PAGE NO.:
DATE: 5/06/2021

# Assignment - I

## 18AU63

1) Design a helical spring to operate for range of force 90N to 135 N. The deflection of spring for load range is 7.5mm. Take c=10 & FOS=2.

Ans Given:-

Max load $F_2 = 135\,N$, Min load $F_1 = 90\,N$.

$y' = 7.5\,mm$

$C = 10$

$FOS = 2$

Let us assume the materials as Chrome vanadium.

From table 20.10.

$\tau_y = 690\,MPa$

$G = 79.34\,GPa$

$$\therefore \tau = \frac{\tau_y}{FOS} = \frac{690}{2} = 345\,N/mm^2$$

Formula 20.31 DDHB

$$\text{max deflection } y = \frac{y' F_2}{F_2 - F_1} = \frac{7.5 \times 135}{135-90} = 22.5\,mm$$

i) Diameter of wire

$$\text{Shear stress } \tau = \frac{8 F_2 D K}{\pi d^3} \quad 20.22\ DDHB$$

PAGE NO. :
DATE : / /

Stress factor $K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$

$= \frac{4\times10-1}{4\times10-4} + \frac{0.615}{10} = 1.1448$

Spring index $C = \frac{D}{d}$ $\therefore D = Cd = 10d$

$\therefore 345 = \frac{8\times135\times10d\times1.1448}{\pi d^3}$

$\therefore d = 3.37mm$

From table 20.8

$d = 3.4mm$

ii) Diameter of coil

mean dia of coil $D = Cd = 10\times3.4 = 34mm$

Outer dia of coil $D_o = D+d = 34+3.4 = 37.4mm$

Inner dia of coil $D_i = D-d = 34-3.4 = 30.6mm$

iii) Number of coils

$y_a = \frac{8F_a D^3 i}{d^4 G}$ formula 20.29.

$22.5 = \frac{8\times135\times34^3\times i}{3.4^4\times79340}$

$\therefore i = 5.62$

PAGE NO.:
DATE: / /

Thus we consider $i = 6 \Rightarrow$ no of active coils.

iv) Free length.

From formula 20.53 (DDHB)

$l_0 \geq (i+n)d + y + a$

$a = 25\%$ of max deflection $= \frac{25}{100} \times 24.022 = \underline{6\,mm}$

Actual max deflection

$$y = \frac{8 \times 135 \times 34^3 \times 6}{3.4^4 \times 79340} = \underline{24.022\,mm}$$

Assuming squared & ground ends.
The number of aditional coil $n = 2$.

$$\therefore l_0 \geq (6+2) \times 3.4 + 24.022 + 6$$

$\geq 57.22\,mm$

v) Pitch
from table 20.14

$$p = \frac{l_0 - 2d}{i} = \frac{57.22 - 2 \times 3.4}{6} = 8.4\,mm$$

PAGE NO. :
DATE : / /

vi) Required stiffness

From formula 20.30 (DDHB)

$$F_o = \frac{F_{max}}{y} = \frac{135}{22.5} = 6 \text{ N/mm}$$

vii) Actual stiffness

~~From formula~~

$$F_o = \frac{d^4 G}{8 i D^3} = \frac{3.4^4 \times 79340}{8.6 \times 343}$$

$$= 5.04 \text{ N/mm}$$

viii) Total length of wire.

$l = \pi D i'$ $\quad i' = i + n$

$\therefore l = 3.14 \times 34 \times 8$

$= 854.513 \text{ mm}$

PAGE NO.:
DATE: / /

2) Design a helical spring to operate for range of Force ~~90~~ 100 N to ~~85~~ 160 N. The deflection of spring for load range is 7.5 mm. Take $C=8$ & FOS $=1.5$

Ans → Given:-

Max load $F_2 = 160\,N$, Min load $F_1 = 100\,N$

$y' = 7.5\,mm$

$C = 8$

$FOS = 1.5$

Lets assume the material as Chrome vanadium.

from table no. 10.

$\tau_y = 690\,MPa = 690\,N/mm^2$

$G = 79340\,MPa$.

