German University in Cairo **Date: Sep / 2008**
Faculty of Engineering and Material Science.
Dr. Mohamed Kamal Shoukry

Bar Code

Engineering Design II, EDPT 602
Spring Semester 2007 / 2008

Make-up of Final Exam

Instructions: **Read Carefully Before Proceeding.**

1- Calculators of any type are allowed.
2- **For all numerical problems, use a maximum of four (4) significant figures**
3- Write your solutions in the space provided.
4- The exam consists of (5) Questions.
5- This exam booklet contains (27) pages including this cover page.
6- Attempt all problems within the time limits.
7- Total time allowed for this exam is **(180)** ***minutes***

Good Luck!

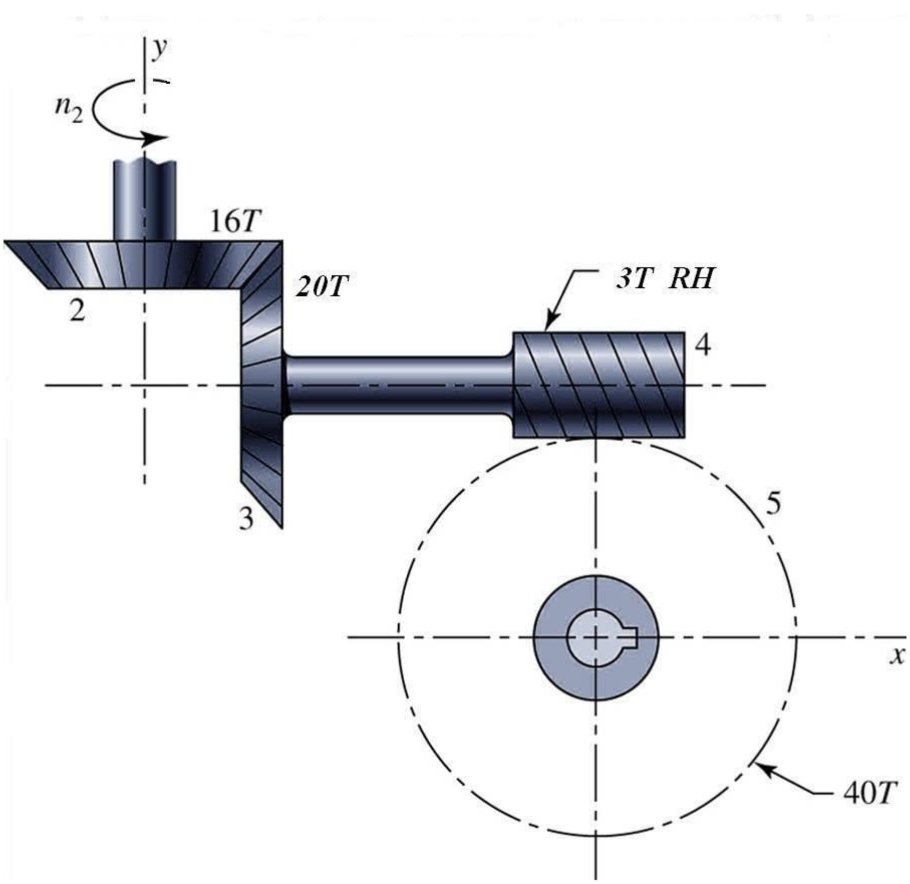
Question #	1	2	3	4	5	Total
Max. Score	25	25	15	15	20	100
Obtained Score						

<u>Question 1:</u> (25 points)

The layout of a two-stage gear reducer is shown in the figure below, the first stage is a pair of bevel gears where the number of teeth of the pinion (2) and gear (3) are 16 and 20 tooth respectively and the module is 4 mm.

