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(A2)



DEVELOPMENT AND ANALYSIS

Gear Calculations

Abstract

By using angle gears and the ball screw jack component, the flap actuator changes rotating input shaft movement over to direct fold movement.

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Flap actuator is an airplane mechanical gadget that permits folding on of airplane wings to be changed. Retractable flaps used control surfaces on the extreme end of the wing and are meant to be stretched out or withdrawn to change the wing's alignment and surface for better efficiency in low-speed flight. This flap actuator is commonly a lead screw kind of system driven by electric or water powered engine. The actuator mounted on the inner parts of the wing with a movable nut on the power screw as shown in the design. The lead screw pivots when the actuator is turned on, making the nut to move in an oscillatory manner with its described length, hence extending and withdrawing of the flap during the operation.

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Introduction

Actuators are important components of both air and spacecraft. In the advancement of aircraft, each segment is planned exactly as per predefined details. Individual parts, systems included are exposed to electronic and mechanical investigation to recognize and determine the best electromechanical highlights for the airplane's plan. This report presents the product design specification analysis for designing a flap aircraft actuator for a light airplane such as Naval aircraft.

Calculations from the Design

Technical specifications for the actuator:

1. Max force: W Newton (7000)
2. Power screw/actuator linear velocity underload: 10.0mm/s
3. Reliability 95%
4. SF: 4-6
5. The compatibility
6. Max. extension: 200mm
7. Life 8years, assume the motor works 10hours per day at top speed.
8. Max. temp: 50°C
9. The gearbox can accommodate any type of Bevel, Spur or Helical gears but not Worm gears.
10. Screw actuator material should be aircraft grade aluminium.
11. Gears and shaft can be steel or aluminium.
12. Max. size of large gear 150mm

Gearbox specification

Table 1: gear box specification

Power output	5KW
Freq. of input rotation	1500 RPM
Freq. of output rotation	500 RPM
Reliability	95%
SF	3-4
Life	8 years
Max temp	50°C
Dimension	300*200*300

have these been calculated?

Have these been decided on

Design analysis and calculations

Minimum Number of Teeth on the Gears

Gearbox Sizing

Max dimensions $200 \times 300 \times 200$

Input Speed 1500 rpm

Output Speed 500 rpm

? These values can be calculated.

$$\text{Train Value } e = \frac{\text{rpm out}}{\text{rpm in}} = \frac{500}{1500} = 0.3$$

In other relation

$$e = \frac{\text{inner dia}}{\text{outer dia}}$$

For the gear to mesh, assuming that total dia of 150mm

$$= d_1 + d_2 = 150\text{mm}$$

Gear pitch, $p = \pi m$

Where m is the module.

With the pressure angle set at 20°

Hence number of teeth $z = D/m$

$$d_1 = 55\text{mm}$$

$$d_2 = 95\text{mm}$$

No. of Teeth

From the Module = 3mm

$$M = \frac{d}{n}$$

$$\text{Input Teeth} = \frac{d_{in}}{mod} = \frac{55}{3} = 19 \text{ Teeth}$$

$$N_{in} = 19$$

$$N_{out} = 95/3 = 32 \text{ Teeth}$$

Torque on Gears

$$T = \frac{\text{Angular Velocity}}{\text{Power}} = \frac{\omega}{\rho} \frac{\rho}{\omega}$$

Small Gear

$$\text{Input Power} = 3000 \text{ W}$$

$$\omega = 1500 \times \frac{2\pi}{60} = 157 \text{ rads}$$

$$T = \frac{3000}{157} = 19.12 \text{ Nm}$$

Output Torque

$$\frac{T_{in}}{d_{in}} = \frac{T_{out}}{d_{out}}$$

$$\frac{55}{95} = \frac{19.12}{d}$$

$$\frac{55}{95} = T_{out}$$

$$T_{out} = \frac{T_{in} \times d_{out}}{d_{in}} = \frac{19.12 \times 95}{55} = 33.03 \text{ Nm}$$

Tangential Forces

$$T = \frac{d}{2} W_t \text{ hence } W_t = \frac{2T_{in}}{d_{in}}$$

$$W_t = \frac{2 \times 33.03}{0.095}$$

$$= 695.37 \text{ N}$$

Analysis for 2 stages of compound gear-train (Gupta, Khurmi 2005),

- Total speed reduction ratio = $\frac{N_{in}}{N_{out}} = \frac{1500}{500} = 3$
- The gear ratios were chosen as 1:3.

- 14 1/2° involute gear teeth type were chosen because they were easily machinability and optimal for heavy loads (Gupta, Khurmi, 2012).
- Gears were determined to design with minimum number of teeth in order to maintain the compactness and lightweight of the system. When G is the gear ratio, A_w is equals to 1 and

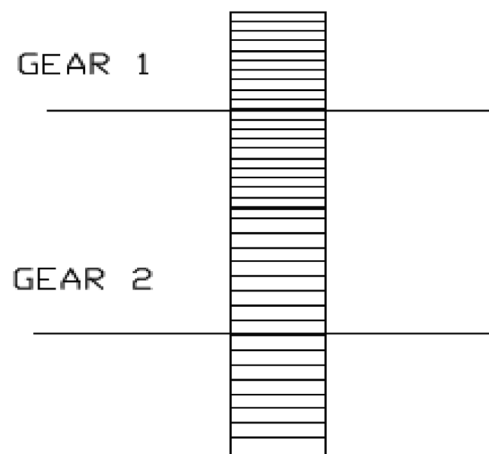
θ is the pressure angle,

GEAR DESIGN

T = Number of Teeth

N = Gear speed in r.p.m.