$\therefore \tau_d = \dfrac{\tau_y}{FOS} = \dfrac{690}{2} = 345\,MPa$

Formula no. 31 DDHB

Max deflection $y = \dfrac{y' F_2}{F_2 - F_1} = \dfrac{7.5 \times 160}{160 - 100} = 20\,mm$

i) Diameter of wire    no. 22 DDHB

Shear stress $\tau = \dfrac{8 F_2 D k}{\pi d^3} =$

PAGE NO.:
DATE: / /

stress factor $k = \frac{4c-1}{4c-4} + \frac{0.615}{c}$

$= \frac{4\times10-1}{4\times10-4} + \frac{0.615}{10} = 1.1448$

spring index $c = \frac{D}{d}$ $\therefore$ $D = cd = \cancel{10}\,8d$.

$\therefore 345 = \frac{8\times160\times8d\times1.1448}{\pi\times d^3}$

$\therefore d^2 = 10.81$

$d = 3.28\text{mm} \approx 3.3\text{mm}$

ii) Dia of coil

mean dia of coil $D = Cd = 10\times3.3 = 33\text{mm}$

Outer dia of coil $D_o = D+d = 33+3.3 = 36.3\text{mm}$

Inner " " " $D_i = D-d = 33-3.3 = 29.7\text{mm}$

iii) Number of coils

$y_a = \frac{8F_a D^3 i}{d^4 G}$ 20.29 DDAB

$20 = \frac{8\times160\times33^3\times i}{3.3^4\times79340}$

$\therefore i = 4.090$

PAGE NO.:
DATE: / /

Thus we consider $i = 5$

iv) Free length 20.53 DDHB

$$l_0 \geq (i+n)d + y + a$$

$$a = 25\% \text{ of max defl} = \frac{20}{100} \times 29.33 = 5.86\,mm$$

Actual ~~de~~ max deflection

$$y = \frac{8 \times 160 \times 33^3 \times 6}{3.3^4 \times 79340} = 29.33\,mm$$

Assuming squared & ground end
the no of additional coil $n = 2$

$$\therefore l_0 \geq (6+2) \times 3.3 + 29.33 + 5.86$$

$$\geq 61.59\,mm$$

v) Pitch

Table 20.14 DDHB

$$P = \frac{l_0 - 2d}{i} = \frac{61.59 - 2 \times 3.3}{6} = 9.16\,mm$$

vi) Required stiffness

20.30 (DDHB)

$$F_0 = \frac{F_{max}}{y} = \frac{160}{40} = 4\,N/mm$$

PAGE NO.:
DATE: / /

vii) Actual ~~deflection~~ stiffness.

$$F_0 = \frac{d^4 G}{8 i D^3} = \frac{3.3^4 \times 79340}{8 \times 5 \times 33^3} = 6.54 \text{ N/mm}$$

viii) Total length of wire.

$l = \pi D i'$ $\quad i' = i + n$

$l = \pi \times 33 \times 7$

$= 725.707 \text{ mm}$

3) Derive the equation for stress in helical spring of circular wire.

**Ans.** Consider a helical compression spring made of circular wire and subjected to an axial load 'F'

Let, D = Mean dia of coil

d = dia of spring wire

i = number of ~~spring~~ active coils.

