# UNIT - I

## DESIGN OF JOINTS AND FASTENERS

Step by Step Procedure of design of Sleeve and Cotter Joint.

### STEP 1: To find dia of Rod (d):

Stress = Load / Area of rod.

From the data book PSG 7.1

$$\sigma_t = \frac{P}{a}$$

$$\text{Area of Rod } (a) = \frac{\pi}{4} \times d^2$$

$$\sigma_t = \frac{P}{\frac{\pi}{4} \times d^2}$$

$$\sigma_t = \frac{4P}{\pi d^2}$$

$$d^2 = \frac{4P}{\sigma_t \times \pi}$$

$$\boxed{d = \sqrt{\frac{4P}{\sigma_t \times \pi}}}$$

STEP: 2 : To find Other dimensions :

From the data book PSG 7.140

STEP: 3: Check for failure:

Check for failure of the Enlarged rod in tension

Induced tensile Stress = $\frac{\text{Load}}{\text{Area}}$.

Area of Enlarged Rod $= \frac{\pi}{4} \times d_1^2 - d_1 \times t$

$$\sigma_t = \frac{P}{\frac{\pi}{4} \times d_1^2 - d_1 \times t}$$

$$\boxed{\sigma_t = \frac{4P}{\pi[d_1^2 - d_1 \times t]}}$$

STEP: 4: Checking for failure in ~~tension~~ of Sleeve in tension:

Induced tensile Stress = $\frac{\text{Load}}{\text{Area}}$.

$$\sigma_t = \frac{P}{A}$$

Area of Sleeve $= \frac{\pi}{4} \times (d_2^2 - d_1^2) - (d_2 - d_1)t$

$$\sigma_t = \frac{P}{\frac{\pi}{4}(d_2^2 - d_1^2) - (d_2 - d_1)t}$$

$$\boxed{\sigma_t = \frac{4P}{\pi(d_2^2 - d_1^2) - (d_2 - d_1)t}}$$

STEP: 5: Checking for failure of Cotter in Crushing:

Induced Crushing Stress $= \frac{\text{Load}}{\text{Area}}$

$$\sigma_c = \frac{P}{A}$$

Area of Cotter (A) $= d_1 \times t$.

$$\boxed{\sigma_c = \frac{P}{d_1 \times t}}$$

## Step: 6: Checking for failure of Cotter in Crushing in Sleeve.

Induced Crushing Stress $= \frac{\text{Load}}{\text{Area}}$.

Area of Cotter in Sleeve $= (d_2 - d_1) t$.

$$\sigma_c = \frac{P}{A}.$$

$$\boxed{\sigma_c = \frac{P}{(d_2 - d_1) t}}$$

## Step 7: Checking for failure of Cotter in Shear:

Induced Shear Stress $= \frac{\text{Load}}{\text{Area}}$.

$$\tau = \frac{P}{A}.$$

Area (A) $= 2 \times b \times t$.

$$\boxed{\tau = \frac{P}{2bt}}$$

STEP 8: Checking for failure of Enlarged rod end in Shear:

Induced Shear Stress = $\frac{\text{Load}}{\text{Area}}$

$$\tau = \frac{P}{A}$$

Area of Enlarged Rod (A) = $2 \times c \times d_1$

$$\boxed{\tau = \frac{P}{2 \times c \times d_1}}$$

STEP 9: Checking for failure of Sleeve in Shear:

Induced Shear Stress = $\frac{\text{Load}}{\text{Area}}$

$$\tau = P/A$$

Area of Sleeve: $2 \times a \times (d_2 - d_1)$

$$\boxed{\tau = \frac{P}{2 \times a \times (d_2 - d_1)}}$$

Problem:

1- Design a Sleeve and Cotter Joint to withstand a tensile load of 60kN. Assuming all the Parts are Made of Same Material. The Permissible Stresses are 60 N/mm² in tension, 125 N/mm² in bearing and 70 N/mm² in Shear.

Given data:

Tensile load $(P) = 60kN \Rightarrow 60 \times 10^3 N$.

Permissible Stresses

Tensile Stress $(\sigma_t) = 60 N/mm^2$

Bearing Stress (or)
Crushing Stress $(\sigma_c) = 125 N/mm^2$

Shear Stress $(\tau) = 70 N/mm^2$

To find:

Design a Sleeve and Cotter Joint

Solution:

Step 1: To find dia of Rod(d):

FDB PSG 7.1

$$\text{Stress} = \frac{\text{Load}}{\text{Area}} \rightarrow \sigma_t = \frac{P}{a}$$

$$a = \frac{\pi}{4} \times d^2$$

$$\sigma_t = \frac{4P}{\pi \times d^2}$$

$$60 = \frac{60 \times 10^3 \times 4}{\pi \times d^2}$$

$$d^2 = \frac{60 \times 10^3 \times 4}{\pi \times 60}$$

$$d^2 = 1273.23$$

$$d = 35.68 \text{ mm}$$

Say

$$\boxed{d = 40 \text{mm.}}$$

Step 2: To find other dimensions:

FDB PSG 7.40

| S.No | DESCRIPTION | EQUATION | VALUES |
|---|---|---|---|
| 1. | Dia of Enlarged Rod | $d_1 = 1.3d$ <br> $\Rightarrow 1.3 \times 40$ | 52 mm. |
| 2. | outer dia of Sleeve | $d_2 = 2.5d$ <br> $\Rightarrow 2.5 \times 40$ | 100 mm. |
| 3. | distance of slot from Sleeve | $a = 1.3 \times d.$ <br> $\Rightarrow 1.3 \times 40$ | 52 mm |
| 4. | Length of the Sleeve | $L = 8 \times d$ <br> $\Rightarrow 8 \times 40$ | 320 mm |

| | | | |
|---|---|---|---|
| 5. | Length of the cover | $l = 4d$<br>$= 4 \times 40$ | 160 mm |
| 6. | Width of the cotter | $b = 1.3 \times d$<br>$\Rightarrow 1.3 \times 40$ | 52 mm |
| 7. | distance of the slot from Rod | $c = 1.4d$<br>$\Rightarrow 1.4 \times 40$ | 56 mm |
| 8. | Length of the Enlarged Rod End | $e = 0.5d$<br>$= 0.5 \times 40$ | 20 mm |
| 9. | Thickness of Cotter | $t = 0.3d$<br>$\Rightarrow 0.3 \times 40$ | 12 mm. |

Step 3: Check for failures:

Check for failure of Enlarged Rod ~~End~~ in tension:

Induced tensile Stress $= \frac{\text{Load}}{\text{Area}}$

$$\text{Area} = \frac{\pi}{4} \times (d_1^2 - d_1 \times t)$$

$$= \frac{\pi}{4} \times [52^2 - 52 \times 12]$$

$$\boxed{A = 1633.628 \text{ mm}^2}$$

$$\sigma_t = \frac{P}{A}$$

$$\Rightarrow \frac{60\times10^3}{1633.625}$$

$$\boxed{\sigma_t = 36.72\ N/mm^2}$$

Therefore $36.72\ N/mm^2 < 60\ N/mm^2$. So the design is safe.

Step 4: Checking for failure of sleeve in tension:

Induced tensile stress $= \frac{\text{Load}}{\text{Area}}$

$$\sigma_t = \frac{P}{A}$$

$$\text{Area } (A) = \frac{\pi}{4}[d_2^2 - d_1^2] - [d_2 - d_1]\times t$$

$$= \frac{\pi}{4}[100^2 - 52^2] - [100 - 52]\times 12$$

$$= 0.785\times[7296 - 576]$$

$$\boxed{A = 5275.2\ mm^2}$$

$$= \frac{60\times10^3}{5275.2}$$

$$\boxed{\sigma_t = 11.37\ N/mm^2}$$

Therefore $11.37\ N/mm^2 < 60\ N/mm^2$. So the design is safe.

Step 5: Checking for failure of Cotter Pin Crushing:

Induced Crushing Stress $= \frac{Load}{Area}$

$$\sigma_c = \frac{P}{A}.$$

$$Area\ (A) = d_1 \times t$$
$$= 52 \times 12$$
$$\boxed{A = 624\ mm^2}$$

$$\sigma_c = \frac{60 \times 10^3}{624}$$
$$\boxed{\sigma_c = 96.15\ N/mm^2}$$

Therefore $96.15\ N/mm^2 < 125\ N/mm^2$. So the design is safe.

Step 6: Checking for failure of Cotter Pin Crushing in sleeve.

Induced Crushing Stress $= \frac{Load}{Area}$

$$\sigma_c = \frac{P}{A}.$$

$$\text{Area } (A) = (d_2 - d_1) \times t$$
$$= (100 - 52) \times 12$$
$$\boxed{A = 576 \text{ mm}^2}$$

$$\Rightarrow \frac{60 \times 10^3}{576}$$
$$\boxed{\sigma_c = 104.16 \text{ N/mm}^2}$$

Therefore $104.16 \text{ N/mm}^2 < 125 \text{ N/mm}^2$. So the design is safe.

Step 7: Checking for failure of Cotter in shear:

$$\text{Induced Shear Stress} = \frac{\text{Load}}{\text{Area}}$$
$$\tau = P/A.$$
$$\text{Area}(A) = 2 \times b \times t$$
$$= 2 \times 52 \times 12$$
$$\boxed{A = 1248 \text{ mm}^2}$$
$$= \frac{60 \times 10^3}{1248}$$
$$\boxed{\tau = 48.07 \text{ N/mm}^2}$$

Therefore $48.07 \text{ N/mm}^2 < 70 \text{ N/mm}^2$. So the design is safe.

Step 8: Checking for failure of Enlarged Rod end in Shear.

$$\text{Induced Shear Stress} = \frac{\text{Load}}{\text{Area}}$$

$$\tau = \frac{P}{A}$$

$$\text{Area } (A) = 2 \times c \times d_1$$

$$= 2 \times 56 \times 52$$

$$\boxed{A = 5824\ mm^2}$$

$$\tau = \frac{60 \times 10^3}{5824}$$

$$\boxed{\tau = 10.30\ N/mm^2}$$

Therefore $10.30\ N/mm^2 < 70\ N/mm^2$. So the design is safe.

Step 9: Checking for failure of Sleeve in shear:

$$\text{Induced Shear Stress} = \frac{\text{Load}}{\text{Area}}$$

$$\tau = \frac{P}{A}$$

$$\text{Area}(A) = 2 \times a (d_2 - d_1)$$

$$= 2 \times 52 \ (100 - 52)$$

$$\boxed{A = 4992\ mm^2}$$

$$\tau = \frac{60 \times 10^3}{4992}$$

$$\boxed{\tau = 12.021\ N/mm^2}$$

Therefore $12.02\ N/mm^2 < 70\ N/mm^2$. So the design is safe.

Result:

(1) Dia of Rod $(d)$ = 40 mm
(2) Dia of Enlarged Rod $(d_1)$ = 52 mm
(3) Outer dia of Sleeve $(d_2)$ = 100 mm
(4) Distance of Slot from Sleeve $(a)$ = 52 mm
(5) Length of the sleeve $(L)$ = 320 mm
(6) Length of Cover $(l)$ = 160 mm
(7) Width of Cotter $(b)$ = 52 mm
(8) Distance of Slot from Rod $(c)$ = 56 mm.
(9) Length of Enlarged Rod $(e)$ = 20 mm.
(10) Thickness of Cotter $(t)$ = 12 mm
(11) Checking for failure of Rod in tension $(\sigma_t)$ = 36.73 $N/mm^2$
(12) Checking for failure of Sleeve $(\sigma_t)$ = 11.37 $N/mm^2$
(13) Checking for failure of Cotter $(\sigma_c)$ = 96.15 $N/mm^2$
(14) Checking for failure in sleeve $(\sigma_c)$ = 104.16 $N/mm^2$
(15) Checking for failure of Cotter in sleeve $(\tau)$ = 48.07 $N/mm^2$
(16) Checking for failure in Enlarged Rod End $(\tau)$ = 10.30 $N/mm^2$
(17) Checking for failure in Sleeve ~~I~~ $(\tau)$ = 12.00 $N/mm^2$

2. Design a Sleeve and Cotter Joint to Connect two rods for transmitting a maximum tensile load of 100 kN. The Rods, Sleeve and Cotters are Made of Same Material and permissible stress are 65 N/mm² in tension, 130 N/mm² in Compression (or) Crushing (or) bearing and 60 N/mm² in Shear.

**Given data:**

Tensile load (P) = 100 kN ⇒ $100 \times 10^3$ N.

Permissible Stresses.

Tensile Stress ($\sigma_t$) = 65 N/mm²

Crushing (or) Compression Stress ($\sigma_c$) = 130 N/mm²

Shear Stress ($\tau$) = 60 N/mm²

**To find:**

Design a Sleeve and Cotter Joint.

**Solution:**

**Step 1: To find the dia of Rod (d):**

$$\text{Stress} = \frac{\text{Load}}{\text{Area}} \quad \text{FDB PSG 7.1}$$

$$\sigma_t = \frac{P}{a} \Rightarrow \frac{P}{\frac{\pi}{4} \times d^2}$$

$$65 = \frac{4 \times 100 \times 10^3}{\pi \times d^2}$$

$$d^2 = \frac{4 \times 100 \times 10^3}{\pi \times 65}$$

$d^2 = \cancel{6153.846}\ 1958.83$

$d = \cancel{78.44\ mm\ say}\ 44.25\ mm$ say.

$\boxed{d = \cancel{80mm}\ 45\ mm.}$

Step: 2: To find other dimensions:

PDB PSG: 7.40

| S.NO | DESCRIPTION | EQUATION | VALUES |
|---|---|---|---|
| 1 | Dia of Enlarged Rod | $d_1 = 1.3 \times d \Rightarrow 1.3 \times 45$ | 58.5mm |
| 2. | Outer dia of Sleeve | $d_2 = 2.5d \Rightarrow 2.5 \times 45$ | 112.5mm. |
| 3. | distance of Slot from sleeve | $a = 1.3d \Rightarrow 1.3 \times 45$ | 58.5mm |
| 4. | Length of the Sleeve | $L = 8d \Rightarrow 8 \times 45$ | 360mm. |
| 5. | Length of Cover | $l = 4d \Rightarrow 4 \times 45$ | 180mm |
| 6. | Width of Cotter. | $b = 1.3d \Rightarrow 1.3 \times 45$ | 58.5mm |
| 7. | distance of Slot from Rod. | $c = 1.4d \Rightarrow 1.4 \times 45$ | 63mm |
| 8. | Length of Enlarged Rod end | $e = 0.5d \Rightarrow 0.5 \times 45$ | 22.5mm |
| 9. | Thickness of Cotter | $t = 0.3d \Rightarrow 0.3 \times 45$ | 13.5mm |

Step 3: Check for failures.

Checking for failure of Enlarged Rod End in tension:

$$\text{Tensile Stress} = \frac{\text{Load}}{\text{Area}} \Rightarrow \sigma_t = \frac{P}{A}.$$

$$\text{Area}(A) = \frac{\pi}{4}\left[d_1^2 - d_1 \times t\right]$$

$$\Rightarrow \frac{\pi}{4}\left[56.5^2 - 56.5 \times 13.5\right]$$

$$\boxed{A = 2067.56\ mm^2}$$

$$\sigma_t = \frac{100 \times 10^3}{2067.56}$$

$$\boxed{\sigma_t = 48.36\ N/mm^2}$$

Therefore $48.36\ N/mm^2 < 65\ N/mm^2$. So the design is safe.

Step 4: Checking for failure of Sleeve in tension

$$\text{Tensile Stress} = \frac{\text{Load}}{\text{Area}} \Rightarrow \sigma_t = \frac{P}{A}$$

$$\text{Area}(A) = \frac{\pi}{4}\left[\left[d_2^2 - d_1^2\right] - \left[d_2 - d_1\right] \times t\right]$$

$$= \frac{\pi}{4}\left[\left(112.5^2 - 56.5^2\right] - \left[112.5 - 56.5\right] \times 13.5\right]$$

$$= 0.785\left[9234 - 729\right]$$

$$\boxed{A = 6676.425\ mm^2}$$

$$\sigma_t = \frac{100 \times 10^3}{6676.425}$$

$$\boxed{\sigma_t = 14.97 \ N/mm^2}$$

Therefore $14.97 \ N/mm^2 < 65 \ N/mm^2$. So the design is safe.

Step 5: Checking for failure of Cotter in Crushing:

$$\text{Crushing Stress} = \frac{\text{Load}}{\text{Area}} \Rightarrow \sigma_c = \frac{P}{A}.$$

$$\text{Area}(A) = d_1 \times t.$$

$$\Rightarrow 58.5 \times 13.5$$

$$\boxed{A = 789.75 mm^2}$$

$$\sigma_c = \frac{100 \times 10^3}{789.75}$$

$$\boxed{\sigma_c = 126.62 \ N/mm^2}$$

Therefore $126.62 \ N/mm^2 < 130 \ N/mm^2$. So the design is safe.

Step 6: Checking for failure of Cotter in Crushing in Sleeve

$$\text{Crushing Stress} = \frac{\text{Load}}{\text{Area}} \Rightarrow \sigma_c = \frac{P}{A}.$$

$$\text{Area}(A) = (d_2 - d_1) t.$$

$$\Rightarrow (112.5 - 58.5) \times 13.5$$

$$\boxed{A = 729 mm^2}$$

$$\Rightarrow \frac{100 \times 10^3}{729}$$

$$\boxed{\sigma_c = 137.17\ N/mm^2}$$

Therefore $137.17\ N/mm^2 > 130\ N/mm^2$. So the design is not safe. Now Take $\sigma_c = 130\ N/mm^2$

$$\sigma_c = \frac{P}{A} \Rightarrow \frac{P}{(d_2 - d_1)t}$$

$$130 = \frac{100 \times 10^3}{[112.5 - 58.5]\,t}$$

$$t = \frac{100 \times 10^3}{[54] \times 130} \Rightarrow \frac{100 \times 10^3}{7020}$$

$t = 14.245mm$ say

$$\boxed{t = 14.25mm}$$

Step 7: Checking for failure of cotter in shear.

$$\text{Shear Stress} = \frac{\text{Load}}{\text{Area}} \Rightarrow \tau = \frac{P}{A}$$

$$\text{Area }(A) = 2 \times b \times t$$

$$\Rightarrow 2 \times 58.5 \times 14.5$$

$$\boxed{A = 1696.5\ mm^2}$$

$$\tau = \frac{100 \times 10^3}{1696.5}$$

$$\boxed{\tau = 58.94\ N/mm^2}$$

Therefore $58.94\ N/mm^2 < 60\ N/mm^2$. So the design is safe.

Step 8: Checking for failure of Enlarged Rod end in Shear.

$$\text{Shear Stress} = \frac{\text{Load}}{\text{Area}} \Rightarrow \tau = \frac{P}{A}.$$

$$\text{Area } (A) = 2 \times c \times d_1$$

$$= 2 \times 63 \times 58.5$$

$$\boxed{A = 7371\ mm^2}$$

$$\tau = \frac{100 \times 10^3}{7371}$$

$$\boxed{\tau = 13.56\ N/mm^2}$$

Therefore $13.56\ N/mm^2 < 60\ N/mm^2$. So the design is safe.

Step 9: Checking for failure of Sleeve in Shear.

$$\text{Shear Stress} = \frac{\text{Load}}{\text{Area}} \Rightarrow \tau = \frac{P}{A}.$$

$$\text{Area } (A) = 2 \times a \times (d_2 - d_1)$$

$$= 2 \times 56.5 \times (112.5 - 56.5)$$

$$\boxed{A = 6316\ mm^2}$$

$$\tau = \frac{100 \times 10^3}{6316}$$

$$\boxed{\tau = 15.82\ N/mm^2}$$

Therefore $15.82\ N/mm^2 < 60\ N/mm^2$. So the design is safe.

Result:

1. Dia of Rod $(d)$ = 45mm
2. Dia of Enlarged Rod $(d_1)$ = 56.5mm
3. Outer dia of Sleeve $(d_2)$ = 112.5mm
4. distance of slot from Sleeve $(a)$ = 58.5mm
5. Length of the sleeve $(L)$ = 360mm
6. Length of Cover $(l)$ = 160mm
7. Width of Cotter $(b)$ = 56.5mm
8. distance of slot from Rod $(c)$ = 63mm
9. Length of Enlarged Rod $(e)$ = 22.5mm
10. Thickness of Cotter $(t)$ = 13.5mm.
11. Checking for failure at Enlarged Rod end $(\sigma_F)$ = 48.36 N/mm²
12. Checking for failure in sleeve $(\sigma_F)$ = 14.97 N/mm²
13. Checking for failure of Cotter $(\sigma_C)$ = 126.62 N/mm²
14. Checking for failure of Cotter in Sleeve $(\sigma_C)$ = 137.17 N/mm²
15. So Thickness $(t)$ = 14.5mm.
16. Checking for failure of Cotter in shear $(\tau)$ = 53.94 N/mm²
17. Checking for failure in Enlarged Rod End $(\tau)$ = 13.56 N/mm²
18. Checking for failure of sleeve in shear $(\tau)$ = 15.82 N/mm².

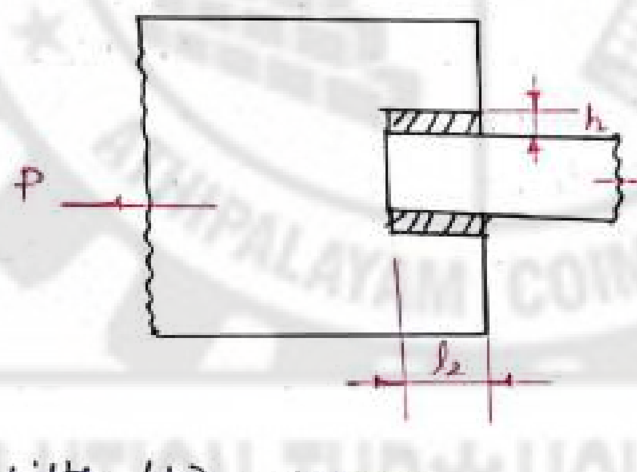
WELDING.

# DOUBLE PARALLEL WELD:

## Problem:

(1) A Plate 100mm Wide and 10mm thick is to be Welded to another plate by means of double parallel fillets. The plates are subject to a Static load of 80kN. Find the length of weld if allowable Shear Stress in the weld does not exceed $55 N/mm^2$.

GIVEN Data:




Width (b) = 100mm

thick (t) = 10mm (or) Size of weld (h)

Load (P) = 80kN = $80 \times 10^3 N$.

Shear stress ($\tau$) = $55 N/mm^2$

TO FIND:

Length of the Weld ($l_2$)

SOLUTION:

FDB PSG 11.30 For double parallel weld.

$$\tau = \frac{0.707 P}{h \times l_2}$$

$$55 = \frac{0.707 \times 80 \times 10^3}{10 \times l_2}$$

$$l_2 = \frac{0.707 \times 80 \times 10^3}{10 \times 55}$$

$l_2 = 102.83$ mm say

$l_2 = 103$ mm.

By Adding Weld ~~...~~ Constant.

$l_2 = 12.5 + 103$

$$\boxed{l_2 = 115.5 \text{ mm.}}$$

RESULT:

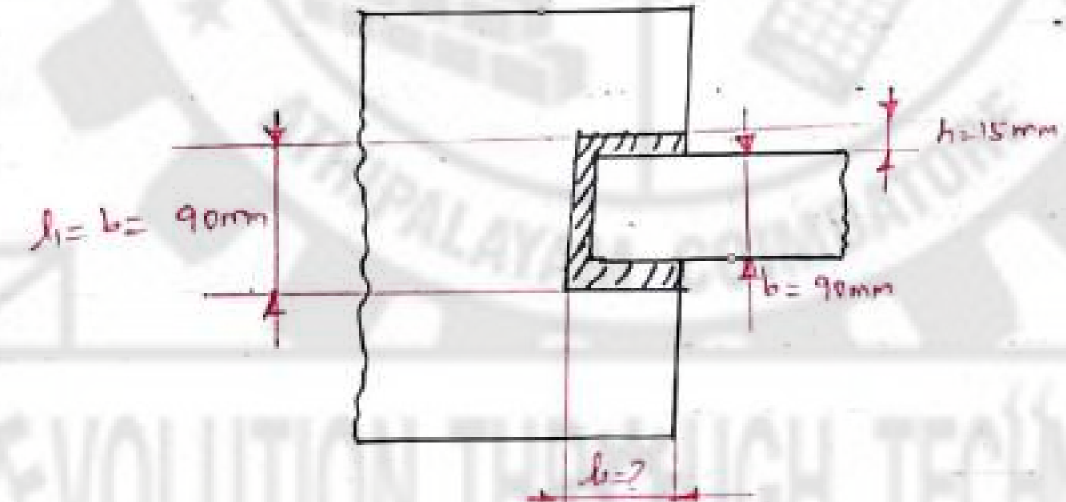
$l_2 = 115.5$ mm

## SINGLE TRANSVERSE WELD AND DOUBLE PARALLEL FILLET WELD:

Problem:

(1) A plate 90mm wide and 15mm thick welded onto another plate by a single transverse weld and a double parallel fillet weld. Find the length of a parallel fillet weld if the plate is loaded by a static tensile load. Take allowable tensile stress as 70N/mm² and shear stress as 55N/mm².

GIVEN DATA:




Width (b) = 90mm

thickness (t) = 15mm (size of weld (h))

tensile stress ($\sigma_t$) = 70 N/mm²

Shear stress ($\tau$) = 55N/mm²

TO FIND:

Length of Weld ($l_2$)

SOLUTION:

FDB PSG 11.3 For single transverse Weld.

$$\sigma_t = \frac{\text{Load}}{\text{Area}} \Rightarrow \frac{P}{a}$$

Area(a) = b×t ⇒ 90×15 = 1350 mm²

$$\sigma_t = \frac{P}{a} \Rightarrow 70 = \frac{P}{1350}$$

$$P = 70 \times 1350$$

$$\boxed{P = 94500\,N.}$$

For single transverse fillet Weld.

$$\sigma_t = \frac{1.414\,P_1}{h \times l_1}$$

$$70 = \frac{1.414 \times P_1}{15 \times 90}$$

$$P_1 = \frac{70 \times 15 \times 90}{1.414}$$

$$\boxed{P_1 = 66831.68\,N.}$$

Total Load = Load taken by transverse Weld + Load taken by double parallel Weld

$$P = P_1 + P_2$$

$$P_2 = P - P_1$$

$P_2 = 94500 - 66831.68$

$$\boxed{P_2 = 27668.32\ N}$$

FDB PSG 11.3 for double parallel weld

$$\tau = \frac{0.707 \times P_2}{h\, l_2}$$

$$l_2 = \frac{0.707 \times P_2}{h \times \tau} = \frac{0.707 \times 27668.32}{15 \times 55}$$

$l_2 = 23.71\ mm$ say

$$\boxed{l_2 = 24\ mm}$$

By adding weld constant.

