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ME8651 Design of Transmission Systems Formula

Flat Belt drive

1) Selection of pulley diameter:

DB · Pg · NO : 7.52

2) Design power in kN

DB · Pg NO : 7.53

3) Selection of a belting

DB · Pg NO : 7.54

4) Load rating correction

DB · Pg NO : 7.54

5) Determination of Belt width :-

$$\text{width of belt} = \frac{\text{Design power}}{\text{Load rating} \times \text{NO of plies}}$$

Standard width D.B Pg.NO : 7.52

6) Determination of pulley width:

D.B · Pg · NO : 7.54

7) Length of the belt (L) :

D.B · Pg · NO : 7.53

____ x ____ x ____

Basic equation for flat belt drive

$$\frac{T_1 - mv^2}{T_2 - mv^2} = e^{\frac{\mu\alpha}{\sin\beta}} = e^{\mu\alpha \cdot \operatorname{cosec}\beta}$$

where,

β - Groove angle (180° for flat belts)

T_1 - tension on the tight side

T_2 - tension on slack side

Power transmitted (P) = $(T_1 - T_2)v$

area = $b \times t$

Mass of belt = Density x Volume

Centrifugal tension $T_c = mv^2$

worm and wheel design

Find in order

$a = 8.44$

$m_x = 8.84$ / std m_x 8.2

$a = 8.44$ or 8.43

$[M\bar{E}] = 8.44$

$V_s = 8.44$ | $z_v = \frac{z}{\cos^3\gamma}$

$\sigma_b \le [\sigma_b]$ all in 8.44

$\sigma_c \le [\sigma_c]$

V-BELT DRIVE

1) Selection of belt section

D.B Pg.NO: 7.58

2) Selection of pulley diameter (d and D)

D.B Pg NO : 7.54

3) Selection of centre distance (c):

DB Pg NO : 7.61

4) Determination of nominal pitch (L)

D.B Pg NO : 7.61 & 7.59, 7.60

5) Selection of various modification factors:-

i) Length correction factor (F_c) - D.B Pg.NO 7.59 - 7.60

ii) correction factor for arc of contact (F_d)

D.B · Pg NO : 7.68

6) iii) Service factor (F_a)

D.B Pg.NO: 7.69

6) maximum power capacity (KW):-

D.B · Pg · NO : 7.62

7) Determination of number of belts (n_b):-

D.B · Pg NO : 7.70

8) Calculation of actual centre distance

C_{actual} = D.B Pg.NO: 7.61

____ x ____ x ____

ADDITIONAL FORMULA :-

Power per belt = $(T_1 - T_2)v$

$$\frac{T_1 - mv^2}{T_2 - mv^2} = e^{\mu\alpha \cdot \operatorname{cosec}\beta}$$

stress induced = $\frac{\text{maximum tension}}{\text{cross-sectional area}}$

$T_c = m \cdot v^2$ / $T_{max} = \sigma \times$ area

CHAIN DRIVE (ROLLER CHAIN)

1) Selection of the transmission ratio

D.B Pg.NO: 7.74

2) Selection of number of teath on the driver sprocket (z_1):

DB · Pg · NO : 7.74

3) Determination of number of teath on driven sprocket (z_2) :

$z_2 = i \times z_1$ [D.B · Pg : NO : 7.74]

Recommended value z_2

4) Selection of standared pitch (P)

$a = (30 - 50)P$

D.B Pg.NO: 7.74

5) Selection of the chain

D.B · Pg NO : 7.71, 7.72, 7.73

6) Total load on the driving side (P_T)

$P_T = P_t + P_c + P_s$

P_t, P_c, P_s - D.B · Pg · NO : 7.78

7) Calculation of service factor (K_s)

$K_S = K_1 \cdot K_2 \cdot K_3 \cdot K_4 \cdot K_5 \cdot K_6$

DB · Pg · NO : 7.76

8) Design load:

Design load = $P_T \times K_S$

9) working factor of safety (FS_ω):-

$$F.S_\omega = \frac{Q}{P_T \times K_S}$$

10) Factor of safety (n') : (check)

D.B · Pg NO : 7.77 $F_{S\omega} > n'$

11) check bearing stress $[\sigma]$:-

$\sigma = \frac{P_t \times K_s}{A}$ $\sigma \angle [\sigma]$ D.B Pg NO: 7.7

if $\sigma > [\sigma]$, Design not safe.