The second stage is a worm gearing set, the worm (4) has 3 teeth and the worm wheel (5) has 40 teeth as shown:

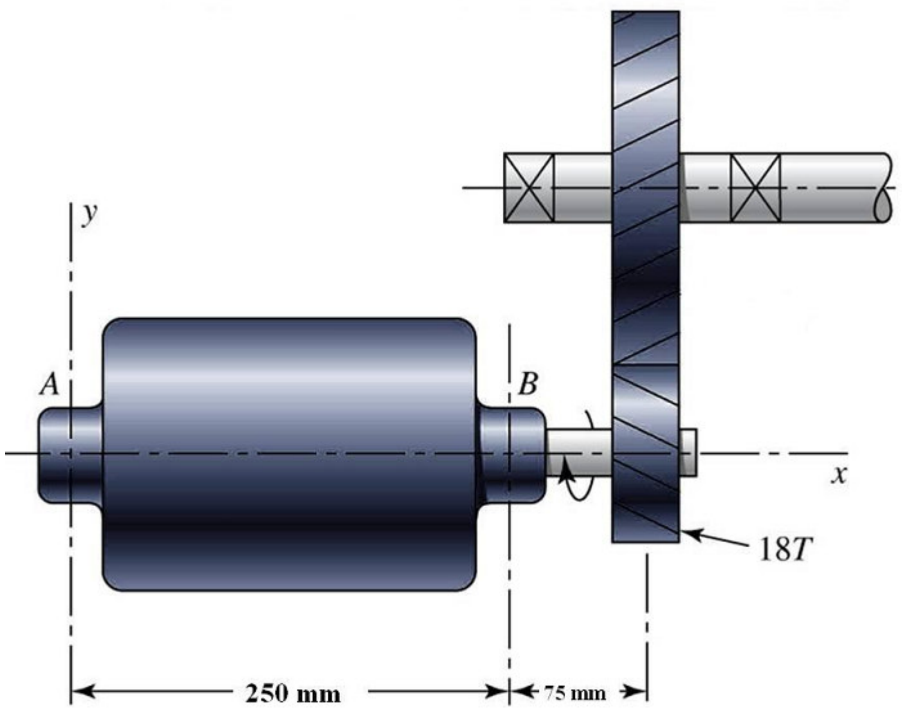
a) If the bevel pinion (2) rotates at 900 rpm in the direction shown, determine the speed (rpm) and the direction of rotation (CW or CCW) of the worm gear (5)
b) Determine the maximum power that could be transmitted through the worm gear set if the worm diameter is 45 mm, has a 14.5^{o} normal pressure angle and the worm wheel diameter is 90 mm, the worm is made of case-hardened forged steel and the gear wheel has sand-cast bronze teeth, the coefficient of friction between the worm and wheel teeth is 0.04.



Question 2: (25 points)

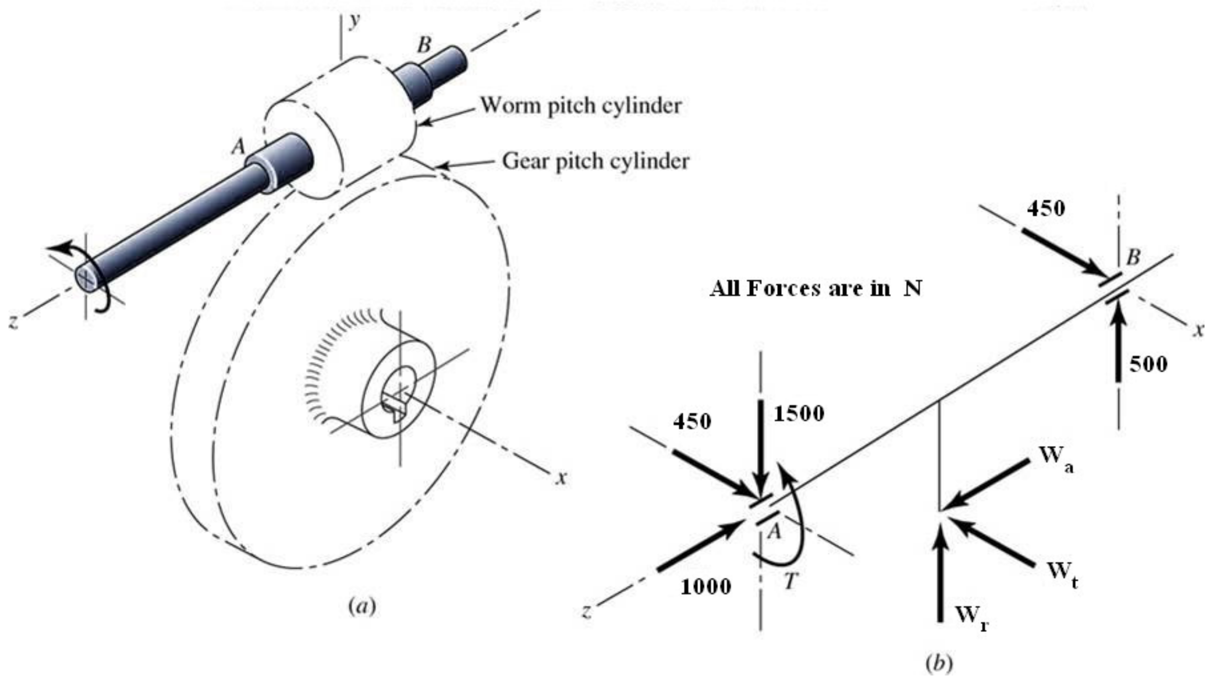
A single stage helical gear reducer is used to transmit the power of 30 kW of an electric motor to a hydraulic pump. The number of teeth of the pinion is 18 teeth and the transmission ratio of the gear set is 3. The normal module is 4 mm and the normal pressure angle is 20^o and the helix angle is 20^o. The gears are made according to a quality number $Q_v = 6$. The face width of the teeth is 42 mm and the gears are made of through hardened grade 1 steel. Determine the required hardness of the pinion and gear (HB) in order to reach a factor of safety of at least 1.5 for both bending and surface fatigue calculations. The motor speed is 1000 rpm and the gear reducer is designed to work for infinite life ($Y_N = Z_N = 0.96$), accurate mounting ($K_H = 1.3$) and uniform driving and driven conditions ($K_o = 1$), and a reliability of 90%. Take $C_p = 191\ (MPa)^{0.5}$ and the AGMA shape factor for surface fatigue $I = 0.19$