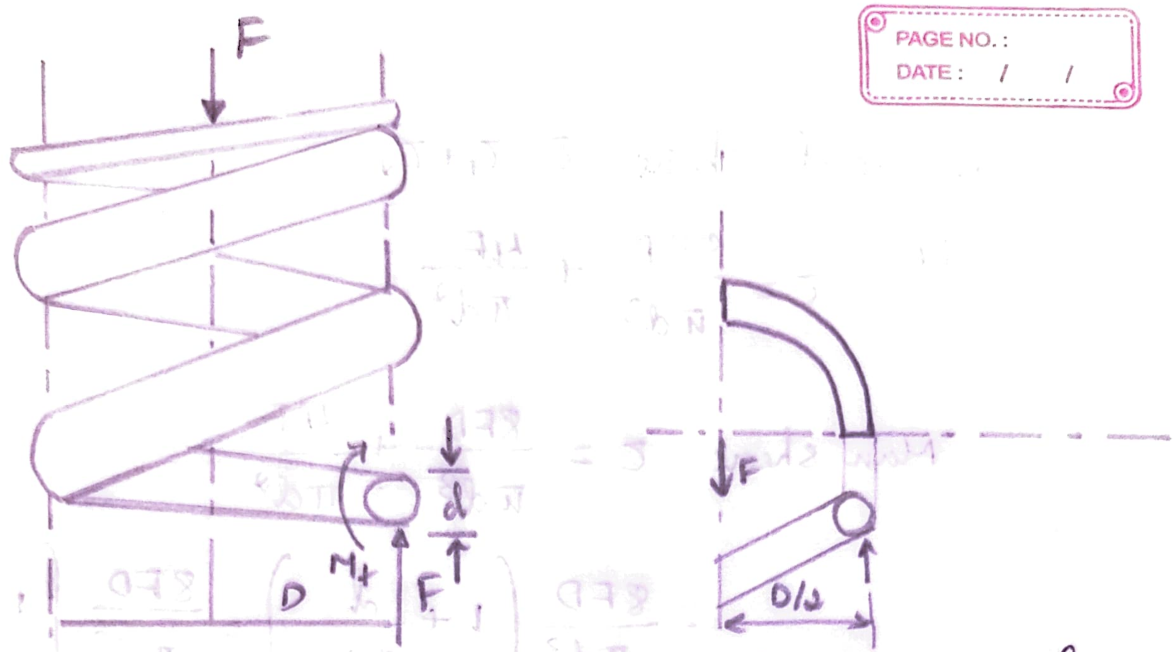
G = Modulus of rigidity.

F = Axial load on the spring.

$\tau$ = Max shear stress on the spring.

C = Spring Index.

y = Deflection of the spring.




Now consider a quadrant of a coil of a round wire spring. The load 'F' acting along the axis of the spring which has a mean diameter 'D' produces a torsional moment 'T' or $M_t$.

$$\therefore M_t \text{ or } T = F \cdot \frac{D}{2}$$

This torsional moment is also equal to $\frac{\pi}{16} \tau_1 d^3$

$$\therefore T = \frac{FD}{2} = \frac{\pi}{16} \tau_1 d^3$$

$$\therefore \tau_1 = \frac{8FD}{\pi d^3} = \text{Pmy shear.}$$

In addition there are 2 stresses acting.

i) Direct stress due to load 'F'

$$= \frac{\text{load}}{\text{C/s Area of wire.}}$$

$$\tau_2 = \frac{F}{\frac{\pi}{4} d^2}$$

$$\therefore \tau_2 = \frac{4F}{\pi d^2}$$

PAGE NO.:
DATE: / /

Resultant shear $\tau = \tau_1 + \tau_2$

i.e; $\tau = \dfrac{8FD}{\pi d^3} + \dfrac{4F}{\pi d^2}$

Max shear $\tau = \dfrac{8FD}{\pi d^3} + \dfrac{4F}{\pi d^2}$

$$= \frac{8FD}{\pi d^3}\left(1 + \frac{d}{2D}\right) = \frac{8FD}{\pi d^3}\left[1 + \frac{1}{2c}\right]$$