$l_2 = 12.5 + 24$

$$\boxed{l_2 = 36.5\ mm}$$

RESULT:

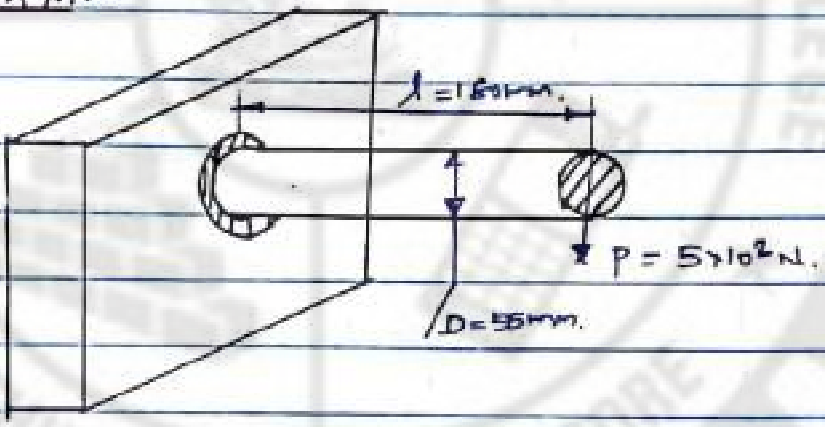
$l_2 = 36.5\ mm.$

## FILLET WELD FOR A CIRCULAR SHAFT.

Problem:

1. A Steel bar of 55mm diameter and 180 mm long is weld Perpendicular to a Solid plate by fillet weld around the Circumference of a bar. The bar is loaded with 5kN at the free end. Determine the size of the Weld, if the allowable stress in the Weld is 90N/mm²

GIVEN DATA:




Dia of bar (D) = 55mm

Length of bar (l) = 180mm

load (P) = 5kN ⇒ $5\times10^3$ N.

Stress in the Weld ($\sigma$) = 90 N/mm²

TO FIND:

Size of the Weld (h)

Solution:

Bending Moment ($M_b$) = Load x Length

$(M_b) = P \times l$.

$= 5 \times 10^3 \times 160$

$$\boxed{M_b = 900 \times 10^3 \text{ N.mm.}}$$

FDB pg PSG 11.3

$$\sigma_t = \frac{5.66 \times M_b}{h \times D^2 \times \pi}$$

$$90 = \frac{5.66 \times 900 \times 10^3}{h \times 55 \times \pi}$$

$$h = \frac{5.66 \times 900 \times 10^3}{90 \times 55^2 \times \pi}$$

$h = 5.9556 \text{ mm} \simeq 6 \text{ mm.}$

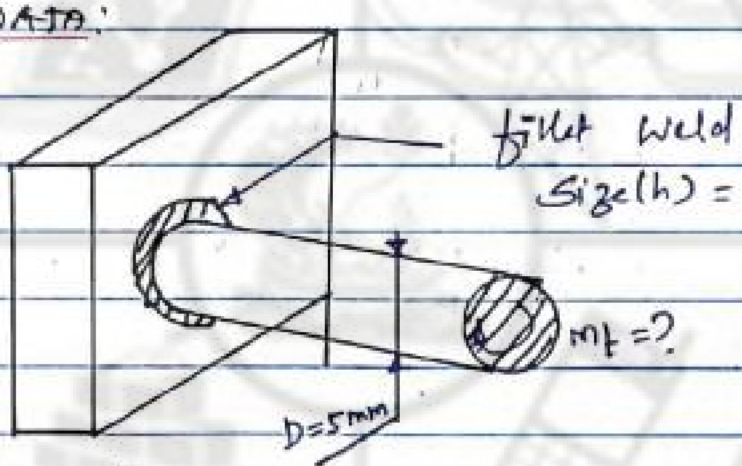
$$\boxed{h = 6 \text{ mm}}$$

RESULT:

$h = 6$ mm.

2. A Circular Steel bar of 5 mm diameter is Welded to a flat plate by 8mm fillet Weld. If the Permissible Shear Stress in the Weld material is 70 mpa. Calculate the Maximum torque that the Welded joint can Stand.

GIVEN DATA:




Diameter (D) = 5mm.

Size of the Weld (h) = 8mm.

Shear Stress ($\tau$) = 70 mpa ⇒ 70 N/mm²

To find:

Maximum torque.

Solution:

PDB pg 11·3

$$\tau = \frac{2.83 \times M_t}{h \times D^2 \times \pi} \Rightarrow 70 = \frac{2.83 \times M_t}{8 \times 5^2 \times \pi}$$

$$M_t = \frac{70 \times 8 \times 5^2 \times \pi}{2.83}$$

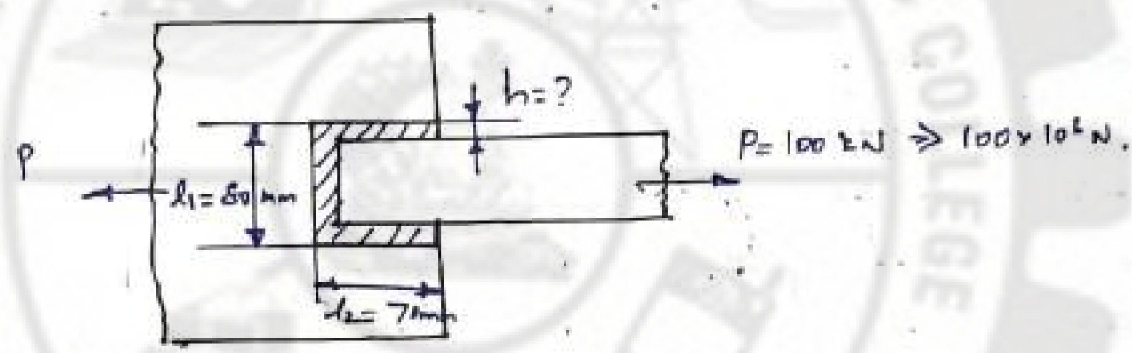
$$\boxed{M_t = 15541.44 \text{ N.mm}}$$

RESULT:

$M_t$ = 15541.44. N.mm.

## MODEL FOR FINDING SIZE OF WELD

(1) Calculate the leg size of the weld for the welding joint shown in figure. The static tensile load acting on the plates is 100 kN. Assume the permissible shear stress and tensile stress of the weld as 70 N/mm².




**GIVEN DATA:**

Load (P) = 100 kN ⇒ $100\times10^3$ N.

Permissible Shear Stress ($\tau$) = 70 N/mm²

Permissible Tensile Stress ($\sigma_t$) = 70 N/mm²

Length of transverse weld ($l_1$) = 80mm.

Length of parallel weld ($l_2$) = 70mm

**TO FIND:**

Size of weld(h)

**SOLUTION:**

PDB pg 11.3 for single transverse weld.

$$\sigma_t = \frac{1.414 \times P}{h \times l}.$$

$$70 = \frac{1.414 \times P_1}{h \times 80} \Rightarrow P_1 = \frac{70 \times h \times 80}{1.414}$$

$$\boxed{P_1 = 3960.39\,h \rightarrow (1)}$$

FDB pg 11.3 for double Parallel weld.

$$\tau = \frac{0.707 \times P_2}{h \times l_2} \Rightarrow 70 = \frac{0.707 \times P_2}{h \times 70}$$

$$P_2 = \frac{70 \times h \times 70}{0.707}$$

$$\boxed{P_2 = 6930.693\,h \rightarrow (2)}$$

$$P = P_1 + P_2 \rightarrow (3)$$

Sub (1) and (2) in (3) we get.

$$100 \times 10^3 = 3960.39\,h + 6930.693\,h$$

$$10891.083\,h = 100 \times 10^3$$

$$h = \frac{100 \times 10^3}{10891.083}$$

$$\boxed{\begin{array}{l} h = 9.18\,mm \text{ say} \\ h = 10\,mm \end{array}}$$

RESULT:

Size of the weld $(h) = 10\,mm$.

# UNIT - II

## DESIGN OF SHAFTS, KEYS AND COUPLINGS.

### Step by Step Procedure of Design of Protective type flange Couplings:

**STEP 1: To find Torque ($M_t$):**

FDB PSG 7.96

$$\boxed{M_t \text{ (or) } T = \frac{9.73 \times 10^6 \times kW}{n}}$$

**STEP 2: To find dia of Shaft (d):**

FDB PSG 7.57 (or) 7.23

$$\tau = \frac{16 \times M_t}{\pi \times d^3} \qquad M_t = \frac{\pi}{16} \times d^3 \times \tau.$$

$$\boxed{d^3 = \frac{16 \times M_t}{\pi \times \tau}}$$

**STEP 3: To find other dimensions from Empirical Relations:**

FDB PSG 7.134.

$D = 2d$     $n$ = 3 for d upto 40 mm
4 for d upto 100 mm
6 for d upto 180 mm.

$L = 1.5d$

$D_1 = 3d$

$D_2 = 4d$     $t_p = d/4$.

$t_f = d/2$

**STEP 4: To find width of the key (w) and thickness of key (h):**

FDB PSG 5.16

**STEP 5: To find dia of bolt ($d_1$):**

FDB PSG 7.135

$$T = n \times (\pi/4) \times d_1^2 \times \tau_b \times D_1/2$$

**STEP 6: Checking for bolt in crushing:**

FDB PSG 7.135

$$T = n \times d_1 \times t_f \times \sigma_{cb} \times (D_1/2)$$

**STEP 7: Checking for flange in shearing:**

FDB PSG 7.135

$$T = \pi \times D^2/2 \times \tau_f \times t_f.$$

**STEP 8: Checking for hub in shearing:**

FDB PSG 7.135

$$T = \frac{\pi}{16} \times \tau_h \times \frac{D^4 - d^4}{D}$$

**STEP 9: Checking for key in shearing:**

FDB PSG 7.135

$T = [illegible] \tau_k \times (d/2)$ $\quad T = l \times w \times \tau_k \times (d/2)$

**STEP 10: Checking for key in crushing:**

FDB PSG 7.135 (or) 7.136

$$T = t/2 \times \sigma_c \times d/2 \times l.$$

Problem 1

1. Design a protective type flange coupling two shafts to transmit 7.5kw at 720 rpm. The permissible shear stress for shaft, bolt and key materials is 33 N/mm². Permissible crushing strength for bolt and key material is 60 N/mm². and permissible shear stress for cast iron is 15 N/mm².

GIVEN DATA:

Power (kw) = 7.5 kw

Speed (n) = 720 rpm

Permissible shear stress for shaft, bolt and key

$$\tau_s = \tau_b = \tau_k = 33 \text{ N/mm}^2.$$

Permissible crushing strength for bolt and key material

$$\sigma_{cb} = \sigma_{ck} = 60 \text{ N/mm}^2$$

Permissible shear stress for cast iron flange

$$\tau_{cf} = 15 \text{ N/mm}^2.$$

TO FIND:

Design a protective type flange coupling.

SOLUTION:

STEP 1: To find torque ($M_t$) (or) T

FDB PSG 7.96

$$M_t \text{ (or) } T = \frac{9.73 \times 10^6 \times kw}{n}$$

$$= \frac{9.73 \times 10^6 \times 7.5}{720}$$

$\boxed{M_{t} = T = 101.35 \times 10^{3}\ N.mm}$

STEP 2: To find the dia of Shaft (d)

FDB PSG 7.67 (or) 7.02

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

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

$$d^3 = \frac{101.35 \times 10^3 \times 16}{\pi \times 33}$$

$$d^3 = 15.64 \times 10^3$$

$$\boxed{d = 25.007}$$

FDB PSG 7.25

$$\boxed{d = 28\ mm}$$

STEP 3: To find other dimensions from Emphirical Relations:

FDB PSG 7.134

1. $D = 2d \Rightarrow 2 \times 28 = 56mm$
2. $L = 1.5d \Rightarrow 1.5 \times 28 = 42mm$
3. $D_1 = 3d \Rightarrow 3 \times 28 = 84mm$
4. $D_2 = 4d \Rightarrow 4 \times 28 = 112mm$
5. $t_f = d/2 \Rightarrow 28/2 = 14mm$
6. $n = 3$ for dia upto 40mm
7. $t_p = d/4 \Rightarrow 28/4 = 7mm$

STEP 4: To find width of the key (w) and thickness of key (h):

FDB PSG 5.16.

For Shaft dia from 22mm To 30mm.

Width (b) (or) (w) = 8mm.

Thickness of key (h or t) = 7mm.

STEP 5: To find dia of bolt ($d_1$):

FDB PSG 7.135

$$T = n\left(\frac{\pi}{4}\right) \times d_1^2 \times \tau_b \times \left(\frac{D_1}{2}\right)$$

$$101.35 \times 10^3 = 3 \times \frac{\pi}{4} \times d_1^2 \times 33 \times \frac{84}{2}$$

$$101.35 \times 10^2 = 3265.68 \times d_1^2$$

$$d_1^2 = \frac{101.35 \times 10^3}{3265.68}$$

$$d_1^2 = 31.0348$$

$$d_1 = 5.57$$

FDB PSG 5.42

$d_1 = 6mm$

$d_1 = M6$

STEP 6: Checking for Bolt in Crushing:

FDB PSG 7.135

$$T = n \times d_1 \times t_f \times \sigma_{cb} \times (D_1/2)$$

$$101.35 \times 10^3 = 3 \times 6 \times 14 \times \sigma_{cb} \times 84/2$$

$$101.35 \times 10^3 = 10584 \times \sigma_{cb}$$

$$\sigma_{cb} = \frac{101.35 \times 10^3}{10584}$$

$$\sigma_{cb} = 9.57 \text{ N/mm}^2$$

∴ Therefore $9.57 \text{ N/mm}^2 < 60 \text{ N/mm}^2$. So the design is safe.

STEP 7: Checking for flange in shearing:

FDB PSG 7.135

$$T = \pi \times (D^2/2) \times \tau_f \times t_f$$

$$101.35 \times 10^3 = \pi \times 56^2/2 \times \tau_f \times 14$$

$$101.35 \times 10^3 = 68.96 \times 10^3 \times \tau_f$$

$$\tau_f = \frac{101.35 \times 10^3}{68.96 \times 10^3}$$

$$\tau_f = 1.46 \text{ N/mm}^2$$

Therefore $1.46 \text{ N/mm}^2 < 15 \text{ N/mm}^2$. So the design is safe.

STEP 8: Checking for hub in Shearing:

FDB PSG 7.135

$$T = \pi/16 \times \tau_h \times \left(\frac{D^4 - d^4}{D}\right)$$

$$101.35 \times 10^3 = \frac{\pi}{16} \times \tau_h \times \left[\frac{56^4 - 28^4}{56}\right]$$

$$101.35 \times 10^3 = 32.326 \times 10^3 \times \tau_h.$$

$$\tau_h = \frac{101.35\times10^3}{32.326\times10^3}$$

$$\tau_h = 3.135\ N/mm^2.$$

Therefore $3.135\ N/mm^2 < 15\ N/mm^2$. So the design is safe.

STEP 9: Checking for key in Shearing:

FDB PSG 7.135

$$T = l w \times \tau_k \times d/2$$

$$101.35\times10^3 = 8\times\tau_k\times 28/2 \times 42$$

$$101.35\times10^3 = \overset{4704}{\cancel{112}}\times\tau_k$$

$$\tau_k = \frac{101.35\times10^3}{4704}$$

$$\tau_k = 21.54\ N/mm^2.$$

Therefore $21.54\ N/mm^2 < 33\ N/mm^2$. So the design is safe.

STEP: 10: Checking for key in Crushing:

FDB ~~PSG~~ PSG 7.135 (or) 7.136

$$T = l\times\left(\frac{t}{2}\right)\times\sigma_c\times\left(\frac{d}{2}\right)$$

$$101.35\times10^3 = \frac{42\times7\times\sigma_{ck}\times28}{2\times2}$$

$$\sigma_{ck} = \frac{101.85 \times 10^3 \times 2 \times 2}{42 \times 7 \times 28}$$

$$\sigma_{ck} = 49.48 \ N/mm^2$$

Therefore 49.48 N/mm² < 60 N/mm². So the design is safe.

RESULT:

(1) Torque ($M_t$ (or) T) = $101.85 \times 10^3$ N-mm.
(2) Dia of Rod (d) = 26 mm.
(3) Width (w (or) b) = 8 mm.
(4) Thickness of key (h or t) = 7 mm.
(5) Dia of bolt ($d_1$) = $M_6$.
(6) Checking for B.H in crushing ($\sigma_{cb}$) = 9.57 N/mm²
(7) Checking for flange in shearing ($\tau_f$) = 1.46 N/mm²
(8) Checking for hub in shearing ($\tau_h$) = 3.135 N/mm²
(9) Checking for key in shearing ($\tau_k$) = 21.54 N/mm²
(10) Checking for key in crushing ($\sigma_{ck}$) = 49.46 N/mm².

## Step by Step Procedure for Design a Knuckle Joint:

**STEP 1: To find the dia of Rod (d):**

FDB PSG 7.1

Tensile Stress = $\frac{\text{Load}}{\text{Area of rod}}$

$$\boxed{\sigma_t = P/a.}$$

Area of Rod (a) = $\pi/4 \times d^2$

$$\sigma_t = \frac{P}{\frac{\pi}{4} \times d^2}$$

FDB PSG 7.87

$$\boxed{\sigma_t = \frac{4P}{\pi d^2}}$$

**STEP 2: To find Emphirical Relations:**

FDB PSG 7.139

**STEP 3: To find other Equations:**

FDB PSG 7.139

**STEP 4: Check for failure:**

Checking for ~~the knuckle pin~~ failure of the knuckle pin in Shear.

Induced Shear Stress = $\frac{\text{Load}}{\text{Shearing area of knuckle pin.}}$

Area (A) = $2 \times \frac{\pi}{4} \times d_1^2$

$$\tau = \frac{P}{A}$$

$$\tau = \frac{P}{2 \times \frac{\pi}{4} \times d_1^2}$$

$$\boxed{\tau = \frac{4P}{2 \times \pi \times d_1^2}}$$

Step 5: Checking for failure of the knuckle pin in crushing:

Single Eye

Induced crushing stress = $\frac{\text{Load}}{\text{Area}}$

$$\sigma_c = \frac{P}{A}.$$

$$\text{Area}(A) = d_1 \times t.$$

$$\boxed{\sigma_c = \frac{P}{d_1 \times t.}}$$

Step 6: Checking for failure of knuckle in crushing

Double Eye.

Induced crushing stress = $\frac{\text{Load}}{\text{Area.}}$

$$\sigma_c = \frac{P}{A.}$$

$$\text{Area}(A) = 2 \times d_1 \times t_1$$

$$\boxed{\sigma_c = \frac{P}{2 d_1 t_1.}}$$

Step 7: Checking for failure of the single Eye in tension.

Induced tensile stress = $\frac{\text{Load}}{\text{Area}}$

$$\sigma_t = \frac{P}{A}$$

$$\text{Area}(A) = (d_2 - d_1)t$$

$$\boxed{\sigma_t = \frac{P}{(d_2-d_1)t}}$$

Step 8: Checking for the failure of the single eye in shear.

Induced shear stress = $\frac{\text{Load}}{\text{Area}}$

$$\tau = \frac{P}{A}$$

$$\text{Area}(A) = (d_2 - d_1)t$$

$$\boxed{\tau = \frac{P}{(d_2-d_1)t}}$$

Step 9: Checking for the failure of the double Eye in tension.

Induced tensile stress = Load/Area.

$\sigma_t = P/A$.

$$\text{Area}(A) = 2\times(d_2 - d_1)t$$

$$\boxed{\sigma_t = \frac{P}{2\times(d_2-d_1)t_1}}$$

Problem:

(1) Design a knuckle Joint to transmit a load of 60kN. Take allowable Stresses as 60 MPa in tension, 75 MPa in Compression and 40 MPa in Shear.

Given data:

Load (P) = 60 kN = $60 \times 10^3$ N.

Tensile Stress ($\sigma_t$) = 60 MPa = 60 N/mm²

Compression (or) Crushing Stress ($\sigma_c$) = 75 MPa = 75 N/mm²

Shear Stress ($\tau$) = 40 MPa = 40 N/mm².

To find:

Design a knuckle Joint.

Solution:

Step 1: To find dia of Rod (d):

FDB pg 7.1

$$\text{Tensile Stress} = \frac{\text{Load}}{\text{Area.}}$$

$$\sigma_t = \frac{P}{A} \Rightarrow \frac{4P}{\pi \times d^2}$$

$$60 = \frac{4 \times 60 \times 10^3}{\pi \times d^2}$$

$$d^2 = \frac{4 \times 60 \times 10^3}{\pi \times 60}$$

$$d^2 = 1273.239$$

$d = \sqrt{1273.239}$

$$\boxed{\begin{array}{l} d = 35.68 \text{ mm say} \\ d = 40 \text{ mm.} \end{array}}$$

Step 2: To find Emphirical Relations:

FDB PSG 7.139

| S.No | Nomenclature | Equation | Values |
|---|---|---|---|
| 1 | Diameter of the Pin ($d_1$) | $d_1 = d \Rightarrow d_1 = 40$ | 40 mm |
| 2. | Outer dia of the Eye ($d_2$) | $d_2 = 2d \Rightarrow 2 \times 40$ | 80 mm |
| 3. | diameter of Pin head ($d_3$) | $d_3 = 1.5d \Rightarrow 1.5 \times 40$ | 60 mm |
| 4. | Thickness of the Eye ($t$) | $t = 1.25d \Rightarrow 1.25 \times 40$ | 50 mm |
| 5. | Thickness of the fork ($t_1$) | $t_1 = 0.75d \Rightarrow 0.75 \times 40$ | 30 mm |
| 6. | Thickness of Pin head ($t_2$) | $t_2 = 0.5d \Rightarrow 0.5 \times 40$ | 20 mm. |

Step 3: To find Other Equations:

FDB PSG 7.139

| S.No | Nomenclature | Equation | Values |
|---|---|---|---|
| 1. | Size across flats of forked end | $1.2d \Rightarrow 1.2 \times 40$ | 46 mm |
| 2. | Length of forked end | $1.2d \Rightarrow 1.2 \times 40$ | 46 mm |
| 3. | Size across Rod end | $1.1d \Rightarrow 1.1 \times 40$ | 44 mm |
| 4. | Length of Rod end | $1.2d \Rightarrow 1.2 \times 40$ | 46 mm |
| 5. | Length of Single eye | $4d \Rightarrow 4 \times 40$ | 160 mm |
| 6. | Length of double eye | $4.5d \Rightarrow 4.5 \times 40$ | 180 mm |

Step 4: Check for failures:

Checking for the failure of knuckle pin in shear.

$$\text{Induced Shear Stress} = \frac{\text{Load}}{\text{Area}}$$

$$\tau = P/A.$$

$$\text{Area }(A) = 2 \times \frac{\pi}{4} \times d_1^2 \Rightarrow 2 \times \frac{\pi}{4} \times 40^2$$

$$A = 2513.27 \text{ mm}^2$$

$$\tau = \frac{60 \times 10^3}{2513.27}$$

$$\boxed{\tau = 23.87 \text{ N/mm}^2}$$

Therefore 23.87 N/mm² < 40 N/mm². So the design is Safe.

Step 5: Checking for failure of knuckle pin in crushing single eye.

$$\text{Induced crushing Stress} = \frac{\text{Load}}{\text{Area}}$$

$$\sigma_c = \frac{P}{A}$$

$$\text{Area}(A) = d_1 \times t$$

$$= 40 \times 50$$

$$A = 2000 \text{ mm}^2$$

$$\sigma_c = \frac{60 \times 10^3}{2000}$$

$$\boxed{\sigma_c = 30 \text{ N/mm}^2}$$

Therefore 30 N/mm² < 75 N/mm². So the design is safe.

Step 6: Checking for failure of the ~~single~~ eye ~~in tension~~. Knuckle pin in crushing double eye

Induced crushing stress = $\frac{\text{Load}}{\text{Area}}$

$$\sigma_c = \frac{P}{A}$$

$$\text{Area}(A) = 2 \times d_1 \times b_1$$

$$= 2 \times 40 \times 30$$

$$A = 2400 \text{ mm}^2$$

$$\sigma_c = \frac{60 \times 10^3}{2400}$$

$$\boxed{\sigma_c = 25 \text{ N/mm}^2}$$

Therefore $25 \text{ N/mm}^2 < 75 \text{ N/mm}^2$. So the design is safe.

Step 7: Checking for failure of the single eye in tension:

Induced tensile stress = Load / Area.

$$\sigma_t = P/A.$$

$$\text{Area}(A) = (d_2 - d_1)t.$$

$$= (80 - 40)50.$$

$$A = 2000 \text{ mm}^2$$

$$\sigma_t = \frac{60 \times 10^3}{2000}$$

$$\boxed{\sigma_t = 30 \text{ N/mm}^2}$$

Therefore $30 \text{ N/mm}^2 < 60 \text{ N/mm}^2$. So the design is safe.

Step 8: Checking for failure of single eye in shear:

Induced Shear Stress = $\frac{Load}{Area}$

$$\tau = \frac{P}{A}$$

$$Area\,(A) = (d_2 - d_1)t$$
$$= (80 - 40) \times 50$$
$$A = 2000\,mm^2$$
$$\tau = \frac{60 \times 10^3}{2000}$$
$$\boxed{\tau = 30\,N/mm^2}$$

Therefore $30\,N/mm^2 < 40\,N/mm^2$. So the design is safe.

Step 9: Checking for failure of double eye in tension:

Induced tensile Stress = $\frac{Load}{Area}$

$$\sigma_t = \frac{P}{A}$$

$$Area\,(A) = 2(d_2 - d_1)t_1$$
$$= 2 \times (80 - 40)\,30$$
$$A = 2400\,mm^2$$
$$\sigma_t = \frac{60 \times 10^3}{2400}$$
$$\boxed{\sigma_t = 25\,N/mm^2}$$

Therefore $25 N/mm^2 < 40 N/mm^2$. So the design is safe.

Result:

1. Dia of Rod $(d)$ = 40 mm.
2. Dia of Pin $(d_1)$ = 40 mm
3. outer dia of Eye $(d_2)$ = 80 mm
4. dia of Pin head $(d_3)$ = 60 mm.
5. Thickness of Eye $(t)$ = 50 mm
6. Thickness of fork $(t_1)$ = 30 mm
7. Thickness of Pin head $(t_2)$ = 20 mm.
8. failure of knuckle pin in shear $(\tau)$ = $23.67 N/mm^2$
9. failure of knuckle pin in crushing single eye $(\sigma_c)$ = $30 N/mm^2$
10. failure of knuckle pin in crushing double eye $(\sigma_c)$ = $25 N/mm^2$
11. failure of single eye in tension $(\sigma_t)$ = $30 N/mm^2$
12. failure of single eye in shear $(\tau)$ = $30 N/mm^2$
13. failure of double eye in tension $(\sigma_t)$ = $25 N/mm^2$

# UNIT - III

## DESIGN OF FLAT AND V- BELTS.

### Step by Step procedure for flat Belts:

**STEP 1: To find Speed Ratio (i):**

FDB PSG 7.61 & 7.74

$$i = \frac{D}{d} = \frac{n_1}{n_2}$$

**STEP 2: To find Belt Speed (V):**

FDB PSG 8.15

$$V = \frac{\pi d n_1}{60 \times 1000}$$

(or)

$$V = \frac{\pi D n_2}{60 \times 1000}$$

**STEP 3: To find arc of Contact ($\theta$)**

FDB PSG 7.54.