D_G =Diameter of the Gear in mm: (RG=Radius of the gear)



Are you only using two gears?

Figure 9: Gear arrangement

The transmission ratio of Gear drive = 2: 1

$$\frac{N_1}{N_2} = \frac{D_{G1}}{D_{G2}}$$

$$\frac{2}{1} = \frac{D_{G1}}{D_{G2}}$$

Assumptions:

$$D_{G1} = 2D_{G2}$$

Diameter of gear 1 (D_{G1}) = 100 mm

Diameter of gear 2 (D_{G2}) = 150 mm

$$\text{Module, } m = \frac{D}{T}$$

Selecting Module from That Standard Module, $m = 4$.

Number of teeth on gear 1 and 2,

$$T_{G1} = \frac{100}{4} = 25$$

Assumptions: -

This can be calculated.

1. Speed of the gear 1 = 1500 r.p.m.

Table 2-Gear Characteristics

$$T_{G2} = \frac{150}{4} = 37.5$$

Fig 10: No: of teeth vs. Involute factor Y

MATERIALS SELECTION

For this design 20° Stub involute system gears are used, because it has a strong tooth to take heavy loads.

Using the table of mechanical properties of some gear steels, EN8 Hardened steel was chosen.

Assume that the Ultimate tensile stress $(\sigma_{ut}) = 110 \text{ kg/mm}^2 = 1078.75 \text{ N/mm}^2$ MPa

Allowable static stress $(\sigma_0) = \frac{\sigma_{ut}}{3}$

$$(\sigma_0) = \frac{1078.75 \text{ N/mm}^2}{3}$$

$$(\sigma_0) = 359.58 \text{ N/mm}^2$$

Strength factor of gear wheel 1

$$\sigma_{01} = \sigma_0 \times y$$

$$\sigma_{01} = 359.58 \times 0.133$$

$$\sigma_{01} = 47.824 \text{ N/mm}^2$$
 ✓

Strength factor of gear wheel 2

$$\sigma_{02} = \sigma_0 \times y$$

$$\sigma_{02} = 359.58 \times 0.151$$

$$\sigma_{02} = 54.296 \text{ N/mm}^2$$

Here, the strength factor for gear wheel 1 is less than gear wheel 2. ($\sigma_{01} < \sigma_{02}$)

So, gear 1 is weaker.

Assuming the gear 1 and 3 get the same torque.

Torque on gear 1 (T_{G1}) = 199.46×10^3 N-mm.

Circular pitch (P_c) = $\pi m = \pi \times 4 \text{ mm}$

$$(P_c) = 12.566 \text{ mm}$$

$$\text{Tangential load } (W_{T1}) = \frac{2T}{D} = \frac{2 \times 199.46}{100 \times 10^{-3}}$$

$$(W_{T1}) = 3989.2N$$

$$\text{Normal load on the tooth } (W_{N1}) = \frac{W_T}{\cos \phi} = \frac{3989.2N}{\cos 20^\circ}$$

$$(W) = 4245.28N$$

this can be calculated.

Let assume the normal pressure between the teeth is 40 N/mm , therefore necessary face width of gear 1,

$$b = \frac{4245.28}{40} = 106.132 \approx 106 \text{ mm}$$

Pitch line velocity of gear wheel 1

$$v = r\omega$$

$$= \frac{100 \times 10^{-3}}{2} \times \frac{2\pi}{60} \times 359.5$$

$$= 1.882 \text{ m/s}$$

Value of Deformation Factor C

Explain this

$$C = 952 \times 10^3 \text{ N/m.}$$

$$W_{11} = \frac{21v(b.C + W_{T1})}{21v + \sqrt{bC + W_{T1}}}$$

$$W_{11} = \frac{21 \times 1.882 \times (106 \times 952 + 3989.2)}{21 \times 1.882 + \sqrt{(106 \times 952) + 3989.2}}$$

$$W_{11} = 11408.45N$$

Dynamic Tooth Load

$$W_D = W_{T1} + W_{11}$$

$$W_D = (3989.2 + 11408.45)$$

$$W_D = 15397.65N$$

Static Tooth Load (Endurance Strength)

$$W_{S1} = \sigma_e \times b \times \pi \times m \times y$$

From steel Hardness conversion table

$$40 \text{ HRC} = 373 \text{ B.H.N}$$

$$\sigma_e = 1.75 \times 373 \text{ MPa}$$

$$\sigma_e = 652.75N/mm^2$$

$$W_{S1} = 652.75 \frac{N}{mm^2} \times 106mm \times \pi \times 4 \text{ mm} \times 0.133$$

$$W_{S1} = 36809.88N$$

For Safety,

$$W_{s1} \geq W_D$$

The condition satisfies for the obtained values

Wear Tooth Load

$$W_w = D_p \times b \times Q \times k$$

Where,

W_w = Maximum or limiting load for wear in N

D_p = Pitch circle diameter of the pinion in mm

b =Face width of the pinion in mm

Q =Ratio factor

K = *Load*-stress factor (also known as material combination factor) in N/mm^2 .

1. $Q = \frac{2 \times V.R}{V.R + 1}$

$$V.R = \frac{T_{G1}}{T_{G2}} = \frac{50}{25} = 2$$

$$Q = \frac{2 \times 2}{2 + 1} = \frac{4}{3} = 1.333$$

2. $K = \frac{(\sigma_{es})^2 \times \sin \phi}{1.4} \left\{ \frac{1}{E_p} + \frac{1}{E_g} \right\}$

Where,

σ_{es} =Surface endurance limit in MPa or N/mm^2

ϕ =Pressure angle

E_p = *Young's* modulus for the material of the pinion in N/mm^2

E_g = *Young's* modulus for the material of the gear in N/mm^2 .

