (i) & (ii).

$$113.517 m_n^2 K_v = \frac{6035.8}{m_n}$$

$$m_n^3 k_v = 53.171$$

Mean pitch line velocity of weaker member

$$V_m = \frac{\pi d_2 n_2}{60000} = \frac{\pi m_n Z_2 n_2}{60000 \cos\beta} = \frac{\pi m_n 20 \times 800}{60000 \times \cos 26} = 3.7284 m_n \text{ m/s}$$

Trial 1

Select module $m_n = 5$mm (Select standard module from Table No 2.3

$\therefore V_m = 3.7284 \times 5 = 18.642$ m/s

velocity factor $k_v = \frac{6}{6+V_m}$ $(\because V_m < 20$ m/sec$)$

$$= \frac{6}{6+18.642} = 0.2435$$

Now, from equation (iii)

$$(5)^3(0.2435) > 53.171$$

$$30.4375 < 53.17$$

$\therefore$ Not suitable

Trial: 2

Select module $m_n = 6$mm

$$V_m = 3.7284 \times 6 = 22.3704 \text{ m/sec}$$

velocity factor $k_v = C_v = \frac{5.6}{5.6+\sqrt{V_m}}$ since $V_m > 20$ m/sec

$$= \frac{5.6}{5.6+\sqrt{22.37}} = 0.542 \,//$$

From equ (iii)

$$(6)^3(0.542) > 53.71$$

$$117.1 > 53.71$$

Hence suitable
∴ Normal module $m_n = 6mm$

c) Check for the stress

Allowable stress $\sigma_{all} = (\sigma_{o2} k_v)_{all} = 31 \times 0.54 = 16.74 N/mm^2$

Induced stress $\sigma_{ind} = (\sigma_{o2} k_v)_{ind} = \frac{F_{t2} C_v}{b y_2 p_n}$ -- Eqn 2.286

$$= \frac{\left(\frac{6035.8}{6}\right) \times 1.25}{(10 \times 6)(0.1457)(\pi \times 6)} = 7.63 N/mm^2$$

Since $(\sigma_{o2} k_v)_{ind} < (\sigma_{o2} k_v)_{all}$ the design is safe.
Also in order to avoid the breakage of gear tooth due to bending, the beam strength should be more than the effective force between the meshing teeth.

Effective force $F_{eff} = \frac{F_{t2} C_s}{k_v} = \frac{F_{t2}}{k_v}$ since $C_s$ is already considered

$$= \frac{(6035.8/6)}{0.542} = 1856.032 N$$

Beam strength of weaker member

$$F_{b2} = \frac{\sigma_{o2} b y_2 p_n}{C_w} = \frac{(31)(10 \times 6)(0.1457)(\pi \times 6)}{1.25} = 4086.6 N$$

$F_{b2} > F_{eff}$

$$\therefore FOS = \frac{F_{b2}}{F_{eff}} = \frac{4086.6}{1856.032} = 2.2$$

The design is satisfactory and hence the module in normal plane should be equal to 6mm
i.e. Normal module $m_n = 6mm$

iii) Dimensions

Module in normal plane $m_n = 6mm$

Module in diametral plane $m_t = \frac{m_n}{\cos\beta} = \frac{6}{\cos 26} = 6.6756mm$

Face width $b = 10m_n = 10 \times 6 = 60mm$

$$b_{min} = \frac{\pi m_n}{\sin\beta} = \frac{\pi \times 6}{\sin 26} = 43mm$$

Since $b > 43mm$, safe

Normal pitch $P_n = \pi m_n = \pi \times 6 = 18.85mm$

Axial pitch $P_t = \pi m_t = \pi \times 6.6756 = 20.972\ mm$

Pitch circle diameter of pinion

$$d_1 = \frac{m_n z_1}{\cos\beta} = \frac{6 \times 20}{\cos 26} = 133.5mm$$

Pitch circle diameter of gear

$$d_2 = \frac{m_n z_2}{\cos\beta} = \frac{6 \times 80}{\cos 26} = 534\ mm$$

$\therefore$ Centre distance $a = \frac{d_1 + d_2}{2} = \frac{133.5 + 534}{2} = 333.75mm$ //