12) actual length of chain (L) D.B · 7.75

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GEAR DESIGN USING LEWIS AND BUCKINGHAMS EQUATION

SPUR GEAR

1) Selection of material:
D.B.Pg.NO: 1.40

2) Calculation of z_1 and z_2:
If not given $z_1 \geq 17$, say 18
$z_2 = i \times z_1$
i - gear ratio.

3) tangential load on tooth (F_t)
D.B Pg NO: 8.50

4) Initial dynamic load (F_d):
D.B Pg.NO: 8.50

5) Beam strength (F_s)
D.B. Pg.NO: 8.50

6) Cal. of module (m)
$F_s \geq F_d$
standard the module value using D.B. Pg.NO: 8.2

7) calculate of b, d, v
$b = 10m$
$d_1 = z \cdot m$
$v = \pi d_1 N_1 / 60$

8) Recal. beam strength (F_s):
D.B. Pg. NO: 8.50 using std module.

9) Accurate dynamic load (F_d)
D.B. Pg.NO: 8.51

10) check for beam strength
$F_d \leq F_s$ Design safe

11) check for wear load (F_w)
$F_d \leq F_w$ Design safe

12) Basic dimension of pinion & Gear:
D.B Pg NO: 8.22

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HELICAL GEAR

1) Selection of material
D.B. Pg NO: 1.40

2) Calculation of z_1 and z_2:
$i = \frac{z_2}{z_1}$

3) tangential load on tooth (F_t)
D.B. Pg.NO: 8.50 (or) 8.51

4) Initial dynamic load (F_d):
D.B Pg.NO: 8.50 & 8.51

5) Beam strength (F_s):
D.B. Pg.NO: 8.51

6) Normal module (m_n):
$F_s \geq F_d$
Std the m_n using D.B Pg.NO: 8.2

7) calculation of b, d, & v
$b = 10 m_n$
$d_1 = \frac{m_n}{\cos\beta} \times z_1$
helical angle $\beta = 15°$ assume

8) Recal beam strength (F_s):
D.B Pg.NO: 8.51

9) Accurate dynamic load (F_d):
D.B Pg NO: 8.51

10) check for beam strength (or tooth breakage):
$F_d \leq F_s$ Design safe. fail rise the module value

11) cal. maximum wear load (F_w):
D.B. Pg.NO: 8.51

12) check for wear load:
$F_d \leq F_w$. Design safe. fail rise the module value

13) Basic dimension of pinion & gear
D.B Pg.NO: 8.22

common assume
$z_1 = 18$, $\psi = 0.3$, $v = 15$ m/s assume

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BEVEL GEAR

1) Selection of material:
D.B Pg.NO: 8.4

2) Calculation of z_1 and z_2 & pitch angle:
$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$ / $\tan\delta_2 = i$, $\delta_1 + \delta_2 = 90°$

3) virtual Number of teeth (z_v):
D.B. Pg.NO: 8.39

4) tangential load on tooth (F_t)
D.B. Pg.NO:

5) dynamic load (F_d):
D.B. Pg NO: 8.52

6) Beam strength (F_s):
D.B Pg.NO: 8.52

7) Transverse module (m_t):
$F_s \geq F_d$
Std the m_t using D.B Pg NO: 8.2

8) cal. b, d, & v:
$b = 10 m_t$
$d_1 = m_t \times z_1$
$v = \frac{\pi d_1 N_1}{60}$