Young modulus for EN8 steel = $185 \times 10^3 N/mm^2$

The surface endurance limit for steel

$$\sigma_{es} = (2.8 \times 373 - 70) N/mm^2$$

$$\sigma_{es} = 974.4 N/mm^2$$

$$K = \frac{(974.4)^2 \times \sin 20}{14} \left\{ \frac{1}{180 \times 10^3} + \frac{1}{180 \times 10^3} \right\}$$

$$K = 2.577 N/mm^2$$

$$w_w = 100 \text{ mm} \times 106 \text{ mm} \times 1.333 \times 2.577 N/mm^2$$

The condition satisfies for the obtained values

$$w_w = 36412.49 N$$

For Safety,

$$W_w \geq W_D$$

$$N_{G5} = 198.6 \text{ r.p.m.}$$

$$150 \text{ mm} = \frac{D_{G3}}{2} + D_{G4} + \frac{D_{G5}}{2}$$

$$D_{G4} = 150 - (30 + 60)$$

$$D_{G4} = 60 \text{ mm}$$

Table 3-Gear Characteristics

$$\text{Module, } m = \frac{D}{T}$$

Selecting Module from the Standard Module, $m = 4$.

$$T_{G4} = \frac{60}{4} = 15$$

Number of teeth on gear 1 and 2,

$$T_{G3} = \frac{60}{4} = 15 \quad T_{G4} = \frac{60}{4} = 15 \quad T_{G5} = \frac{120}{4} = 30$$

MATERIALS SELECTION

For this design 20° Stub involute system gears are used, because it has a strong tooth to take heavy loads. Using the table of mechanical properties of some gear steels, EN8 Hardened steel is chosen.

$$\text{Ultimate tensile stress } (\sigma_{ut}) = 110 \text{ kg/mm}^2 = 1078.75 \text{ N/mm}^2$$

$$\text{Allowable static stress } (\sigma_0) = \frac{\sigma_{ut}}{3}$$

$$(\sigma_0) = \frac{1078.75 \text{ N/mm}^2}{3}$$

$$(\sigma_0) = 359.58 \text{ N/mm}^2$$

Strength factor of gear wheel 3 & 4

$$\sigma_{03} = \sigma_0 \times y$$

$$\sigma_{03} = 359.58 \times 0.111$$

$$\sigma_{03} = 39.91 \text{ N/mm}^2$$

Strength factor of gear wheel 5

$$\sigma_{05} = \sigma_0 \times y$$

$$\sigma_{05} = 359.58 \times 0.139$$

$$\sigma_{05} = 49.98 \text{ N/mm}^2$$

Here, the strength factor for gear wheel 3 is less than gear wheel 4. So, gear wheels 3 & 4 are weaker. Out of those two gears, number 4 is the the pinion. Assuming the torque received on gear 3 completely transferred onto gear 4.

$$\text{Torque on gear 4 } (T_{G4}) = 199.46 \times 10^3 \text{ N-mm.}$$

$$\text{Circular pitch } (P_c) = \pi m$$

$$(P_c) = 12.566 \text{ mm}$$

$$\text{Tangential load, } W_{T4} = \frac{2T}{D} = \frac{2 \times 199.46}{60 \times 10^{-3}}$$

$$W_{T4} = 6648.67N$$

$$\text{Normal load on the tooth, } W_{N1} = \frac{W_T}{\cos \phi} = \frac{6648.67N}{\cos 20^\circ}$$

$$W_{N1} = 7075.37N$$

Pressure between the tooth is 40 N/mm^2 , face width of gear 1,

$$b = \frac{7075.37}{40} = 176.88 \text{ mm} \approx 177 \text{ mm}$$

Pitch line velocity of gear wheel 1

$$v = r\omega$$

$$= \frac{60 \times 10^{-3}}{2} \times \frac{2\pi}{60} \times 359.5$$

$$= 1.129 \text{ m/s}$$

Value of Deformation Factor C

$$C = 952 \times 10^3 \text{ N/m.}$$

$$W_{14} = \frac{21v(b.C + W_{T4})}{21v + \sqrt{bC + W_{T4}}}$$

$$W_{14} = \frac{21 \times 1.129 \times (177 \times 952 + 6648.67)}{21 \times 1.129 + \sqrt{(177 \times 952) + 6648.67}}$$

$$W_{14} = 9390.53N$$

Dynamic Tooth Load

$$W_{D4} = W_{T4} + W_{14}$$

$$W_{D4} = (6648.67 + 9350.53)$$

$$W_{D4} = 15999.2N$$

Static Tooth Load (Endurance Strength)

$$W_{S4} = \sigma_e \times b \times \pi \times m \times y$$

From steel Hardness conversion table,

$$40 \text{ HRC} = 373 \text{ B.H.N}$$

$$\sigma_e = 1.75 \times 373 \text{ MPa}$$

$$\sigma_e = 652.75 \text{ N/mm}^2$$

$$W_{S4} = 652.75 \frac{\text{N}}{\text{mm}^2} \times 177 \text{ mm} \times \pi \times 4 \text{ mm} \times 0.111$$

$$W_{S4} = 161158.41 \text{ N}$$

For Safety,

$$W_{S4} \geq W_{D4}$$

α The condition satisfies for the obtained values

Wear Tooth Load

$$W_w = D_p \times b \times Q \times k$$

Where,

W_w = Maximum or limiting load for wear in N

D_p = Pitch circle diameter of the pinion in mm

b = Face width of the pinion in mm

Q = Ratio factor

K = Load-stress factor (also known as material combination factor) in N/mm^2

$$1. Q = \frac{2 \times V.R}{V.R + 1}$$

$$V.R = \frac{T_{G1}}{T_{G2}} = \frac{30}{15} = 2$$

$$Q = \frac{2 \times 2}{2 + 1} = \frac{4}{3} = 1.333$$

$$2. K = \frac{(\sigma_{es})^2 \times \sin \phi}{1.4} \left\{ \frac{1}{E_p} + \frac{1}{E_g} \right\}$$

Where,

σ_{es} = Surface endurance limit in MPa or N/mm^2

ϕ = Pressure angle

E_p = Young's modulus for the material of the pinion in N/mm^2

E_g = Young's modulus for the material of the gear in N/mm^2 .