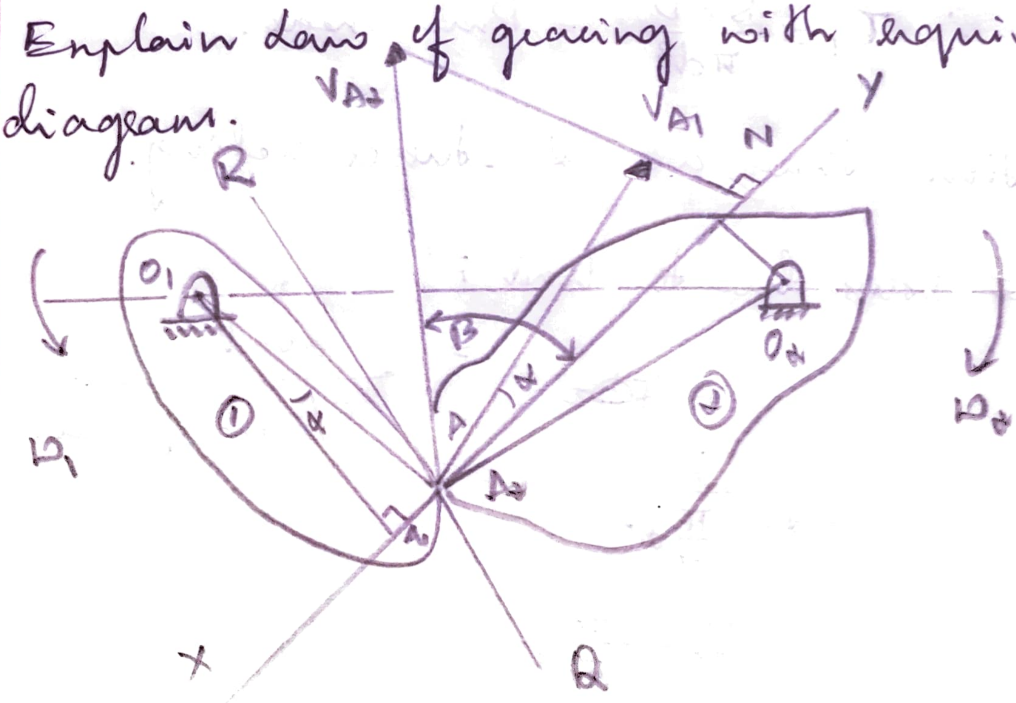
$\therefore \tau = \dfrac{8FD}{\pi d^3} \cdot K_s$ $\left(\because c = \dfrac{D}{d}\right)$

$K_s = 1 + \dfrac{1}{2c}$ = Shear stress factor.

$\therefore$ Max shear stress induced in the wire $\tau = \dfrac{8FD}{\pi d^3} \cdot k$

where $k = \dfrac{4c-1}{4c-4} + \dfrac{0.615}{c}$

4) Explain law of gearing with required equations and diagram.



PAGE NO.:
DATE: / /

when the tooth profiles are shaped, and if they produce a constant angular velocity ratio during ~~teeth~~ meshing, then they are said to have conjugate action.
Involute tooth profile is one of them which gives conjugate action.

* Now let us consider two curved bodies 1 & 2 rotating about centres $O_1$ & $O_2$ which is in contact at A.

* $A_1$ and $A_2$ are two coincident points. $A_1$ lying on body 1 and $A_2$ lying on body 2.

* RQ and XY are common tangent and common normal at the point ~~at the~~ of contact respectively.

* $W_1$ and $W_2$ are the angular velocities of $A_1$ and $A_2$ respectively.

* The linear velocities at the point of contact at the instant be $V_{A1}$ and $V_{A2}$.

* The direction of these are $\perp^{r}$ to the line joining $O_1$ to $A_1$ and $O_2$ to $A_2$ respectively.

* Let the normal to the intersection of the line joining the centres of rotation 1 & 2 be P.

* Let $O_2N$ be a perpendicular to the common normal from $O_2$ and $O_1M$ be a perpendicular to the common normal from $O_1$.
* If two bodies remain in contact then the linear velocities of $A_1$ & $A_2$ along the normal must be equal.

$$V_{A_1} \cdot \cos\alpha = V_{A_2} \cdot \cos\beta$$

$$\omega_1 \cdot O_1A_1 \cos\alpha = \omega_2 \cdot O_2A_2 \cdot \cos\beta$$

$$\therefore \omega_1 \cdot O_1A_1 \frac{O_1M}{O_1A_1} = \omega_2 O_2A_2 \cdot \frac{O_2N}{O_2A_2}$$

$$\therefore \frac{\omega_1}{\omega_2} = \frac{O_2N}{O_1M} = \text{velocity ratio.}$$

also triangles $O_1MP$ and $O_2NP$ are similar.

$$\therefore \frac{O_2N}{O_1P}$$

$$\therefore \text{velocity ratio} = \frac{\omega_1}{\omega_2} = \frac{O_2P}{O_1P}$$

Thus for constant angular velocity ratio of gearing, the normal at the point of contact divide the line joining the ~~contact~~ centres of rotation in the inverse ratio of the angular velocities.