For pressure angle $20^\circ$ full depth involute system

Addendum $h_a = 1m_n = 1 \times 6 = 6mm$

Dedendum $h_f = 1.25m_n = 1.25 \times 6 = 7.5mm$

Whole depth $h = 2.25m_n = 2.25 \times 6 = 13.5\ mm$

Clearance $c = 0.25m_n = 0.25 \times 6 = 1.5mm$

Tooth thickness $s = \frac{\pi}{2} m_n = \frac{\pi}{2} \times 6 = 9.425mm$

Working depth $h' = 2m_n = 2 \times 6 = 12mm$

Outside or Addendum circle diameter of pinion

$$d_{a_1} = d_1 + 2h_a$$
$$= 133.5 + 2 \times 6 = 145.5mm$$

Addendum circle diameter of gear

$$d_{a_2} = d_2 + 2h_a$$
$$= 534 + 2 \times 6 = 546mm$$

Root or dedendum circle diameter of pinion

$$d_{f_1} = d_1 - 2h_f$$
$$= 133.5 - 2 \times 7.5 = 118.5mm$$

Root or dedendum circle diameter of gear

$$d_{f_2} = d_2 - 2h_f$$
$$= 534 - 2 \times 7.5$$
$$= 519\ mm$$ //

iv) Checking

a) Dynamic loading

According to Buckingham's equation

Dynamic load $F_d = F_t + \dfrac{21v_m(F_t + bC\cos^2\beta \cdot c)\cos\beta}{21v_m + \sqrt{F_t + bC\cos^2\beta}}$

For $v_m = 22.3704$ m/s

Error $f = 0.015$ mm

$C = 118.084$ N/mm

$F_d = 1006 + \dfrac{21 \times 22.3704[1006 + 60 \times 118.084\cos^2 26]\cos 26}{21 \times 22.3704 + \sqrt{1006 + 60 \times 118.084\cos^2 26}}$

$F_d = 6155.27$ N

b) wear load

According to Buckingham's equation

wear load $F_w = \dfrac{d_1 b Q K}{\cos^2\beta}$

~~Ratio~~ Ratio factor $Q = \dfrac{2Z_{2v}}{Z_{1v} + Z_{2v}} = \dfrac{2Z_2}{Z_1 + Z_2} = \dfrac{2 \times 80}{20 + 80} = 1.6$

For safer design

$F_w > F_d$

$\dfrac{d_1 b Q K}{\cos^2\beta} > F_d$

$\dfrac{133.5 \times 60 \times 1.6 \times K}{\cos^2 26} > 6155.27$

$K \geq 0.388$ N/mm$^2$

For 20° FD and $K \geq 0.388$ N/mm$^2$

Surface hardness for pinion = 200 BHN

Surface hardness for gear = 150 BHN

8) Given data

Pressure angle, $\phi = 20°$

Shaft angle = $90°$

velocity ratio, $i = 4:1$

Pinion teeth, $Z_p = 20$

Gear teeth, $Z_g = 80$

Power transmitted, $P = 40kW$

Speed of pinion, $N_p = 720rpm$

Material

Pinion: C30 steel (heat treated)

Gear : Cast steel (0.2%C untreated)