9) Recal. beam strength (F_s):
D.B. Pg.NO: 8.52

10) Accurate dynamic load (F_d)
D.B Pg NO: 8.51

11) check for beam strength:
$F_d \leq F_s$ Design safe. fail, rise the module value

12) cal. maximum wear load: (F_w):
$F_w = \frac{0.75 \times d_1 \times b \times Q' \times K_w}{\cos\delta_1}$

13) Check for wear load:
$F_d \leq F_w$

14) Basic dimension of Bevel gear and wheel
D.B. Pg NO: 8.38

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SPUR GEAR:

1) calculation of Gear ratio (i):

$$i = \frac{N_1}{N_2} = \frac{Z_2}{Z_1}$$

2) Selection of materials:

D.B. Pg.NO: 1.40 (If not given)

3) Gear life (N)

If not given, assume (20,000 hrs)

4) Initial design torque (M_t):

D.B. Pg.NO: 8.15

5) calculation of E_{eq}, $[\sigma_b]$ and $[\sigma_c]$

E_{eq} = D.B. Pg.NO: 8.14

$[\sigma_b]$ = D.B Pg.NO: 8.18 (or) 8.5

$[\sigma_c]$ = D.B Pg.NO: 8.16 (or) 8.5

6) centre distance (a)

a = D.B Pg NO: 8.13

7) Selection of no of teeth on pinion (Z_1) & Gear (Z_2):

assume $Z_1 \geq 17$, say 18 ; $Z_2 = i \times Z_1$

8) module (m)

D.B Pg.NO: 8.22 / Std module - D.B 8.2

9) Revision of centre distance (a):

D.B Pg.NO: 8.22

10) calculation of b, d, v and ψ_p:

$b = \psi a$ / $d_1 = m \cdot Z_1$; $V = \frac{\pi d_1 N_1}{60}$, $\psi_p = \frac{b}{d_1}$

11) Selection of quality of Gears:

D.B Pg.NO: 8.3

12) Revision of design torque $[M_t]$:

D.B. Pg.NO: 8.15 & 8.16

13) check for bending σ_b : D.B. Pg NO: 8.13A

$\sigma_b \leq [\sigma_b]$ safe

14) check for wear strength σ_c : D.B Pg NO: 8.13

$\sigma_c \leq [\sigma_b]$ safe

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GEAR USING GEAR LIFE METHOD

HELICAL GEAR

1) calculation of Gear ratio (i) [assume β not given β = 15°]

$i = N_1/N_2 = Z_2/Z_1$

2) Selection of materials:

D.B. Pg.NO: 1.40 (If not given)

3) Gear life: (N)

If not given, assume (20,000 hrs)

4) Initial design torque $[M_t]$:

D.B Pg NO: 8.15

5) calculation of E_{eq}, $[\sigma_b]$ and $[\sigma_c]$:

E_{eq} = D.B Pg.NO: 8.14

$[\sigma_b]$ = D.B Pg.NO: 8.18 (or) 8.5

$[\sigma_c]$ = D.B Pg.NO: 8.16 (or) 8.5

6) centre distance (a):

a = D.B Pg.NO: 8.13

7) Selection of no of teeth on pinion (Z_1) & gear (Z_2)

assume $Z_1 \geq 17$, say 18

$Z_2 = i \times Z_1$

8) Normal module (m_n):

D.B Pg NO: 8.22 / Std module - D.B. 8.2

9) Revision of centre distance (a):

D.B. Pg.NO: 8.22

10) calculation of b, d, v and ψ_p

$b = \psi \times a$ $P_a = \frac{\pi d_1}{Z_1 \times \tan\beta}$, $d_1 = \frac{m_n}{\cos\beta} \times Z_1$

$v = \frac{\pi d_1 N_1}{60}$, $\psi_p = b/d_1$

11) Select the suitable quality of Gear:

D.B Pg.NO: 8.3

12) Revision of design torque $[M_t]$:

D.B. Pg.NO: 8.15 & 8.16

13) check for bending σ_b : D.B Pg: 8.13A

$\sigma_b \leq [\sigma_b]$ safe, not safe raise module

14) check for wear strength σ_c : D.B Pg: 8.13

$\sigma_c \leq [\sigma_c]$ safe. Not safe rise the module

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BEVEL GEAR

1) Gear ratio: (i)

$i = N_1/N_2 = \frac{\text{Pinion}}{\text{Gear}}$ | $\delta_1 = \tan^{-1}(i)$; $\delta_1 + \delta_2 = 90°$

2) Selection of materials:

If not given (D.B Pg NO: 1.40 & 8.4)

3) Gear life: (N)

Not given assume (20,000 hrs)

4) Initial design torque $[M_t]$:

D.B Pg.NO: 8.15

5) calculation of E_{eq} $[\sigma_b]$ and $[\sigma_c]$:

E_{eq} = D.B. Pg NO: 8.14

$[\sigma_b]$ = D.B. Pg.NO: 8.18 (or) 8.5

$[\sigma_c]$ = D.B. Pg.NO: 8.16 (or) 8.5

6) cone distance (R):

R = D.B Pg NO: 8.13

ψ_y = 3 initially assumed

7) no of teeth on pinion and Gear:

$Z_1 \geq 17$, say 18 ; $Z_2 = i \times Z_1$

$Z_{V1} = Z_1/\cos\delta_1$ / $Z_{V2} = \frac{Z_2}{\cos\delta_2}$

8) transverse module (m_t):

m_t = 8.38 from D.B/Std in 8.2 D.B/8.13A

9) revision of cone distance (R):

D.B. Pg.NO: 8.38

10) Calculation b, m_{av}, d_{iav}, V and ψ_y:

$b = R/\psi_y$ | $m_{av} = m_t - \frac{b \sin\delta_1}{Z_1}$

$d_{iav} = m_{av} \times Z_1$ | $V = \frac{\pi d_{iav} \times N_1}{60}$

$\psi_y = b/d_{iav}$

11) Is Quality: D.B NO: 8.3

12) Revision of design torque: D.B Pg.NO: 8.15 & 8.16

13) checking for bending: D.B Pg.NO: 8.13A

$\sigma_b \leq [\sigma_b]$ safe, not safe rise m_t

14) checking for wear strength D.B Pg.NO: 8.13

$\sigma_c \leq [\sigma_c]$ safe. Not, rise m_t

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UNIT IV

GEAR BOX:

1) Step ratio (or series ratio or progression ratio) ϕ:

$$\frac{N_{max}}{N_{min}} = \phi^{n-1}$$

where n - number of steps of speed

2) Select the series of preferred number:

D.B. Pg.No: 7.20

3) Structural formula:

$$n = P_1(x_1) \cdot P_2(x_2) \cdot P_3(x_3)$$

1st stage 2nd stage 3rd stage

$x_1 = 1$, $x_2 = P_1$ $x_3 = P_1 \cdot P_2$

Preferred structural formulas:

6 speeds - 3(1) 2(3)

9 speeds - 3(1) 3(3) — speed

12 speeds - 3(1) 2(3) 2(6) — Interval

Basic Rules for Gear box design

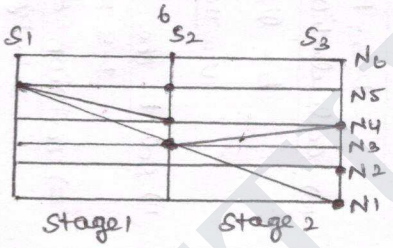
$$\frac{N_{min}}{N_{input}} \geq \frac{1}{4}$$