For open Belt drive:

$$\text{Arc of Contact } (\theta) = 180° - \frac{(D-d) \times 60°}{C}$$

For Crossed Belt drive:

$$\theta = 180° + \frac{(D+d) \times 60°}{C}$$

STEP 4: To find Correction load (Pd)

$$\text{Correction load} = \frac{\text{Given Power} \times \text{Load Correction factor} \times \text{Arc of Contact factor}}{\text{Pulley Correction factor}}$$

FDB PSG 7.53 – Load Correction factor ($k_s$)

FDB PSG 7.54 – Arc of Contact factor ($k_\alpha$)

– Pulley Correction factor ($k_d$)

STEP 5: To find ~~Load~~ Rating: Load Rating For mm:

FDB PSG 7.54.

STEP 6: To find Load Rating at Velocity V(m/s):

FDB PSG 7.54.

$$\text{Load Rating at } v \text{ m/s} = \text{Load Rating at 10 m/s} \times V/10$$

STEP 7: To find Millimeter plies of belt:

FDB PSG 7.54.

$$\text{Millimeter plies of Belt} = \frac{\text{Corrected Load}}{\text{Load Rating Per mm Per ply at Belt Speed.}}$$

STEP 8: To find Belt Width (b):

$$\text{Belt Width (b)} = \frac{\text{Millimeter plies of belt}}{\text{No of plies.}}$$

If No of Plies Not given Assume.

6 (or) 5 (or) 4. And also

Standard belt Width (bstd) FDB PSG 7.52 table.

STEP 9: To find pully Width (B):

FDB psg 7.54 table.
And also Standard pully width is taken from psg 7.55

STEP 10: To find Belt Length (L):

FDB psg 7.53

For open Belt drive:

$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$$

For Crossed Belt drive:

$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D+d)^2}{4C}.$$

STEP 11: ~~8~~ RESULT:

Sketch & Specifications.

## OPEN BELT DRIVE

### Type 1 : DIAMETER IS GIVEN.

1. Design a flat belt drive to transmit 22.5 kW at 740 rpm to an alluminium rolling Machine. The Speed Ratio is 3. The distance between the pulleys is 3 m. Diameter of rolling Machine pulley is 1.2 m. Use Manufacturer's data.

Assume the following.

1. Load Correction factor $k_c = 1.5$
2. Pulley Correction factor $k_d = 0.9$
3. Open belt drive.
4. Use fabric - high Speed duck Belt.

**GIVEN DATA:**

Type of Belt - Open Belt drive.

Power transmitted ($P_o$) = 22.5 kW, Speed Ratio ($i$) = 3

Speed of Smaller pulley (or) driver Pulley ($n_1$) = 740 rpm.

Centre distance between pulleys ($C$) = 3 m

$= 3 \times 10^3$ mm.

Diameter of Rolling Machine Pulley ($D$) = 1.2 m

$= 1.2 \times 10^3$ mm.

Load Correction factor Or Service factor ($k_c$) = 1.5

Pulley Correction factor ($k_d$) = 0.9

Use fabric high Speed duck belt.

**TO FIND:**

Design a flat belt drive.

## SOLUTION:

STEP 1: To find Speed Ratio (i):

FDB pg 7.61 & 7.74.

$$i = \frac{D}{d} = \frac{n_1}{n_2}$$

$$3 = \frac{1.2 \times 10^3}{d}$$

$$d = \frac{1.2 \times 10^3}{3}$$

$$\boxed{d = 400 mm.}$$

FDB pg. 7.54.

$$d_{std} = 400 mm.$$

$$i = \frac{n_1}{n_2}$$

$$3 = \frac{740}{n_2}$$

$$n_2 = \frac{740}{3}$$

$$\boxed{n_2 = 246.67 rpm.}$$

STEP 2: To find belt Speed (V):

FDB PSG 8.54 15

$$V = \frac{\pi D n_2}{60\times1000} = \frac{\pi \times 1.2\times10^3 \times 246.67}{60\times1000}$$

$$\boxed{V = 15.498 \text{ m/s}}$$

STEP 3: To find arc of Contact ($\theta$)

FDB PSG 7.54 for open Belt drive.

$$\theta = 180 - \frac{(D-d)\times60}{C}$$

$$= 180 - \frac{(1.2\times10^3 - 400)60}{3\times10^3}$$

$$= 180 - 16$$

$$\boxed{\theta = 164^\circ}$$

~~To~~ STEP 4: To find Correction Load ($P_d$)

$$\text{Correction Load} = \frac{\text{Given Power} \times \text{Load Correction factor} \times \text{Arc of Contact factor}}{\text{Pulley Correction factor.}}$$

To find Arc of Contact factor:

For $\theta = 164^\circ$ FDB PSG 7.54

| | | | | |
|---|---|---|---|---|
| For $\theta$ | $160^\circ$ | $170^\circ$ | $10^\circ$ | $4^\circ$ |
| - | 1.08 | 1.04 | 0.04 | 0.016. |

$164^\circ = 160^\circ + 4^\circ$

Arc of Contact factor $= 1.08 + 0.016 \Rightarrow 1.096$.

$$\Rightarrow \frac{22.5 \times 1.5 \times 1.096}{0.9}$$

$$P_d = 41.1\,kw$$

STEP 5: To find Load Rating Per mm:

FDB PSG 7.54. For fabric High Speed duck belt.

at V 10 m/s = 0.023 kw

STEP 6: To find Load Rating at V m/s:

FDB PSG 7.54

$$\left.\begin{array}{l}\text{Load Rating at}\\ V\ m/s\end{array}\right\} = \text{Load Rating at } V\ 10\ m/s \times \frac{V}{10}$$

$$= \frac{0.023 \times 15.498}{10}$$

$$= 0.0356$$

STEP 7: To find Millimeter plies of Belt:

FDB PSG 7.54

$$\text{Millimeter plies of Belt} = \frac{\text{Correction Load } (P_d)}{\text{Load Rating per mm ply at Belt Speed}}$$

$$= \frac{41.1}{0.0356}$$

$\boxed{\text{Millimeter plies of Belt} = 1153.02}$

STEP 8: To find Belt Width (b)

$$\text{Belt Width (b)} = \frac{\text{Millimeter Plies of Belt}}{\text{No of plies}}$$

Assume No of Plies = 6

$$= \frac{1153.02}{6}$$

$$\boxed{b = 192.17\,mm}$$

FDB PSG 7.52

$$\boxed{b_{std} = 200\,mm}$$

STEP 9: To find pulley width (B):

FDB PSG 7.54 for Belt Width (b) - 200 mm

$$B = 200 + 25$$

$$B = 225\,mm$$

FDB PSG 7.55

$$\boxed{B_{std} = 250\,mm}$$

STEP 10: To find Belt Length (L):

FDB PSG 7.53 For open Belt drive

$$L = 2c + \left[\frac{\pi}{2}(D+d)\right] + \frac{(D-d)^2}{4c}$$

$$= 2 \times 3000 + \frac{\pi}{2}(1200+400) + \frac{(1200-400)^2}{4 \times 3000}$$

$$= 6000 + 2513.27 + 53.33$$

$$\boxed{\begin{aligned} L &= 8566.60\text{mm say} \\ L &= 8570\text{mm} \end{aligned}}$$

To find correction length(Lc) ($L_c$)

FDB PSG 7.53 for 6 plies belt length is cut by 1%.

Therefore.

$$L_c = L \times (100-1)/100$$

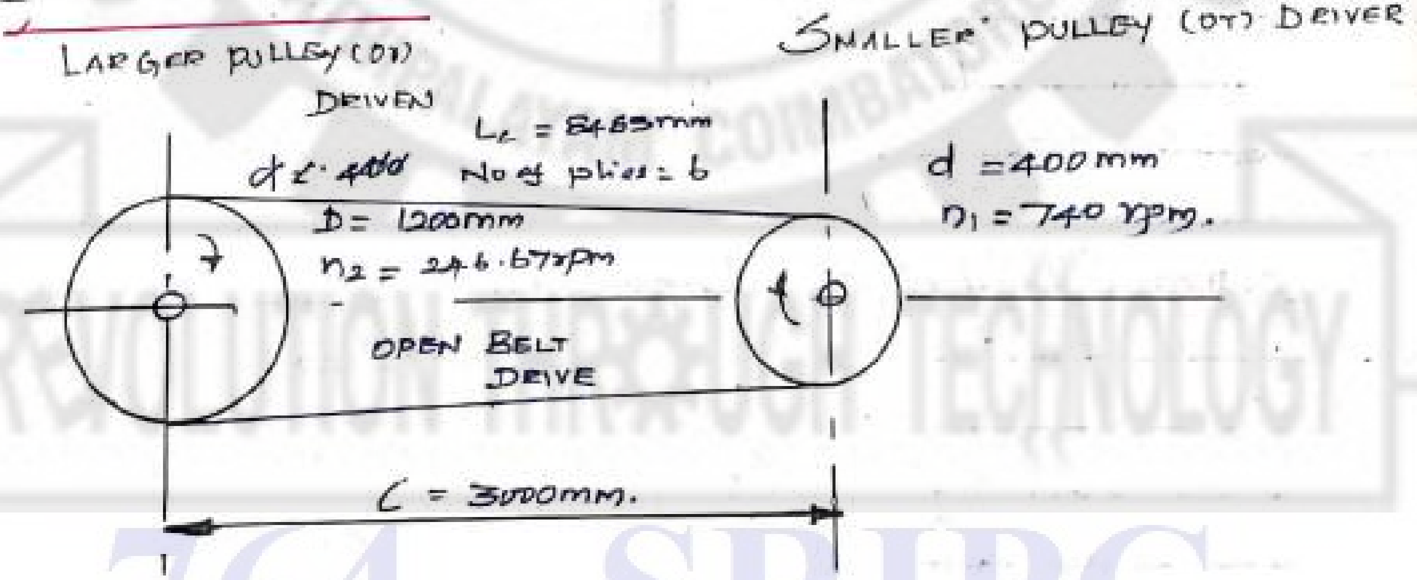
$$= 8570 \times (100-1)/100$$

$$= 8570 \times 99/100.$$

$$L_c = 8484.3 \text{ say}$$

$$\boxed{L_c = 8465\text{mm.}}$$

STEP 11: RESULT:



1. Type of drive - Open Belt drive.
2. Dia of Smaller pulley (d) = 400mm
3. Speed of Smaller pulley ($n_1$) = 740rpm.
4. Dia of Larger pulley (ϕD) = 1200mm.
5. Speed of Larger pulley ($n_2$) = 246.67rpm.
6. Centre distance between pulleys (c) = 3000 mm.
7. Speed of belt (V) = 15.498 m/s
8. Standard Belt width (b) = 200mm.
9. Standard pulley width (B) = 250mm.
10. Correction length ($L_c$) = 84.65mm.
11. No of plies = 6.

# MODEL FOR FINDING BELT TENSIONS:

2. Design a fabric belt to transmit 11kw at 420rpm. of an Engine to a line Shaft at 20 RPS. Engine pulley diameter is 550mm. and the Centre distance is 2m. Assume Co-efficient of friction $\mu = 0.2$ and number of Plies = 6.

GIVEN DATA:

Type of Belt drive — Open belt drive.

Power transmitted (P) = 11kw

Speed of Smaller pulley $(n_1)$ = 20 RPS
= 20×60 = 1200 rpm.

Speed of larger pulley $(n_2)$ = 420 rpm.

Diameter of Engine pulley (or) larger pulley (D) = 550mm.

Centre distance (C) = 2m → $2\times10^3$ mm

Co-efficient of friction ($\mu$) = 0.2

No of plies = 6.

TO FIND:

Design a flat Belt drive

SOLUTION:

STEP 1: To find Speed Ratio (i):

PSG 7.61 & 7.74.

$$i = \frac{D}{d} = \frac{n_1}{n_2}.$$

$$i = \frac{n_1}{n_2} = \frac{1200}{420}$$

$$\boxed{i = 2.857}$$

$i = \frac{D}{d} \Rightarrow 2.857 = \frac{550}{d}$

$\therefore d = 550/2.857$

$d = 192.50$ mm.

FDB PSG - 7.54.

$\boxed{d_{std} = 200mm}$

STEP 2: To find belt Speed (V) FDB PSG : 8.15

$$V = \frac{\pi \times D \times n_2}{60 \times 1000} = \frac{\pi \times 550 \times 420}{60 \times 1000}$$

$$\boxed{V = 12.095 \ m/s}$$

STEP 3: To find Arc of Contact ($\theta$):

FDB PSG : 7.54 for Open belt drive.

$$\theta = 180 - \frac{(D-d) \times 60}{C}$$

$$\theta = 180 - \frac{(550 - 200) \times 60}{2000}$$

$$\theta = 180 - 10.5 \Rightarrow 169.5^\circ \text{ Say}$$

$$\boxed{\theta = 170^\circ}$$

STEP 4: To find Correction Load (Pd).

Correction Load = $\frac{\text{Given Power} \times \text{Load Correction factor} \times \text{Arc of contact factor}}{\text{pulley Correction factor.}}$

$$= \frac{P_0 \times k_s \times k_\theta}{k_d}.$$

To find Arc of Contact factor ($k_\theta$).

FDB psg 7.54 For $\theta = 170°$

$k_\theta = 1.04$.

To find Load Correction factor (or) Service factor ($k_c$).

FDB psg 7.53 For Steady load.

$k_c = 1.2$

To find pulley Correction factor: ($k_d$)

Assume pulley Correction factor if not Mentioned......

$k_d = 1$.

$= 11 \times 1.2 \times 1.04$

$P_d = 13.728$ kw.

STEP 5: To find Load Rating per mm:

FDB psg 7.54.

Load Rating per mm, width per ply at 180° arc of contact at 10 m/s.

High Speed duck belt $= 0.023$ kw.

STEP 6: To find Millimeter plies of Belt:

FDB psg 7.54.

$$\text{Millimeter plies of Belt} = \frac{\text{Correction Load } (P_d)}{\text{Load Rating per mm ply at Belt speed}}$$

$=$

Step 6: To find Load Rating at V m/s:

FDB pg 7.54.

Load Rating = Load Rating at 10 m/s × V/10

= 0.023 × 12.095/10

Load Rating = 0.0278.

Step 7: To find Millimeter plies of Belt:

FDB psg: 7.54.

$$\text{Millimeter plies of Belt} = \frac{\text{Correction Load}}{\text{Load Rating per mm ply at Belt Speed}}$$

$$= \frac{13.728}{0.0278}$$

$$= 493.46.$$

Step 8: To find Belt Width (b):

$$\text{Belt Width (b)} = \frac{\text{Millimeter plies of Belt}}{\text{No of plies}}$$

$$= \frac{493.48}{6}$$

$$(b) = 82.247 \text{ mm}$$

FDB psg 7.52

$b_{std}$ = 100 mm.

Step 9: To find pulley width (B):

FDB psg 7.54 for Belt Width $b_{std}$ = 100 mm.

B = 100 + 13 = 113 mm

FDB psg 7.52

$B_{std}$ = 125 mm.

## STEP 10: To find Belt Length (L):

FDB pg 7.53 for open Belt drive.

$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$$

$$= 2 \times 2000 + \frac{\pi}{2}(550+200) + \frac{(550-200)^2}{4 \times 2000}$$

$$= 4000 + 1178.097 + 15.3125$$

$$= 5193.40 \text{ say}$$

$$L = 5200 \text{ mm.}$$

## To find Correction length ($L_c$)

FDB pg 7.53 For 6 plies. belt lengths cut by 1%.

Therefore.

$$L_c = L \times (100-1)/100$$

$$= 5200 \times \left(\frac{100-1}{100}\right)$$

$$= 5200 \times 99/100.$$

$$= 5148 \text{ mm say.}$$

$$L_c = 5150 \text{ mm.}$$

Adding Steps to find Belt tension

## STEP 11: To find Torque ($M_t$ (or) $T$):

FDB pg 7.56

$$M_t \text{ (or) } T = \frac{9.73 \times 10^6 \times kW}{n_1} = \frac{9.73 \times 10^6 \times 11}{1200}$$

$M_t$ (or) $T = 89.19 \times 10^3$ N.mm.

STEP 12: To find tangential force (P):

FDB PSG 7.95 & 7.106 & 8.57.

$$F_t \text{ (or) } P = \frac{2 \times M_t}{d} = \frac{2 \times 89.19 \times 10^3}{200}$$

$$P = 891.91 \text{ N.}$$

STEP 13: To find Belt Tensions:

FDB PSG 9.16

$$T_1 = P\left[\frac{e^{\mu\alpha}}{e^{\mu\alpha} - 1}\right]$$

$\theta = 170°$

$$= \frac{170 \times \pi}{180}$$

$\alpha = 2.967$ rad.

$$= 891.91 \times \left[\frac{e^{0.2 \times 2.967}}{e^{0.2 \times 2.967} - 1}\right]$$

$$= 891.91 \times \frac{1.810}{1.810 - 1} \Rightarrow 891.91 \times \frac{1.810}{0.810}$$

$$= 891.91 \times 2.23$$

$$T_1 = 1993.15 \text{ N.}$$

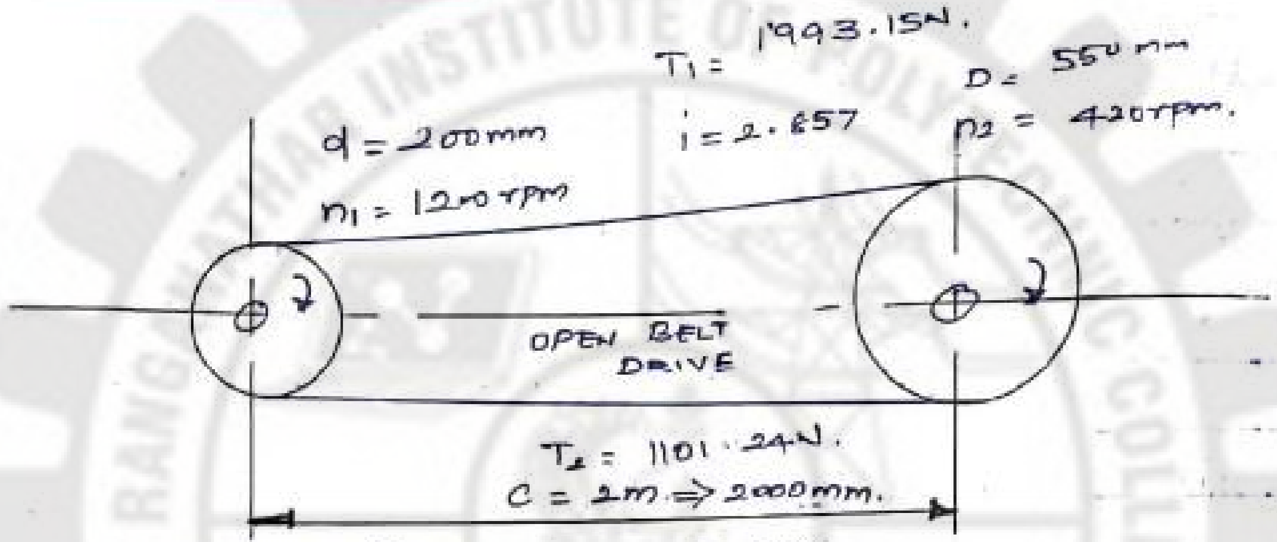
FDB PSG 9.16

$$T_2 = T_1 - P$$

$$= 1993.15 - 891.91$$

$$T_2 = 1101.24 \text{ N.}$$

## STEP 14: RESULT:




(1) Speed Ratio (i) = 2.657

(2) Dia of Smaller pulley (d) = 200mm.

(3) Belt Speed (V) = 12.095 m/s

(4) Arc of Contact ($\theta$) = 170°

(5) Correction Load ($P_d$) = 13.728 kW

(6) Load Rating = 0.0278

(7) Millimeter plies of Belt = 493.48

(8) Belt Width (b) = 100mm.

(9) Pulley Width (B) = 125mm

(10) Length of Belt (L) = 5200 mm.

(11) Correction length ($L_c$) = 5150mm.

(12) Torque ($M_t$ or T) = $89.19 \times 10^3$ N.mm.

(13) Tangential force (P) = 891.91N.

(14) ~~Tension~~ Belt tension ($T_1$) = 1993.15N.

(15) Belt tension ($T_2$) = 1101.24N.

## TYPE 2: DIAMETER IS NOT GIVEN.

1. Select a flat belt from Manufacturers Catalogue to transmit power of 15 kW at 1200 rpm. The Speed of the driven pulley is 450 rpm. Maximum Centre distance between the shaft is 2m. Assume Steady load.

GIVEN DATA:

Power transmitted ($P_0$) = 15kW

Speed of Smaller pulley ($n_1$) = 1200rpm

Speed of larger pulley ($n_2$) = 450rpm.

Centre distance (C) = 2m ⇒ $2\times10^3$ mm.

TO FIND:

Design a flat Belt drive.

SOLUTION:

Assume open flat Belt drive.

STEP 1: To find Speed Ratio (i):

FDB psg 7.61 & 7.74

$$i = \frac{D}{d} = \frac{n_1}{n_2}.$$

$$i = \frac{1200}{450} = 2.67 \qquad i = 2.67$$

To find the diameter of pulleys:

$$i = \frac{D}{d}.$$

FDB psg 7.53 Assume the Belt Velocity From 17.5m/s To 22.5m/s

V = 16 m/s.

FDB PSG 8.15

$$V = \frac{\pi D n_2}{60 \times 1000}$$

$$16 = \frac{\pi \times D \times 450}{60 \times 1000} \Rightarrow D = \frac{16 \times 60 \times 1000}{\pi \times 450}$$

$$D = 763.94 \text{ mm}$$

FDB PSG 7.54

$$D_{std} = 800 \text{mm.}$$

$$i = \frac{D}{d} \Rightarrow d = \frac{D}{i}$$

$$i \times d = \frac{800}{2.67}$$

$$d = 299.62 \text{ mm.}$$

FDB PSG 7.54

$$d_{std} = 315 \text{mm.}$$

STEP 2: To find belt Speed (V)

FDB PSG 8.15

$$V = \frac{\pi \times D \times n_2}{60 \times 1000} = \frac{\pi \times 800 \times 450}{60 \times 1000}$$

$$V = 18.849 \text{ m/s.}$$

STEP3: To find Arc of Contact ($\theta$):

FDB psg 7.54 for open belt drive.

$$\theta = 180^\circ - \frac{(D-d)\times 60}{C}$$

$$= 180^\circ - \frac{(800-315)\times 60}{2000}$$

$$= 180^\circ - 14.55$$

$$\theta = 165.45^\circ \approx \text{say}.$$

$$\theta = 166^\circ$$

STEP4: To find Correction load ($P_d$):

$$\text{Correction Load} = \frac{\text{Given power} \times \text{Load Correction factor} \times \text{Arc of Contact factor}}{\text{pulley Correction factor}}$$

$$P_d = \frac{P_R \times K_s \times K_\alpha}{K_d}$$

To find arc of contact factor: ($K_\alpha$)

FDB psg 7.54 for 166°.

| | | | | | |
|---|---|---|---|---|---|
| Degree | 160° | 170 | 10 | 1° | 6° |
| Value | 1.08 | 1.04 | 0.04 | 0.004 | 0.024 |

$166^\circ \Rightarrow 160^\circ + 6^\circ$

$= 1.08 + 0.024$

$166^\circ = 1.104.$

To find Load Correction factor.

FDB psg 7.53 for Steady load.

$K_s = 1.2$

If pulley correction factor is not given neglect it.

$= 15 \times 1.104 \times 1.2$

$P_d = 19.872$ kW

## STEP 5: To find Load Rating per mm:

FDB psg 7.54.

Load Rating per mm width per ply at 180° arc of Contact at 10 m/s

For Hi-speed duck Belt = 0.023 kW.

## Step 6: To find Load Rating at v m/s:

FDB psg 7.54

$$\text{Load Rating} = \text{Load Rating at } 10\text{ m/s} \times \frac{V}{10}$$

$$= \frac{0.023 \times 18.829}{10}$$

Load Rating = 0.0433.

## STEP 7: To find Millimeter plies of Belt:

FDB psg 7.54

$$\text{Millimeter plies of Belt} = \frac{\text{Correction load}}{\text{Load Rating Per mm ply at belt Speed.}}$$

$$= \frac{19.872}{0.0433}$$

$$= 458.37$$

STEP 8: To find Belt Width (b):

$$\text{Belt Width (b)} = \frac{\text{Millimeter plies of Belt}}{\text{No of plies.}}$$

Assume No of plies = 6.

$$= \frac{458.37}{6}$$

$$b = 76.39 \text{ mm.}$$

FDB pg 7.52

$$b_{std} = 100 \text{ mm.}$$

STEP 9: To find pulley width (B):

FDB pg 7.54 for Belt width (b) = 100 mm.

$$B = 100 + 13$$

$$B = 113 \text{ mm}$$

FDB pg 7.52

$$B_{std} = 125 \text{ mm.}$$

STEP 10: To find Belt Length (L):

FDB pg 7.53 for open Belt drive.

$$L = 2c + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4c}.$$

$$= 2 \times 2000 + \frac{\pi}{2}(800+315) + \frac{(800-315)^2}{4 \times 2000}$$

$$= 4000 + 1751.43 + 29.40$$

$$= 5780.83 \text{ mm. Say}$$

$$L = 5785 \text{ mm.}$$

To find Correction Length ($L_c$)

PDB pg 7.53 For 6 plies belt length cut by 1%.
Therefore.

$$L_c = L \times (100-1)/100$$

$$= 5765 \times (100-1)/100$$

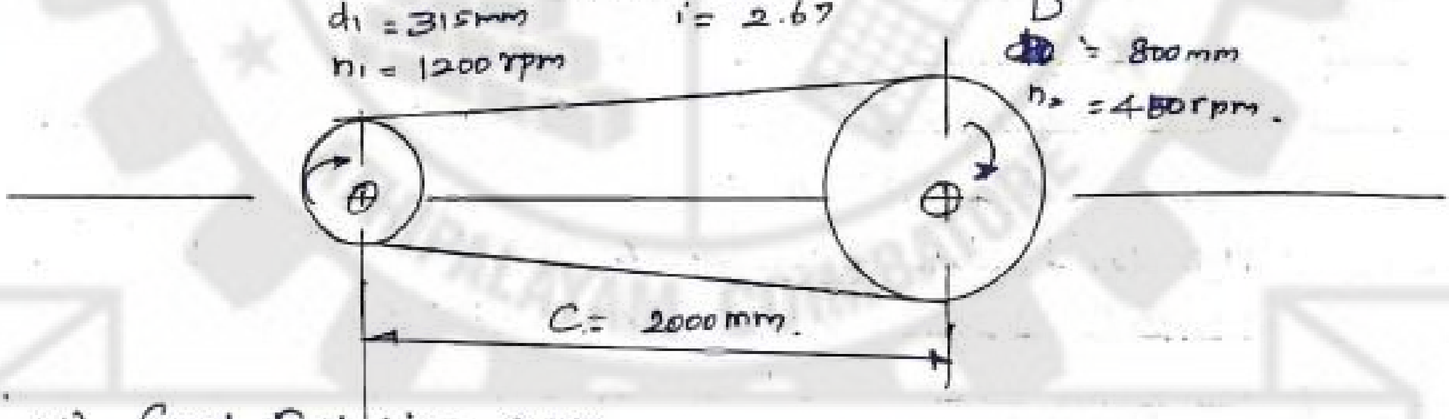
$$L_c = 5785 \times 99/100$$

$$L_c = 5727.15 \text{ mm say}.$$

$$L_c = 5730 \text{ mm}.$$

STEP 11: RESULT:

OPEN BELT DRIVE




(1) Speed Ratio (i) = 2.67
(2) Smaller pulley diameter ($d_1$) = 315 mm.
(3) Larger pulley diameter (D) = 800 mm
(4) Arc of Contact ($\theta$) = 166°
(5) Correction load ($P_d$) = 19.872 kW
(6) Load Rating = 0.0433
(7) Millimeter plies of Belt = 456.37
(8) Belt Width (b) = 100 mm
(9) pulley Width (B) = 125 mm
(10) Length (L) = 5765 mm
(11) Correction Length ($L_c$) = 5730 mm.