PAGE NO.:
DATE: / /

Thus law of gearing states that for constant angular velocity ratio of the two gears, the common normal at the point of contact of the two meshing teeth must pass through a fixed point on the line joining the centre of rotation.

PAGE NO.:
DATE: / /

5) Design a spur gear to transmit power of 15kW at 1000rpm. Check suitability and saffty. Suggest suitable BHN, Take $i = 3.5$.

Ans → Given:-

$P = 15\,kW$, $N_1 = 1000\,rpm$, safety = ? BHN = ? $i = 3.5$

Transmission ratio $i = 3.5$.

Calculating no of teeth ---(23.6) DDHB

assuming profile as 20° FDI ∴ $z_1 > 14$

Lets take $z_1 = 20$

WKT; $z_2 = i z_1$ ∴ $z_2 = 3.5 \times 20 = 70$

∴ $z_1 = 20$ & $z_2 = 70$

$n_1 = 1000\,rpm$

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

$3.5 = \frac{1000}{n_2}$

∴ $n_2 = 285.71\,rpm$

Lewis form Factor.

For 20° FDI $y = 0.154 - \frac{0.912}{z}$ 23.116 DDHB

Material ⇒ SAE 2320 for both gear and pinion 23.18 DDHB

∴ $\sigma = 345\,MPa$

Pinion = $y_1 = 0.154 - \frac{0.812}{20} = 0.1084$.

Gear = $y_2 = 0.154 - \frac{0.912}{70} = 0.1409$.

| Particular | $\sigma$ | $y$ | $(\sigma \times y)$ | Remarks |
| --- | --- | --- | --- | --- |
| * Pinion | 345 | 0.1084 | 37.39 | Weaker. |
| * Gear | 345 | 0.1409 | 48.63 | |

"Pinion is weaker member one design will be based on pinion".

Tangential load ---- 23.97 (DDHB)

$$F_t = \frac{9550 \times 1000 \times P \times C_s}{n_1 \times r_1}, \quad \left[r = \frac{mz}{2}\right] \left[C_s = 1.5\right]$$

$$\left[\because r_1 = \frac{mz_1}{2}\right]$$

$$= \frac{9550 \times 1000 \times 15 \times 1.5 \times 2}{1000 \times m \times 20}$$

$$\therefore F_t = 21.487 \times 10^3 / m \longrightarrow (1)$$

Lewis tangential tooth load,

$$F_t = \sigma \times b \times y \times p \times C_v \qquad [y = y_1]$$

$$= 345 \times 10 \times m \times 0.1084 \times \pi \times m$$

$b = 9.5m - 12.5m$

$\therefore b = 10m$

$p = \pi \times m.$

$$F_t = 1.174 \times 10^3 \, m^2 C_v \longrightarrow (2)$$

Equating (1) = (2)

$$\frac{21.487 \times \cancel{10^3}}{m} = 1.174 \times \cancel{10^3} \, m^2 C_v$$

$$\therefore m^3 C_v = 18.302 \longrightarrow (3)$$

Mean velocity

$$V_m = \frac{\pi \times d_1 \times n_1}{60 \times 1000} = \frac{\pi \times m z_1 \times n_1}{60 \times 1000} = \frac{\pi \times m \times \cancel{20} \times \cancel{1000}}{\cancel{360}_{3} \times \cancel{1000}}$$

$\therefore V_m = 1.047m$ m/s

Table 23.3 DDHB

Trial ① $m = 4$ mm

$V_m = 1.047 \times 4 = 4.188$ m/s $< 7.5$ m/s.