Note: See the attached data sheets; assume any missing factor or coefficient to be unity.



Question 3: (15 points)

The force analysis on the worm shaft ,of the worm gearing set shown below, determined the reactions at bearings A & B as described on the figure where bearing A is intended to take the axial force. The shaft is rotating at 1000 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 tables) where the diameter of the shaft at the bearings is 30 mm.



<u>Question 4:</u> (15 points)

If the bearing at B, of the previous problem, is replaced by a sliding bearing of length 30 mm and a radial clearance of 0.02 mm. Using SAE oil at an inlet temperature of 50^{o}C, find the temperature rise and the power loss at this bearing.

Question 5: (20 points)

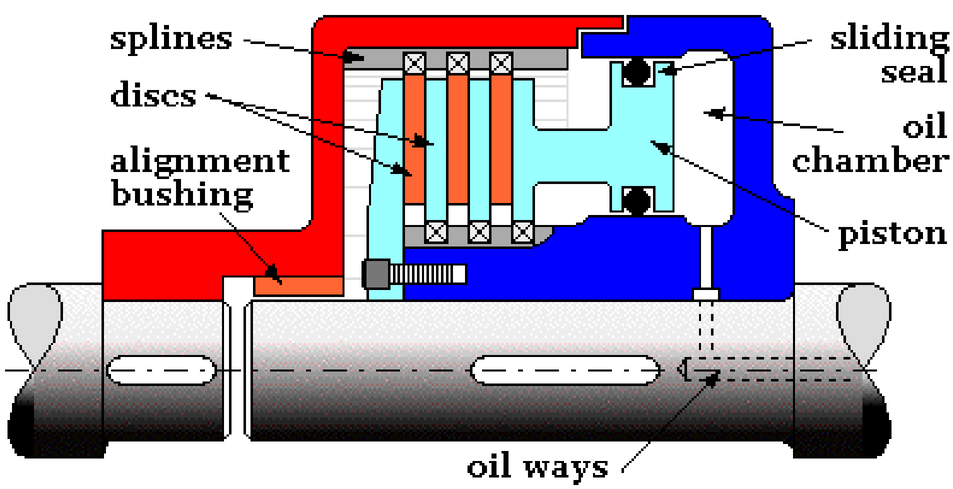
A hydraulically operated multidisk plate clutch is shown below; the clutch has an effective disk outer diameter (D) of 170 mm and an inner diameter (d) of 100 mm. The coefficient of friction is 0.25 in the six planes of sliding and the limiting pressure is 1.1 MPa.

a) Using the uniform wear theory, estimate the axial force F and the torque T.

b) Let the effective inner diameter of the friction plates (d) be a variable, while all other parameters (D, f , m, p_{max}) are the same, and complete the following table.

d (mm)	50	75	100	125	150
T (N.m)					

c) Determine the ratio d/D that produces the maximum torque capacity.