CROSSED BELT DRIVE:

1. Design a crossed belt drive to transmit 22 kW at 1060 rpm. The speed of driven pulley is 360 rpm. and the centre distance between the pulleys is 1.6 m. Assume the intermittent load and the number of plies = 6

GIVEN DATA:

Type of drive = Crossed Belt drive.

Power transmitted ($P_t$) = 22 kW

Speed of Smaller pulley ($n_1$) = 1060 rpm.

Speed of larger pulley ($n_2$) = 360 rpm.

Centre distance between pulleys ($C$) = 1.6 m ⇒ $1.6 \times 10^3$ mm

Assume intermittent load.

Number of plies = 6.

TO FIND:

Design a crossed flat belt drive.

SOLUTION:

Take crossed flat belt drive.

STEP 1: To find Speed Ratio ($i$):

FDB PSG 7.61 & 7.74.

$$i = \frac{D}{d} = \frac{n_1}{n_2}.$$

$$i = \frac{n_1}{n_2} = \frac{1060}{360}$$

$$i = 3.$$

$$i = \frac{D}{d}.$$

FDB pg 7.53 Assume the Belt Velocity as $(V) = 18m/s$.

Therefore

$$V = \frac{\pi D n_2}{60 \times 1000} \Rightarrow 18 = \frac{\pi \times D \times 360}{60 \times 1000}$$

$$D = \frac{18 \times 60 \times 1000}{\pi \times 360}$$

$$D = 954.92 mm$$

FDB pg 7.54.

$$D_{std} = 1000 mm.$$

$$i = \frac{D}{d}.$$

$$3 = \frac{1000}{d}.$$

$$d = \frac{1000}{3}$$

$$d = 333.33 mm.$$

FDB pg 7.54

$$d_{std} = 355 mm.$$

STEP2: To find Belt Velocity (or) Speed (V):

FDB pg 8.15.

$$V = \frac{\pi \times D \times n_2}{60 \times 1000}$$

$$= \frac{\pi \times 1000 \times 360}{60 \times 1000}$$

$V = 16.64\ m/s.$

STEP 3: To find Arc of Contact ($\theta$):

FDB PSG 7.54. For Crossed Belt drive.

$$\theta = 180° + \frac{(D+d)\times 60}{C}$$

$$= 180° + \frac{(1000+355)\times 60}{1.6\times 10^3}$$

$$= 180° + 50.81$$

$$= 230.81$$

$$\theta = 231°$$

STEP 4: To find Correction load ($P_d$):

Correction load = Given power × Load Correction factor × Arc of Contact factor.

Pulley Correction factor.

$$P_d = \frac{P_o \times k_c \times k_\alpha}{k_d}$$

To find arc of Contact factor ($k_\alpha$)

FDB PSG 7.54 for 231°

| | | | | |
|---|---|---|---|---|
| Degree | 220° | 240° | 10° | 1° |
| Value | 0.86 | 0.64 | 0.02 | 0.002 |

For

$231° = 230+1 \Rightarrow 0.86 + 0.002$

$k_\alpha = 0.862.$

To find Load Correction factor ($k_c$):

FDB PSG 7.54. For Intermittent load.

$k_c = 1.3.$

Neglect pulley Correction factor if its not given.

$P_d = P_o \times k_s \times k_a$

$= 22 \times 1.3 \times 0.862$

$P_d = 24.65$ kw

5 STEP5: To find Load Rating Per mm:

FDB PSG 7.54.

Load Rating Per mm Width Per ply at 180° arc of Contact at 10 m/s

For Hi Speed duck Belt = 0.023 kw.

STEP6: To find Load Rating at V m/s:

FDB PSG 7.54.

$$\text{Load Rating} = \text{Load Rating at } 10\text{ m/s} \times \frac{V}{10}$$

$$= 0.023 \times \frac{18.64}{10}$$

$$\text{Load Rating} = 0.0433.$$

STEP7: To find Millimeter Plies of Belt:

FDB PSG 7.54.

$$\text{Millimeter Plies of Belt} = \frac{\text{Correction load } (P_d)}{\text{Load Rating per mm ply at Belt speed}}$$

$$= \frac{24.65}{0.0433}$$

$$= 568.86.$$

## STEP 8: To find Belt Width (b):

$$\text{Belt Width } (b) = \frac{\text{Millimeter plies of belt}}{\text{No of Plies}}$$

$$= \frac{566.86}{6}$$

$$b = 94.61\,mm.$$

FDB PSG 7.52.

$$b_{std} = 100\,mm.$$

## STEP 9: To find Pulley Width (B):

FDB PSG 7.54 for Belt width (b) = 100 mm.

$$B = 100 + 13.$$

$$B = 113\,mm.$$

FDB PSG 7.52.

$$B_{std} = 125\,mm.$$

## STEP 10: To find belt length (L):

FDB PSG 7.53 for Crossed Belt drive.

$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D+d)^2}{4C}$$

$$= 2\times1.6\times10^3 + \frac{\pi}{2}(1000+355) + \frac{(1000+355)^2}{4\times1.6\times10^3}$$

$$= 3200 + 2128.42 + 286.87$$

$$L = 5615.29\,mm \approx 5620\,mm.$$

## To find Correction length ($L_c$).

FDB PSG 7.53 for 6 plies belt length cut by 1%.

Therefore.

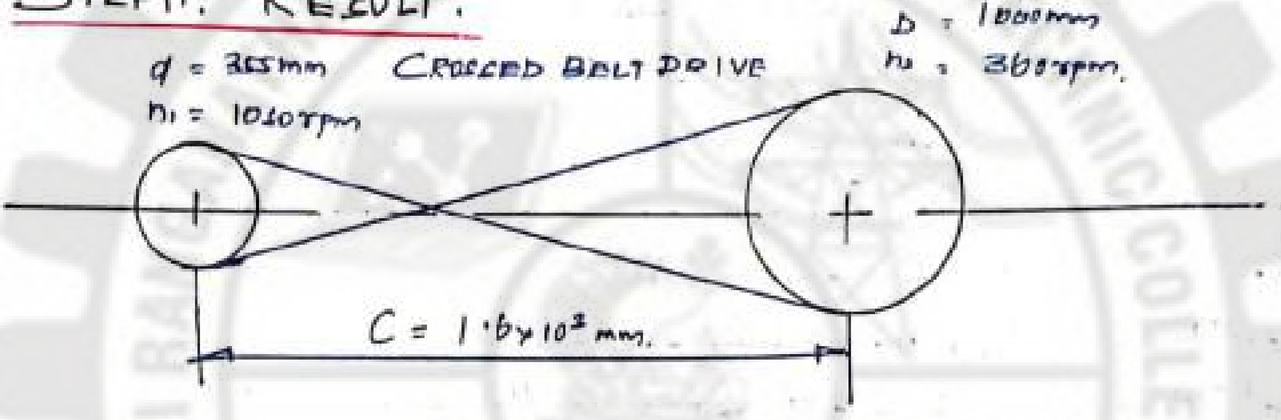
$$L_c = L\times(100-1)/100$$

$L_c = 5620 \times 99/100$

$L_c = 5563.8 \approx 5565mm.$

$L_c = 5565mm.$

STEP 11: RESULT:




(1) Speed Ratio (i) = 3.
(2) Dia of Smaller pulley (d) = 355mm.
(3) Dia of larger pulley (D) = ~~1000rpm~~. 1000mm.
(4) Speed of Smaller pulley ($n_1$) = 1080 rpm
(5) Speed of Larger pully ($n_2$) = 360 rpm.
(6) Belt Speed (V) = 18.64 m/s
(7) Arc of Contact ($\theta$) = 231°.
(8) Correction Load (Pd) = 24.65kw
(9) Load Rating = 0.0433.
(10) Millimeter plies of Belt = 568.66
(11) Belt Width (b) = 100mm.
(12) pulley Width (B) = 125mm.
(13) Belt Length (L) = 5620mm.
(14) Correction Belt Length ($L_c$) = 5565mm.

# V- BELT.

STEP BY STEP PROCEDURE FOR DESIGN OF V-BELT DRIVE:

**STEP 1: To find the type of Belt:**

FDB PSG 7.58 SELECT THE TYPE OF BELT.

**STEP 2: To find Speed Ratio (i):**

FDB PSG 7.61

$$i = D/d$$

$$D = d(n_1/n_2)\,\eta$$

$\eta$ (Efficiency) = 0.98 Assumed.

**STEP 3: To find Belt Speed (S) or (V):**

FDB PSG 8.15.

$$S \text{ (or) } V = \frac{\pi d n_1}{60\times1000}$$

(or)

$$S \text{ (or) } V = \frac{\pi D n_2}{60\times1000}$$

**STEP 4: To find Nominal Pitch length of belt (L):**

FDB PSG 7.53 For Open Belt drive & 7.61.

$$L = 2c + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$$

From PSG 7.56 to 7.60 Standard length of belt is Selected.

If c is not given Select it from $\frac{C}{D}$ Ratio Refer from 7.61 table.

STEP 5: To find the equivalent dia of smaller pulley ($d_e$).

FDB PSG 7.62

$$d_e = d_p \times F_b$$

$$d_e = d \times F_b$$

Where,

$F_b$ = Smaller pulley diameter factor. It is noted from PSG 7.62 table depending upon speed ratio.

STEP 6: To find power rating of V-Belt:

FDB PSG 7.62.

Select the formulae for the type of Belt.

(or)

Directly note the values from PSG 7.63 to 7.67 tables.

STEP 7: To find arc of contact: ($\theta$):

FDB PSG 7.68

$$\text{Arc of contact } (\theta) = 180° - 60° \times \left(\frac{D-d}{C}\right)$$

After calculating $\theta$, corresponding correction factor $F_d$ for V-V drive is noted from PSG 7.68

STEP 8: To find Number of belts (n):

FDB PSG 7.70

Number of Belts $(n) = \frac{P \times F_a}{kW \times F_c \times F_d}$.

$F_c$ – Correction factor for length. It is noted from PSG 7.58 to 7.60 table.

$F_a$ – Correction factor for industrial service. It is noted from PSG 7.69 table.

STEP 9: To find New centre distance (C):

FDB PSG 7.61.

$$C' = A + \sqrt{A^2 - B}$$

Where $A = \frac{L}{4} - \frac{\pi (D+d)}{8}$

$$B = \frac{(D-d)^2}{8}$$

Problems:

Type 1: DIA IS GIVEN.

1. Design V- belt drive to the following Specification.

Power to be transmitted (P) = 75 kW
Speed of driving Wheel = 1440 rpm.
Speed of driven Wheel = 400 rpm.
Diameter of driving Wheel = 300 mm.
Centre distance = ~~2000 mm~~. 2500 mm.
Small pulley factor $F_b$ = 1.14.
Service factor $F_a$ = 1.3
Correction factor for length $F_c$ = 1.07.

GIVEN DATA:

Power transmitted ($P$) = 75 kW
Speed of driving Wheel ($n_1$) = 1440 rpm.
Speed of driven Wheel ($n_2$) = 400 rpm.
Diameter of driving Wheel ($d_1$) = 300 mm.
Centre distance ($C$) = 2500 mm.
Small pulley factor ($F_b$) = 1.14.
Service factor ($F_a$) = 1.3
Correction factor for length ($F_c$) = 1.07.

TO FIND:

Design a V - Belt drive.

SOLUTION:

STEP 1: To find the type of Belt:

FDB PSG 7.58 For 75 kw power transmitted.

| Cross Section Symbol | Usal load of drive (kw) | Recommed Minimum Pulley diameter (d) (mm) | Nominal top Width W, mm |
|---|---|---|---|
| D | 92 - 150 | 355 | 32 |

| Nominal thickness T, mm | Weight Per Metre (kgf) |
|---|---|
| 19 | 0.596 |

STEP 2: To find Speed Ratio (i):

FDB PSG 7.61.

$$D = d\left(\frac{n_1}{n_2}\right)\eta$$

Assume $\eta = 0.98$.

$$D = 300 \times \frac{1440}{400} \times 0.98$$

$$D = 1058.4\,mm$$

FDB PSG 7.54

$$D_{std} = 1120\,mm$$

$$i = \frac{D}{d} = \frac{1120}{300}$$

$$i = 3.73.$$

STEP 3: To find Belt Speed (s) or (V):

FDB PSG 8.15.

$$S = \frac{\pi d n_1}{60 \times 1000} = \frac{\pi \times 300 \times 1440}{60 \times 1000}$$

$$S \text{ (or) } V = 22.62\,m/s.$$

STEP4: To find Nominal Pitch length of Belt (L):

FDB PSG 7.52 & 7.61

$$L = 2c + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4c}$$

$$= 2\times2500 + \frac{\pi}{2}(1120+300) + \frac{(1120-300)^2}{4\times2500}$$

$$= 5000 + 2230.53 + 67.24$$

$$L = 7297.77$$

FDB PSG 7.60

$L = 7648$ mm.

STEP5: To find Equivalent dia of Smaller pulley: ($d_e$):

FDB PSG 7.62

$d_e = d_p \times F_b$.

$d_e = d \times F_b$.

$= 300 \times 1.14$

$d_e = 342$ mm.

Step 6: To find Belt Rating of V-Belt:

FDB PSG 7.62 For Type of Belt – D.

Belt Cross section Symbol | Formula.

D $kW = \left[3.22 \times S^{-0.09} - \frac{506.7}{d_e} - 4.78 \times 10^{-4} S^2\right]$

$$kW = \left[3.22 \times S^{-0.09} - \frac{506.7}{d_e} - 4.78 \times 10^{-4} S^2\right]$$

$$= \left[3.22 \times 22.62^{-0.09} - \frac{506.7}{342} - 4.78 \times 10^{-4} \times 22.62^2\right] \times 22.62$$

$$= (2.43 - 1.48 - 0.244) \times 22.62$$

$$= 0.705 \times 22.62$$

$$kW = 15.9567$$

Step 7: To find arc of Contact ($\theta$):

FDB PSG 7.68

$$\text{Arc of Contact } (\theta) = 180 - 60 \left(\frac{D-d}{C}\right)$$

$$= 180 - 60 \left(\frac{1120 - 300}{2500}\right)$$

$$= 180 - \cancel{19.68}\ 19.68$$

$$= 160.32^\circ$$

FDB PSG 7.68

$$\theta = 163^\circ$$

| Arc of Contact in degree | Correction factor (Fd) V-V drive. |
|---|---|
| 163 | 0.96 |

$F_d = 0.96$

## STEP 8: To find Number of belts (n):

FDB pg 7.70

$$\text{Number of Belts } (n) = \frac{P \times F_a}{kw \times F_c \times F_d}$$

$$= \frac{75 \times 1.3}{15.9567 \times 1.07 \times 0.96}$$

$$= 5.948 \text{ say}$$

$$n = 6 \text{ belts.}$$

## STEP 9: To find New Centre distance (C):

FDB pg 7.61

$$A = \frac{L}{4} - \frac{\pi (D+d)}{8}$$

$$= \frac{7648}{4} - \frac{\pi (1120+300)}{8} = 1912 - 557.63$$

$$A = 1354.36 \text{ mm.}$$

$$B = \frac{(D-d)^2}{8} = \frac{(1120-300)^2}{8}$$

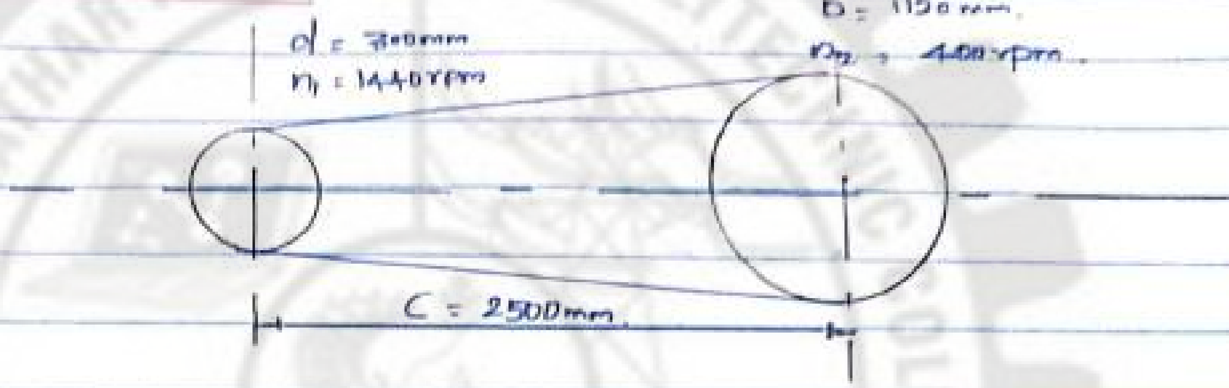
$$B = 84050 \text{ mm.}$$

$$C' = A + \sqrt{A^2 - B} = 1354.36 + \sqrt{1354.36^2 - 84000}$$

$C' = 2677.32$ mm say

$C' = 2680$ mm.

STEP 10: RESULT:




(1) Speed Ratio (i) = 3.73.

(2) Diameter of driven pulley (D) = 1120mm.

(3) Belt speed (v) = ~~22.62~~ 22.62 m/s

(4) Length (L) = 7648 mm.

(5) Equivalent dia ($d_e$) = 342mm.

(6) Belt Rating (kW) = 15.9567

(7) Arc of Contact ($\theta$) = 163°

(8) Arc of Contact factor ($F_d$) = 0.96

(9) New Centre distance (C') = 2680 mm.

TYPE: 2 DIA IS NOT GIVEN.

3. V belt drive is to transmit 15 kW to a Compressor. The Motor Speed 1200 rev/min and the Compressor pulley runs 400 rev/min. Determine the Size and Number of Belts Required.

GIVEN DATA:

Power transmitted (P) = 15 kW

Speed of the Motor pulley (or) Smaller pulley ($n_1$) = 1200 rpm.

Speed of the Compressor pulley (or) Larger pulley ($n_2$) = 400 rpm.

TO FIND:

Design a V-Belt drive.

SOLUTION:

STEP 1: To find the type of Belt:

FDB pg 7.58 For P = 15 kW.

| Cross section Symbol. | Usual load of drive kW | Recommended Minimum pulley diameter (d) mm | Nominal top width W mm. | Nominal thickness T mm. |
|---|---|---|---|---|
| C | 7.5-75 | 200 | 22 | 14 |

| Weight Per Meter kg. |
|---|
| 0.343. |

STEP 2: To Find Speed Ratio (i):

FDB Pg 7.61.

$i = D/d$ (or) $D = d\,(n_1/n_2)\,\eta$.

Assume $\eta = 0.98$

Take d = 200 from table. So,

$$D = 200\left(\frac{1200}{400}\right) \times 0.98$$

$D = 588$ mm.

Scanned with CamScanner

FDB PSG 7.54.

$D_{std} = 630$ mm.

Now $i = \frac{D}{d} = \frac{630}{200}$

$i = 3.15$

**STEP 3: To Find Belt Speed (S) or (V):**

FDB PSG 8.15.

$$S = \frac{\pi d n_1}{60 \times 1000} = \frac{\pi \times 200 \times 1200}{60 \times 1000}$$

$S = 12.566$ m/s.

**STEP 4: To find Nominal pitch length of Belt (L):**

FDB PSG 7.53 & 7.61

$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$$

Now C centre distance between the pulleys is not given so

FDB PSG 7.61. For $i = 3.15$.

$\frac{C}{D} = 1. \Rightarrow \frac{C}{630} = 1.$

$C = 1 \times 630$

$C = 630$ mm.

$$= 2 \times 630 + \frac{\pi}{2}(630 + 200) + \frac{(630 - 200)^2}{4 \times 630}$$

$$= 1260 + 1303.7 + 73.37$$

$L = 2637.07$ mm.

FDB PSG 7.60 For C type of Belt.

$L = 2723$ mm.

STEP 5: To find the Equivalent diameter of Smaller Pulley ($d_e$):

FDB PSG 7.62

$$d_e = d_p \times F_b \quad (\text{or}) \quad d \times F_b$$

FDB PSG 7.62 For $i = 3.15$

$$F_b = 1.14$$

$$= 200 \times 1.14$$

$$d_e = 228 \text{ mm.}$$

STEP 6: To find Belt Rating of V-Belt:

FDB PSG 7.62 For type of Belt – C

$$kW = \left(1.47 S^{-0.09} - \frac{142.7}{d_e} - 2.34 \times 10^{-4} S^2\right) S.$$

$$= \left(1.47 \times 12.566^{-0.09} - \frac{142.7}{228} - 2.34 \times 10^{-4} \times 12.566^2\right) 12.566$$

$$= (1.1705 - 0.6256 - 0.0369)\, 12.566$$

$kW = 6.381$.

STEP 7: To find Arc of Contact ($\theta$):

FDB PSG 7.68

$$\theta = 180^\circ - 60^\circ\left(\frac{D-d}{C}\right)$$

$$= 180^\circ - 60\left(\frac{630 - 200}{630}\right)$$

$\theta = 139.047°$

FDB PSG 7.68

$\theta = 142°$

$F_a = 0.90$

STEP 8: To find Number of bolts (n):

FDB PSG 7.70

$$n = \frac{P \times F_a}{kW \times F_c \times F_d}$$

FDB PSG 7.60 For Length (l) = 2723 mm.

$F_c = 0.94$

FDB PSG 7.69 Fa For industrial Service 16hrs/day.

$F_a = 1.2$.

$$= \frac{15 \times 1.2}{6.361 \times 0.94 \times 0.90}$$

$= 3.33$ Say.

$n = 4$ bolts.

STEP 9: To find New Centre distance (c').

FDB PSG 7.61.

$C' = A + \sqrt{A^2 - B}$.

$$A = \frac{L}{4} - \pi\left(\frac{D+d}{8}\right) = \frac{2723}{4} - \pi\left(\frac{630+200}{8}\right)$$

$= 680.75 - 325.94$

$A = 354.81$ mm.

$$B = \frac{(D-d)^2}{8} = \frac{(630-200)^2}{8}$$

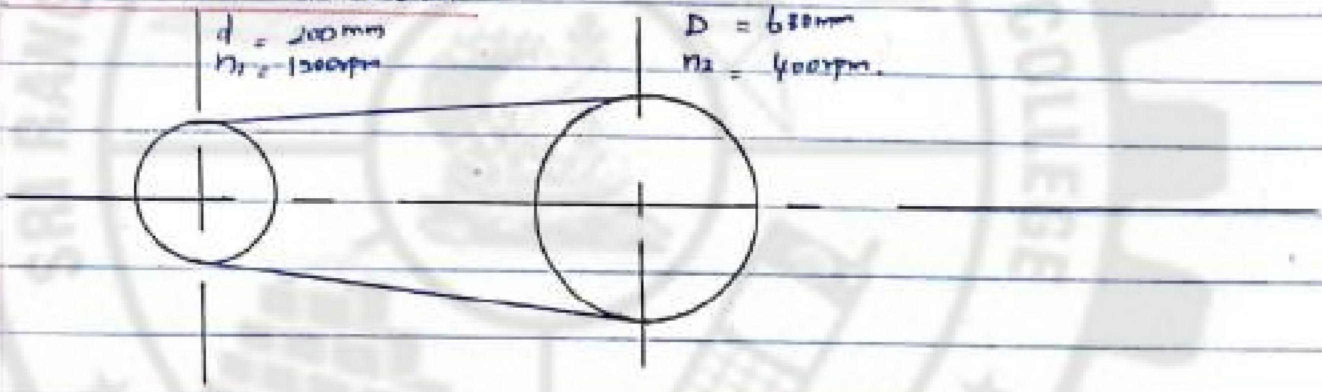
$$B = 23112.5\,mm$$

$$C' = 354.81 + \sqrt{354.81^2 - 23112.5}$$

$$= 354.81 + 320.57$$

$$C' = 675.39\,mm.$$

STEP 10: RESULT:




(1) Speed Ratio (i) = 3.15
(2) Diameter of larger pulley (D) = 630mm.
(3) Diameter of Smaller pulley (d) = 200mm.
(4) Belt speed (s) = 12.56m/s
(5) Length of the Belt (L) = 2723mm
(6) Equivalent diameter ($d_e$) = 228mm
(7) Arc of Contact ($\theta$) = 142°
(8) Number of belts (n) = 4 belts
(9) New centre distance (c') = 675.39mm.

2. Design a V-belt drive using Manufacturer's data to the following Specification.

Power transmitted = 7.5 kW

Speed of driving pulley = 1000 rpm.

Speed of driven pulley = 300 rpm.

Diameter of driving pulley = 150 mm.

Diameter of driven pulley = 500 mm.

Centre distance between pulleys = 925 mm.

Service = 16 hrs/day.

GIVEN DATA:

Power transmitted (P) = 7.5 kW

Speed of driving pulley ($n_1$) = 1000 rpm. ($n_1$)

Speed of driven pulley ($n_2$) = 300 rpm. ($n_2$)

diameter of driving pulley (d) = 150 mm

diameter of driven pulley (D) = 500 mm

Centre distance (C) = 925 mm.

Service = 16 hrs/day.

TO FIND:

Design a V-belt drive.

SOLUTION:

STEP 1: To find the type of Belt:

FDB PSG 7.58

| Cross Section Symbol | Veal load of drive kW | Recommended Minimum pulley diameter (d) | Nominal top width W, mm | Nominal thickness T mm. |
|---|---|---|---|---|
| B | 2 – 15 | – | 17 | 11 |

Weight Per meter.

0.189

STEP2: To find Speed Ratio: (i):

FDB psg 7.61.

$i = D/d = 500/150$

$i = 3.33$

STEP3: To find belt speed (s) or (v):

FDB psg 8.15

$$S = \frac{\pi d n_1}{60 \times 1000} = \frac{\pi \times 150 \times 1000}{60 \times 1000}$$

$S = 7.854$ m/s.

STEP4: To find Nominal Pitch length of Belt (L):

FDB psg 7.53 & 7.61.

$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$$

$$= 2 \times 925 + \frac{\pi}{2}(500+150) + \frac{(500-150)^2}{4 \times 925}$$

$$= 1850 + 1021.01 + 33.108$$

$L = 2904.11$ mm

FDB psg 7.60

$L = 3091$ mm.

