German University in Cairo Date: May / 2014
Faculty of Engineering and Material Science.
Design and Production Engineering Dept.
Dr. Mohamed Kamal Shoukry

Bar Code

Student major	**MATS**	**DPE**	**MCTR**

Engineering Design II, EDPT 602
Spring Semester 2013/ 2014
Final Exam

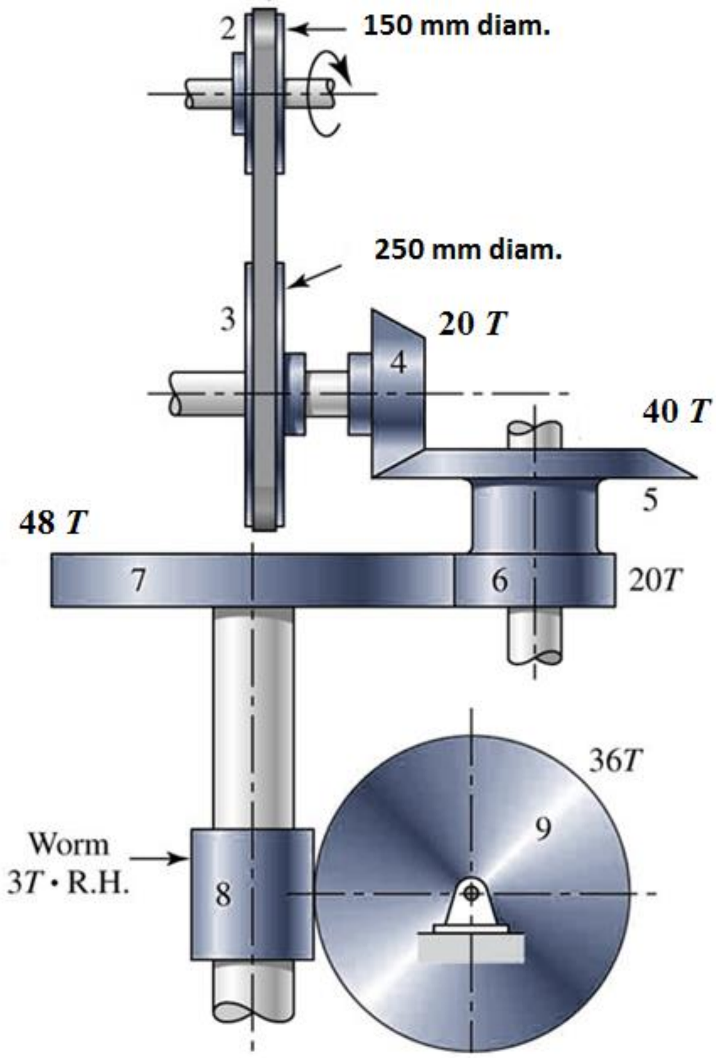
Instructions: **Read Carefully Before Proceeding.**

1- Only non-programmable Calculators are allowed.
2- **For all numerical problems, use a minimum of three (3) significant figures**
3- Write your solutions in the space provided.
4- **The exam consists of (5) Questions.**
5- This exam booklet contains (14) pages and (9) pages for formula sheets.
6- Attempt all problems within the time limits.
7- Total time allowed for this exam is **(180)** ***minutes***

Good Luck!

Question #	1	2	3	4	5	Total
Max. Score	25	25	20	15	15	100
Obtained Score						

Question 1: (25 points)



The transmission mechanism shown in the figure consists of two belt pulleys and three sets of gears. The number of teeth of the gears and the diameters of the pulleys are shown. Pulley (2) rotates at 2000 rpm in the direction indicated by the arrow on the pulley shaft. The 3-tooth worm (8) is to drive a 36-tooth gear wheel (9). The worm has a right –hand helix, an axial pitch of 7.5 mm, a 14.5^{o} normal pressure angle and a pitch diameter of 30 mm. The coefficient of friction between the worm and worm wheel teeth is 0.04.

a) Determine the speed (rpm) and the direction of rotation (CW or CCW) of the gear wheel (9).
b) Determine the output torque that could be taken from gear wheel (9) if the power of the motor driving pulley (2) is 500 W.

