University of British Columbia

MECH 325 - Mechanical Design I

Assignment 1

Gear Train Design

GROUP C2

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Velocity = 12345 mm/sec

Cost = $1245

Performance Metric = 12345 mm/$s

1 Summary

2 Appendix

2.1 Power Screw Analysis

The objective of this section is to verify the power screw is self-locking and find the minimum required torque and rotational speed needed to lift the 2500 kg load at 4 mm/sec. Self-locking is ensured when the coefficient of thread friction is greater than a threshold defined as follows:

$$f > tan(\lambda) \tag{1}$$

$$tan\lambda = \frac{l}{\pi d_m} = 0.0335$$

Since $f = 0.08 > \tan\lambda = 0.03$, the current design is self-locking.

The torque required to lift a load with gravitational force F is:

$$\tau = \frac{F d_m}{2}\left(\frac{l + \pi f d_m}{\pi d_m - f l}\right) \tag{2}$$

Parameters			
Symbol	Value	Units	Description
F	2500×9.81	N	Axial compressive force
d	60	mm	Major diameter
d_m	57	mm	Mean diameter, $d - \frac{l}{2}$
l	6	mm	Pitch
f	0.08	N/A	Friction Coefficient

A torque of 79.5 Nm is required to lift the load where efficiency losses in the power screw is accounted for by the friction coefficient, f.

2.2 Worm Gear Analysis

Worm gears are used in high torque applications but they are subject to efficiency losses during operation. Calculating this value allows us to find the gear train required to raise the load.

$$\eta = \frac{cos\phi_n - f tan\lambda}{cos\phi_n + f cot\lambda} \tag{3}$$

The helix angle component is as follows:

$$\tan\lambda = \frac{p_x}{\pi d_p} = 0.4074366 \tag{4}$$

We find the worm gear is therefore 80.34% efficient.

Parameters			
Symbol	Value	Units	Description
N_G	18	N/A	Number of teeth on worm drive gear
N_w	2	N/A	2-thread worm
ϕ_n	14.5	degrees	Pressure angle
l	16	mm	pitch
p_x	32	mm	Axial pitch
d_p	25	mm	Worm pitch diameter
d_s	20	mm	Worm shaft diameter

The 2-threaded worm gear engages with an 18 teeth worm drive nut. For each 9 revolution of the worm gear, the nut completes 1 revolution, resulting in a 9 fold torque increase.

$$\frac{\tau}{9\eta_{worm}} = \frac{79.5}{9(0.8034)} = 11.0Nm$$

Therefore, the system requires 11.0 Nm to reach the worm gear.

2.3 Motor Torque and Gear Reduction

The motor provided has the following torque-speed curve. The maximum power output occurs when the motor operates at 2500 rpm with 2.5 Nm of torque.

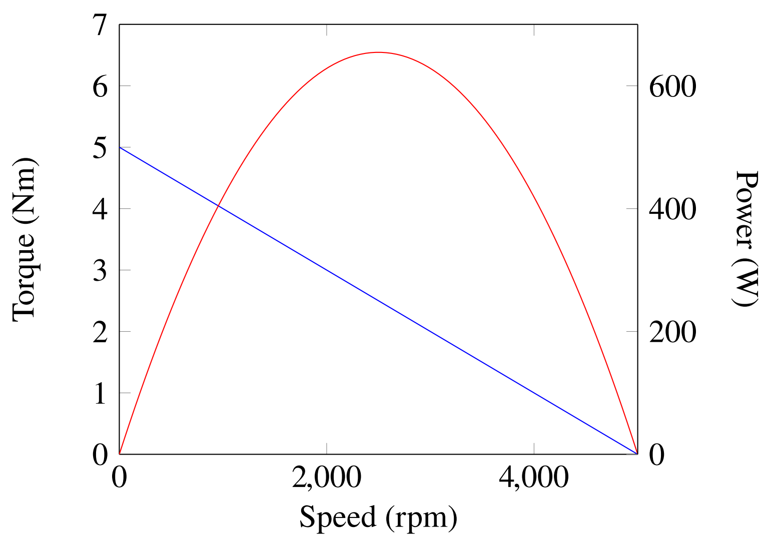

Figure 1: Torque and power curve for chosen motor. Blue shows the torque and red shows the power output. Power is maximized at 2500 rpm.

The relationship to determine train value given torque specifications, e is:

$$e = \frac{T_{out}}{T_{in} f_g} = \frac{11}{2.5} f_g = 4.4 f_g \tag{5}$$

where T_{out} = torque output, T_{in} = torque input, and f_g = gear efficiency.
Spur gears are selected for their efficiency, between 94 - 98 %, and simplicity. We assume the lower bound of efficiency.

$$\text{(simple two-gear system)} \qquad e = \frac{4.4}{0.94} = 4.68 \tag{6}$$

$$\text{(compound four-gear system)} \qquad e = \frac{4.4}{(0.94)(0.94)} = 5.0 \tag{7}$$

If only two gears are used, select gear ratio to be at least 4.68. If a compound four gear system is appropriate, select gear ratio to be at least 5.0.

2.4 Bending & Contact Stress Analysis

Detailed stress analysis was performed on each gear and pinion considered for the design of the gear train. Major stress factors were determined to come from the bending and contact stresses exerted on the gear/pinion system during meshing. The following equation was used calculate and verify the allowable bending stress:

$$\sigma_{bending} = W^t K_o K_v K_s \frac{P_d}{F} \frac{K_s K_B}{J} \tag{8}$$

Parameters		
Symbol	Units	Description
W^t	lbf	Tangential transmitted load
K_o	N/A	Overload factor
K_v	N/A	Dynamic factor
K_s	N/A	Size factor
P_d	N/A	Transverse Diametral pitch
F	mm	Face Width of narrower member
K_m	N/a	Load-distribution factor
K_B	N/A	Rim-thickness factor
J	N/A	Geometry factor (bending strength)

The contact stress measurement was taken into account by the following equation, along with the following new parameters:

$$\sigma_{contact} = C_p \sqrt{W^t K_o K_v K_s \frac{K_m}{d_P F} \frac{C_f}{I}} \tag{9}$$

Parameters		
Symbol	Units	Description
C_p	$\sqrt{lbf/in^2}$	Elastic Coefficient
C_f	N/A	Surface condition factor
d_P	mm	Pitch diameter
I	N/A	Geometry Factor (pitting resistance)

Note: Many of the factors are constants that were obtained from reading values off graphs and tables. Other values depend on the material, geometry, and physical properties of the gear and pinion. Furthermore, all the parameters in the stress equations are in U.S. customary units.

2.5 Safety Factor Analysis

To ensure the gear train system maintains an acceptable level of safety (2.2 minimum), bending and contact safety factors were calculated to validate design specifications.

$$S_{F(bending)} = \frac{S_t}{\sigma_b}\frac{Y_N}{K_T K_R} \tag{10}$$

$$S_{H(contact)} = \frac{S_c}{\sigma_c}\frac{Z_N C_H}{K_T K_R} \tag{11}$$

Parameters		
Symbol	Units	Description
S_t	bf/in^2	AGMA bending strength
S_c	lbf/in^2	AGMA surface endurance strength
Y_N	N/A	Stress cycle factor (bending strength)
Z_N	N/A	Stress cycle factor (pitting resistance)
C_H	N/A	Hardness-ratio factor
σ_b	lbf/in^2	Bending stress
σ_c	lbf/in^2	Contact stress
K_T	N/A	Temperature factor
K_R	N/A	Reliability factor