STEP5: To find Equivalent dia of Smaller Pulley ($d_e$)

FDB PSG 7.62

$d_e = d_p \times F_b$ (or) $d \times F_b$

FDB psg 7.62. For $i = 3.33$

$F_b = 1.14$

$= 150 \times 1.14$

$d_e = 171$ mm.

## Step 6: To find Belt Rating of V - Belt:

FDB PSG 7.62 For Type of Belt - B

Symbol

Formula

B $\quad kW = \left[0.79\,S^{-0.09} - \frac{50.6}{d_e} - 1.32\times10^{-4} S^2\right] S$

$$kW = \left[0.79\,S^{-0.09} - \frac{50.6}{d_e} - 1.32\times10^{-4} S^2\right] S$$

$$= \left[0.79\times 7.654^{-0.09} - \frac{50.6}{171} - 1.32\times10^{-4}\times 7.654^{2}\right] 7.654$$

$$= \left[0.656 - 0.297 - 8.142\times10^{-3}\right] 7.654$$

$$kW = 2.755$$

## STEP 7: To find Arc of Contact ($\theta$):

FDB PSG 7.68

$$\theta = 180 - 60\,\frac{(D-d)}{C}$$

$$= 180 - 60\,\frac{(500-150)}{925}$$

$$= 180 - 22.7$$

$$\theta = 157.29^\circ$$

FDB PSG 7.68 $\theta = 160^\circ$

For $160^\circ$ $F_d = 0.95$

## STEP 8: To find Number of belts (n)

FDB PSG 7.70

$$n = \frac{P \times F_a}{kw \times F_c \times F_d}$$

$F_d = 0.95$

FDB PSG 7.60 $F_c$ for Nominal pitch length 3091mm

$F_c = 1.07$.

FDB PSG 7.69 $F_a$ for industrial service 16hrs/day.

$F_a = 1.2$

$$= \frac{7.5 \times 1.2}{2.755 \times 1.07 \times 0.95}$$

$= 3.2137$

$n = 4$ belts.

## STEP 9: To find New centre distance (c'):

FDB PSG 7.61.

$$c' = A + \sqrt{A^2 - B}$$

$$A = L/4 - \pi \times (D+d)/8.$$

$$= \frac{3091}{4} - \pi \times (500+150)/8 \Rightarrow 772.75 - 255.25$$

$A = 517.49$mm.

$$B = \frac{(D-d)^2}{8} = \frac{(500-150)^2}{8} = 15312.5\text{mm}.$$

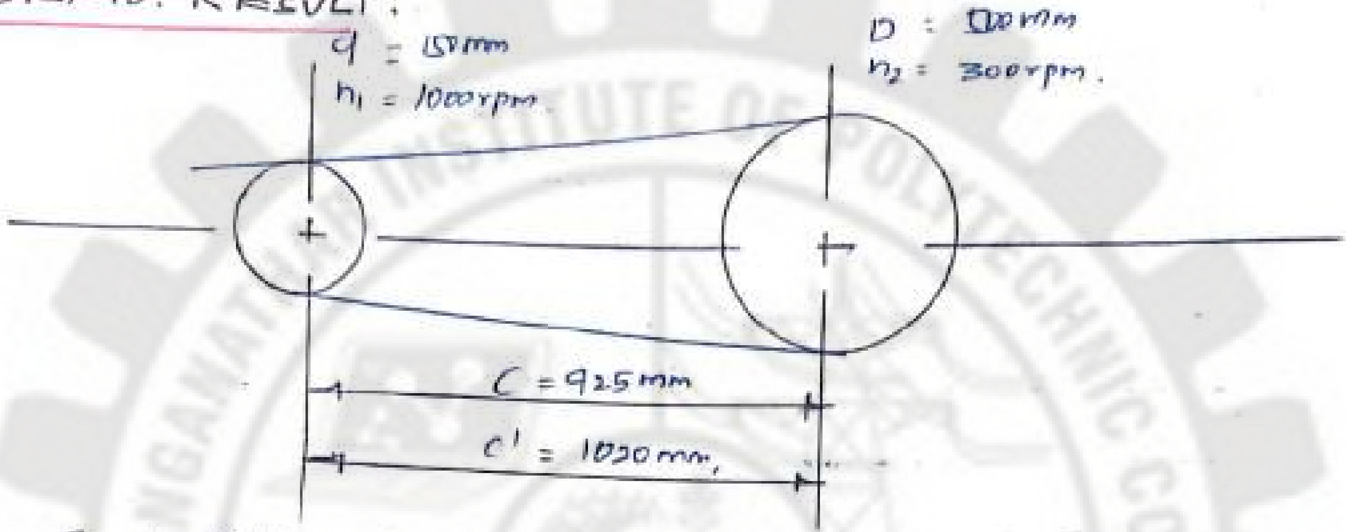
$$c' = 517.49 + \sqrt{517.49^2 - 15312.5}$$

$= 517.49 + 502.47$

$c' = 1019.96$mm. Say.

FDB $c' = 1020$mm.

STEP 10: RESULT:




(1) Speed Ratio (i) = 3.33

(2) Belt speed (s) = 7.854 m/s

(3) Nominal pitch length (L) = 3091 mm.

(4) Equivalent dia of Smaller pulley ($d_e$) = 171mm.

(5) Belt Rating (kw) = 2.755.

(6) Arc of Contact (θ) = 160°

(7) Number of Belt (n) = 4 belts.

(8) New Centre distance (c) = 1020mm.

UNIT - IV

# DESIGN OF BEARINGS.

## DESIGN PROCEDURE FOR JOURNAL BEARING:

### STEP 1: To find Bearing Pressure (P):

FDB PSG 7.45

$$P = \frac{W}{LD} \text{ kgf/cm}^2.$$

From this Equation, Working Pressure is Calculated It is Checked with allowable pressure given in FDB PSG 7.31 table.

Note:

If D is not given, Assume.

$$D = 100\text{mm} = \frac{100}{10} = 10\text{cm}.$$

If L is not given, Select Suitable L/D ratio from. FDB PSG 7.31 table.

### STEP 2: To find Velocity (v):

$$V = \frac{\pi D n}{100} \text{ m/min.}$$

If n is not given assume n = 1000 rpm.

STEP 3: To find co-efficient of friction ($\mu$):

FDB PSG 7.34.

$$\mu = \left[\frac{33.25}{10^{10}} \times \frac{Zn}{P} \times \frac{D}{C}\right] + K.$$

Note: If C is not given. Assume.

$$\frac{C}{D} = 0.001.$$

$$C = 0.001 \times D$$

If Z is not given, Assume suitable SAE type of oil and oil temperature.

Then note Z from the Data book PSG 7.41 graph for the corresponding SAE type of oil and its temperature.

STEP 4: To find Heat generated (or) Power lost ($H_g$).

FDB PSG 7.34.

$$H_g = \mu \times W \times V. \quad kgf\ m/min.$$

STEP 5: To find Heat dissipated ($H_d$).

FDB PSG 7.34.

$$H_d = \frac{(\Delta t + 18)^2 LD}{K} \quad kgf\ m/min.$$

STEP 6: To find amount of Cooling Required:

$$\left.\begin{array}{c}\text{Amount of Artificial Cooling} \\ \text{Required}\end{array}\right\} = H_g - H_d \text{ kgf m/min.}$$

STEP 7: To find Weight of Cooling oil Required:

$$\left.\begin{array}{c}\text{Weight of Cooling oil} \\ \text{Required } (W_c)\end{array}\right\} = \frac{(H_g - H_d) \times 100}{c' \Delta t_0}$$

Problem:

1. A Journal Bearing is proposed for a Centrifugal pump. The diameter of journal is 150mm and the load on it is 40kN. and its Speed is 900rpm. Design and give the Complete Calculation of the Bearing.

GIVEN DATA:

Machine - Centrifugal pump.

Diameter (D) = 150mm $\Rightarrow \frac{150}{10} = 15cm.$

Load (W) = ~~4kN~~ 40 kN. $\Rightarrow 40\times10^3 N.$

To Convert from N to kgf

$\Rightarrow \frac{40\times10^3}{10}$

W = 4000 kgf.

Speed (n) = 900 rpm.

TO FIND:

Design the Journal Bearing.

SOLUTION:

STEP 1: To find Bearing pressure (P):

PDB pg 7.45.

$$P = \frac{W}{LD}$$

$$\Rightarrow \frac{4000}{L \times 15}$$

To find Length (L):

FDB pg 7.31.

| Machine | $L/D$ | $Zn/p$ Minimum. |
| --- | --- | --- |
| Centrifugal pump | 1.0 – 2.0 | 2844.5. |

Therefore.

$$\frac{L}{D} = 1.6 \Rightarrow L = 1.6 \times D \Rightarrow 1.6 \times 15.$$

$$L = 24 \text{ cm}.$$

$$= \frac{4000}{24 \times 15}$$

$$P = 11.11 \text{ kg/cm}^2.$$

Step 2: To find Velocity (v):

$$V = \frac{\pi D n}{100} = \frac{\pi \times 15 \times 900}{100}.$$

$$V = 424.115 \text{ m/min}$$

Step3: To find Co-efficient of friction ($\mu$):

FDB pg 7.34.

$$\mu = \left[\frac{33.25}{10^{10}} \times \frac{Zn}{P} \times \frac{D}{C}\right] + k.$$

FDB pg 7.34.

for l/D Ratio 1.6 $k = 0.002$ from graph.

Assume $C/D = 0.001$.

$C = 0.001 \times D$

$= 0.001 \times 15$.

$C = 0.015$ cm.

Select lubricant oil SAE40 at temperature 60°C

FDB pg 7.41 Graph.

Assume For SAE40.

$Z = 45$ Centipoise

Therefore.

$$\frac{Zn}{P} \Rightarrow \frac{45 \times 900}{11.11} = 3645.36$$

$3645.36 > 2844.5$ so selected oil is safe.

$$\mu = \left[\frac{33.25}{10^{10}} \times \frac{Zn}{P} \times \frac{D}{C}\right] + k.$$

$$= \frac{33.25}{10^{10}} \times 3645.36 \times \frac{15}{0.015} + 0.002$$

$\mu = 0.014$.

STEP 4: To find Heat generated ($H_g$):

FDB PSG 7.34.

$$H_g = \mu \times W \times V.$$

$$= 0.014 \times 4000 \times 424.115.$$

$$H_g = 23750.44 \text{ kgfm/min.}$$

STEP 5: To find Heat dissipated ($H_d$):

FDB PSG 7.34.

$$H_d = \frac{(\Delta t + 18)^2 LD}{K}.$$

FDB PSG 7.48 Assume $t_a = 15°C$

$$\Delta t = \frac{t_o - t_a}{2}$$

$$\Rightarrow \frac{60 - 15}{2} = 22.5$$

Assume $K = 437$ for heavy Construction.

Therefore.

$$H_d = \frac{(22.5 + 16)^2 \times 24 \times 15}{137}$$

$H_d = 1351.235$ kgf m/min

STEP 6: To find amount of Cooling Required:

Amount of Cooling Required $= H_g - H_d$

$= 23750.44 - 1351.235$

$= 22399.20$ kgf m/min

STEP 7: To find Weight of Cooling oil Required:

Weight of Cooling oil Required $(W_c) = \frac{(H_g - H_d) \times 100}{C' \times \Delta t_0}$

For PCl4 7.34 $C' = 1710$ kgf (m) / kgf °C.

Assume $\Delta t_0 = 10°C$

$$= \frac{22399.20 \times 100}{1710 \times 10}$$

$= 13.09$ kgf/min

RESULT:

(1) Bearing Pressure (p) = 11.11 kg/cm2

(2) Velocity (v) = 424.115 m/min.

(3) $\mu$ = 0.014 (Co-efficient of friction):

(4) ~~Hg~~ Heat generated ($H_g$) = 23750.44 kgf m/min.

(5) Heat dissipated ($H_d$) = 1351.235 kgf m/min.

(6) Weight of Cooling oil Required ($W_c$) = 13.09 kgf/min.

02 Design a Suitable Journal Bearing a Centrifugal pump from the following data.

Load on the bearing $= 13.25 kN$.

Dia of the journal $= 80 mm$

Speed $= 1440 rpm$.

Bearing Characteristic Number $= 30 \times 10^{-6}$

Permissible bearing pressure $= 0.7$ to $1.4 N/mm^2$

Average atmospheric temp $= 30°C$

Calculate the Cooling requirements using Lasche's equation Mc Mckee's equation for Calculating the friction Co-efficient. Assume $L/D = 2$. Average temp of oil $t_0 = 75°C$. Temperature rise $\Delta t_0 = 6°C$

Given Data:

$$Load\ (W) = 13.25 kN = 13.25 \times 10^3 N \Rightarrow \frac{13.25 \times 10^3}{10}$$

$$W = 1325\ kgf.$$

$$Diameter\ (D) = 80 mm = \frac{80}{10} = 8 cm$$

$$Speed\ (n) = 1440 rpm$$

$$Bearing\ Characteristic\ Number\ \frac{Zn}{P} = 30 \times 10^{-6}$$

By Converting to PSG Data Book Unit.

$$\Rightarrow 30 \times 10^{-6} \times 10^{6} \Rightarrow 300$$

$$Permissible\ bearing\ pressure,\ P = 0.7 \text{ to } 1.4 N/mm^2$$

$$\Rightarrow 0.7 \text{ to } 1.4 \times 10$$

$$\Rightarrow 07 \text{ to } 14\ kgf/cm^2$$

Atmospheric temp $t_a = 30°C$

$L/D = 2$

$L = 2 \times D \Rightarrow 2 \times 8 = 16cm$

Oil temp $t_0 = 75°C$

Temperature rise $\Delta t_0 = 6°C$

To find:

Design the Bearing.

Solution:

Step 1: To find Bearing Pressure (P)

FDB pg 7.45

$$p = \frac{W}{LD} = \frac{1325}{16 \times 8}$$

$$P = 10.35 \text{ kgf/cm}^2$$

Step 2: To find Velocity (v)

$$V = \frac{\pi D n}{100} = \frac{\pi \times 8 \times 1440}{100}$$

$$V = 361.91 \text{ m/min}$$

Step3: To find co-efficient of friction ($\mu$):

Assume $c/D = 0.001$.

$c = 0.001 \times D = 0.001 \times 8 = 0.008$ cm.

FDB PSG 7.34 Graph

$\frac{L}{D} = 2$ $k = 0.0025$ is noted.

$$\mu = \left[\frac{33.25}{10^{10}} \times \frac{Zn}{P} \times \frac{D}{c}\right] + k$$

$$= \left[\frac{33.25}{10^{10}} \times 3000 \times \frac{8}{0.008}\right] + 0.0025$$

$\mu = 0.012475$

Step4: To find Heat generated ($H_g$)

FDB PSG 7.34

$H_g = \mu W V \Rightarrow 0.012475 \times 1325 \times 361.91$

$H_g =$ ~~5962~~ $5962.146$ kgf m/min

Step5: To find Heat dissipated ($H_d$):

FDB PSG 7.34

$$H_d = \frac{(\Delta t + 16)^2 LD}{k}$$

$$\Delta t = \frac{(t_o - t_a)}{2} = \frac{(75 - 30)}{2} = 22.5°C$$

Assume $k = 437$ for Heavy construction.

$$H_d = \frac{(22.5 + 16)^2 \times 16 \times 8}{437}$$

$H_d = 460.439$ kgf/min

Step 6: To find amount of Cooling Required:

Amount ~~of~~

Amount of Cooling Required = $H_g - H_d$.

$= 5962.146 - 460.439$

$= 5501.707$ kgf m/min.

Step 7: To find Weight of Cooling oil Required:

Weight of Cooling oil Required $(W_c) = \dfrac{(H_g - H_d) \times 100}{C' \times \Delta t_o}$

From pg 7.34

$C' = 17100$

Assume $\Delta t_o = 6°c$

$= \dfrac{(5962.146 - 460.439) \times 100}{17100 \times 6}$

$W_c = 5.36$ kgf / min

Result:

1. Bearing pressure (P) = 10.35 kgf/cm²
2. Velocity (V) = 361.91 m/min
3. Co-efficient of friction ($\mu$) = 0.012475
4. Heat generated ($H_g$) = 5962.146 kgf m/min
5. Heat dissipated ($H_d$) = 460.439 kgf m/min
6. Weight of Cooling oil Required ($W_c$) = 5.36 kgf/min.

03. A Journal Bearing 300mm long and 150mm dia Carries a Radial load of 9kN at 1200rpm. The power lost in friction is 6kW. Viscosity of an oil at Room temperature is 0.016 pa-s. Find the diametral Clearance.

Given data:

Length (L) = 300 mm ⇒ $\frac{300}{10}$ = 30cm

Diameter (D) = 150mm ⇒ $\frac{150}{10}$ = 15cm

Load (W) = 9kN = $9\times10^3$N ⇒ $\frac{9\times10^3}{10}$ = 900kgf

Speed (n) = 1200rpm

power lost in friction & Heat generated ($H_g$) = 6kW

= $6\times10^3$ W ⇒ $6\times10^3$ N m/s

= $\frac{6\times10^3\times60}{10}$ = 36000 kgf m/min

To find:

Diametral Clearance (c)

Solution:

Step 1: To find Bearing pressure (P):

PDB pg 7.45

$$P = \frac{W}{LD} = \frac{900}{30\times15}$$

$$P = 2\ kgf/cm^2$$

Step2: To find Velocity (v):

$$V = \frac{\pi D n}{100} = \frac{\pi \times 15 \times 1200}{100}$$

$$V = 565.486 \text{ m/min}$$

Step 3: To find Co-efficient of friction ($\mu$):

FDB pg 7.34

$$H_g = \mu \times W \times V.$$

$$36000 = \mu \times 900 \times 465.486$$

$$\mu = \frac{36000}{900 \times 465.486}$$

$$\mu = 0.0707$$

Step4: To find Diametral Clearance (C) from Co-efficient of friction formular:

FDB pg 7.34.

$$\mu = \left[\frac{33.25}{10^{10}}\left(\frac{Zn}{P}\right)\left(\frac{D}{C}\right)\right] + K.$$

Assume $K = 0.0025$.

$$0.0707 = \left[\frac{33.25}{10^{10}}\left(\frac{15 \times 1200}{2}\right)\left(\frac{15}{C}\right) + 0.0025\right.$$

$$0.0707 = \left[\frac{5.3865 \times 10^{-4}}{c}\right] + 0.0025$$

$$\frac{5.3865 \times 10^{-4}}{c} = 0.0707 - 0.0025$$

$$\frac{5.3865 \times 10^{-4}}{c} = 0.0682$$

$$c = \frac{5.3865 \times 10^{-4}}{0.0682}$$

$$c = 7.898 \times 10^{-3} \text{ cm.}$$

Recult:

$P = 2$ kgf/cm²

$V = 565.466$ m/min

$\mu = 0.0707$

$c = 7.898 \times 10^{-3}$ cm.

Q4: A 75mm journal bearing 100mm long is Subjected to 2.5 kN at 600rpm. If the room temperature is 24°c What Viscosity of oil should be used to limit the bearing Surface temperature at 55°c. Take $\frac{D}{c} = 1000$. Bearing is used for light and Medium Construction

Given data:

Diameter (D) $= 75mm = \frac{75}{10} = 7.5cm$.

Length (L) $= 100mm = \frac{100}{10} = 10cm$.

Load, (W) $= 2.5kN = 2.5 \times 10^3 N$.

$= \frac{2.5 \times 10^3}{10} = 250 kgf$.

Speed (n) $= 600$ rpm.

Room air temperature ($t_a$) $= 24°c$

Bearing Surface temperature ($t_b$) $= 55°c$

$\frac{D}{c} = 1000$

For light and Medium Construction (k) $= 775$

To find:

Viscosity of oil (Z).

Solution:

Step 1 : To find Bearing Pressure (P):

FDB pg 7.15

$$P = \frac{W}{LD} = \frac{250}{10 \times 7.5}$$

$$P = 3.33 \text{ kgf/cm}^2.$$

Step 2: To find Velocity (V):

$$V = \frac{\pi D n}{100} = \frac{\pi \times 7.5 \times 600}{100}$$

$$V = 141.3716 \text{ m/min}$$

Step 3: To find Heat dissipated (Hd):

FDB pg 7.24.

$$\Delta t = t_b - t_a = 55 - 24 = 31°c.$$

$$Hd = \frac{(\Delta t + 16)^2 LD}{K} \quad \cancel{\Delta t \; + 16}$$

$$= \frac{(31+76)^2 \times 10 \times 7.5}{775}$$

$$Hd = 147.19 \text{ kgf m/min} \quad 232.25 \text{ kgf m/min}$$

Assume $Hg = Hd$.

Step4: To find Co-efficient of friction from Heat generation formula:

$$H_g = \mu \times W \times V.$$

$$147.19 = \mu \times 250 \times 141.3716$$

$$\mu = \frac{\cancel{147.19}\ 232.35}{250 \times 141.3716}$$

$$\mu = \cancel{4.164 \times 10^{-3}}\ 6.57 \times 10^{-3}$$

Step5: To find Viscosity of oil (z) from ($\mu$):

FDDPSG 7.34. Assume $k = 0.002$

$$\mu = \left[\frac{33.25}{10^{10}} \times \left(\frac{Zn}{P}\right) \times \left(\frac{D}{C}\right)\right] + k.$$

$$\cancel{4.164 \times 10^{-3}}\ 6.57 \times 10^{-3} = \left[\frac{33.25}{10^{10}} \times \left(\frac{Z \times 600}{3.33}\right) \times 1000\right) + 0.002$$

$$\cancel{4.164 \times 10^{-3}}\ 6.57 \times 10^{-3} = (5.99 \times 10^{-4} \times Z) + 0.002$$

$$5.99 \times 10^{-4} \times Z = \cancel{4.164 \times 10^{-3}}\ 6.57 \times 10^{-3} - 0.002$$

$$5.99 \times 10^{-4} \times Z = \cancel{2.164 \times 10^{-3}}\ 4.57 \times 10^{-3}$$

$$Z = \frac{\cancel{2.164 \times 10^{-3}}\ 4.57 \times 10^{-3}}{5.99 \times 10^{-4}}$$

$$Z = \cancel{3.6\ CP}\ 7.62\ CP$$

Result:

$P = 3.33\ kgf/cm^2$

$V = 141.3716\ m/min$

$Hg = Hd = \cancel{147.19\ kgf\ m/min}\ 252.55\ kgf\ m/min$

$\eta = \cancel{4.164 \times 10^{-3}}\ 6.57 \times 10^{-3}$

$Z = \cancel{8.6cp}.\ 7.62\ cp.$

Q6. A 80mm long journal bearing supports a load of 2800N on a 50mm diameter shaft. The bearing has a radial clearance of 0.05mm and the viscosity of the oil is 0.021 kg/m-s at the operating temperature. If the bearing is capable of dissipating 80 J/s, determine the maximum safe speed.

Given data.

Length $(L) = 80mm = \frac{80}{10} = 8cm$.

Load $(W) = 2800 N = \frac{2800}{10} = 280 \, kgf$.

Diameter $(D) = 50mm = \frac{50}{10} = 5cm$.

Radial clearance $(c_r) = 0.05mm = \frac{0.05}{10} = 0.005cm$.

Diametral clearance $(c) = 2 \times c_r = 2 \times 0.005 = 0.01cm$.

Viscosity $(Z) = 0.021 \, kg/m\text{-}s = 0.021 \times 10^3 \, C.P.$

Heat dissipated $(H_d) = 80 \, Nm/s = \frac{80 \times 60}{10}$

$= 480 \, kgf \, m/min.$

To find:

Safe speed $(n)$

Solution:

Step 1: To find Bearing ~~Speed~~ Pressure (p):

FDB PSG 7.45

$$P = \frac{W}{LD} = \frac{280}{8\times5} =$$

$$P = 7 \text{ kgf/cm}^2.$$

Step 2: To find Co-efficient of friction ($\mu$):

$\frac{L}{D}$, $\mu$

FDB PSG 7.34

$$\frac{L}{D} = 8/5 = 1.6$$

FDB PSG 7.34 graph for $L/D = 1.6$

$k = 0.002$

$$\mu = \left[\frac{33.25}{10^{10}} \times \frac{Zn}{P} \times \frac{D}{C}\right] + k.$$

$$= \left[\frac{33.25}{10^{10}} \times \frac{21\times n}{7} \times \frac{5}{0.01}\right] + 0.002$$

$$\mu = \left[4.9875\times10^{-6} \times n\right] + 0.002 \rightarrow (1)$$

Step 3: To find Velocity (v):

FDB PSG 8.15

$$V = \frac{\pi D n}{100} = \frac{\pi\times 5\times n}{100}$$

$$V = 0.157\times n \rightarrow (2)$$

Step 4: To find Heat generated ($H_g$)

Assume $H_g = H_d$.

$H_g = H_d = 460 \ kg_f \ m/min$

$H_g = \mu \times W \times V$

$460 = [[4.9675 \times 10^{-6} \times n] + 0.002] \times 280 \times 0.157 \times n$

~~460 =~~

$460 = [[4.9675 \times 10^{-6} \times n] + 0.002] \times 43.96 n$

$460 = 2.1925 \times 10^{-4} n^2 + 0.0879 n$

$\underbrace{2.1925 \times 10^{-4}}_{a} n^2 + \underbrace{0.0879}_{b} n - \underbrace{460}_{c} = 0$

By using Equation Mode.

$n = 1292.68 \ rpm$

~~n = 1293~~ $n = 1293 \ rpm$

Result:

$p = 7 \ kg_f/cm^2$

$n = 1293 \ rpm$.

06. The following data refers to a 360° hydrodynamic bearing. Radial load = 3.2 kN. Journal speed = 1490 rpm Journal diameter = 50mm ; Bearing length = 50 mm. Radial Clearance = 0.05mm Viscosity of oil = 0.025 N-s/m². Assuming the total heat generated in the bearing is carried by the total oil flow in the bearing. Calculate Co-efficient of friction, power lost in friction, Minimum oil film thickness, flow requirement in lpm and temperature rise.

Given data:

360° - full Journal Bearing.

$$Load\ (W) = 3.2\,kN \Rightarrow 3.2\times10^3\,N = \frac{3.2\times10^3}{10} = 320\ kgf.$$

$$Speed\ (n) = 1490\,rpm = \frac{1490}{60} = 24.83\,rps.$$

$$Dia\ (D) = 50\,mm = \frac{50}{10} = 5\,cm.$$

$$Length\ (L) = 50\,mm = \frac{50}{10} = 5\,cm$$

$$L/D = 5/5 = 1.$$

$$Radial\ Clearance\ (C_r) = 0.05\,mm = \frac{0.05}{10} = 0.005\,cm.$$

$$Diametral\ Clearance\ (C) = 2\times C_r \Rightarrow 2\times0.005 \Rightarrow 0.01\,cm$$

$$Viscosity\ (Z) = 0.025\ N\text{-}s/m^2 \Rightarrow 0.025\times10^3 = 25\,cp.$$

To find:

Design a Journal Bearing.