1) Determination of Pitch Cone angles

$$\tan\delta_p = \frac{Z_p}{Z_g} = \frac{20}{80} = 0.25$$

$$\delta_p = 14.04$$

$$\delta_g = 90° - 14.04°$$

$$\delta_g = 75.96°$$

2) Torque on Pinion

$$T = \frac{9550P}{N} = \frac{9550 \times 40}{720}$$

$$T = 530.5 \text{ N-m} = 530500 \text{ N-mm}$$

3) Tangential tooth load

$$F_t = \frac{2T}{d_p}$$

Since $d_p = mZ_p = 20m$

$$F_t = \frac{2 \times 530500}{20m}$$

$$F_t = \frac{53050}{m}$$

4.) Lewis Form Factor

For 20° FDI

$$y = 0.154 - \frac{0.912}{z}$$

For pinion

$$y_p = 0.154 - \frac{0.912}{20}$$

$$y_p = 0.1084$$

5.) Cone distance

$$L = \frac{m}{2}\sqrt{z_p^2 + z_g^2}$$

$$= \frac{m}{2}\sqrt{(20)^2 + (80)^2}$$

$$L = 41.23$$

Assume face width

$$b = 10m$$

Check: $b < L/3$

$$10m < 13.74m$$

6.) Beam strength

$$F_t = \sigma_d b \pi m y \left(1 - \frac{b}{L}\right)$$

$$1 - \frac{b}{L} = 1 - \frac{10m}{41.23m} = 0.757$$

Allowable stress for heat treated C30:

$$\sigma_d = 200\,MPa$$

Substituting,

$$\frac{53050}{m} = 200(10m)(\pi m)\,\cancel{0.180}\,(0.1084)\,(0.757)$$

Solving $m \simeq 6mm$

7.) Final Dimensions

Pitch diameters

$d_p = 20m = 120mm$

$d_g = 80m = 480mm$

Face width

$b = 10m = 60mm$

Cone distance

$L = 41.23m = 247.4mm$

Pitch cone angles

$\delta_p = 14.04°$

$\delta_g = 75.96°$

8.) Wear strength

wear load

$F_w = d_p b Q K$

$Q = \frac{2z_g}{z_p + z_g} = \frac{160}{100} = 1.6$

For steel-steel combination

$k = 1 N/mm^2$

$F_w = 120 \times 60 \times 1.6$

$F_w = 11520 N$

Since beam strength > wear load.

9) a) Petroff's Equation.

Consider a vertical shaft rotating in a guide bearing It is assumed that the bearing

i) Carries a very small load

ii) The clearance c is completely filled with oil

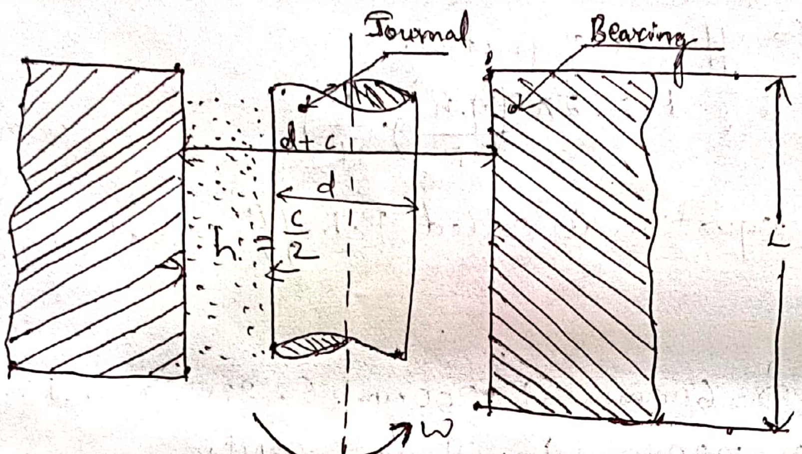
iii) The end leakage is completely negligible

iv) The oil used is of high viscosity

v) The journal revolves at very high speeds

vi) The bearing run concentrically

Let




Let $d$ = diameter of journal or shaft

$c$ = Diametral clearance

$n'$ = Speed of shaft or journal in rps

$= n/60 =$

$L$ = Length of bearing

$\psi = c/d$ = diametral clearance ratio.

$\eta$ = Viscosity of oil in Pas

$v$ = Velocity $= \dfrac{\pi d n}{60} = \pi d n'$ m/s

Shear stress $\tau = \eta \cdot \dfrac{v}{h} = \eta \cdot \dfrac{\pi d n'}{\left(\frac{c}{2}\right)} = \dfrac{2\pi d \eta . n'}{c}$

Surface area $A = \pi d L$

$\therefore$ Force $F = \tau A = \dfrac{(2\pi d \eta . n')}{c} (\pi d L) = \dfrac{2\pi^2 d^2 n' . \eta . L}{c}$

Torque $M_t$ = Force x radius $= F . \dfrac{d}{2} = \dfrac{2\pi^2 d^2 n' . \eta . L}{c} . \dfrac{d}{2}$