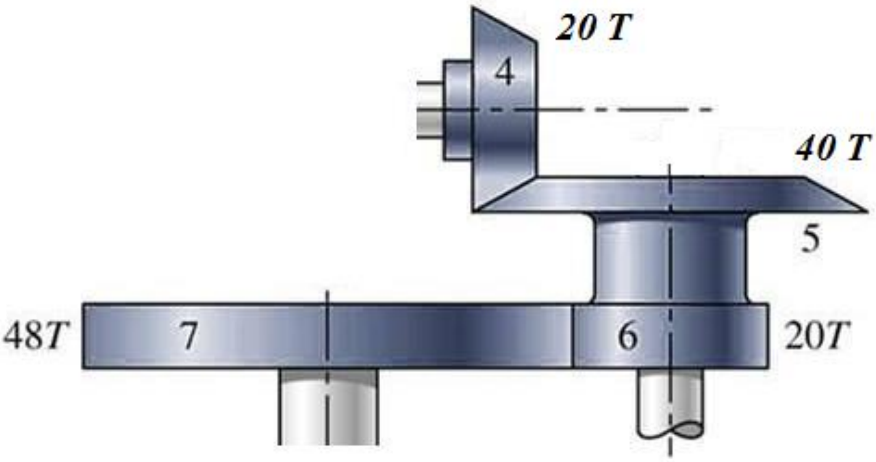
Answer:

Question 2: (25 points)

The two-stage gear train shown in the following figure is composed of a bevel gear set and a helical gear set. The power transmitted through this gear train is 3.5 kW. The number of teeth of the gears are shown on figure. The pinion (4) is driven at 2000 rpm, the module is 4 mm, the pressure angle is 20° and the face width is 35 mm. Both bevel gears are mounted outboard (K_H = 1.25) and are formed by hobbing (Q_v = 6) and are made of grade 1 steel hardened to 240 HB. The gears are intended to work for infinite life ($Y_N = Z_N = 0.95$), for general industrial use, and for uniform driving and moderate shock driven machinery (K_o = 1.5) (Hint: Take $K_s = 1$, $C_{xc} = 1.5$ and $C_p = 191\ (MPa)^{0.5}$)

a) For a reliability of 90% determine the factors of safety guarding against bending and surface fatigue failures of **the bevel gear set.**

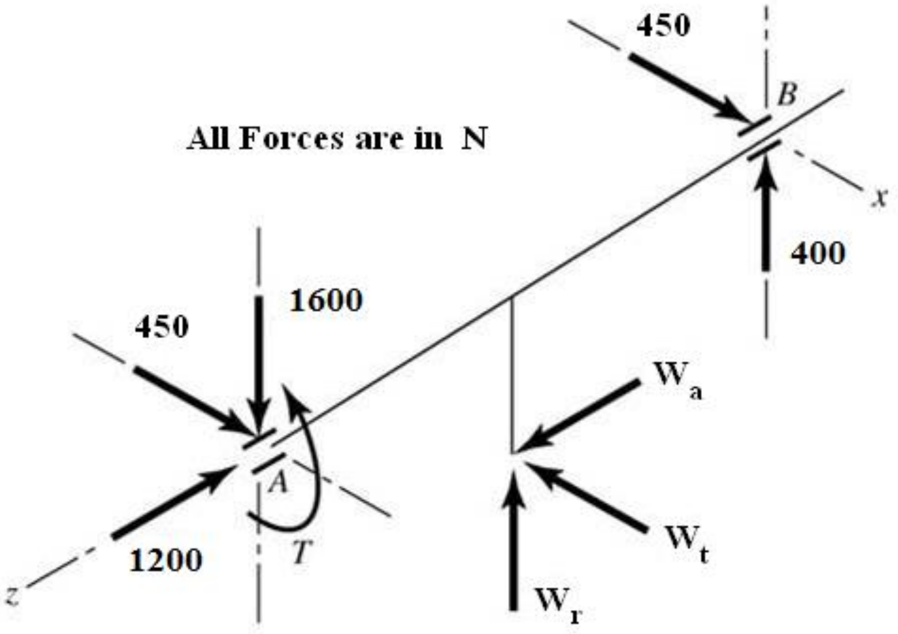
b) If the bevel pinion (4) is rotating in the clockwise direction (when seen from the left hand side), what is the recommended direction (sense) of the helical teeth of the helical pinion (6) (RH or LH) in order to minimize the resultant axial force on the intermediate shaft. (show the directions of the tangential, radial and axial forces on pinion (6) at the mesh point).