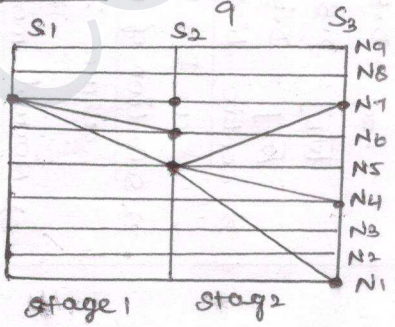
$$\frac{N_{max}}{N_{input}} \leq 2$$

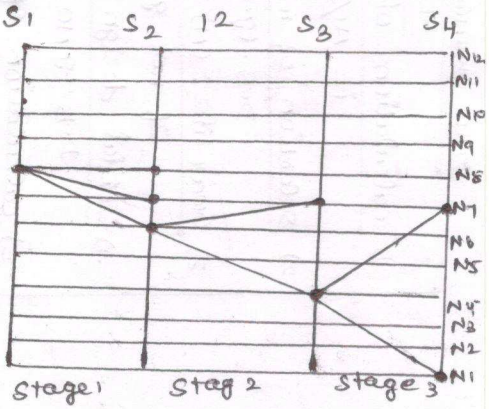
In all stage except in the first stage

$N_{max} \geq N_{input} \geq N_{min}$

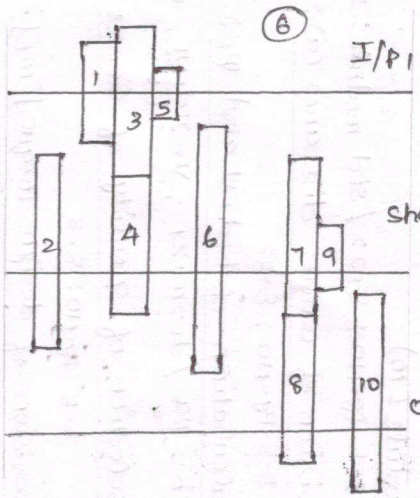
Ray diagram (or speed diagram)

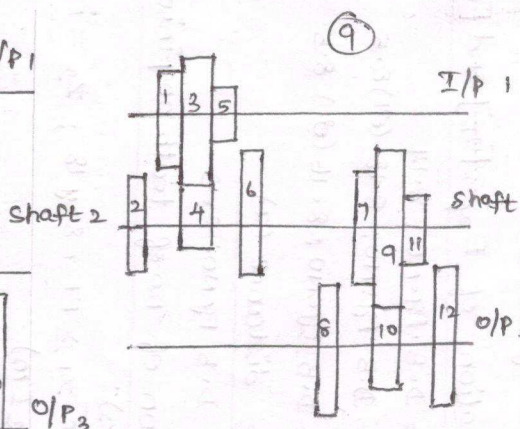


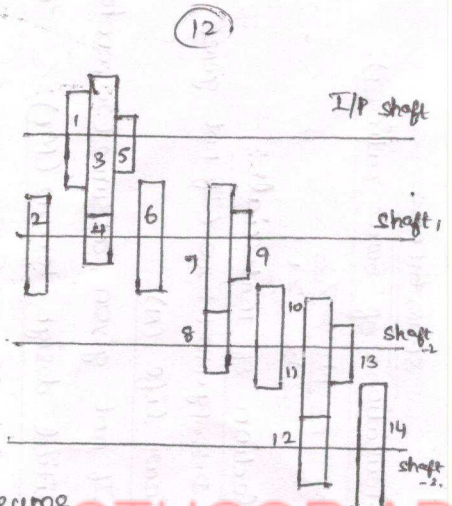




KINEMATIC LAYOUT (OR KINEMATIC ARRANGEMENT):







(X) For every stage assume minimum tooth $z \geq 17, 20$ say.

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Design of a single plate clutch

Let
T = Torque transmitted by clutch
P = Intensity of axial pressure acting on contact surfaces,
r_1 = External radius of friction surface
r_2 = Internal radius of friction surface &
μ = coefficient of friction.