—(1)

$$\therefore M_t = \frac{\pi^2 d^2 n' . \eta . L}{\left(\frac{c}{d}\right)} = \frac{\pi^2 d^2 n' \eta . L}{\psi}$$

But $M_t = (\mu W)\frac{d}{2}$ where $\mu W$ = Friction force

$W = P.A = P.L.d$ = Load

$P$ = Bearing pressure in $N/m^2$

$$\therefore M_t = \mu . (PLd) . \frac{d}{2} \qquad \text{—— (2)}$$

Equating (1) & (2)

$$\mu (PLd) . \frac{d}{2} = \frac{\pi^2 d^2 n' . \eta . L}{\psi}$$

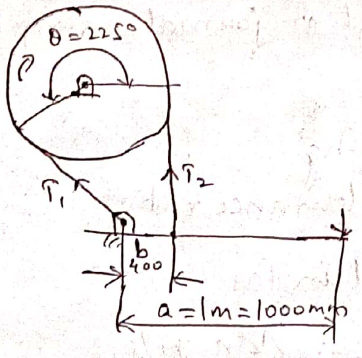
$\therefore$ Co-efficient of friction

$$\mu = 2\pi^2 \left(\frac{\eta . n'}{P}\right) . \left(\frac{1}{\psi}\right)$$

This equation is called Petroff's equation.

2) b) Given : $D = 600mm$ $R = 300mm$, $\theta = 225°$, $N = 15kW$

$n = 720rpm$ $h_1 = 3b$, $\sigma_b = 80MPa$




Torque transmitted $M_t = 9550 \times 1000 \times \frac{N}{n}$

$$= 9550 \times 1000 \times \frac{15}{720} = 198958.33 \text{ N-mm}$$

Also, $M_t = F_o . R$

$198958.33 = F_o \times 300$

$\therefore$ Tangential force $F_o = 663.2$ N

$$e^{\mu\theta} = e^{0.3 \times 225 \times \pi/180} = 3.2482, \text{ Assume } \mu = 0.3$$

For cw rotation, actuating force

$$F = \frac{F_0 . b}{a}\left[\frac{1}{e^{\mu\theta}-1}\right]$$

$$= \frac{663.2 \times 400}{1000}\left[\frac{1}{3.2482-1}\right]$$

For ccw rotation, actuating force

$$F = \frac{F_0 . b}{a}\left[\frac{e^{\mu\theta}}{e^{\mu\theta}-1}\right]$$

$$= \frac{663.2 \times 400}{1000}\left[\frac{3.2482}{3.2482-1}\right] = 383.3\,N$$

$\therefore F_{max} = 383.3\,N$

Considering the lever as a cantilever and neglecting the effect of $T_1$ or $T_2$, maximum bending moment $M_b$

$M_b = F_{max} \cdot a = 383.3 \times 1000 = 383300\,N\text{-}mm$

we have, $\frac{M_b}{I} = \frac{\sigma_b}{c}$ where $I = \frac{b_1 h_1^3}{12} = \frac{b_1(3b_1)^3}{12} = \frac{27b_1^4}{12}$

$$c_1 = \frac{h_1}{2} = \frac{3b_1}{2} = 1.5b_1$$

$$\frac{383300}{\frac{27b_1^4}{12}} = \frac{80}{1.5b_1} \qquad \therefore b_1 = 14.727\,mm$$

Select, width of lever $b_1 = 15\,mm$

Thickness of lever $h_1 = 45\,mm$

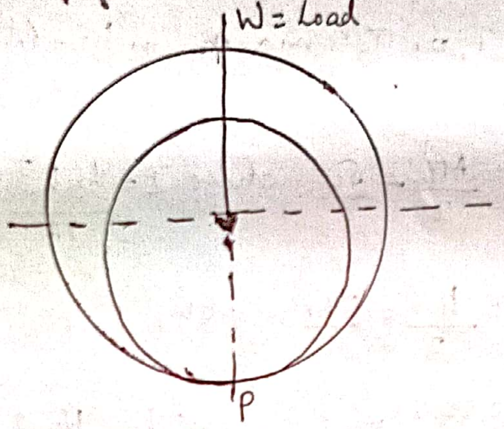
10. a) Hydrodynamic Theory of Lubrication

Perfect lubrication (thick film) is lubrication that maintains a complete film of lubricant between the surfaces of the shaft and bearing.