Answer:

Question 3: (20 points)

The force analysis on a shaft is shown schematically in the following figure. The reactions on the shaft at bearings A & B are described on the figure where bearing A is intended to take the axial force W_a . The shaft is rotating at 1250 rpm, for a life of 25 kh, 90% reliability and an application factor of 1.25. Select two identical taper roller bearings at A and B (from the given table) where the diameter of the shaft at the bearings is 30 mm.



Bearing	d	D	T	C (N)	C_o (N)	Y	e
32006X	30	55	17	35800	44000	1.4	0.43
30206	30	62	17.25	40200	44000	1.6	0.37
32206	30	62	21.25	50100	57000	1.6	0.37
33206	30	62	25	64400	76500	1.7	0.35

Answer:

Question 4: (15 points)

The layout of a leather belt drive transmitting 15 kW power is shown in the figure. The diameter of the small pulley is 270 mm and is running at 1500 rpm while the large pulley is rotating at 500 rpm. The thickness of the flat belt is 5 mm, the width is 150 mm and the density of leather is 0.95 gm/cm^3 .

The coefficient of friction between the belt and pulleys is 0.35 and the contact angles between the belt and both the small and large pulleys are $\theta_s = 150^\circ$ and $\theta_l = 210^\circ$ respectively.

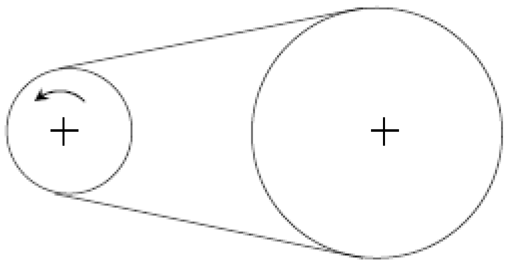
If the tensile stresses in the belt should not exceed 2.5 MPa.

Determine:

a) The diameter of the large pulley

b) The safety factor guarding against tensile failure of the belt

Answer:



Question 5: (15 points)

An automotive single-plate clutch consists of two pairs of contacting surfaces. The outer diameter of the friction disc is 270 mm. The coefficient of friction is 0.3 and the allowable bearing pressure of the friction material is 0.3MPa. The clutch is transmitting a torque of 530 N.m, assuming uniform wear calaculate:

1) the inner diameter of the friction disc to the nearest mm, knowing that it should be , for constructional details, larger than 100 mm.
2) the spring force required to keep the clutch engaged

Answer:

Engineering Design II

Formula Sheets:

1- Bevel Gears

$$d = mN \qquad \tan\gamma = \frac{N_p}{N_G}, \qquad \tan\Gamma = \frac{N_G}{N_p}$$

$$r_{av_p} = r_p - \frac{F}{2}\sin\gamma$$

$$W_r^* = W_t^* \tan\varphi\cos\gamma$$

$$W_a^* = W_t^* \tan\varphi\sin\gamma$$

Use W without (*) for stress analysis and use r instead of r_{av}

Strength :

Bending:

$$\sigma = \frac{W_t}{bmJ} K_v K_0 K_s K_H$$

$$\sigma^{\backslash}_{FP} = 0.3H_B + 14.48 \text{ MPa} \quad \text{for grade 1}$$

$$= 0.33H_B + 41.24 \text{ MPa} \quad \text{for grade 2}$$

The corrected strength:

$$\sigma_{FP} = \sigma^{\backslash}_{FP} \frac{Y_N}{Y_\theta Y_Z}$$

Contact Strength:

$$\sigma_c = C_p \sqrt{\frac{W_t}{bd_p I} K_v K_o K_s K_H C_{xc}}$$

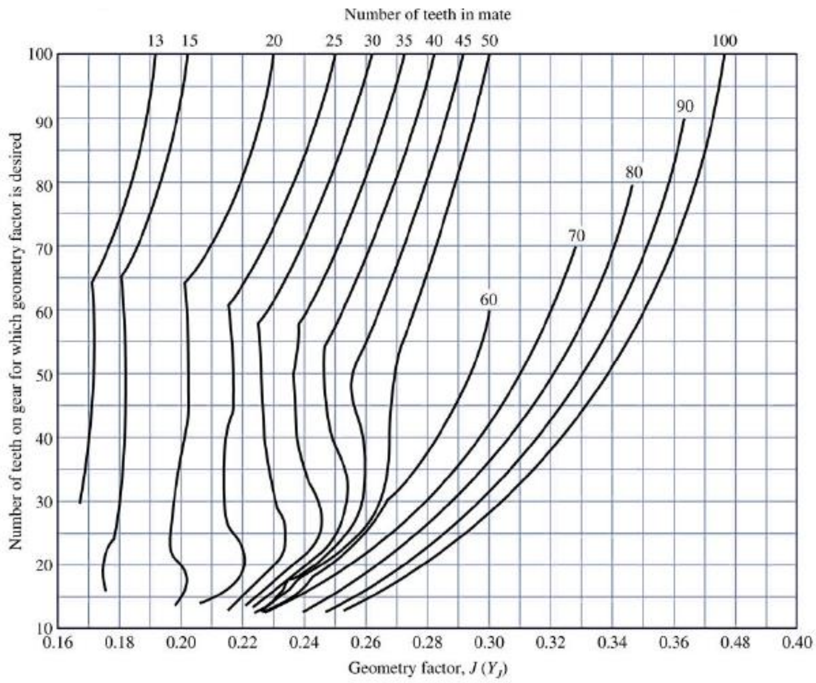
$$\sigma^{\backslash}_{HP} = 2.35H_B + 162.89 \text{ MPa} \quad \text{for grade 1}$$

$$= 2.51H_B + 203.86 \text{ MPA} \quad \text{for grade 2}$$

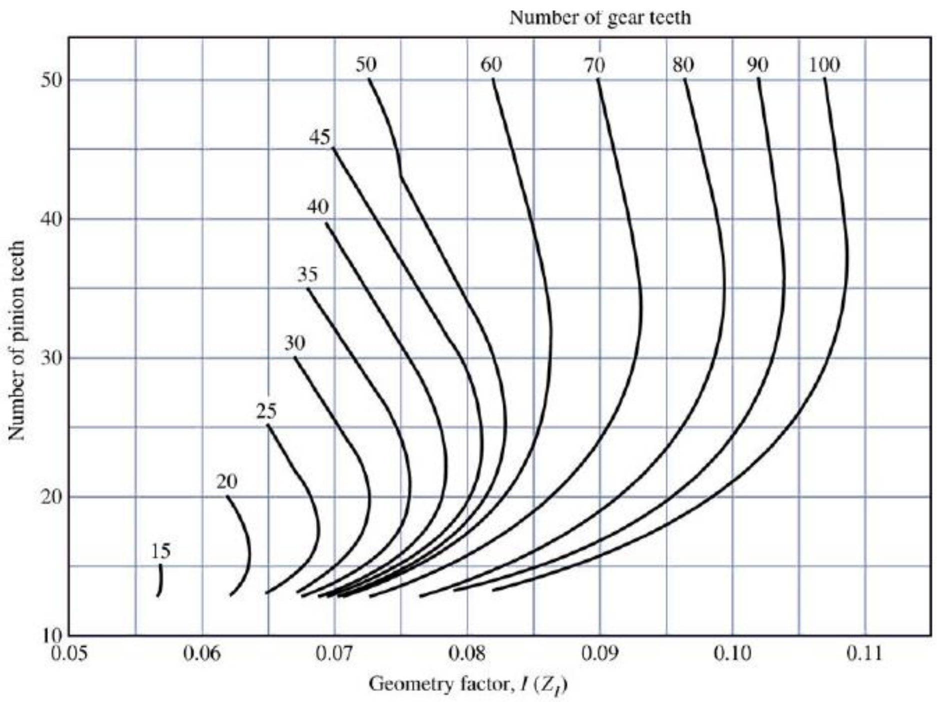
The corrected contact strength

$$\sigma_{\mathrm{HP}} = \sigma^{\backslash}_{HP} \frac{Z_N C_H}{Y_\theta Y_Z}$$

J-Factor



I-Factor



Dynamic factor K_v:

$$K_v = \left(\frac{A+\sqrt{200V}}{A}\right)^B, \quad \text{V is the tangential velocity in m/s}$$

$$A = 50 + 56(1-B)$$

$$B = 0.25(12 - Q_v)^{\frac{2}{3}}$$

Reliability factor:

Reliability	Y_Z
0.9999	1.5
0.999	1.25
0.99	1
0.9	0.85
0.5	0.7

2- Helical Gears

$$d = mN = \frac{m_n}{\cos\psi} N$$

$$W_r = W_t \tan\varphi_t$$

$$\tan\varphi_n = \tan\varphi_t \cos\psi$$

$$W_a = W_t \tan\psi$$

Bending Stress:

$$\sigma = \frac{W_t}{bmJ} K_v K_0 K_s K_H K_B$$

$$\sigma_{FP}^{\backslash} = 0.533\,\text{HB} + 88.3 \quad \text{MPa for grade 1}$$

$$\sigma_{FP}^{\backslash} = 0.703\,\text{HB} + 113 \quad \text{MPa for grade 2}$$

$$\sigma_{FP} = \sigma_{FP}^{\backslash}\,(Y_N / Y_\theta\, Y_Z)$$

Contact Stress:

$$\sigma_c = C_p \sqrt{\frac{W_t}{b d_p I} K_v K_o K_s K_H C_f}$$

$$I = \frac{\cos\varphi_t \sin\varphi_t}{2 m_N} \frac{m_G}{m_G + 1}$$

$$\mathrm{m_N} = \frac{\mathrm{p_N}}{0.95\, \mathrm{Z}}$$

$$\mathrm{p_N} = p_n \cos\varphi_n$$

$$p_n = \pi m_n$$

$$Z = \sqrt{(r_p + a_p)^2 - (r_p \cos\varphi)^2} + \sqrt{(r_g + a_g)^2 - (r_g \cos\varphi)^2} - C \sin\varphi$$

r_p, r_g : are the pitch circle radii of pinion and gear.
a_p, a_g : are the addenda of pinion and gear
C : is the center distance

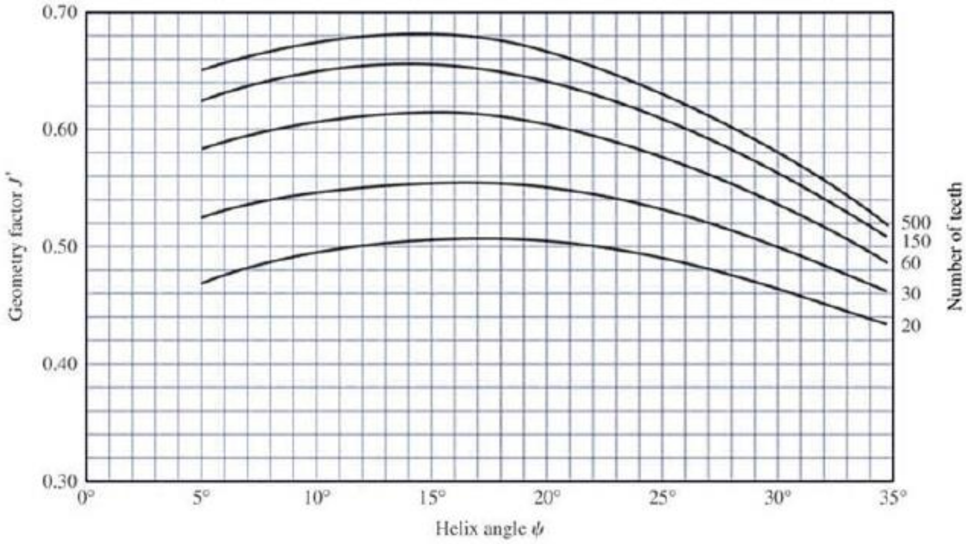
$a_p = a_g = m_n$ and $\varphi = \varphi_t$

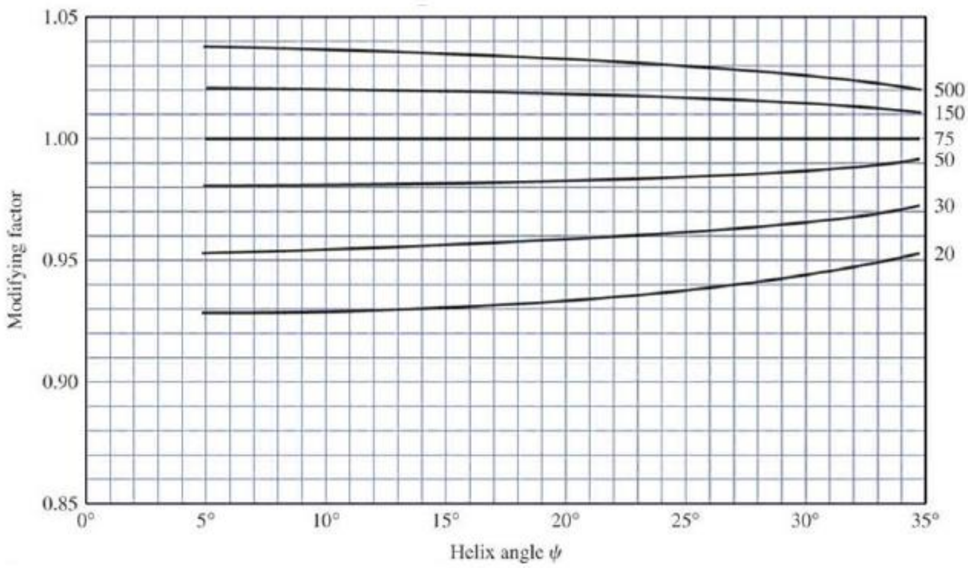
Strength:

$$\sigma^{\backslash}_{HP} = 2.22\,\mathrm{HB} + 200 \quad \mathrm{MPa\ for\ grade\ 1}$$
$$\sigma^{\backslash}_{HP} = 2.41\,\mathrm{HB} + 237 \quad \mathrm{MPa\ for\ grade\ 2}$$

$$\sigma_{HP} = \sigma^{\backslash}_{HP} \frac{Z_N C_H}{Y_\theta Y_Z}$$

J-Factor





3- Rolling Bearings

$$\frac{L_1}{L_2} = \left(\frac{F_2}{F_1}\right)^a \qquad\qquad F_2 = F_1\left(\frac{L_1}{L_2}\right)^{\frac{1}{a}}$$

For 90% Reliability: $$C_R = k_A F_{eq}\left[\left(\frac{L_D}{L_R}\right)\left(\frac{n_D}{n_R}\right)\right]^{1/a}$$

Take L_R = 500 hours, and n_R = 100/3 rpm, a =3 for ball bearings and a = 10/3 for roller and taper bearings

4- Worm Gearing:

$$W_{Wt} = -W_{Ga} = W_x$$
$$W_{Wr} = -W_{Gr} = W_y$$
$$W_{Wa} = -W_{Gt} = W_z$$

$$W_x = W\left(\cos\phi_n \sin\lambda + \mu\cos\lambda\right)$$
$$W_y = W\sin\phi_n$$
$$W_z = W\left(\cos\phi_n \cos\lambda - \mu\sin\lambda\right)$$

$$\overline{\mathbf{V}}_W = \overline{\mathbf{V}}_G + \overline{\mathbf{V}}_S$$

$$H = H_o + H_l$$

$$V_S = \frac{V_W}{\cos\lambda}$$

$$W_{tg} = \beta\left(K_s d_g^{0.8} F_e K_m K_v\right)$$

$$\tan\lambda = \frac{L}{\pi d_W} \qquad L = p_x N_W \qquad d_G = \frac{N_G p_t}{\pi}$$

Note that: $d_g == d_G$

The conversion factor $\beta = 0.0131$

The loss power $H_l = \mu W \times V_s$ $\qquad W_f = \mu W$

The output power $H_o = W_{tg} \times V_{tg}$

Table 1: Material factor (K_s) for cylindrical worm gears

Face width, mm	Sand-Cast Bronze	Static-Chill Cast Bronze	Centrifugal Cast Bronze
Up to 75	700	800	1000
100	665	780	975
125	640	760	940
150	600	720	900
175	570	680	850
200	530	640	800
225	500	600	750

Table 2: Ratio correction factor (K_m)

Tr. Ratio	K_m	Tr. Ratio	K_m	Tr. Ratio	K_m
3.0	0.500	8.0	0.724	30	0.825
3.5	0.554	9.0	0.744	40	0.815
4.0	0.593	10	0.760	50	0.785
4.5	0.620	12	0.783	60	0.745
5.0	0.645	14	0.799	70	0.687
6.0	0.679	16	0.809	80	0.622
7.0	0.706	20	0.820	100	0.490

Table 3: Velocity factor (K_v)

V_s(m/s)	K_v	V_s(m/s)	K_v	V_s(m/s)	K_v
0.005	0.649	1.50	0.47	7.2	0.216
0.008	0.646	1.80	0.45	8	0.200
0.050	0.644	2.00	0.42	9	0.187
0.100	0.638	2.25	0.395	10	0.175
0.150	0.631	2.50	0.375	11	0.168
0.200	0.625	2.80	0.360	12	0.156
0.300	0.615	3.00	0.340	13	0.148
0.400	0.600	3.60	0.310	14	0.140
0.500	0.590	4.00	0.285	16	0.134
0.750	0.560	4.50	0.265	20	0.106
1.000	0.530	5.00	0.258	25	0.089
1.250	0.500	6.00	0.235	30	0.079