Both gear and pinion are made of the same material. So, both young modulus values are equal.

Young modulus for EN8 steel = $185 \times 10^3 N/mm^2$

The surface endurance limit for steel

$$\sigma_{es} = (2.8 \times 373 - 70) N/mm^2$$

$$\sigma_{es} = 974.4 N/mm^2$$

$$K = \frac{(974.4)^2 \times \sin 20}{14} \left\{ \frac{1}{180 \times 10^3} + \frac{1}{180 \times 10^3} \right\}$$

$$K = 2.577 N/mm^2$$

$$W_{W4} = 60 \text{ mm} \times 177 \text{ mm} \times 1.333 \times 2.577 N/mm^2$$

$$W_{W4} = 36481.197 N$$

For Safety,

$$W_{W4} \geq W_{D4}$$

The condition satisfies for the obtained values

$$\text{Minimum no. of teeth for engine pinion} = \frac{2A_w}{G \left[\sqrt{1 + \frac{1}{G} \left[\frac{1}{G} + 2 \right]} \sin^2 \theta - 1 \right]}$$

$$\text{Minimum no. of teeth for engine pinion} = \frac{2}{3 \times \sqrt{1 + \frac{1}{3} \times \left[\frac{1}{3} + 2 \right]} \sin^2 14.5^\circ - 3}$$

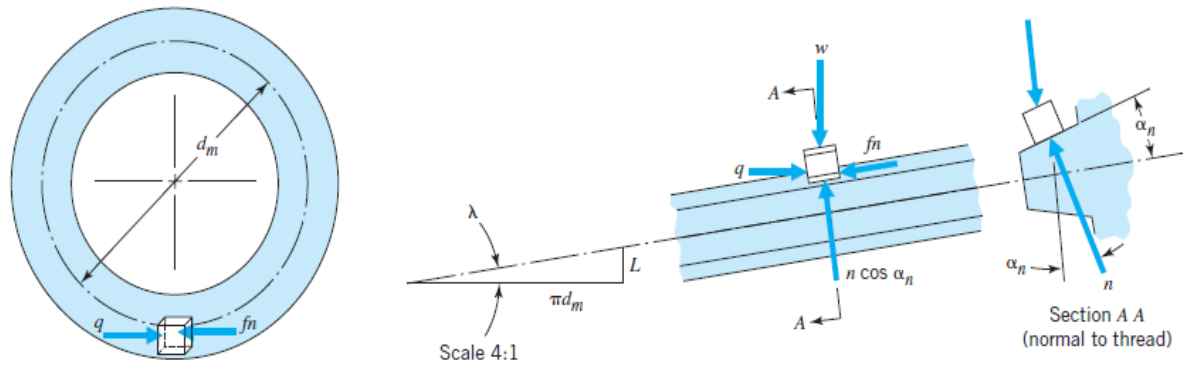
$$\text{Minimum no. of teeth for engine pinion} = 28$$

$$\text{No. of teeth for engine gear} = 28 \times 3 = 84$$

?

Power, Screw and Linear Actuator Screw Calculations.

Major reason for using power screw for the design, is to get higher MA for the drive system.



http://www.uobabylon.edu.iq/eprints/publication_7_4903_752.pdf

$$\tan \lambda = \frac{L}{\pi d_m}$$

Where

λ = lead angle

L = lead

d_m = mean diameter of contact of the thread

Hence.

Take lead angle = 20° and mean diameter = 70mm

$$L = \pi d_m (\tan \lambda) = \pi \times 70 \times \tan 20^\circ$$

$$= 0.08004\text{m}$$

Forces acting on the system as a result of:

- Load,
- Axial Load,
- Nominal force,
- Friction forces
- Tangential forces.

Σ Tangential forces.

$$\Sigma F_t = 0: \quad q - n(f \cos \lambda + \cos \alpha_n \sin \lambda) = 0$$

Σ Axial forces.

$$\Sigma F_a = 0: \quad w + n(f \sin \lambda - \cos \alpha_n \cos \lambda) = 0$$

Σ Normal force.

$$n = \frac{w}{\cos \alpha_n \cos \lambda - f \sin \lambda}$$

$$q = w \frac{f \cos \lambda + \cos \alpha_n \sin \lambda}{\cos \alpha_n \cos \lambda - f \sin \lambda}$$