Solution:

Step 1: To find Bearing pressure (P):

$$P = \frac{W}{LD} = \frac{320}{5\times5}$$

$$P = 12.8 \text{ kg/cm}^2.$$

Step 2: To find Minimum oil film thickness Variable:

FDB PSG 7.36

$$\left.\begin{array}{r}\text{Minimum oil film thickness}\\ \text{Variable}\end{array}\right\} = \frac{2h_0}{c}$$

$$\frac{2h_0}{c} = 0.4 \Rightarrow \frac{0.4\times c}{2} \Rightarrow \frac{0.4\times 0.01}{2} \Rightarrow 2\times10^{-3}\text{ cm.}$$

$$h_0 = 2\times10^{-2}\text{ cm.}$$

Step 3: To find Viscosity of oil from Sommerfield Number:

FDB PSG 7.34

$$S = \frac{z'n'}{P}\left(\frac{D}{c}\right)^2$$

$$z' = \frac{z}{9.81\times10^{7}} \quad \text{FDB PSG 7.34}$$

$$= \frac{25}{9.81\times10^{7}}$$

$$z' = 2.546\times10^{-7}\text{ kg s/cm}^2.$$

$$\Rightarrow \left[\frac{2.546\times10^{-7}\times 24.63}{12.6}\right]\left[\frac{5}{0.01}\right]^2$$

$$= 0.12$$

FDBPCG 7.36 For 360° Full Journal Bearing.

| $\frac{L}{D}$ | $\frac{2h_0}{C}$ | S | $\mu \times \frac{D}{C}$ | $\frac{4q}{Dcn'L}$ | $\frac{q_s}{q}$ | $\frac{\rho c' \Delta t_0}{P}$ |
|---|---|---|---|---|---|---|
| 1 | 0.4 | 0.121 | 3.22 | 4.33 | 0.66 | 14.2 |

$P/P_{max} = 0.415$

Step 4: To find Co-efficient of friction ($\mu$):

$$\mu \frac{D}{c} = 3.22$$

$$\mu \times \frac{5}{0.01} = 3.22$$

$$\mu = \frac{3.22\times 0.01}{5}$$

$$\mu = 6.44\times10^{-3}$$

Step 5: To find Velocity ($v$):

$$V = \frac{\pi D n}{100} = \frac{\cancel{\pi}\times 5\times 1490}{100}$$

$$V = 234.05 \text{ m/min}$$

Step 6: To find Heat lost in friction ($H_g$):

$$H_g = \mu\times W\times V \Rightarrow 6.44\times10^{-3}\times 320\times 234.05$$

$$H_g = 432.33 \text{ kgfm/min}$$

Step 7: To find oil flow ($q$):

$$\frac{4q}{Dcn'L} = 4.33$$

$$q = \frac{4.33 \times D \times c \times n' \times L}{4}$$

$$= \frac{4.33 \times 5 \times 0.01 \times 24.62 \times 5}{4}$$

$$q = 6.719 \ cm^3/s.$$

Step 8: To find temperature rise ($\Delta t_o$):

$$\frac{e \rho c' \Delta t_o}{p} = 14.2$$

$$\Delta t_o = \frac{14.2 \, p}{\rho c' \Delta}$$

FDB pg 7.36

$\rho = 0.00083$

$c' = 17100.$

$$= \frac{14.2 \times 12.6}{0.00083 \times 17100.}$$

~~$\Delta t_o = 12.6°c$~~ $\Delta t_o = 12.6°c$

Step 9: To find $P_{max}$:

$$\frac{P}{P_{max}} = 0.415$$

$$P_{max} = \frac{P}{0.415} = \frac{12.6}{0.415}$$

$P_{max} = 30.64$ kg/cm²

Result:

$P = 12.6$ kg/cm²

$h_o = 2 \times 10^{-3}$ cm

$Z' = 2.548 \times 10^{-7}$ kgf c/cm²

$S = 0.12$

$\mu = 6.44 \times 10^{-3}$

$V = 234.05$ m/min

$H_g = 432.23$ kgf m/min

$q = 6.719$ cm³/s

$\Delta t_o = 12.6$ °c

$P_{max} = 30.64$ kg/cm²

# UNIT - V.

## DESIGN OF LEVERS AND SPUR GEARS.

TYPE: 1 MODULE IS GIVEN:

DESIGN OF SPUR GEARS:

STEP 1 : SELECTION OF MATERIAL:

If not given assume suitable Materials for pinion & gear [Refer PSG 8.5]

When both pinion & gear are Made of Same Material pinion is weaker than gear. So pinion Should be designed.

When different Materials are used product $[\sigma_b]_1 y_1$ for the pinion and product $[\sigma_{b_2}] \cdot y_2$ for the gear are Calculated. If $\sigma_{b_1} y_1$ is less pinion Should be designed. If $[\sigma_{b_2}] y_2$ is Lesser gear should be designed.

NOTE :

To find $y_1$ & $y_2$ refer PSG 8.50.

STEP 2: TO FIND LEWIS FORM FACTOR ($y_1$)

FOR PSG 8.50.

$$y_1 = 0.154 - \left[\frac{0.912}{z_1}\right]$$

STEP 3: TO FIND MEAN PITCH LINE VELOCITY ($V_m$)

FDB pg 8.15

$$V_m = \frac{\pi d_1 n_1}{60 \times 1000}$$

FDB pg 8.22

$$d_1 = m z_1.$$

STEP 4: TO FIND VELOCITY FACTOR ($C_v$).

FDB pg 8.51 Velocity factor $C_v$ is calculated.

STEP 5: TO FIND AXIAL FORCE (or) BEAM STRENGTH [$F_a$ or $F_c$]

FDB pg . 8.50

$$F_a \text{ (or) } F_s = [\sigma_b] \cdot b \cdot y \cdot P_c.$$

FDB pg 8.50

$$P_c = \frac{\pi d}{z}$$

STEP 6: TO FIND DYNAMIC LOAD ($F_d$)?

FDB pg 8.51.

$F_c \geq F_d$ Take.

$$F_s = F_d.$$

STEP 7: TO FIND TANGENTIAL LOAD ($F_t$):

FDB pg 8.50.

$$F_d = F_t \times C_v.$$

$$F_t = \frac{F_d}{C_v}.$$

STEP 8: TO FIND POWER ($P_0$):

$$F_t = \frac{P_0 \cdot k_0}{V_m.}$$

Where $k_0$ is the Service factor. $k_0$ is not given

Assume $k_0 = 1$.

PROBLEM:

1. A Pinion runs at 600rpm drives a gear at a speed Ratio of 4:1. Allowable Static stress of pinion & gear Material is 85 N/mm². Pinion has 16 teeth of module 8mm. Teeth are 20° FD System. Face width 90mm. Find the power transmitted.

Given Data:

Speed of pinion $(N_1)$ = 600 rpm

Speed Ratio $(i)$ = 4:1 $\Rightarrow \frac{4}{1} = 4$.

Allowable Static Stress for Pinion Material.

$(\sigma_b)_1 = 85 N/mm^2$.

Number of teeth of pinion $(z_1)$ = 16.

Module $(m)$ = 8mm

Face width $(b)$ = 90mm.

Teeth are 20° FD System

To find:

Power transmitted $(P_0)$:

Solution:

Step 1: ~~To~~ ~~fi~~ Selection of material:

Pinion and gear are made of Same Material so design the pinion

Step2: To find Lewis form factor ($y_1$):

FDB pg 8.50 for 20° involute.

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

$$= 0.154 - 0.912/16$$

$$y_1 = 0.097$$

Step3: To find Mean Pitch line Velocity ($V_m$):

FDB pg 8.15

$$V_m = \frac{\pi d_1 n_1}{60 \times 1000}$$

To find $d_1$:

FDB pg 8.22.

$$d_1 = m z_1 \Rightarrow 8 \times 16 = 128\,mm.$$

$$= \frac{\pi \times 128 \times 600}{60 \times 1000}$$

$$V_m = 4.021\ m/sec.$$

Step4: To find ~~Pitch~~ Velocity factor ($C_v$):

FDB pg 8.51 for $V_m \leq 10\,m/s$ for Commercially Cut Wheels.

$$C_v = \frac{3+V_m}{3} \Rightarrow \frac{3+4.021}{3}.$$

$$C_v = 2.34.$$

Step 5: To find axial force (or) Beam Strength ($F_a$ or $F_c$)

FDB psg 8.50

$F_a$ or $F_c = (\sigma_{b_1}) \cdot b \cdot y \cdot P_c$.

To find $P_c$:

FDB psg 8.50

$$P_c = \frac{\pi d_1}{Z_1} = \frac{\pi \times 128}{16}$$

$$P_c = 25.1327.$$

$$= 85 \times 90 \times 0.097 \times 25.1327$$

$$F_s = 18649.72\,N.$$

Step 6: To find the dynamic load ($F_d$):

FDB psg 8.51

$F_s \geq F_d$. Take $F_s = F_d$.

$$F_d = 18649.72\,N.$$

Step 7: To find tangential load ($F_t$):

FDB psg 8.50

$$F_d = F_t \times C_v.$$

$$F_t = \frac{F_d}{C_v} = \frac{18649.72}{2.34}$$

$$F_t = 7969.965\,N.$$

Step 8: To find Power ($P_o$):

$$F_t = \frac{P_o \cdot k_o}{V_m}$$

$k_o$ is not given. Assume $k_o = 1$.

$$P_o = \frac{F_t \times V_m}{k_o} = \frac{7969.965 \times 4.021}{1}$$

$P_o = 32047.16$ Watts.

Step 9: Result:

1. $y_1 = 0.097$
2. $d_1 = 128$ mm
3. $V_m = 4.021$ m/sec
4. $C_v = 2.34$.
5. $F_d$ (or) $F_c = 16649.72$ N.
6. $F_s = F_d = 16649.72$ N
7. $F_t = 7969.965$ N.
8. $P_o = 32047.16$ Watts.

TYPE:2 NUMBER OF TEETH GIVEN:

STEP 1: SELECTION OF MATERIAL:

Do the same Procedure in Type-1.

STEP 2: TO FIND LEWIS FORM FACTOR: ($y_1$):

FDB pg 8.50 for 20° Full depth system.

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

STEP 3: To find Mean Pitch Line Velocity ($V_m$):

FDB pg 8.15

$$V_m = \frac{\pi d_1 n_1}{60 \times 1000}$$

To find $d_1$: FDB pg 8.22.

$$d_1 = m \times z_1$$

STEP 4: To find tangential load (static): ($F_t$):

$$F_t = \frac{P_o \cdot k_o}{V_m}$$

If $k_o$ is not given. Assume $k_o = 1$.

STEP 5: To find Velocity factor ($C_v$).

FDB pg 8.51 Velocity factor $C_v$ is Calculated.

STEP 6: To find dynamic load ($F_d$):

FDB psg 8.50

$$F_d = F_t \times C_v$$

STEP 7: To find axial force ($F_n$) (or) beam strength ($F_s$):

FDB psg 8.50

$$F_n \text{ (or) } F_s = \sigma_b \cdot b \cdot y \cdot P_c$$

STEP 8: To find Module (m):

FDB psg 8.51

$$F_s \geq F_d.$$

STEP 9: To find b, $d_1$, & $V_m$:

FDB psg 8.22.

$$d_1 = m z_1$$

$$d_2 = m z_2$$

$$b = 10m.$$

$$V_m = x \cdot m.$$

STEP 10: To find $F_c$ & $F_t$:

From the problem:

$$F_c = x \times m^2.$$

$$F_t = x / m.$$

STEP 11: To find dynamic load using Buckingham's Equation ($F_d$):

FDB psg 8.51

$$F_d = F_t + \left[ \frac{0.164 V_m (C_b + F_t)}{0.164 V_m + 1.485 (\sqrt{C_b + F_t})} \right]$$

To find Value of C:

Value of C = Value of C (FDB pg 8.53 (Table 41)) x Precision of gears.

Where $V_m$ is the mean velocity in m/sec Convert to m/min Therefore

$$V_m = V_m \text{ in m/sec} \times 60$$

Check 1:

FDB pg 8.51 $F_s \geq F_d$.

Step 12: ~~Contan~~ To find Wear load ($F_w$):

FDB pg 8.51

$$F_w = d_1 \times Q \times k \times b$$

To find Q:- FDB pg 8.51

$$Q = \frac{2i}{i+1}$$

To find k: FDB pg 8.51

$$k = \frac{[\sigma_c]^2 \cdot \sin\alpha \left[\frac{1}{E_1} + \frac{1}{E_2}\right]}{1.4}$$

To find $[\sigma_c]$ FDB PSG 8.5 Stress in kg/cm² convert to N/mm² $\rightarrow \dfrac{\text{Value of FDB PSG 8.5}}{10}$

To find $E_1$ & $E_2$ FDB PSG 1.1

$\alpha$ = Pressure angle (14.5° or 20°)

Check 2:

FDB PSG 8.51 $F_w \geq F_d$.

STEP:12 Basic dimensions of Pinion and gear wheels:

FDB PSG 8.22 Table.

Problem:

Design a Spur gear drive to connect an Electric Motor to a Reciprocating pump both being Mounted on a same bed. Speed of the motor is 1440 rpm. Speed Reduction desired is 10:1. Motor power is 36.8 kw. The gears are to have 20° pressure angles. The Minium number of teeth on the pinion is 24.

Given data:

Speed of the Motor pinion $(n_1) = 1440$ rpm.

Speed Reduction Ratio $i = 10:1 \Rightarrow 10/1 \Rightarrow 10$

Motor power $(P_0) = 36.8$ kw $\Rightarrow 36.8 \times 10^3$ W

Teeth 20° pressure angle.

Number of teeth on pinion $(z_1) = 24$.

Number of teeth on Gear $(z_2) \Rightarrow i \times z_1$ (PSG 8.22)

$$z_2 = 10 \times 24$$

$$z_2 = 240 \text{ teeth}$$

To find:

Design of Spur gear.

Solution:

STEP 1: Selection of Material:

Both pinion & Gear are Made of Same Material, So design for pinion.

STEP 2: To find Lewis form factor ($y_1$)

PSG 8.50

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

$$= 0.154 - \frac{0.912}{24}$$

$$y_1 = 0.116$$

STEP 3: To find Mean pitch line Velocity ($V_m$):

PSG 8.15

$$V_m = \frac{\pi d_1 n_1}{60 \times 1000}$$

To find $d_1$: PSG 8.22

$$d_1 = m \cdot z_1 \Rightarrow m \times 24$$

$$d_1 = 24m$$

$$= \frac{\pi \times 24 \times m \times 1440}{60 \times 1000}$$

$$V_m = \cancel{1.8095m} \; 1.8095m$$

STEP 4: To find tangential load ($F_t$):

$$F_t = \frac{P_0 \cdot k_0}{V_m}$$

Assume $k_0 = 1$ if not given

$$= \frac{36800 \times 1}{1.8095m}$$

$$F_t = \frac{20337.109}{m}$$

STEP5: To find Velocity factor ($C_v$):

For PSG 8.51, for $V_m$ = 5 – 20 m/s & for Carefully cut gears.

$$C_v = \frac{6+V_m}{6}$$

To calculate $C_v$, $V_m$ may be taken as from 10 to 15 m/s.

So Assume $V_m$ = 12 m/s.

$$C_v = \frac{6+12}{6}$$

$$C_v = 3.$$

STEP: 6: To find dynamic Load ($F_d$):

For PSG 8.50:

$$F_d = F_t \times C_v$$

$$= \frac{20337.69}{m} \times 3.$$

$$F_d = \frac{61011.307}{m} \rightarrow (1)$$

STEP 7: To find axial force or Beam strength. ($F_a$ (or) $F_c$)

FDB PSG 8.50.

$$F_a \text{ or } F_c = \sigma_b \cdot b \cdot y \cdot P_c$$

To find d.

FDB psg 8.50

$$P_c = \frac{\pi d}{z} \Rightarrow \frac{\pi d_1}{z_1}$$

Now sub $d_1 = m z_1$

$$\Rightarrow \frac{\pi \times m \times \cancel{z_1}}{\cancel{z_1}}$$

$$P_c = \pi \times m$$

Assume Pinion is made of Steel. FDB psg 8.5.

C 45 steel.

$$\sigma_{b_1} = 1400 \text{ kgf/cm}^2$$

$$= \frac{1400}{10} = 140 \text{ N/mm}^2$$

FDB psg 8.51

$$b = 3 \text{ to } 4 \text{ times } P_c$$

Take $b = 3.2 P_c$

$$= 3.2 \times \pi \times m$$

$$b = 10.05 m \approx 10 m$$

$$\therefore b = 10 m$$

Therefore

$$F_c = 140 \times 10 m \times 0.116 \times \pi \times m$$

$F_s = 510.195 m^2 \rightarrow (2)$

Step 8: To find Module (m):

FDB PSG 8.51

$F_s \geq F_d$

Now From Equation (1) & 2 we get.

$$510.195 m^2 \geq \frac{61011.227}{m}$$

$$m^3 \geq \frac{61011.227}{510.195}$$

$$m^3 \geq 119.58$$

$$m \geq \sqrt[3]{119.58}$$

$m \geq 4.9 \approx 5mm$ [FDB PSG 8.2 Table.]

$$m = 5mm.$$

Step 9: To find b, d, & $V_m$ :

FDB PSG 8.22

$d_1 = m z_1 = 5 \times 24 = 120 mm.$

$d_2 = m z_2 = 5 \times 240 = 1200 mm.$

$b = 10 m = 10 \times 5 = 50 mm.$

$V_m = 1.8095 m = 1.8095 \times 5 = 9.0475 m/s.$

Step 10: To find $F_s$ & $F_t$:

From the Problem

$$F_s = 510.195 m^2 = 510.195 \times 5^2$$

$$F_s = 12754.875 \text{ N}.$$

$$F_t = \frac{20337.109}{m} = \frac{20337.109}{5}$$

$$F_t = 4067.42 \text{ N}.$$

Step 11: To find dynamic load using Buckinghams Equation ($F_d$):

FOB PSG 8.51

$$F_d = F_t + \left[\frac{0.164 V_m [C b + F_t]}{0.164 V_m + 1.485 \times \sqrt{(Cb + F_t)}}\right]$$

Where $V_m$ in m/min so

$$V_m = V_m \text{ in m/s} \times 60 = 9.0475 \times 60$$

$$V_m = 542.85 \text{ m/min}.$$

FOB PSG 8.53 table. For module $(m) = 5mm$

Precision of gears $(e) = 0.0125$.

FOB PSG 8.53 table.

Value of $C = 11860e$ [Steel pinion & Steel gear, 20° Full depth]

Therefore

Value of $C = 11860 \times 0.0125$.

$$C = 148.25 \text{ N/mm}^2.$$

$$F_d = 4067.42 + \left[ \frac{0.164 \times 542.85 \left[148.25 \times 50 + 4067.42\right]}{0.164 \times 542.85 + 1.485 \times \sqrt{148.25 \times 50 + 4067.42}} \right]$$

$$= 4067.42 + \left[ \frac{89.07 \times 11479.92}{89.027 + 1.485 \sqrt{11479.42}} \right]$$

$$= ~~4067~~ \; 4067.42 + \frac{1022022.828}{246.136}$$

$$= 4067.42 + 4118.80$$

$$F_d = 8186.22\,N.$$

Check 1:

FDB PSG 8.51 $F_s \geq F_d$ for severe service.

$\cancel{12874}$ $12754.875\,N \geq 8186.22\,N.$

So the design is safe.

Step 2: To find wear load ($F_W$):

FDB PSG 8.51

$$F_W = d_1 \times Q \times k \times b.$$

To find Q: FDB PSG 8.51

$$Q = \frac{2i}{i+1} \Rightarrow \frac{2\times10}{10+1} = \frac{20}{11}$$

$$Q = 1.818$$

To find k: FDB PSG 8.51

$$k = \frac{\sigma_{c1}^2 \sin\alpha \left[\frac{1}{E_1} + \frac{1}{E_2}\right]}{1.4}$$

~~From~~ FDB PSG 8.5 for steel, $\sigma_{c1}$ = ~~5000~~ 6000 kgf/cm².

$= \frac{6000}{10} = 600$ N/mm² [for safer design increased to 6000 from 5000]

FDB PSG 1.1 $E = 2\times10^5$ N/mm² for Pinion and gear.

Pressure angle $\alpha = 20°$.

$$= \frac{600^2 \cdot \sin 20° \left[\frac{1}{2\times10^5} + \frac{1}{2\times10^5}\right]}{1.4}$$

$k$ = ~~0.610 N/mm²~~ $k = 0.879$ N/mm²

$$F_w = 120 \times 1.818 \times \cancel{0.\ldots}^{0.879} \times 50$$

$$F_w = 9588.132\,N.$$

Check(2):

FOB PSG 8.51 $\quad F_w \geq F_d$.

$$9588.132\,N \geq 8186.02\,N$$

So the design is safe.

Step 12: Basic dimensions of pinion & gear:

FOB PSG 8.22 Table

| Nomenclature | Notation | Units | Spur gears |
|---|---|---|---|
| ~~Centre distance~~ Module | $m$ | mm | $m = 5\,mm$. |
| Centre distance | $a$ | mm | $a = \frac{m(z_1+z_2)}{2}$ <br> $= \frac{5\times(24+240)}{2}$ <br> $a = 660\,mm$. |
| Tooth depth | $h$ | mm | $h = 2.25\,m$ <br> $= 2.25\times 5$ <br> $h = 11.25\,mm$ |
| Pitch diameter | $d$ | mm | $d_1 = m z_1 = 5\times 24$ <br> $= 120\,mm$ <br> $d_2 = m z_2 = 5\times 240$ <br> $= 1200\,mm$ |
| Number of teeth | $z$ | – | $z_1 = 24$ <br> $z_2 = i\times z_1 = 10\times 24$ <br> $= 240$ |

Step 13: Result:

TYPE : 3. MODULE IS NOT GIVEN ; CENTRE DISTANCE (a) GIVEN.

(1) A gear drive is required to transmit a Maximum power of 25kW. The Velocity ratio is 1:2 and rpm of the pinion is 200. The approximate centre distance between the shafts may be taken as 600mm. The teeth has 20° Stub involute profiles. The Material used for the gear is C.I. Find the Module, face width, Number of teeth on each gear. Check your design for dynamic and Wear loads.

GIVEN DATA:

Power transmitted $(P) = 25kW = 25 \times 10^3 W$.

Velocity ratio $(i) = 2:1 \Rightarrow 2/1 = 2$

Speed of pinion $(n_1) = 200$ rpm.

Centre distance $(a) = 600$mm.

Teeth are 20° Stub profiles.

Both pinion and gear wheels are ~~steel~~ Made of C.I. So pinion is Weaker Wheel. Design pinion.

To find:

Design a Spur gear drive

Solution:

Step 1: Selection of Material:

Both pinion & Gear are Made of same Material (C.I). Pinion is weaker. So design pinion.

Step 2: To find Lewis form factor ($y_1$):

FDB pg 8.50 For 20° Full depth system.

$$y_1 = 0.175 - \frac{0.912}{z_1}$$

~~To~~ To find Number of teeth ($z_1$ & $z_2$):

Assume $z_1 = 50$ teeth

~~PSG~~ FDB pg 8.22.

$$z_2 = i z_1$$

$$= 2 \times 50$$

$$z_2 = 100 \text{ teeth}$$

$$= 0.175 - \frac{0.912}{50}$$

$$y_1 = 0.156$$

Step3 : To find Mean pitch Velocity : ($V_m$) :

FDB psg 8.15

$$V_m = \frac{\pi d_1 n_1}{60 \times 1000}$$

To find diameter ($d_1$ & $d_2$) :

FDB psg 8.22.

$d_1 = m z_1$ $\quad$ $d_2 = m z_2$.

To find Module (m) :

FDB psg 8.22.

$$a = \frac{m(z_1 + z_2)}{2}$$

$$600 = \frac{m(50 + 100)}{2}$$

$$\frac{600 \times 2}{150} = m$$

$\boxed{m = 8 \text{ mm}}$ FDB psg 8.2.

$$V_m = \frac{\pi \times 400 \times 200}{60 \times 1000}$$

$V_m = 4.188$ m/s.

$d_1 = 8 \times 50$ $\quad$ $d_2 = 8 \times 100$

$d_1 = 400$ mm $\quad$ $d_2 = 800$ mm.

Step 4: To find tangential load ($F_t$):

$$F_t = \frac{P_0 \cdot k_0}{V_m}$$

If $k_0$ is not given. Assume $k_0 = 1$.

$$= \frac{25 \times 10^3 \times 1}{4.188}$$

$$F_t = 5969.44 N.$$

Step 5: To find Velocity factor ($C_v$):

From PSG 8.51 For 20° FD system.

$$C_v = \frac{6 + V_m}{6} = \frac{6 + 4.188}{6}$$

$$C_v = 1.698$$

Step 6: To find dynamic load ($F_d$):

From PSG 8.50

$$F_d = F_t \times C_v \Rightarrow 5969.44 \times 1.698$$

$$F_d = 10136.10 N.$$

Step 7: To find axial force (or) Beam strength ( Fa or Fc ).

FDB pg 8·51

$$F_a \text{ (or) } F_c = \sigma_{b_1} \cdot b \cdot y_1 \cdot P_c$$

$$P_c = \frac{\pi d_1}{z_1} = \frac{\pi m z_1}{z_1} = \pi m \quad \text{FDB pg 8·50}$$

$$\Rightarrow \sigma_{b_1} \cdot b \cdot y_1 \cdot \pi \cdot m$$

FDB pg 8·5

$$\sigma_{b_1} = 550 \text{ kgf/cm}^2 \Rightarrow \frac{550}{10} = 55 \text{ N/mm}^2$$

FDB pg 8·51

$b =$ 3 to 4 times of $P_c$ (or) $10m$.

$= 3.2 \times \pi \times m \Rightarrow 80.42 \text{ mm} \approx 80 \text{mm}$

$\Rightarrow 10 \times 8 = 80 \text{mm}$

$b = 80 \text{mm}$

$$= 55 \times 80 \times 0.156 \times \pi \times 8$$

$$F_c = 17251.71 \text{ N}.$$

Step 8: To find dynamic load using Buckingham's Equation (Fd):

FDB pg 8·51.

$$F_d = F_t + \left[ \frac{0.164 V_m \left[ C_b + F_t \right]}{0.164 V_m + 1.485 \sqrt{C_b + F_t}} \right]$$

Where $V_m$ in m/min

$V_m = V_m$ in m/min $\times 60 = 4.186 \times 60$

$V_m = 251.28$ m/s

To find value of c:

For PSG 8.53 table

$C = 6150.0$

For PSG 8.53 For m = 8mm.

$e = 0.019$

$= 6150 \times 0.019$

$C = 116.85$ N/mm

$$F_d = 5969.44 + \left[\frac{0.164 \times 251.28 \times 116.85 \times 80 + 5969.44}{0.164 \times 251.28 + 1.465\sqrt{116.85 \times 64}\sqrt{5969.44}}\right]$$

$$= 5969.44 + \frac{631216.385}{224.998}$$

$= 5969.44 + 2805.43$

$F_d = 8774.87$N.

Check 2: For PSG 8.51

$F_s \geq F_d$

$17251.11\text{N} \geq 8774.87\text{N}$.