5-Clutches:

Uniform Wear theory:

$$F = \int_{d/2}^{D/2} 2\pi p r dr = \pi p_{\max} d \int_{d/2}^{D/2} dr = \frac{\pi p_{\max} d}{2}(D - d)$$

$$T = m.\frac{fF}{4}(D + d)$$

Uniform pressure theory:

$$F = \frac{\pi p}{4}(D^2 - d^2)$$

$$T = m.\frac{fF}{3}\frac{(D^3 - d^3)}{(D^2 - d^2)}$$

6- Flat Belt Drive:

$$\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\theta}$$

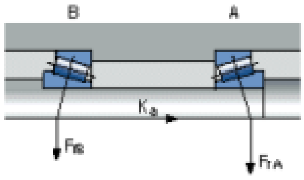
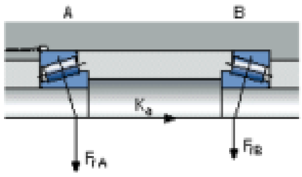
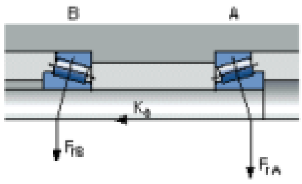
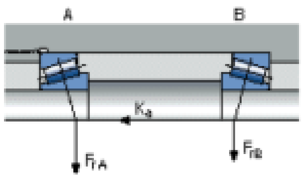
P_1 : Tension of tight side.

P_2 : Tension of loose side.

m : Mass of unit length (of one meter) of the belt (kg).

f : Coefficient of friction between belt & pulley..

7- Tapper Bearings:

Bearing arrangement		Load case	Axial loads	
Back-to-back		1a) $\frac{F_{rA}}{Y_A} \geq \frac{F_{rB}}{Y_B}$ $K_a \geq 0$	$F_{aA} = \frac{0,5\,F_{rA}}{Y_A}$	$F_{aB} = F_{aA} + K_a$
		1b) $\frac{F_{rA}}{Y_A} < \frac{F_{rB}}{Y_B}$ $K_a \geq 0,5\left(\frac{F_{rB}}{Y_B} - \frac{F_{rA}}{Y_A}\right)$	$F_{aA} = \frac{0,5\,F_{rA}}{Y_A}$	$F_{aB} = F_{aA} + K_a$
Face-to-face		1c) $\frac{F_{rA}}{Y_A} < \frac{F_{rB}}{Y_B}$ $K_a < 0,5\left(\frac{F_{rB}}{Y_B} - \frac{F_{rA}}{Y_A}\right)$	$F_{aA} = F_{aB} - K_a$	$F_{aB} = \frac{0,5\,F_{rB}}{Y_B}$
Back-to-back		2a) $\frac{F_{rA}}{Y_A} \leq \frac{F_{rB}}{Y_B}$ $K_a \geq 0$	$F_{aA} = F_{aB} + K_a$	$F_{aB} = \frac{0,5\,F_{rB}}{Y_B}$
		2b) $\frac{F_{rA}}{Y_A} > \frac{F_{rB}}{Y_B}$ $K_a \geq 0,5\left(\frac{F_{rA}}{Y_A} - \frac{F_{rB}}{Y_B}\right)$	$F_{aA} = F_{aB} + K_a$	$F_{aB} = \frac{0,5\,F_{rB}}{Y_B}$
Face-to-face		2c) $\frac{F_{rA}}{Y_A} > \frac{F_{rB}}{Y_B}$ $K_a < 0,5\left(\frac{F_{rA}}{Y_A} - \frac{F_{rB}}{Y_B}\right)$	$F_{aA} = \frac{0,5\,F_{rA}}{Y_A}$	$F_{aB} = F_{aA} - K_a$

$$F_{eq} = F_r \qquad \text{when} \quad F_a / F_r \leq e$$

$$F_{eq} = 0.4\, F_r + Y\, F_a \qquad \text{when} \quad F_a / F_r > e$$

$$C_R = k_A F_{eq}\left[\left(\frac{L_D}{L_R}\right)\left(\frac{n_D}{n_R}\right)\frac{1}{6.84}\right]^{1/a} \frac{1}{[\ln(1/R)]^{1/1.17a}}$$