Data 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:

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

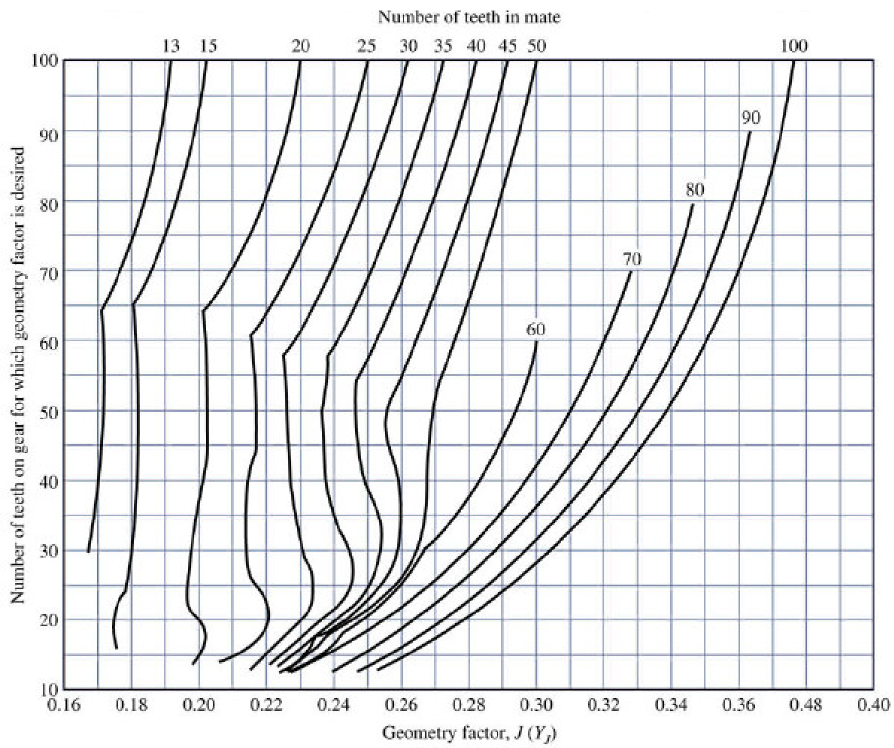
$$\sigma^{\backslash}_{HP} = 2.35H_B + 162.89 \text{ MPa} \quad \text{for grade 1}$$