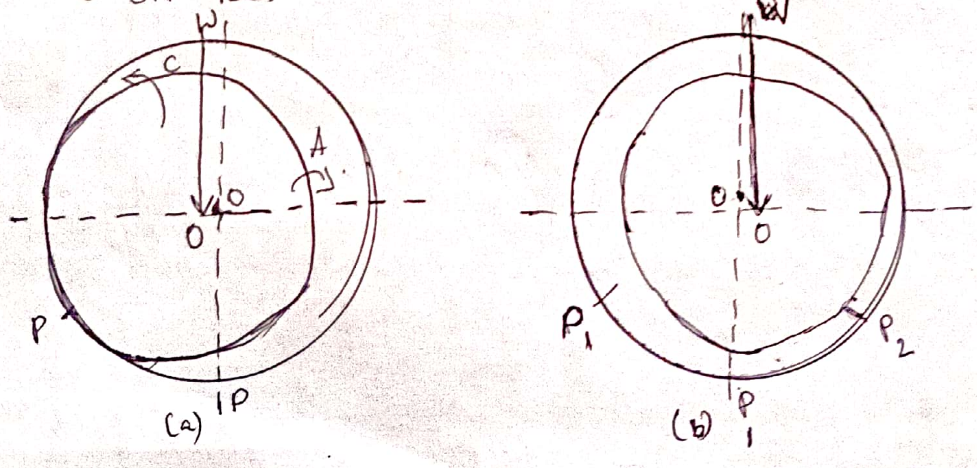
Boundary lubrication (thin film) is the type where metal to metal contact can sometimes occur with no lubricant there is a continuous abrasive action between the two materials, which causes additional wear and generation of heat

The action of lubricant in a plain journal bearing according to hydrodynamic theory is as below,

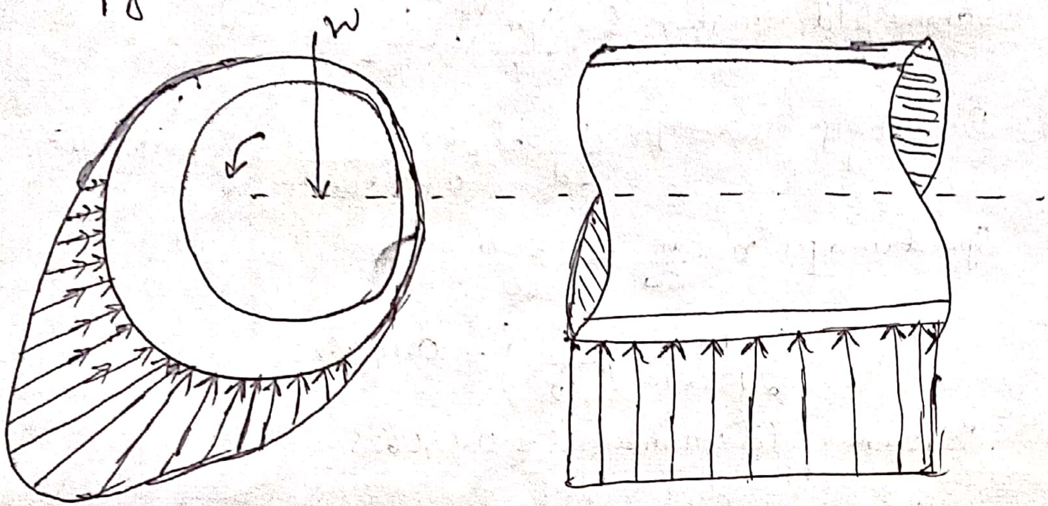
When the journal is at rest, it comes in contact with the bearing at its lowest point P leaving a crescent shaped space above, which is filled with lubricant as shown in fig.