$\therefore C_v = \frac{3}{3 + V_m} = \frac{3}{3 + 4.188} = 0.4173.$ 23.134a DDHB

② $\Rightarrow m^3 C_v > 18.302$

$4^3 \times 0.417 > 18.302$

$\Rightarrow 26.74 > 18.302$

$\therefore$ It passes the requirement.

From T (23.1) DDHB

Addendum $\Rightarrow 1 \times m = 1 \times 4 = 4$ mm.

Deddendum $\Rightarrow 1.25 \times m = 1.25 \times 4 = 5$ mm.

working depth $\rightarrow 2 \times m = 2 \times 4 = 8$ mm.

Total depth $\Rightarrow 2.25 \times m = 2.25 \times 4 = 9$ mm.

Tooth thickness $\Rightarrow (\pi/2) \times m = (\pi/2) \times 4 = 6.28$ mm.

minimum

Clearance ⇒ $0.25 \times m = 0.25 \times 4 = 1mm$.

Approx fillet radius → $0.3 \times m = 0.3 \times 4 = 1.2mm$.

Backlash ⇒ 0mm

Pitch dia

⇒ Pinion ⇒ $z_1 \times m = 20 \times 4 = 80mm$.

⇒ Gear ⇒ $z_2 \times m = 70 \times 4 = 280mm$.

Outside dia

⇒ Pinion ⇒ $(z_1+2)m = (20+2) \times 4 = 88mm$.

⇒ Gear ⇒ $(z_2+2)m = (70+2) \times 4 = 288mm$.

Check for stress

Allow stress ; $\sigma_{allow} = \sigma \times C_v = 345 \times 0.4173$

$= 143.96$ MPa

Induced stress ; $\sigma_{Find} = \dfrac{F_t}{b y_2 P} = \dfrac{(21.48 \times 10^3/4)}{10 \times 4 \times 0.1084 \times \pi \times 4}$

$\therefore \sigma_{ind} = 98.58$ MPa

PAGE NO.:
DATE: / /

∴ $\sigma_{allow} > \sigma_{ind}$ thus the design is safe.

Check for factor of saftey.

WKT;

$$n = \frac{\sigma_{allow}}{\sigma_{ind}}$$

$$= \frac{143.96}{98.58} = 1.460$$

BHN :- (23.37B) 00HB.

Having stress value of 345 MPa ≈ 413.98 MPa

BHN of pinion = 200

BHN of gear = 150

6) Design a spur gear to transmit power of 20kW at 1500 rpm. Check suitability and saftey. Suggest suitable BHN, take i = 4.

1.

Data:

$P = 20\,kW$, $N_1 = 1500$ rpm, safety = ? BHN = ?

$i = 4$

$i = 4$

from 23.6 DDHB

assuming $\alpha = 20°$

$z_1 \geq 14$

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

$\therefore n_2 = \frac{n_1}{i} = \frac{1500}{4}$

$n_2 = 375$ rpm

Lets take $z_1 = 20$

wkt,

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

$z_1 = 20$ & $z_2 = 80$

Lewis form factor.

for 20° FDI $y = 0.154 - \frac{0.912}{z}$

from 23.116 DDHB.

material ⇒ SAE 2320 for both gear & pinion.

$\therefore \sigma = 345$ MPa $\longrightarrow$ 23.18 DDHB.

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

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

| Particular | $\sigma$ | $y$ | $(\sigma \times y)$ |
|---|---|---|---|
| Pinion | 345 | 0.1084 | 37.39 → weaker |
| Gear | 345 | 0.1426 | 50.48 |

Thus pinion is weaker member.

Design

Tangential load.

$$F_t = \frac{9550 \times 1000 \times P \times C_s}{n_1 \times d_1} \qquad \left(d = \frac{mz}{2}\right) \left(C_s = 1.5\right) \quad \therefore \left(d_1 = \frac{mz_1}{2}\right) = \frac{m \times 20}{2}$$

$$F_t = \frac{9550 \times 1000 \times 20 \times 1.5 \times 2}{1500 \times m \times 20}$$

$$\therefore F_t = \frac{19100}{m}$$

$$\therefore F_t = 19.10 \times 10^3 / m \rightarrow (1)$$

PAGE NO. :
DATE : / /

Lewis tangential tooth load.