1) Area of the elemental ring = $2\pi r \cdot dr$
2) Axial force on the ring δW = Pressure × Area
$= P \times 2\pi r \cdot dr$
3) Frictional force acting on the ring
$F_r = \mu \cdot \delta W = \mu P \times 2\pi r \cdot dr$
4) Frictional torque acting on the ring
$T_r = F_r \times r$
$= 2\pi \mu p r^2 \cdot dr$

CONSIDERING UNIFORM PRESSURE:-

Area of the friction surface
$A = \pi(r_1^2 - r_2^2)$
uniform intensity of pressure
$P = \frac{W}{A} = \frac{W}{\pi(r_1^2 - r_2^2)}$

$T = \frac{2}{3} \times \mu W \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right] = \mu W R$

$R = \frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right]$

CONSIDERING UNIFORM WEAR

$P = \frac{C}{r}$

$P_1 \cdot r_1 = P_2 \cdot r_2 = C$

$C = \frac{W}{2\pi(r_1 - r_2)}$

$T = \frac{1}{2} \times \mu \cdot W (r_1 + r_2)$
$= \mu \cdot W \cdot R$

$R = \frac{r_1 + r_2}{2}$

DESIGN OF A MULTIPLATE CLUTCH

Let
n_1 = Number of disc on the driving shaft &
n_2 = Number of disc on the driven shaft
∴ Number of pair of contact surfaces
$n = n_1 + n_2 - 1$
Frictional torque acting on the clutch
$T = n \cdot \mu \cdot W \cdot R$
where,
R = Mean radius of friction surface

$R = \frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right]$ for uniform pressure

For uniform wear,
$R = \frac{r_1 + r_2}{2}$

SERVICE FACTORS

Design torque = Nominal torque × Service factor
$[T] = T \times K_S$
$K_S = K_1 + K_2 + K_3 + K_4$ [PSG. D.B.Pg. NO: 7.90 & 7.91]

DESIGN OF CENTRIFUGAL CLUTCH

1. Mass of the shoes:

Let n = Number of shoes,
m = Mass of each shoe,
R_1 = Inside radius of the pulley rim,
r - Distance of centre of Gravity of the shoes from the centre of spider
N - Running speed of pulley
ω - $2\pi N/60$
ω_1 - Angular speed at which engagement begins to take place &
μ - coefficient of friction b/w shoe and rim.

1) Centrifugal force acting on each shoe, $F_c = m\omega^2 r$
2) Spring force exerted by each spring on the shoe $F_S = m\omega_1^2 \times r$
3) Net outward force on shoe = $F_c - F_S$
$= m\omega^2 r - m\omega_1^2 r$
4) Frictional force acting on each shoe $F = \mu(F_c - F_S)$
5) Frictional torque acting on each shoe $T_r = F \times R = \mu(F_c - F_S)R$

Total frictional torque transmitted $T = n \times \mu(F_c - F_S)R$
$= n \cdot F \cdot R$

2) SIZE OF THE SHOES

Let: l - contact length of shoes
b - width of shoes
R - contact radius of shoes
θ - Angle subtended by shoes at the centre of spider, &
P - Intensity of pressure

Area $A = l \times b$
Net force acting on the shoe = $F_c - F_S = A \times P$
$= l \times b \times P$

DESIGN OF A CONE CLUTCH:-

Let,
P_n - Normal intensity of pressure on friction surface
α - Semi-angle of cone or face angle of the cone
r_1 - outer radius of friction surface
r_2 - Inner radius of friction surface
$R = \frac{r_1 + r_2}{2}$
μ - coefficient of friction

$P_n = \frac{W}{\pi[r_1^2 - r_2^2]}$

$T = \frac{2}{3}\mu W \cdot \operatorname{cosec}\alpha \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right]$

$T = \mu \cdot W \cdot R \operatorname{cosec}\alpha$

$R = \frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right]$

$P \cdot r = C$ [or] $P = \frac{C}{r}$

$C = \frac{W}{2\pi(r_1 - r_2)}$

$T = \mu \cdot W \cdot R \cdot \operatorname{cosec}\alpha$

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