When the shaft rotates in counter clockwise direction as shown in fig (a) below as it tends to move above slightly and the point of P contact will move to $P_1$. As the speed of the shaft increases it carries the oil with it from the wide space 'A' towards 'C' and the fluid pressure in the oil rises

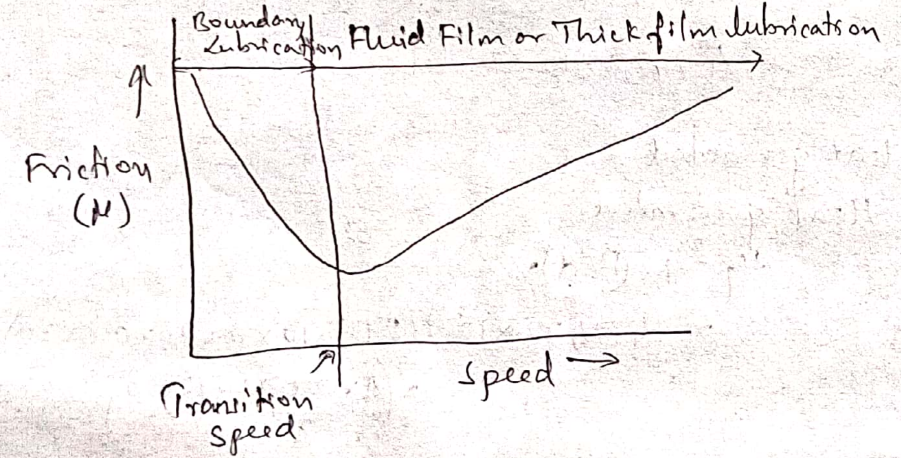


when the speed is higher, the pressure is sufficient to carry the weight of journal. Then there is no metal to metal contact, but the shaft is supported by the oil, being nearest to the bearing surface at a point $P_2$ on the off side. as shown in fig(b). The point of maximum fluid pressure and position of the shaft will be as shown in fig.




Under the above conditions the fluid friction in the lubricant is substituted for sliding friction between the journal and the bearing. Fig above shows the ideal variation of pressure in the converging film in radial and axial directions.

The fig shows the variations of friction with speed of rotation.




If the speed of rotation is less than a critical value, depending on the load, lubrication and surface finish, complete separation of the surfaces by a film is not possible and some of the load is taken by the contact of asperities on the surfaces. This condition is called <u>boundary lubrication</u>

10) b) Given: $L = 75mm = 0.075m$ $d = 75mm = 0.075m$

$W = 12kN = 12 \times 10^3 N$ ; $n = 1800 rpm$, $\frac{d}{c} = 1000$

$\psi = 10^{-3}$ ; $\eta = 0.01 Pas$

Full Journal

Soln

a) Sommerfeld number

Sommerfeld number $S = \left(\frac{\eta n'}{P}\right) \cdot \frac{1}{\psi^2}$

Bearing pressure $P = \frac{W}{Ld} = \frac{12 \times 10^3}{0.075 \times 0.075} = 2.133 \times 10^6 N/m^2$

Speed in rps $n' = \frac{n}{60} = \frac{1800}{60} = 30$

i.e. $S = \left(\frac{0.01 \times 30}{2.133 \times 10^6}\right)\left(\frac{1}{10^{-3}}\right)^2 = 0.140625$

$\therefore$ Sommerfeld number $S = 0.140625$

b) Co-efficient of friction by Mckee equation

Co-efficient of friction $\mu = k_a \left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right) \times 10^{-10} + \Delta\mu$

where $k_a = 1.95 \times 10^{11}$ for full journal bearing

$\Delta\mu = 0.002$

$\therefore \mu = (1.95 \times 10^{11})\left(\frac{0.01 \times 30}{2.133 \times 10^6}\right)\left(\frac{1}{10^{-3}}\right) \times 10^{-10} + 0.002$

$= 4.7422 \times 10^{-3}$

c) Heat generated

Heat generated

$H_g = \mu (PLd) v$

$= (4.7422 \times 10^{-3})(2.133 \times 10^6 \times 0.075 \times 0.0075)$

$\times \left(\frac{\pi \times 0.075 \times 1800}{60}\right)$

$H_g = 402.26$ watts.

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