$F_t = \sigma \times b \times y \times p \times C_v$

$= 345 \times 10 \times m \times 0.1084 \times \pi \times m \times C_v$

DD#B
2393
$D = 10m$
$p = \pi m$

$\therefore F_t = 1.174 \times 10^3 \; m^2 C_v \longrightarrow (2)$

equating (1) = (2)

$\frac{19.100 \times \cancel{10^3}}{m} = 1.174 \times \cancel{10^3} m^2 C_v$

$\therefore m^3 C_v = 16.269 \longrightarrow (3)$

mean velocity

$V_m = \frac{\pi \times d_1 \times n_1}{60 \times 1000}$

$= \frac{\pi \times m\, z_1 \times n_1}{60 \times 1000}$

$= \frac{\pi \times m \times 20 \times 1500}{60 \times 1000}$

$\therefore V_m = 1.570\, m$ m/s

PAGE NO.:
DATE: / /

From table 23.3 DDHB

trial ① $m = 4mm$

~~$V_m = \pi \times 0.4 \times 4 = 4.1$~~

1.57

$V_m = 1.570 \times 4 = 6.28 < 7.5 m/s.$

② ~~$\Rightarrow m^3 C_v$~~

$$C_v = \frac{3}{3 + V_m} = \frac{3}{3 + 6.28} = 0.323$$ $\rightarrow$ 23.1844 DDHB.

③ $\Rightarrow m^3 C_v > 18.300$

$$4^3 \times 0.323 > 18.300$$

$$= 20.672 > 18.300$$

∴ It passes the requirement.

From table (23.1) DDHB.

Addendum $\Rightarrow$ $1 \times m = 1 \times 4 = 4mm$

Deddendum $\Rightarrow$ $1.25 \times m = 1.25 \times 4 = 5 mm$

work depth $\Rightarrow$ $2 \times m = 2 \times 4 = 8 mm.$

total depth $\Rightarrow$ $2.25 \times m = 2.25 \times 4 = 9 mm.$

Tooth thickness $\Rightarrow$ $(\pi/2) \times m = (\pi/2) \times 4 = 6.29 mm$

minimum Clearance $\Rightarrow 0.25 \times m = 0.25 \times 4 = 1mm$

Approx fillet radius $\Rightarrow 0.3 \times m = 0.3 \times 4 = 1.2mm$.

Backlash $\Rightarrow$ 0mm

Pitch dia

$\Rightarrow$ Pinion $\Rightarrow Z_1 \times m = 20 \times 4 = 80mm$.

$\Rightarrow$ Gear $\Rightarrow Z_2 \times m = 80 \times 4 = 320mm$ //

Outside dia

$\Rightarrow$ Pinion $= (Z_1 + 2)m = (20 + 2) \times 4 = 88mm$

Gear $= (Z_2 + 2)m = (80 + 2) \times 4 = 328mm$ //

## Stress

$\sigma_{allow} = \sigma \times C_v = 345 \times 0.323 = 111.435$ MPa. //

Induced Stress ; $\sigma_{ind} = \dfrac{F_t}{b y_p p} = \dfrac{(19.100 \times 10^3 / 4)}{10 \times 4 \times 0.1084 \times \pi \times 4}$

$=$

$\sigma_{ind} = 87.63$ MPa //

∴ $\sigma_{allow} > \sigma_{find}$ ∴ Design is saff.

~~Step~~ 8:- ~~$\phi$~~

For factor of saffty.

WKT $n = \dfrac{\sigma_{allow}}{\sigma_{ind}} = \dfrac{111.435}{87.63} = 1.27$

<u>BHN</u> → 23.378 DDHB.

having stress value of 345 MPa

$\approx$ 413.95 MPa

BHN of pinion = 200

BHN of gear = 150