Thus

$$T = Q \frac{d_m}{2} = \frac{W d_m}{2} \frac{f \cos \lambda + \cos \alpha_n \sin \lambda}{\cos \alpha_n \cos \lambda - f \sin \lambda}$$

$$T = F \times a$$

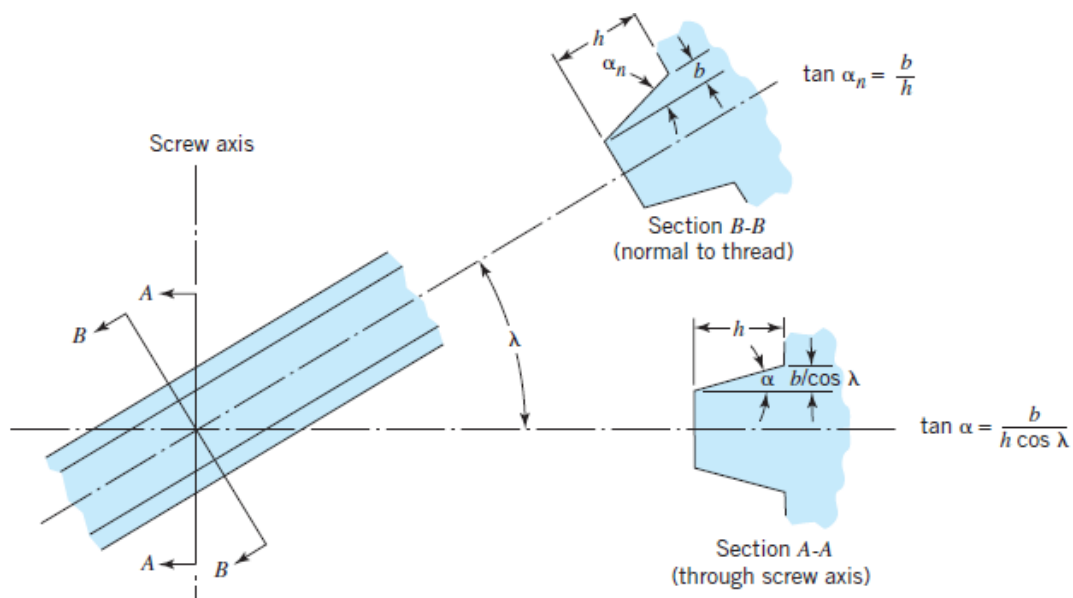
Power generated by the screw in design when it has fine threads in every inch.

$$P = 1/5 = 0.2 \text{ in} \approx 5\text{mm}$$

Due to double thread,

$$L = 2 \times P = 2 \times 5 = 10\text{mm}.$$

$$\text{Mean diameter} = d - p/2 = 70 - 2.5 = 67.5\text{mm}$$



http://www.uobabylon.edu.iq/eprints/publication_7_4903_752.pdf

(α and α_n).

$$\tan \alpha_n = \tan \alpha \cos \lambda$$

$$\alpha_n = \tan^{-1}(\tan \alpha \cos \lambda)$$

$$\tan^{-1}(\tan 14.5 \cos 20) = 13.70^\circ$$

$$T = \frac{1000(70) \times 0.4 \cos 20 + \cos 13.70 \sin 20}{2 \cos 13.70 \cos 20 - 0.4 \sin 20}$$

$$= 4698.96 \text{ Nm}$$

Screw Stresses

Torsional stress

$$\tau = \frac{Tc}{J} = \frac{16T}{\pi d r^3}$$

$$= \frac{16 \times 4698.96}{\pi \times 0.5^3} = 119.63 \text{ GPa.}$$

Force.

$$T = F \times a$$

$$= F = T/a$$

$$= \frac{4698.96}{10} = 469.9 \text{ N}$$

EFFICIENCY OF POWER SCREW

$$\eta = \frac{\text{output work}}{\text{input work}} = \frac{w \times L}{2\pi \times T} = 73\%$$

Electro Motor (BLDC Motor)

The design utilizes an electromotor, typical rare earth permanent brushless magnet type. The Electromotor design has been completed in the CAD software program, and the design is based on the surface-mounted radial flux PM Electromotor with specifications derived through electromagnetic motor expressions and equations (Qiao et al. 2018, p.96).

Motor specifications

Number of Phase	5
-----------------	---

Number of Poles	18
-----------------	----

Phase Current

Safety operation constant	4.4
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One open-phase fault	6.3
----------------------	-----

Two open-phase faults	12.4
-----------------------	------

Speed and torque for healthy operation mode	18.0 Nm in 600 revolution per min.
---	------------------------------------

Speed and torque for one open-phase fault	18.1 Nm in 600 revolution per min.
---	------------------------------------

Speed and torque for two open-phase faults	17.8 Nm in 400 revolution per min.
--	------------------------------------

The design is considered to be the most important source of resistance for the system when; Non reverse drag torque, screw shaft seal, rotor seal and bearing torque. Therefore, the required motor torque output can be calculated using the following formula. Cavallie 2002),

$$T_{MOT} = \frac{L_{OP} \cdot P}{2\pi \cdot \eta_{BS}} + T_{DSS} + T_{DNB} + T_{DROTOR}$$

And hence the required motor power input can be calculated through/using rated speed motor efficiency as.