So the design is safe.

Step 11: To find Wear load ($F_w$):

FDB psg 8.51.

$$F_w = d_1 . Q . K . b$$

FDB psg 8.51

$$Q = \frac{2i}{i+1} = \frac{2\times 2}{2+1} = \frac{4}{3} = 1.333.$$

FDB psg 8.51

$$K = \frac{(\sigma_c)^2 \sin\alpha \left[\frac{1}{E_1} + \frac{1}{E_2}\right]}{1.4}$$

FDB psg 8.5 For Cast iron.

$$\sigma_c = 6000\ kgf/cm^2 = \frac{6000}{10} = 600\ N/mm^2.$$

FDB psg 1.1 $E_1$ & $E_2$ = $1\times10^5\ N/mm^2$.

Pressure angle $\alpha = 20$.

$$K = \frac{600^2 \sin(20^\circ) \left[\frac{1}{1\times10^5} + \frac{1}{1\times10^5}\right]}{1.4}$$

$K = 1.759\ N/mm^2$.

$F_w = d_1 . Q . K . b . = 400 \times 1.333 \times 1.759 \times 80$

$F_w = 75031.9\ N.$

Check : 2 :

FDB pg 8.51 $F_w \geq F_d$

$7031.9N > 8774.87N.$

So the design is safe

Step 12: Basic dimensions of Pinion & gear:

FDB pg 8.22 table

| Nomenclature | Notation | Unit | Spur gear. |
|---|---|---|---|
| Module | $m$ | mm | $m = 8mm.$ |
| Center distance | $a$ | mm | $a = 600\ mm.$ |
| Tooth depth | $h$ | mm | $h = 2.25\ m.$<br>$= 2.25 \times 6$<br>$h = 18\ mm.$ |
| Pitch diameter | $d$ | mm | $d_1 = mz_1 = 400mm$<br>$d_2 = mz_2 = 800mm$ |
| Number of teeth | $z$ | - | $z_1 = 50$<br>$z_2 = 100.$ |

Step 13: Results:

# COMPUTER AIDED DESIGN AND GEOMETRIC MODELLING

# Basic definition of CAD

- CAD

  Computer Aided Design may be defined as the use of computer system to help in the creation, modification, analysis and of a design.

# UNIT 5 – COMPUTER AIDED DESIGN

- TOPICS

  1. Shigley's process
  2. CAD activities
  3. Transformations
     i) Translation
     ii) Rotation
     iii) Scaling
  4. Geometric modelling
     i) Wireframe modelling
     ii) Surface modelling
     iii) Solid modelling
        a) CSG
        b) B-rep
  5. Finite Element Analysis

# 1. Shigley’s design process

- Recognition of need
- Definition of problem
- Synthesis
- Analysis and optimization
- Evaluation
- Presentation

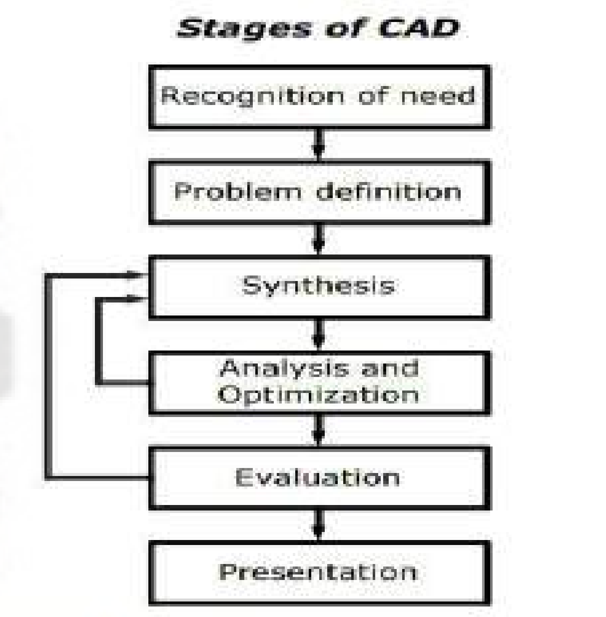



Fig.1.1 Shigley's design process

- Recognition of need
  - ❖It is the process of identifying the same defect in the design, which may need correction for better performance or identifying the new product which may satisfy the customer needs.

- Definition of problem
  - ❖It involves complete specifications of the product be designed. This specification includes.

    Physical characteristics

    size, shape, appearance, weight etc…

    Functional characteristics

    operating performance, cost, quality etc…

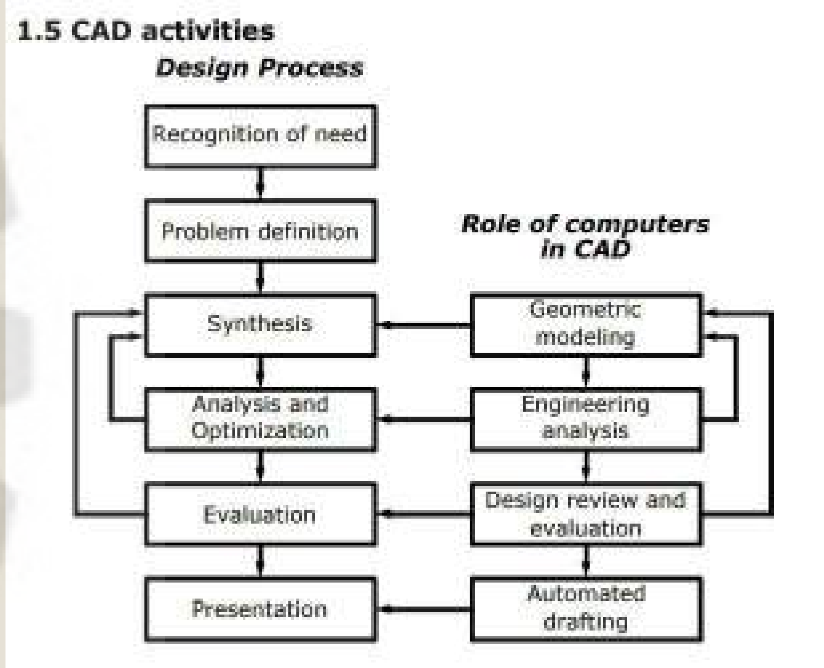
- Synthesis
  - ❖It is the process of developing new concepts about the shape, form and technology used in product by the creativity of the designer or by the research of similar products or design in use.

- Analysis and optimization
  - ❖The developed conceptual design is analysed to check the suitability for the intended purpose. If the developed design is not satisfied, then it is modified, redesigned and analysed till we get the optimized design.

- Evaluation
  - ❖ It is the process of measuring the design against the specification established in problem definition phase. This evaluation often requires the fabrication and testing of a prototype model to assess operating performance, quality, reliability and other criteria.
- Presentation
  - ❖ Documentation of design by means of drawings.
  - ❖ Material specifications.
  - ❖ Sectional views.
  - ❖ Assembly list.
  - ❖ Bill of materials etc…..

# 2. CAD Activities

- Geometric modelling
- Engineering analysis
- Design review and evaluation
- Automated drafting



- Geometric modelling
  - ❖Computer compatible mathematical description of the geometry of an object is called geometric modelling.

    The geometric models may be the any one of the following types.

    1. Wire frame modelling
    2. Surface modelling
    3. Solid modelling

    Example: Auto CAD, Pro-E etc…..

- Engineering analysis
  - ❖The created models can be analysed to check the suitability of the models for intended purpose.

    1. Stress-strain analysis
    2. Heat transfer analysis
    3. Fluid flow analysis
    4. Kinematic analysis
    5. Dynamic analysis
    6. FEA analysis etc…..

    Example: ANSYS, ANSYS CFX etx…

- Design review and evaluation

Checking the accuracy of design.

- ❖Dimensioning and tolerancing
- ❖Layering
- ❖Interference checking in assemblies.
- ❖Animation of designed mechanism by means of kinematics.
- ❖Evaluation of areas and volumes.
- ❖Evaluation of mass and inertia properties.

- Automated drafting

It involves the creation of hard copies of design directly from the CAD database.

- ❖Preparation of detailed drawings.
- ❖Preparation of sectional views.
- ❖Preparation of assembly drawings.
- ❖Preparation of material specification.
- ❖Preparation of bill of materials etc….

# Benefits of CAD

◦ Benefits of CAD in designing of engineering components :

- ✔ Productivity improvement in design.
- ✔ Shorter lead time.
- ✔ More flexibility in design.
- ✔ Fever design errors.
- ✔ Improved design analysis.
- ✔ Standardization of design, drafting and documentation.
- ✔ Easier creation and modification of design.
- ✔ Easier visualization of drawings
- ✔ Preparation of near and more understandable working drawings.
- ✔ Creation of realistic image of component before actually making it.

- **Benefits of CAD in manufacturing :**

- ✔ Tool and fixture design.
- ✔ Computer Aided Process Planning (CAPP).
- ✔ Production Planning and Control (PPC).
- ✔ Preparation of assembly lists and bill of materials.
- ✔ Coding and classification of components.
- ✔ Computer aided inspection.
- ✔ Preparation of NC part programs.
- ✔ Assembly sequence planning

# 3. Transformations

- The capability of any graphic software are mainly depends on the ability to change the orientation, size and shape of created model.

  2D Transformation

  3D Transformation

- The basic geometric transformations are

  1. Translation
  2. Rotation
  3. Scaling

# Translation

It involves moving the geometric elements from one location to another location.

- 2D Translation

It moves the object on X and Y plane along straight line by adding increments in X-axis and Y-axis.

The, Homogeneous representation of above matrix is,

$$\begin{bmatrix} x' \\ y' \\ 1 \end{bmatrix} = \begin{bmatrix} 1 & 0 & t_x \\ 0 & 1 & t_y \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ 1 \end{bmatrix}$$

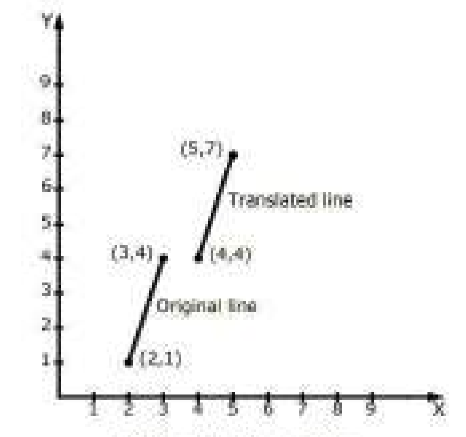



Fig.1.4 Translation

- 3D Translation

It move the object on xyz coordinates along straight line adding increments in x-axis, y-axis and z-axis.

$$\begin{bmatrix} x' \\ y' \\ z' \\ 1 \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & t_x \\ 0 & 1 & 0 & t_y \\ 0 & 0 & 1 & t_z \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix}$$

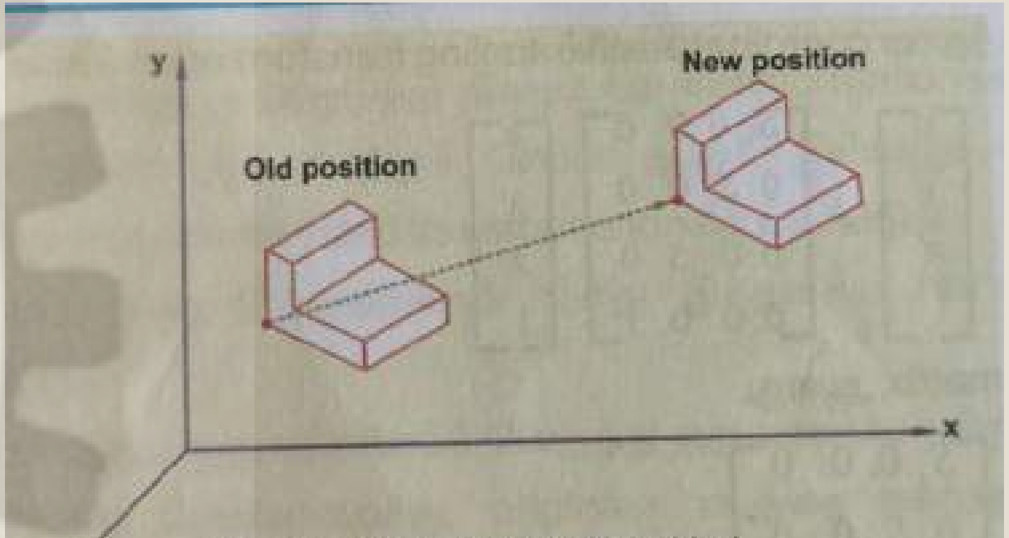



FIG 1.7 3D translation of an object

# Rotation

It is the rigid body transformation that moves the object along the circular path in xy plane or in xyz coordinates without any deformation.

- 2D Rotation

It involves the rotation of an object about its origin by an angle θ. For a positive angle, this rotation is in the counter–clockwise direction. The object is moved while rotating.

The, Homogeneous representation of above matrix is,

$$\begin{bmatrix} x' \\ y' \\ 1 \end{bmatrix} = \begin{bmatrix} cos\theta & 0 & 0 \\ 0 & sin\theta & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ 1 \end{bmatrix}$$

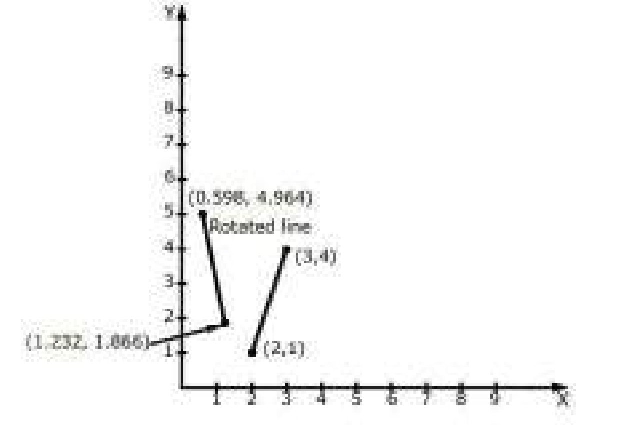



Fig.1.6 Rotation

- 3D Rotation

The, Homogeneous representation of above matrix is,

$$\begin{bmatrix} x' \\ y' \\ z' \\ 1 \end{bmatrix} = \begin{bmatrix} cos\theta & -\sin\theta & 0 & 0 \\ \sin\theta & cos\theta & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix}$$

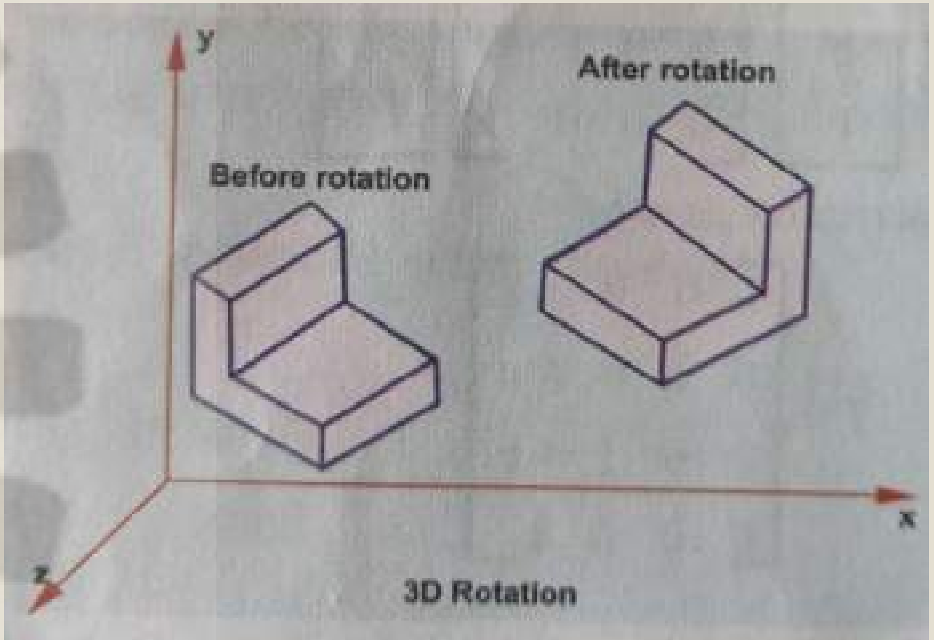



FIG 1.9

# Scaling

It is used to enlarge or reduce the size of the object

- 2D Scaling

The, Homogeneous representation of above matrix is,

$$\begin{bmatrix} x' \\ y' \\ 1 \end{bmatrix} = \begin{bmatrix} s_x & 0 & 0 \\ 0 & s_y & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ 1 \end{bmatrix}$$

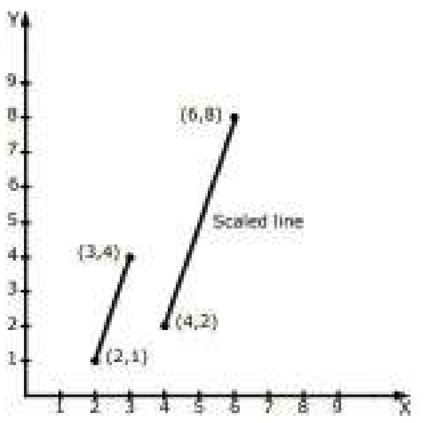


Fig.1.5 Scaling

## 3D Scaling

The, Homogeneous representation of above matrix is,

$$\begin{bmatrix} x' \\ y' \\ z' \\ 1 \end{bmatrix} = \begin{bmatrix} s_x & 0 & 0 & 0 \\ 0 & s_y & 0 & 0 \\ 0 & 0 & s_z & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix}$$

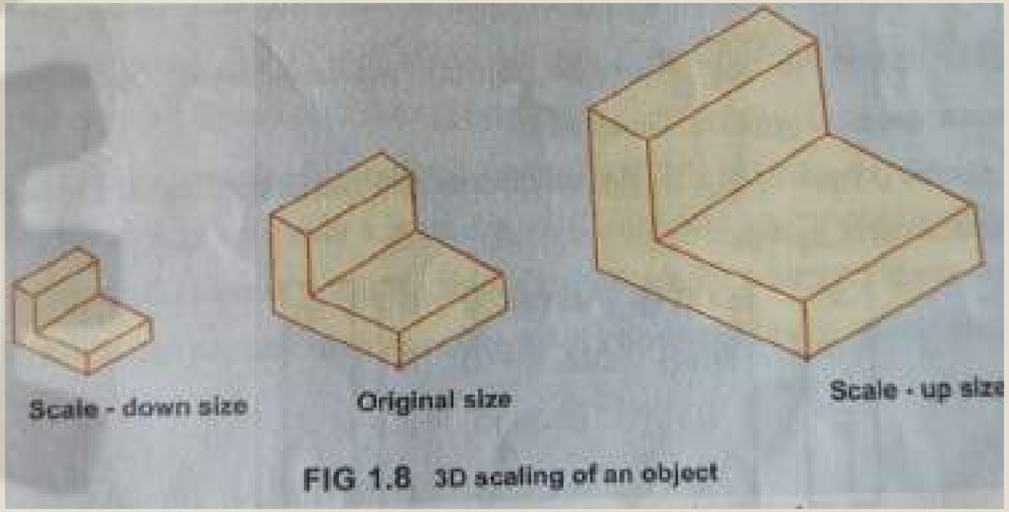


FIG 1.8 3D scaling of an object

# Geometric Modelling

- In CAD, geometric modeling is concerned with the computer compatible mathematical description of the geometry of an object.

There are several methods of Geometric modeling

1. Wire frame modeling
2. Surface modeling
3. Solid modeling

# WIREFRAME MODELING

- In this modelling the object is displayed by interconnecting lines.

2D model – Represent a flat object.

$2^{1}/_{2}$ D model – 3D object to be represented as long as it has no side-wall details.

3D model – Represented 3D object with more complex geometry.

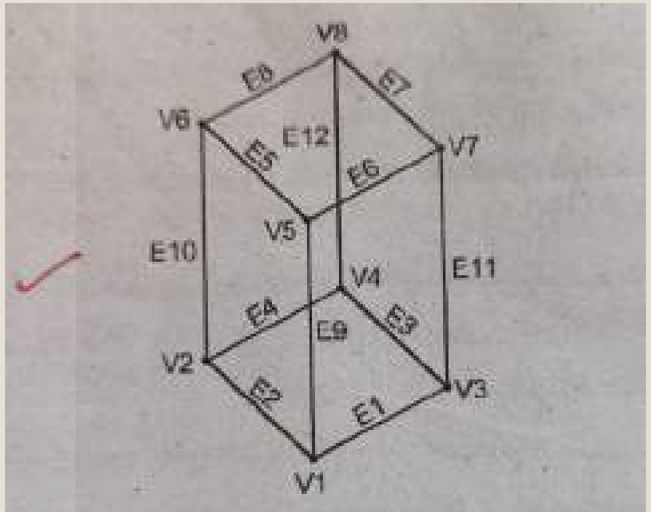



Fig.1.3.1 Wire frame model with straight edges

# SURFACE MODELING

- A surface model of an object is more complete and less confusing representation than its wireframe model.
- A surface model can be built by defining the surface on the wireframe model.
- The boundary of an object may consist of surface, which are bounded by straight lines and curves either single or in combination.

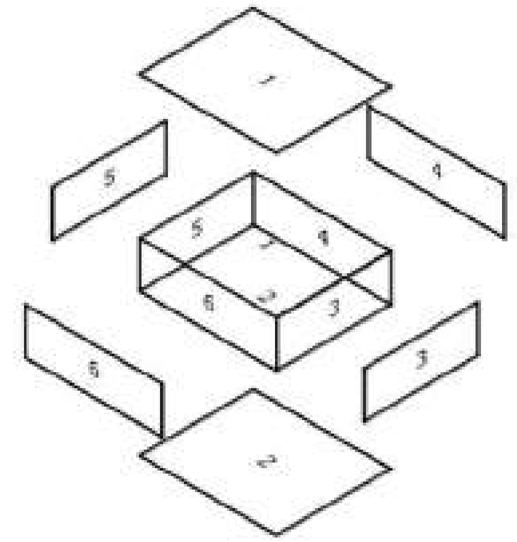


Fig.1.9 Representation of surface modelling

# SOLID MODELING

- The best method for the 3D model construction is the solid modeling technique. It provides the user with complete information about the model.

1. Creating
2. Modifying
3. Inspecting
4. Dimensions

Representation schemes are available for creating solid models

1. Constructive Solid Geometry (CSG)
2. Boundary representation
3. Pure Primitive instancing
4. Generalized sweep
5. Cellular decomposition
6. Hybrid scheme

- Constructive Solid Geometry (CSG)

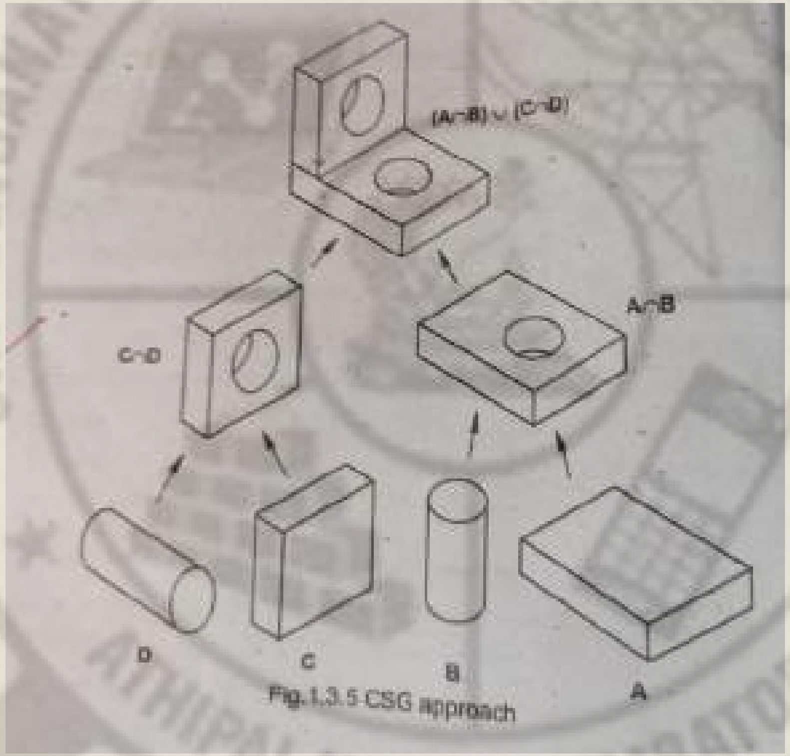
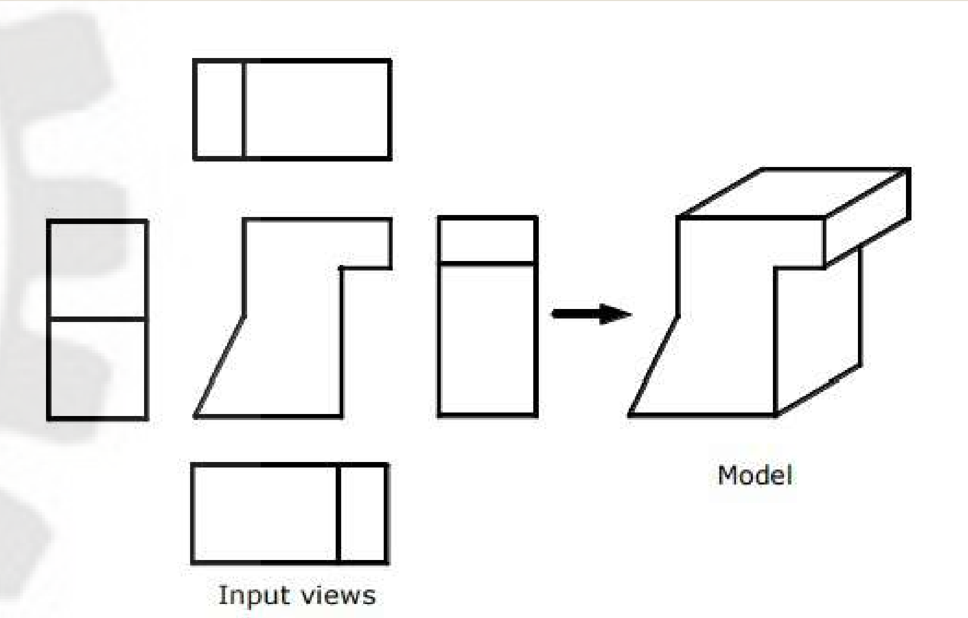


Fig.1.3.5 CSG approach

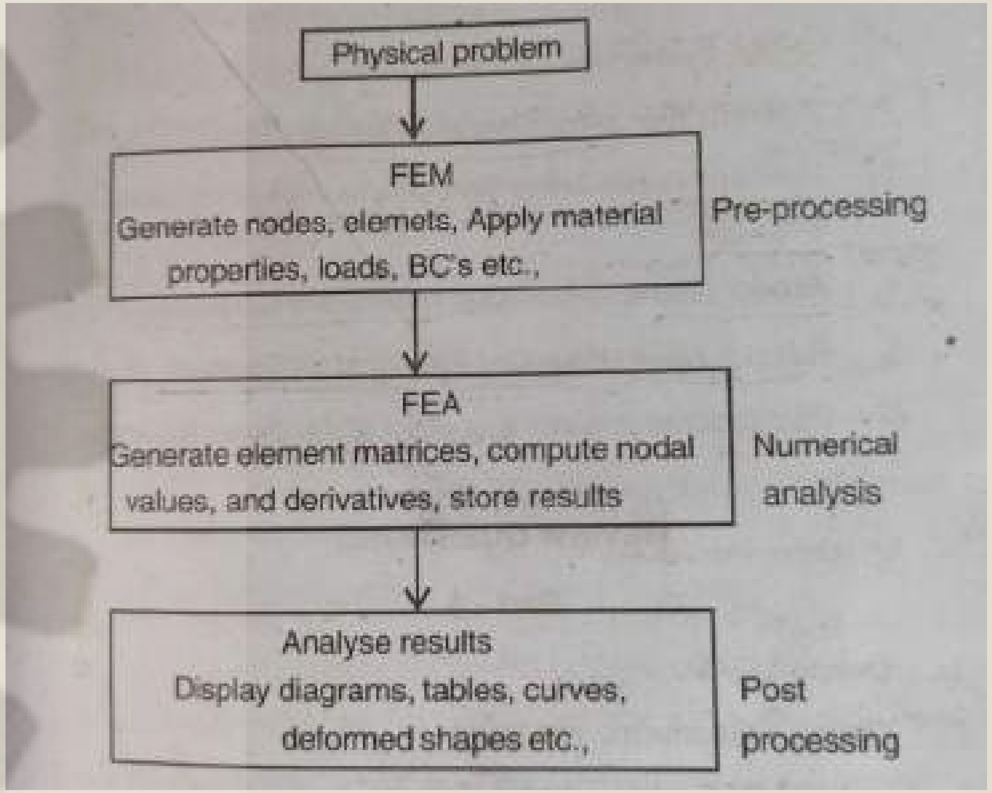
- Boundary Representation (B-rep)



# Finite Element Analysis

- Finite Element Analysis is a computer simulation technique used in engineering analysis to determine the behavior of structures and components under a variety of conditions.