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

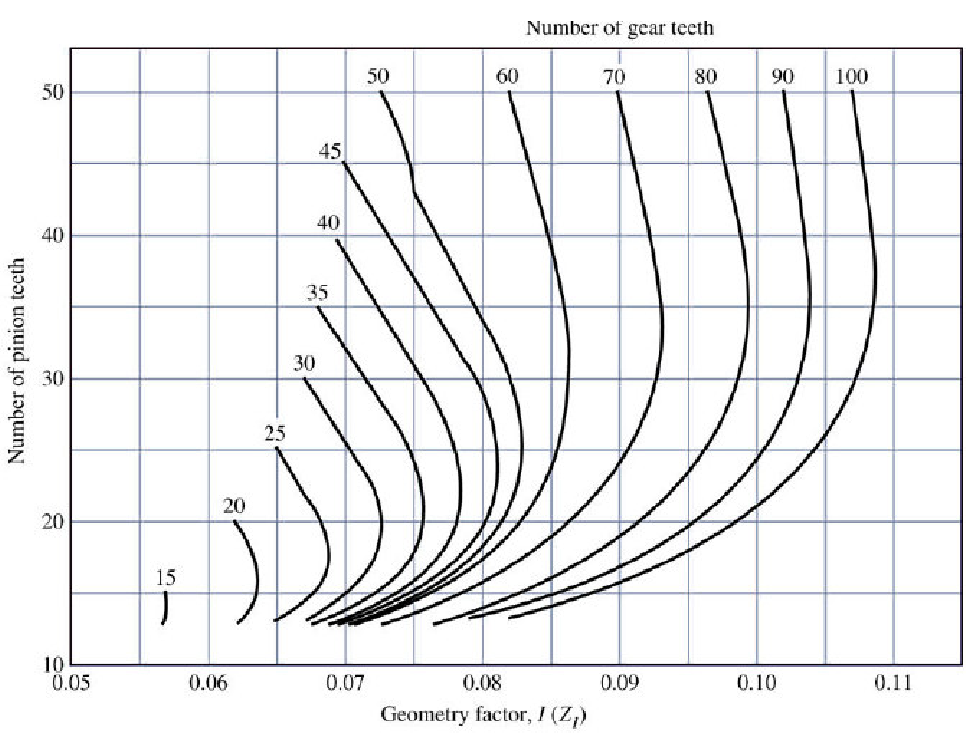
The corrected contact strength

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

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

$$d = mN$$

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

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

Bending Stress:

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

$$\sigma_{FP}' = 0.703 \text{ HB} + 113 \quad \text{MPa}$$

$$\sigma_{FP} = \sigma_{FP}' \left(Y_N / Y_\theta Y_Z\right)$$

Contact Stress:

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

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

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

$$p_N = p_n \cos\varphi_n$$

$$p_n = \pi m_n$$

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

r_p, r_g : are the pitch circle radii of pinion and gear.

a_p, a_g : are the addenda of pinion and gear

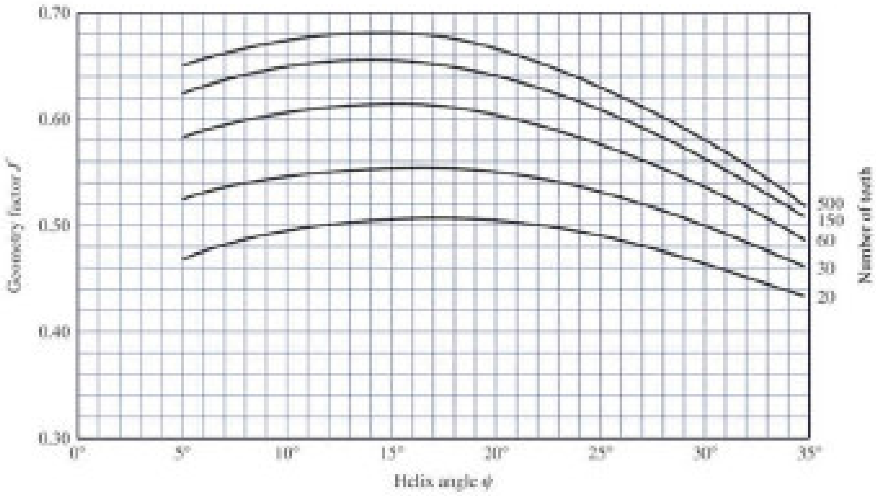
C : is the center distance

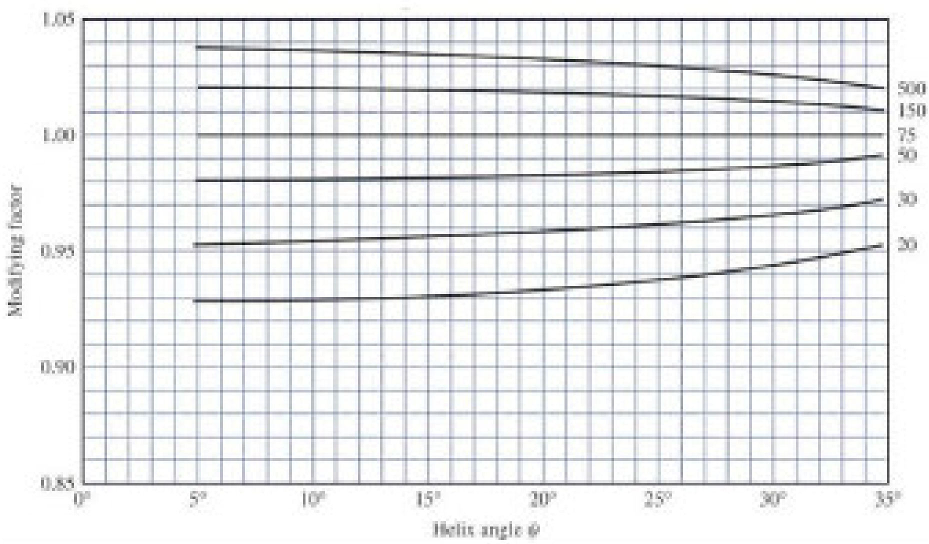
$a_p = a_g = m_n$ and $\varphi = \varphi_t$