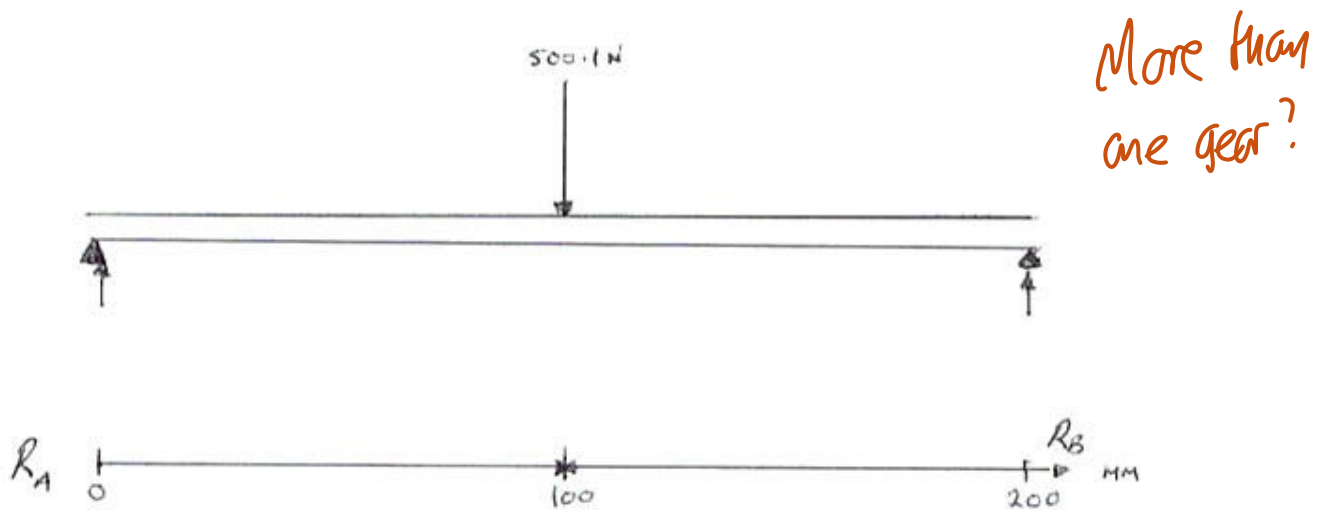
$$P_{IN} = \frac{n \cdot T_{MOT}}{\eta_{MOT}}$$

which.

- P – Power Screw Pitch
 - L_{op} – Operating Load
 - η_{BS} – Power Screw Direct Efficiency
 - T_{DSS} – Bearing Drag Torque and Screw Seal
 - T_{MOT} – Motor Torque
 - η_{MOT} – Motor Efficiency
 - T_{DNB} - No-back Drag Torque
 - n- Speed

Shaft Designing Criteria

For the analysis on material selection, chosen materials are stainless steel, carbon steel and cast iron. Consider the initial length of the shaft as 200mm shown in the sketch below; (Khurumi R S, Gupta 2005).



$$\sum M = 0 \text{ \& } \sum F = 0$$

Consider the moments R_A

$$(500.1 \times 0.1) - (R_B \times 0.2) = 0$$

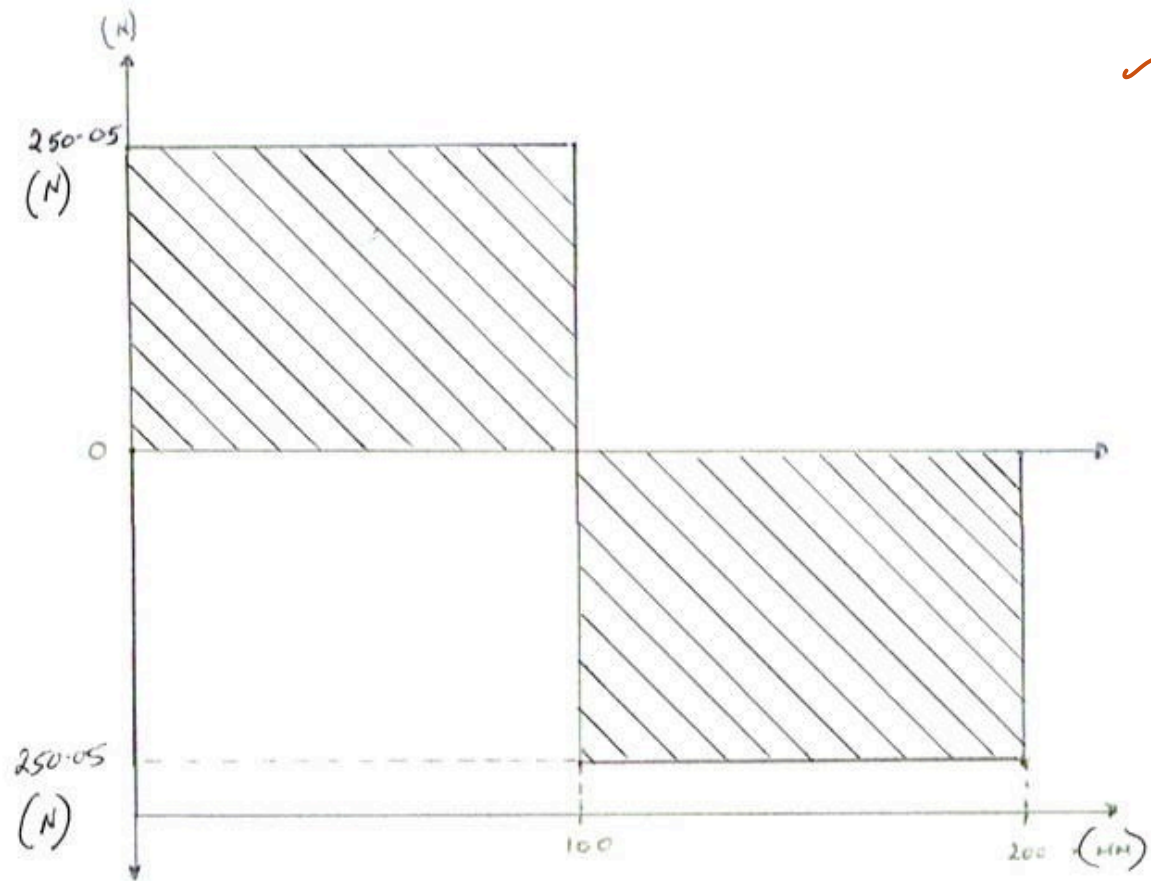
$$R_B = 250.05 \text{ N}$$

Consider moments at R_B

$$(500.1 \times 0.1) = R_A \times 0.2$$

$$R_A = 250.05 \text{ N}$$

Shear moment diagram.