  1. Pre-Processing
  2. Numerical analysis
  3. Post-Processing

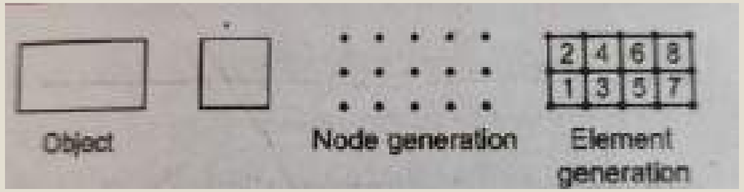


# Basic steps involved in FEA

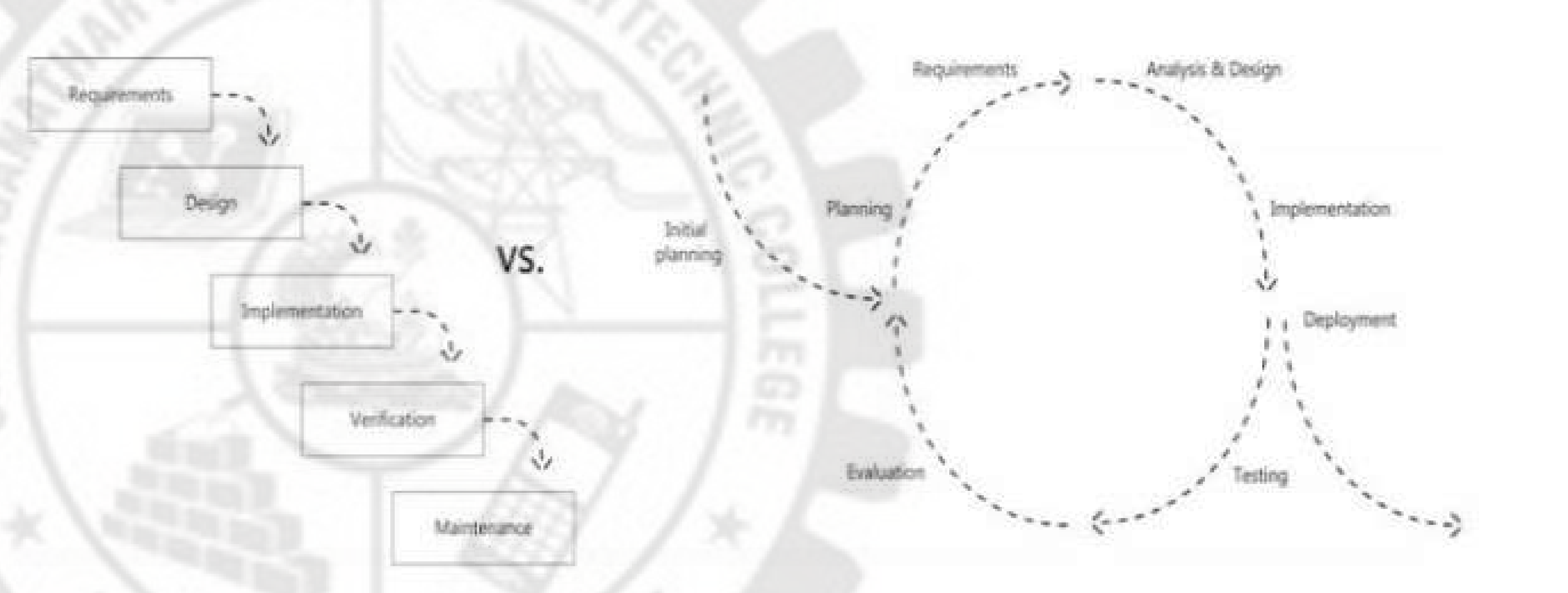
- Discretization of domain
- Approximate the solution within an element
- Develop element matrices and equation

1. Direct method
2. Variation method
3. Weighted residual method
4. Energy method

- Assemble element matrices into global system matrix and equations
- Solve the system of equation to find unknown values
- Interpret the results



# SEQUENCIAL AND CONCURRENT ENGINEERING




Sequential Vs Concurrent Engineering

# SEQUENCIAL AND CONCURRENT ENGINEERING

**Table 1.1. Sequential Vs Concurrent Engineering**

| Sequential Engineering | Concurrent Engineering |
|---|---|
| Sequential engineering is the term used to explain the method of production in a linear system. The various steps are done one after another, with all attention and resources focused on that single task. | In concurrent engineering, various tasks are handled at the same time, and not essentially in the standard order. This means that info found out later in the course can be added to earlier parts, improving them, and also saving time. |
| Sequential engineering is a system by which a group within an organization works sequentially to create new products and services. | Concurrent engineering is a method by which several groups within an organization work simultaneously to create new products and services. |
| The sequential engineering is a linear product design process during which all stages of manufacturing operate in serial. | The concurrent engineering is a non-linear product design process during which all stages of manufacturing operate at the same time. |
| Both process and product design run in serial and take place in the different time. | Both product and process design run in parallel and take place in the same time. |
| Process and Product are not matched to attain optimal matching. | Process and Product are coordinated to attain optimal matching of requirements for effective quality and delivery. |
| Decision making done by only group of experts. | Decision making involves full team involvement. |

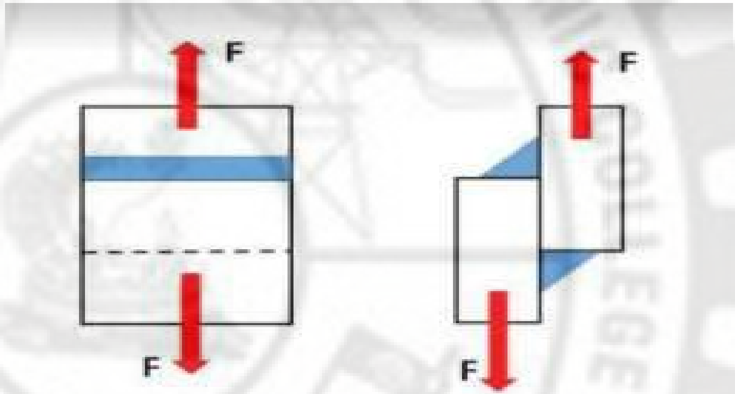
# DESIGN OF JOINTS

Unit I

# DESIGN OF FILLET WELDED JOINTS

Fillet welded joint □□□ □□□□□□□□ □□□□□□□□□□□□□.

1. Parallel fillet weld:



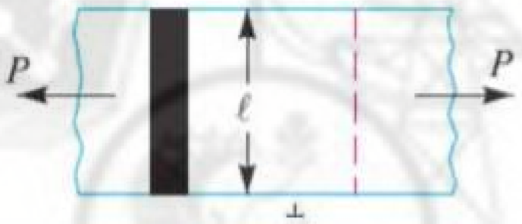

The design equation is, $\tau = 0.707\ P/h\ l$

(PSG Data Book Page No 11.3)

# Transverse fillet weld

a) Single transverse fillet weld:

The design equation is,

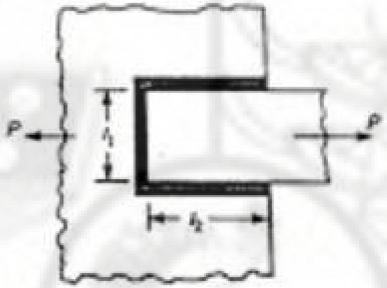



$$\sigma = \frac{1.414\,P}{h\,l_1}$$

where, σ = tensile stress

# Transverse fillet weld

## b) Double transverse fillet weld:




The design equation is,

$$\sigma = \frac{1.414\,P}{h\,l_1}$$

where, σ = tensile stress

# Combination transverse fillet weld

*The design equation is*

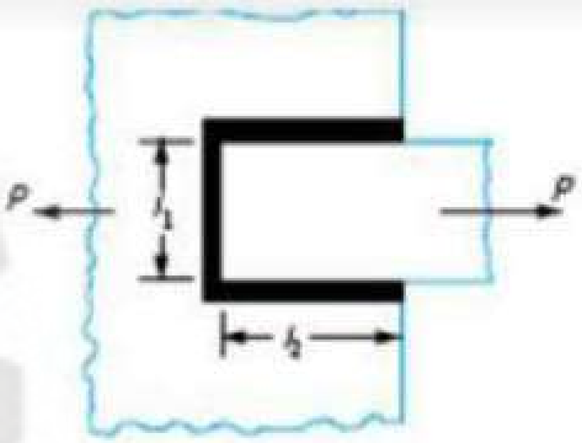
1. $P_1 + P_2 = P$
2. $\sigma_t = \dfrac{1.414\, P_1}{h\, l_1}$
3. $\tau = \dfrac{0.707\, P_2}{h\, l_2}$

*Where,*

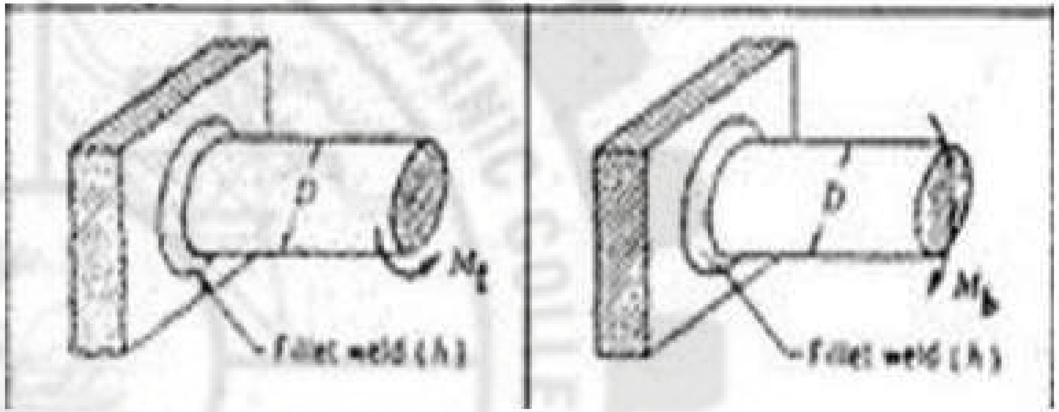
$P_1$ = *Load taken by transverse weld*

$P_2$ = *Load taken by parallel weld*

$P$ = *Total load*



# SPECIAL CASES OF FILLET WELDED JOINTS




$M_t$ = Twisting moment

$$\tau = \frac{2.83\, M_t}{h\, D^2\, \pi}$$

$M_b$ = Bending moment

$$\sigma = \frac{5.66\, M_b}{h\, D^2\, \pi}$$

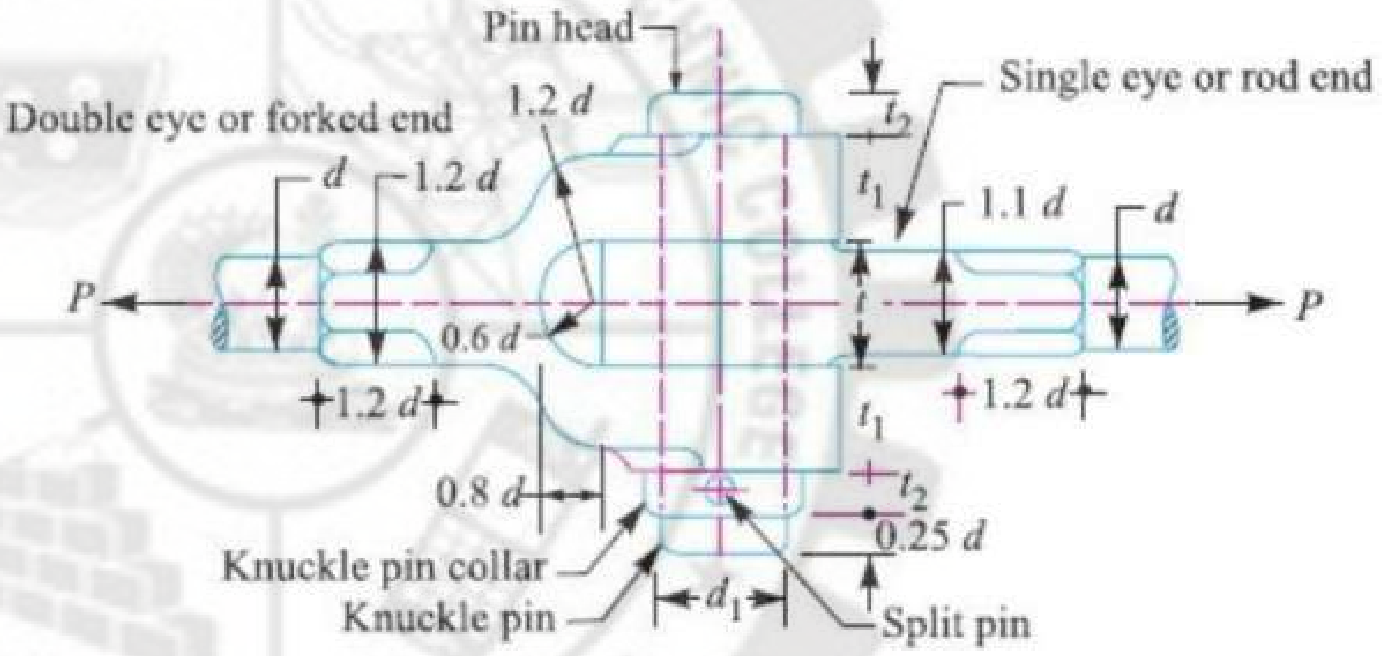
# DESIGN FOR KNUCKLE JOINT

- Step 1: To find the diameter of rod (d)
- Step 2: Calculate diameter and thickness

Check for failures

- Step 3: Check for failure of the knuckle pin in shear stress
- Step 4: Check for failure of the knuckle pin in crushing in single eye
- Step 5: Check for failure of the knuckle pin in crushing in double eye
- Step 6: Check for failure of the single eye end in tension
- Step 7: Check for failure of the single eye end in shearing
- Step 8: Check for failure of the double eye end in tension

# DESIGN FOR KNUCKLE JOINT



# DESIGN FOR KNUCKLE JOINT

**Step 1: To find the diameter of rod (d)**

$$Stress = \frac{\text{Load}}{\text{Area of rod}}$$ (PSG Data Book Page No 7.1)

$$\sigma = \frac{P}{a} \qquad \sigma = \frac{P}{\pi/4\, d^2}$$

$$\sigma = \frac{4P}{\pi d^2}$$

$$d^2 = \frac{4P}{\pi\sigma}$$

$$d = \sqrt{\frac{4P}{\pi\sigma}}$$ (Unit is mm)

where,

P is load

# DESIGN FOR KNUCKLE JOINT

*Step 2: Calculate diameter and thickness*

*(PSG Data Book Page No 7.140)*

| Equation | Nomenclature |
|---|---|
| $d_1 = d$ | $d$ Diameter of the rod |
| $d_2 = 2d$ | $d_1$ Diameter of the pin |
| $d_3 = 1.5d$ | $d_2$ Outer Diameter of the eye |
| $t = 1.25d$ | $d_3$ Diameter of the pin head |
| $t_1 = 0.75d$ | $t$ Thickness of the eye (or) single eye |
| $t_2 = 0.5d$ | $t_1$ Thickness of the fork (or) double eye |
| | $t_2$ Thickness of the pin head |

*Step 3: Check for failure of the knuckle pin in shear stress*

$$\text{Induced shear stress} = \frac{\text{Load}}{\text{Shearing area of knuckle pin}}$$

$$\tau = \frac{P}{2 \times \frac{\pi}{4} \times d_1^{\,2}}$$

where, $P$ is load

$d_1$ is diameter of the pin

$\tau$ is shear

# DESIGN FOR KNUCKLE JOINT

**Step 4: Check for failure of the knuckle pin in crushing in single eye**

$$\text{Induced crushing stress} = \frac{\text{Load}}{\text{crushing area of knuckle pin in single pin}}$$

$$\sigma_c = \frac{P}{d_1 \times t}$$

where, $P$ is load

$t$ is thickness

$d_1$ is diameter of the pin

**Step 5: Check for failure of the knuckle pin in crushing in double eye**

$$\text{Induced crushing stress} = \frac{\text{Load}}{\text{crushing area of knuckle pin in double pin}}$$

$$\sigma_c = \frac{P}{2 \times d_1 \times t_1}$$

where, $P$ is load

$t_1$ is thickness

$d_1$ is diameter of the pin

# DESIGN FOR KNUCKLE JOINT

**Step 6: Check for failure of the single eye end in tension**

$$\text{Induced tensile stress} = \frac{\text{Load}}{\text{tearing area of single eye end}}$$

$$\sigma_c = \frac{P}{(d_2 - d_1) \times t_1}$$

where, $P$ is load

$t_1$ is thickness

$d_1$ is diameter of the pin

$d_2$ is Outer diameter of the eye

**Step 7: Check for failure of the single eye end in shearing**

$$\text{Induced tensile stress} = \frac{\text{Load}}{\text{tearing area of single eye end}}$$

$$\sigma_c = \frac{P}{[(d_2 - d_1) \times t]}$$

where, $P$ is load

$t_1$ is thickness

$t$ is thickness of the eye

$d_2$ is Outer diameter of the eye

# DESIGN FOR KNUCKLE JOINT

**Step 8: Check for failure of the double eye end in tension**

$$\text{Induced tensile stress} = \frac{\text{Load}}{\text{tearing area of double eye end}}$$

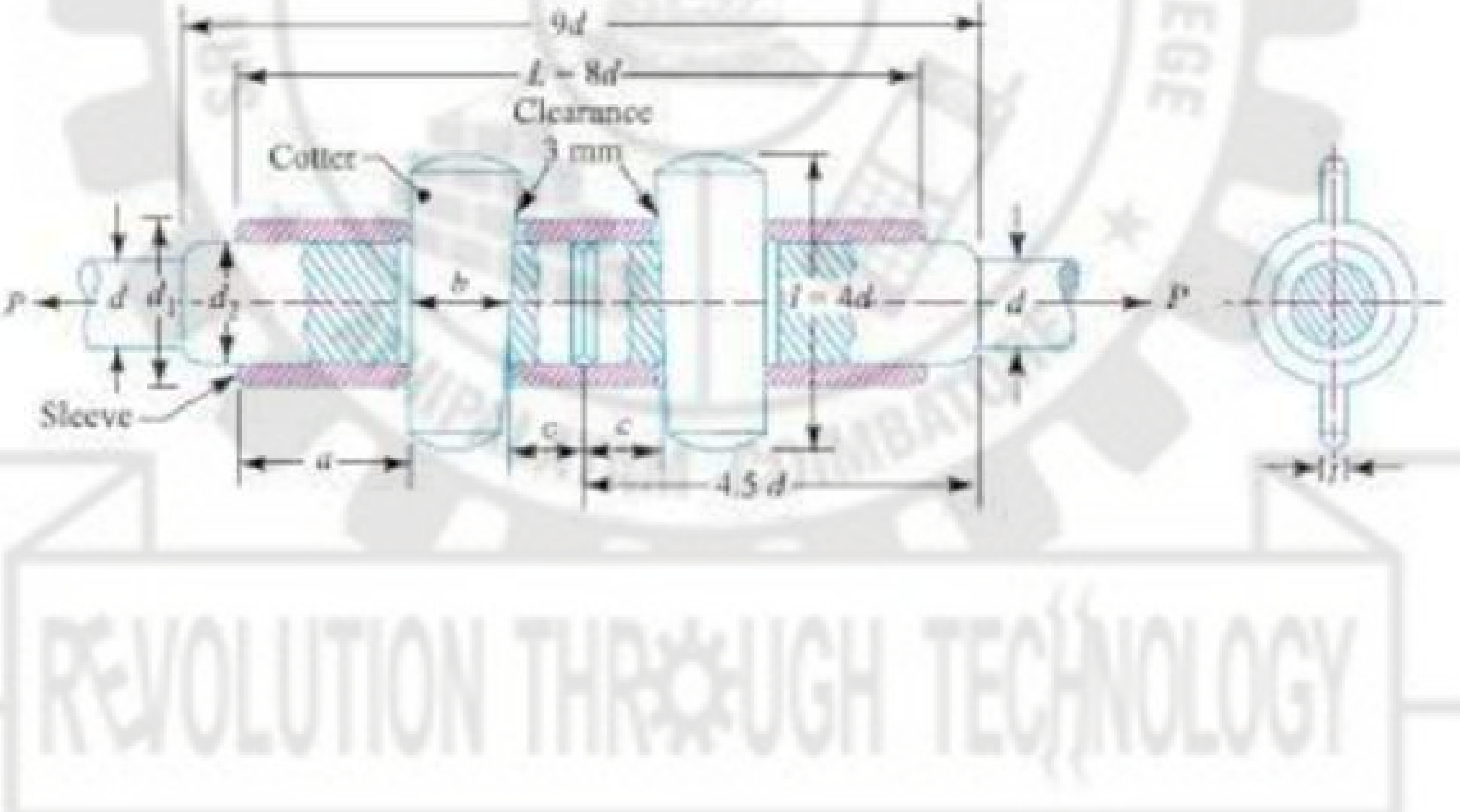
$$\sigma_c = \frac{P}{[2 \times (d_2 - d_1) \times t_1]}$$

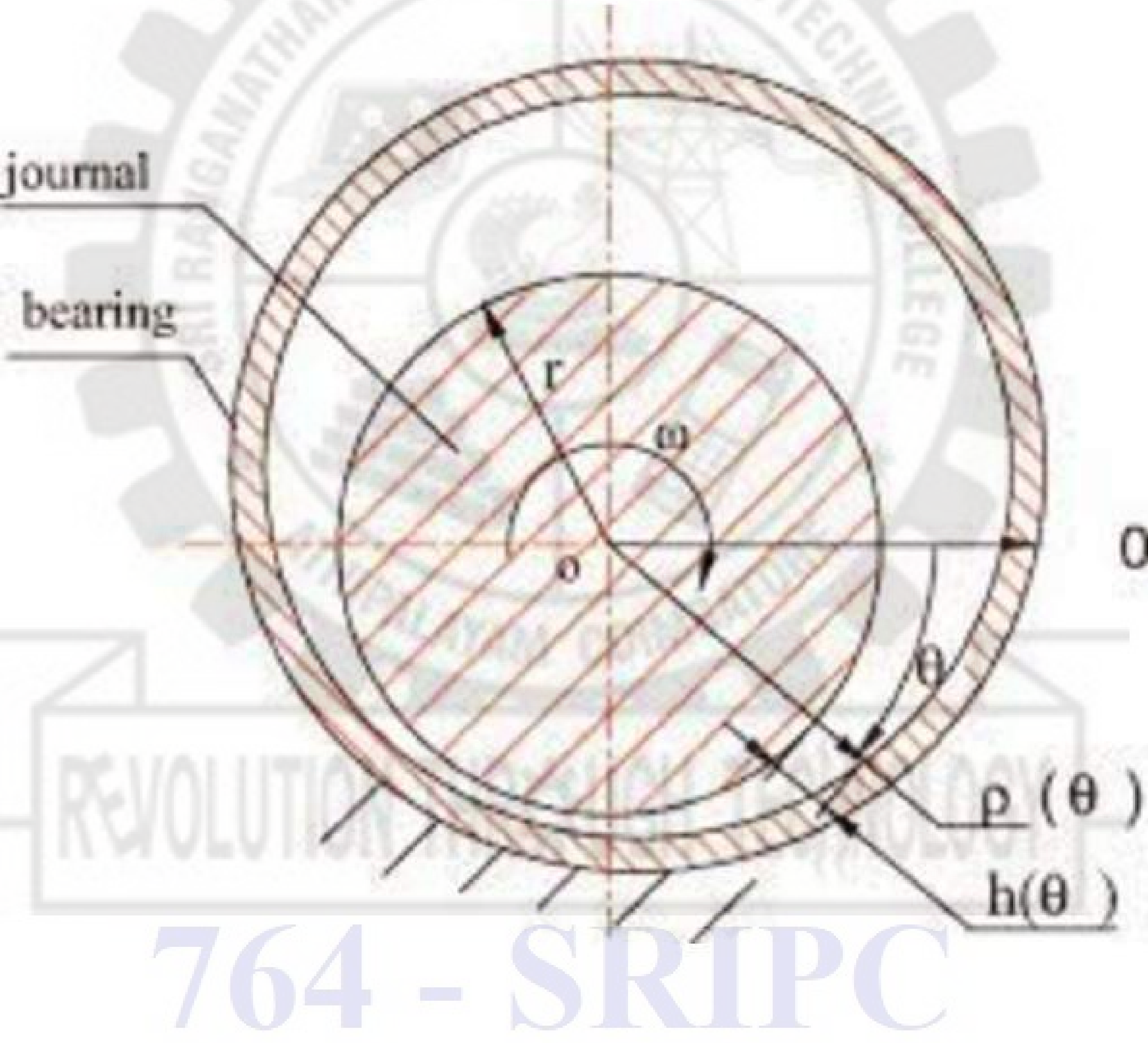
*where, P is load*

$t_1$ *is thickness*

$d_1$ *is diameter of the pin*

$d_2$ *is Outer diameter of the eye*


9d
L = 8d
Clearance
3 mm
Cotter
Sleeve
P
d
$d_1$
$d_2$
b
l = 4d
a
c
c
4.5 d
t


journal
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