Strength:

$\sigma^{\backslash}_{HP}$ = 2.22 HB + 200 MPa

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





3- Bearings

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

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

Take L_R = 500 hours, and n_R = 100/3 rpm

4- Worm Gearing:

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

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

$$\mathbf{V}_W = \mathbf{V}_G + \mathbf{V}_S \qquad H = H_o + H_l$$

$$V_S = \frac{V_W}{\cos\lambda} \qquad W_{tg} = \beta\left(K_s d_g^{0.8} F_e K_m K_v\right)$$

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

Note that: $d_g == d_G$

5-Clutches:

Uniform Wear theory:

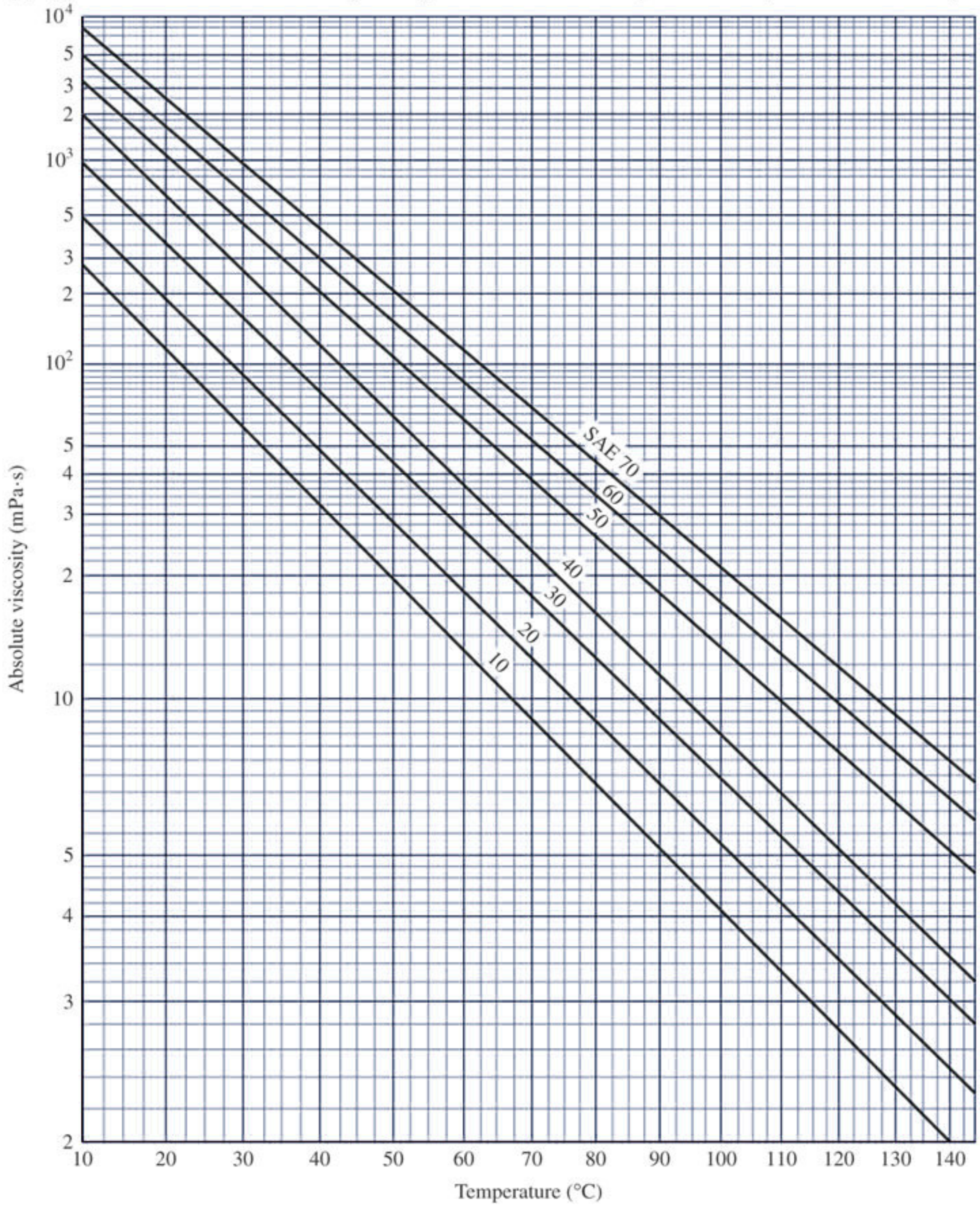
$$F = \int_{d/2}^{D/2} 2\pi p r dr = \pi p_{\max} d \int_{d/2}^{D/2} dr = \frac{\pi p_{\max} d}{2}(D-d)$$