Bending Moment

Maximum Bending Moment at 100mm.

$$M = F \times D$$

$$= 250.05 \times 0.1$$

$$= 25.01 \text{ Nm.}$$

Shaft Calculations

From ACME equations

$$d^3 = \frac{32 \times F \times S}{\pi(1-k)\delta_y} \sqrt{(K_b m)^2 + (K_{tT})^2}$$

Reference made from ACME table.

$$K = 0.2 \quad K_b = 1.5 \quad K_t = 1.0$$

$$\text{And } \sigma_y = 265 \text{ MPa}$$

Therefore, diameter of Shaft =

$$d^3 = \frac{32 \times 4}{(\pi(1-0.2)(265 \times 10^6))} \sqrt{(1.5 \times 25.01)^2 + (1 \times 23.1)^2}$$

$$d^3 = 6.8469 \times 10^6$$

$$D = 0.019 \text{ m} \simeq 19 \text{ mm}$$

Deflection of the Shaft

$$\theta = \frac{FL^2}{16EI} \quad I = \frac{\pi D^4}{64} \quad \text{and} \quad Y = \frac{F_r}{48EL}$$

$$\theta = \frac{500.1 \times 0.2^2}{(16 \times 205 \times 10^9) \times I}$$

$$I = \frac{\pi \times 0.019^4}{64} \quad 6.397 \times 10^{-9}$$

$$\theta = \frac{500.1 \times 0.2^2}{(16 \times 205 \times 10^9) \times (6.397 \times 10^{-9})}$$

$$= 9.534 \times 10^4 \text{ rads}$$

Deflection

$$Y = \frac{FL^3}{48ET} = \frac{500.1 \times 0.2^3}{48(205 \times 10^9)(6.397 \times 10^{-9})}$$

$$= 3.051 \times 10^{-3} \text{ m}$$

$$= 3.05 \text{ mm.}$$

Bearing selection and life calculations

Bearing specification

$$L_{10} = \frac{(C/P)^e \times 10^6}{60 \times N}$$

C = Dynamic Capacity

P = Equivalent Bearing Load

N = Rotating speed in RPM

e = 3.33 for roller bearings and 3.0 for ball bearings

Calculation of Bearing Rating Life

$$L_{10} = \frac{\left(\frac{C}{P}\right)^e \times 10^6}{60 \times N}$$

$L_{10} = (C/p)^3$ for ball bearing and $L_{10} = (C/p)^{10/3}$ roller bearing

$$L_{10} = \frac{\left(\frac{C}{P}\right)^e \times 10^6}{60n} = \frac{50 \times 10^6}{60 \times 3500} \times \left(\frac{10^6}{60} \times 1500\right)$$

Where.

P – Equivalent bearing load (N)

C- Dynamic capacity (dN)

$e = 10/3$ for roller bearings and 3.0 for ball bearings

$$L_{10} = \frac{1}{\left(\frac{T_1}{L_{p1}}\right) + \left(\frac{T_2}{L_{p2}}\right) + \left(\frac{T_n}{L_{pn}}\right)}$$

Where

L_{p1}, L_{p2}, L_{pn} – life in hours for each period of constant speed and load

T_1, T_2, T_n - % of time at different condition, often expressed as decimal

The summation of $T_1 + T_2 + \dots + T_n = 1$

Viscosity ratio

$$\kappa = v/v_1$$

Dynamic viscosity v_1 depends on size (D_{pw}) and speed (n) and is obtained through

$$V_1 = 45000 n^{-0.83} D_{pw}^{-0.5} \text{ for } n < 1000 \text{ rev/min}$$

$$V_1 = 4500 n^{-0.5} D_{pw}^{-0.5} \text{ for } n > 1000 \text{ rev/min}$$

Safety factor on bearings is given by

$$S_0 = C_0/P_0$$

Where

S_0 = safety factor

C_0 = Basic static load rating, radial bearing (C_{0r}), Thrust bearing (C_{0a})

P_0 = static equivalent load (N), radial bearing (P_{0r}), Thrust bearing (P_{0a})

Conditions of the designed actuator

- i. Ball bearing groove; 6308
- ii. Radial load $F_r = 3500$ N
- iii. Axial Load = 1000 N
- iv. Rotational speed = 1500 rpm

With basic load rating C_{0r} , C_r and $f_0 = 24$ kN, 50.9 kN and 13.2 respectively

$$\frac{f_0 F_a}{C_{0r}} = \frac{13.2 \times 1000}{24 \times 10^3} = 0.55$$

$$e = 0.22 + (0.26 - 0.22) \times \frac{0.55 - 0.345}{0.689 - 0.345} = 0.24$$

$$\frac{F_a}{F_r} = \frac{1000}{3500} = 0.29 > e$$

Thus, implies that $X = 0.56$ and $Y = ?$

$$Y = 1.99 - (1.99 - 1.71) \times \frac{0.55 - 0.345}{0.689 - 0.345} = 1.82$$

Therefore

$$P_r = X F_r + Y F_a = (0.56 \times 3500) + (1.82 \times 3500) = 3780 \text{ N}$$

Hence service life with 95% reliability (L_{10h}) is obtained by

$$L_{10} = \frac{\left(\frac{C}{P}\right)^e \times 10^6}{60 \times N} = \frac{10^6}{60 \times 1500} \times \left(\frac{50.9 \times 10^3}{3780}\right)^3$$
$$= 27127 \text{ h}$$

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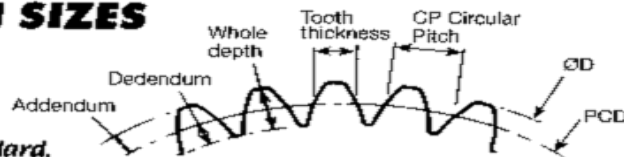
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GEAR TOOTH SIZES



Bold numbers are standard.
Dimensions in mm and inches.

MOD	DP	CIRCULAR PITCH	TOOTH THICKNESS	ADDENDUM (= MODULE)	DEDENDUM	WHOLE DEPTH
0.2 MOD	127.00	0.628	0.314	0.20	0.28	0.48
0.25 MOD	101.60	0.785	0.392	0.25	0.35	0.60
0.254	100 DP	0.0314"	0.0157"	0.010"	0.014"	0.0240"
0.265	96 DP	0.0327"	0.0163"	0.0104"	0.0145"	0.0250"
0.3 MOD	84.67	0.942	0.471	0.30	0.42	0.72
0.318	80 DP	0.0392"	0.0196"	0.0125"	0.0175"	0.0300"
0.353	72 DP	0.0436"	0.0218"	0.0138"	0.0194"	0.0333"
0.397	64 DP	0.0490"	0.0245"	0.0156"	0.02187"	0.0375"
0.4 MOD	63.50	1.256	0.628	0.40	0.56	0.96
0.5 MOD	50.80	1.570	0.785	0.50	0.70	1.20
0.529	48 DP	0.0654"	0.0327"	0.0208"	0.02916"	0.0500"
0.6 MOD	42.33	1.884	0.942	0.60	0.84	1.44
0.635	40 DP	0.0785"	0.0392"	0.025"	0.0350"	0.0600"
0.7 MOD	38.29	2.199	1.099	0.70	0.98	1.68
0.706	36 DP	0.0872"	0.0436"	0.0277"	0.0388"	0.0666"
0.75 MOD	33.87	2.356	1.178	0.75	1.05	1.80
0.794	32 DP	0.0981"	0.04906"	0.0312"	0.0437"	0.0750"
0.8 MOD	31.75	2.513	1.256	0.80	1.12	1.92
0.847	30 DP	0.1047"	0.05235"	0.0333"	0.0466"	0.08"
0.9 MOD	28.22	2.827	1.413	0.90	1.26	2.16
0.907	28 DP	0.1121"	0.05605"	0.0357"	0.0500"	0.0857"
1.00 MOD	25.40	3.141	1.570	1.00	1.400	2.40
1.05	24 DP	0.1308"	0.0654"	0.0416"	0.0583"	0.1000"
1.15	22 DP	0.1427"	0.07135"	0.0454"	0.0636"	0.1090"
1.25 MOD	20.32	3.926	1.963	1.25	1.562	2.81
1.27	20 DP	0.1570"	0.0785"	0.050"	0.0625"	0.1125"
1.411	18 DP	0.1745"	0.08725"	0.0555"	0.0694"	0.1250"
1.50 MOD	16.93	4.712	2.356	1.50	1.875	3.37
1.588	16 DP	0.1963"	0.09815"	0.0625"	0.0781"	0.1406"
1.693	15 DP	0.2094"	0.1047	0.0666"	0.0833"	0.1500"
1.75 MOD	14.51	5.497	2.7485	1.75	2.187	3.93
1.814	14 DP	0.2243"	0.11215"	0.0714"	0.0892"	0.1607"
2.0 MOD	12.70	6.283	3.141	2.00	2.50	4.50
2.117	12 DP	0.2617"	0.13085"	0.0833"	0.1041"	0.1875"
2.25 MOD	11.29	7.068	3.534	2.25	2.812	5.06
2.5 MOD	10 DP	7.853	3.926	2.50	3.125	5.62
2.54	10 DP	0.3141"	0.15705"	0.100"	0.1250"	0.2250"
2.75 MOD	9.24	8.639	4.319	2.75	3.437	6.18
3 MOD	8.47	9.424	4.712	3.00	3.75	6.75
3.175	8 DP	0.3926"	0.1963"	0.125"	0.15625"	0.28125"
4 MOD	6.35	12.566	6.283	4.00	5.00	9.00
4.233	6 DP	0.5235	0.26175"	0.1666"	0.20833"	0.3750"
5 MOD	5.08	15.707	7.853	5.00	6.25	11.25
5.08	5 DP	0.6283"	0.31415"	0.200"	0.2500"	0.4500"
6 MOD	4.23	18.849	9.4245	6.00	7.50	13.50
6.35	4 DP	0.7853"	0.39265"	0.250"	0.3125"	0.5625"
8 MOD	3.175 DP	25.132	12.566	8.00	10.00	18.00

TECHNICAL SECTION | GENERAL

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Figure 2: Table of gear sizing

Materials		Involute tooth form	Tooth Error — mm				
Pinion	Gear		0.01	0.02	0.04	0.06	0.08
cast iron	cast iron	14½°	55	110	220	330	440
steel	cast iron	14½°	76	152	304	456	608
steel	steel	14½°	110	220	440	660	880
cast iron	cast iron	20° full depth	57	114	228	342	456
steel	cast iron	20° full depth	79	158	316	474	632
steel	steel	20° full depth	114	228	456	684	912
cast iron	cast iron	20° stub	59	118	236	354	472
steel	cast iron	20° stub	81	162	324	486	648
steel	steel	20° stub	119	238	476	714	952

Figure 3; Values of Deformation Factor C (kN/m) for Dynamic load