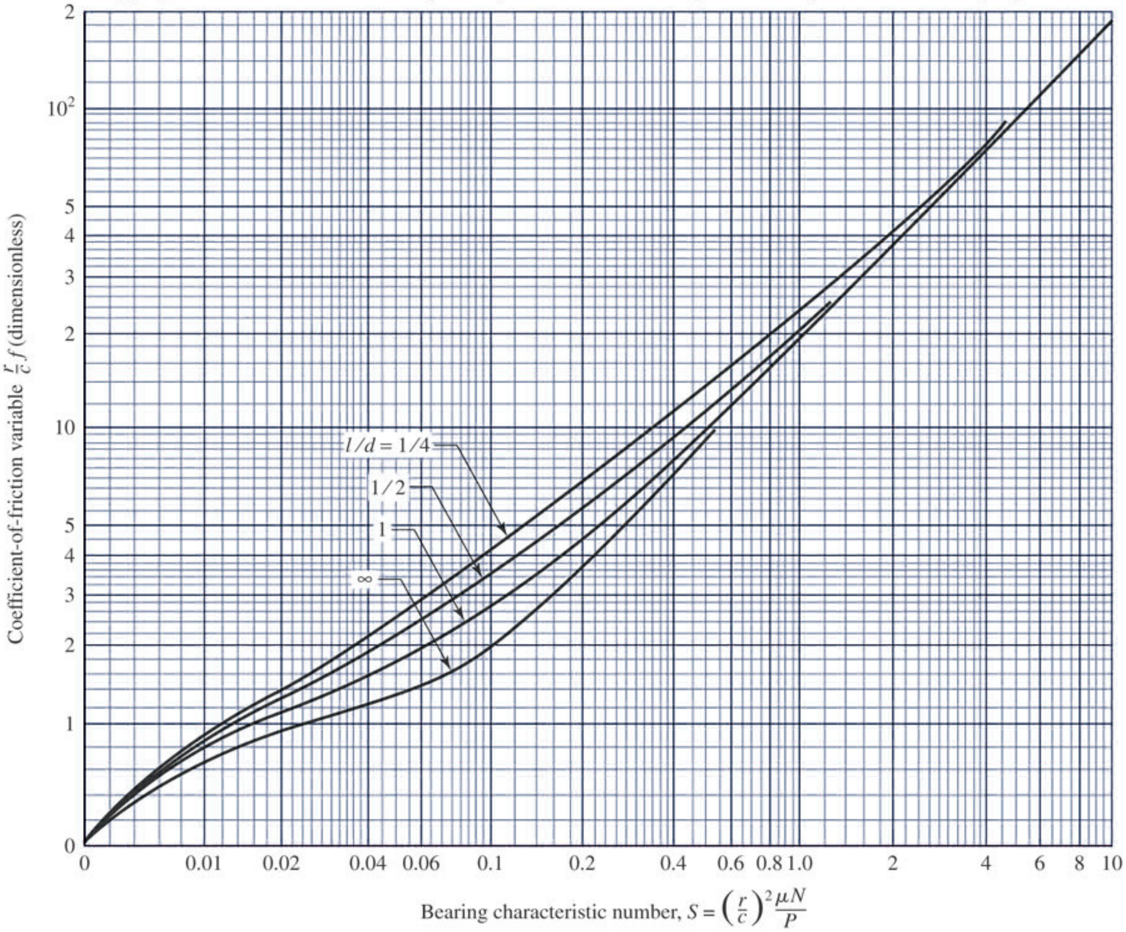
$$T = m.\frac{fF}{4}(D+d)$$

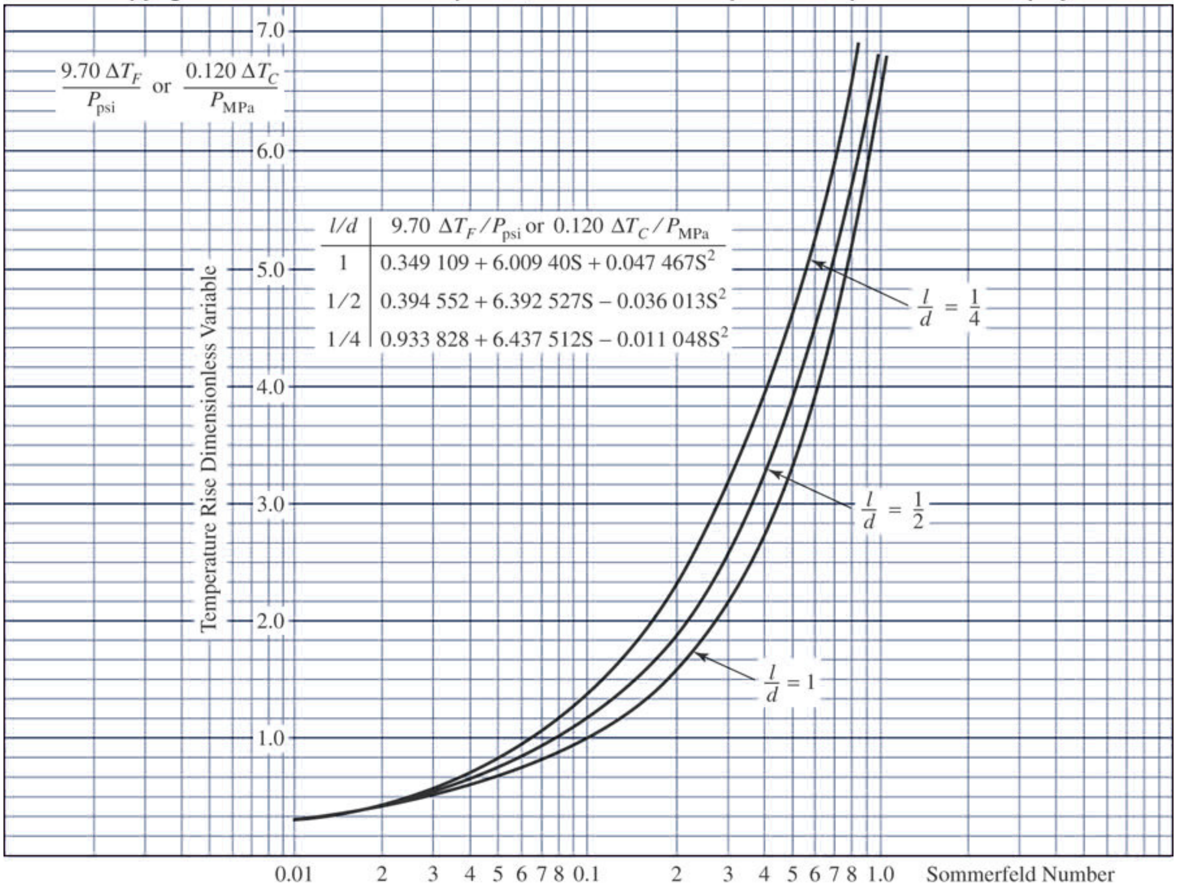
Uniform pressure theory:

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

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


Absolute viscosity (mPa·s)
Temperature (°C)
SAE 70
60
50
40
30
20
10
10^4
10^3
10^2
10
10
20
30
40
50
60
70
80
90
100
110
120
130
140


2
10^2
5
4
3
2
10
5
4
3
2
1
0
$l/d = 1/4$
1/2
1
∞
0
0.01
0.02
0.04
0.06
0.1
0.2
0.4
0.6
0.8
1.0
2
4
6
8
10
Coefficient-of-friction variable $\frac{r}{c}f$ (dimensionless)
Bearing characteristic number, $S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$



$$T_{av} = T_i + \frac{\Delta T}{2}$$

$$T_{\text{var}} = \frac{0.120\ \Delta T}{P}, \qquad p \ \text{ in } \ MPa$$