DEPARTMENT OF MECHANICAL ENGINEERING

VALIDATION REPORT

Power Take-Off Rear Reduction Unit

Mechanical Design B (04-22964)

Module Coordinator: Professor K D Dearn

Group No: 11

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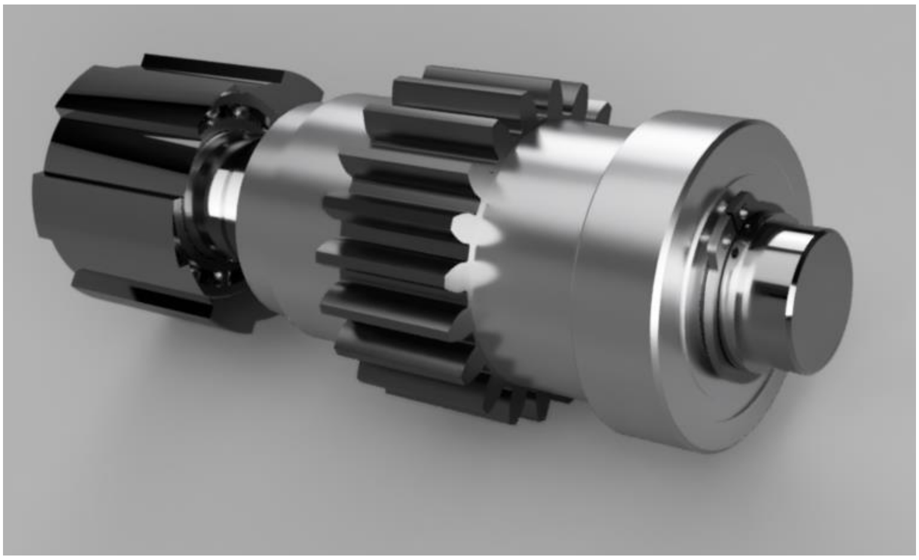


Aero-engine speed reducer

Design brief:

The rotary Wankel is an ideal engine for micro-light aircrafts due to its simple design that has few parts, has the capability to run smoothly and has a high power/weight ratio. The engine is capable of running at such a high speed that it requires a speed reduction gearbox to prevent the propeller from reaching excessive velocities. We have been commissioned for the design of a speed reduction gearbox to be incorporated with a Wankel engine.

1.Performance:

1.1 The engine operates at two settings throughout the flight and the speed reducer must be able to accommodate for these 2 different loads.

1.2 The propeller speed must be at a maximum of 2000 rpm during maximum cruise for a period of 2000 hours. The max cruise power input is 45 hp at 6500 rpm.

1.3 At maximum power for take-off and certain manoeuvres the power input is 50 hp at 7000 rpm for an expected 200 hours of use.

1.4 The propeller should generate a thrust of 82 knots.

1.5 The propeller must rotate clockwise.

2.Lifespan throughout service:

2.1 2000 hours at maximum cruise

2.2 200 hours at maximum power

3.Quantity:

3.1 Minimum production of 200 per annum.

4.Maintenance:

4.1 Every 5 years the transmission should be dismantled, and parts should be inspected for quality purposes.

5.Enviroment

5.1 Corrosion resistance may be considered for the casing using surface protection methods.

5.1 For sites with extreme temperature variations we should consider different viscosity lubricants through-out seasons.

6.Manufacturing

6.1 Spur Gears will be brought in from KHK gears.

6.2 Bearings will be brought in from SFK bearings.

7.Size restrictions:

7.1 The output shaft can extend up to 90 mm from the axial of the engine shaft.

7.2 The distance between the output shaft and engine shaft should be up to 210 mm.

8.Quality and reliability:

8.1 The speed reducer must have high reliability therefore quality materials should be used.

8.2 Gears should be made to a very high standard of machining and inspection.

8.3 Shafts should be designed for an infinite lifespan.

8.4 Standardised bearings and seals will be purchased from established manufacturers to have a strong understanding of their lifespan.

9.Safety:

9.1 Shafts should be inspected to ensure adequate stiffness to minimise the risk of failure.

9.2 To reduce the effects of wear the gears should always be sufficiently lubricated.

10.Packaging:

10.1 Packaging and transportation costs should be kept preferably below 5% of the unit cost.

11.Documentation:

11.1 User to be supplied with an operations and maintenance manual.

12. Shipping

12.1 The product will be dispatched to users by road transportation.

13.Disposal:

13.1 Use sorting methods to separate materials to allow metal parts to be recycled.

Spur Gear design

The initial dimensions that were chosen were based on the GP100 output, this required vast amounts of trial and error until it was believed that the best results had been achieved. To keep the gears at a standard form, a pressure angle of 20° had to be chosen which meant that the minimum required number of teeth for the Pinion would be 17 to prevent any interference. On the other hand, if the number of teeth were too high it would cause conflict as the face width would govern the inion strength and lead to a likelihood of failure. Therefore, choosing a Pinion gear with 20 teeth provided a safe balance and with the required gear ratio of 3.25, this led to the Wheel having 65 teeth. The module calculated by GP100 however was not a standard integer and therefore some of the dimensions from GP100 were dismissed and we decided to source the SS Steel Spur Gears from KHK stock Gears ltd. The same data was inputted and gears with the same number of teeth were chosen, they have the same pressure angle but with a module of 2 as required. The SS2-65 gear will be purchased for the wheel and the SS2-20 refers to the pinion gear. A high carbon-steel was chosen for the Pinion with a large EN to ensure the gear is of good quality and hardness. For the Wheel Pinion a medium carbon steel was chosen, the EN number is still large and at an adequate strength but without compromising too much cost. The manufacturer is well established and provides standardised gears and to improve the quality of gears they incorporate the process of induction hardening.

Gear dimensions:

Gear design specifications/input:

- Centre distance $\leq 90\ mm$
- Gear ratio $= 3.25$(see calculations below)
- Maximum cruise input = $6500\ rpm, 45\ hp$. Required output: $2000\ rpm$
- Maximum power input = $7000\ rpm, 50\ hp$.
- Quality type = ground.
- Pressure angle = 20°

Property	**SS2-20**	**SS2-65**
Number of teeth	20	65
Pitch diameter	40 mm	130 mm
Outside diameter	44 mm	134 mm
Addendum/ dedendum	2 mm/ 1.25 mm	2 mm/1.25 mm
Material	EN 39	EN 36
Tooth depth	4.5 mm	4.5 mm
Mass	0.234 kg	2.264 kg

Gear calculations:

In the following calculations the capital letters refer to wheel parameters and the lower case refers to the pinion.

Gear ratio calculation: Gear ratio: $\frac{output\ speed}{input\ speed} = \frac{6500}{2000} = 3.25$

Circular pitch: $P_c = \frac{\Pi d}{t} = \frac{40\Pi}{20} = 2\Pi \;=\; 6.28\ mm$

Module: m = $\frac{d}{t} = \frac{40}{20} = 2\ mm$

Diametral pitch: $D_p = \frac{t}{d} = \frac{20}{40} = 0.5\ mm = 0.02\ in$

Centre distance: $C_D = 0.5(D + d) = 0.5(134 + \ 44) = 89.00\ mm$

Working depth: h_k = A + a = $4.00\ mm$

Contact ratio: $c_r = 1.60$

Tooth thickness: $t_h = \frac{p_c}{2} = 3.33\ mm$

Face width: f_w = 10.09 x module = $20.18\ mm$

GP100 Data

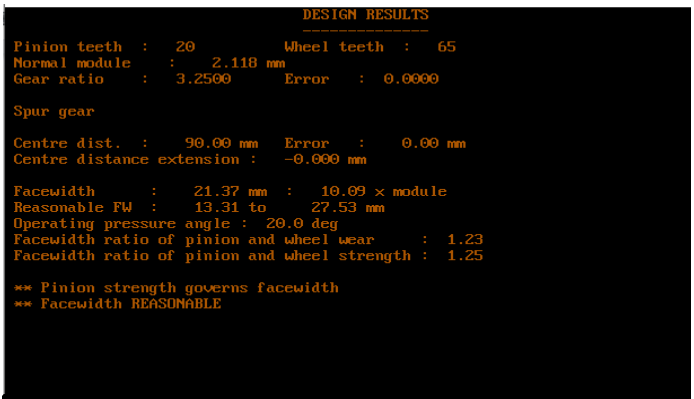


Figure 1: GP100 design results for the pinion and wheel gears.

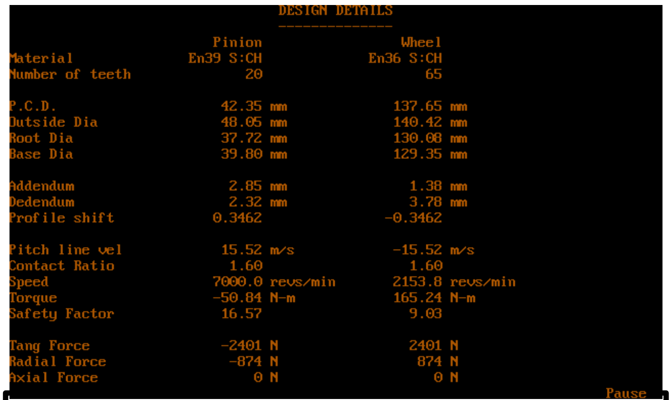


Figure 2: GP100 Design details for the pinion and wheel gears.

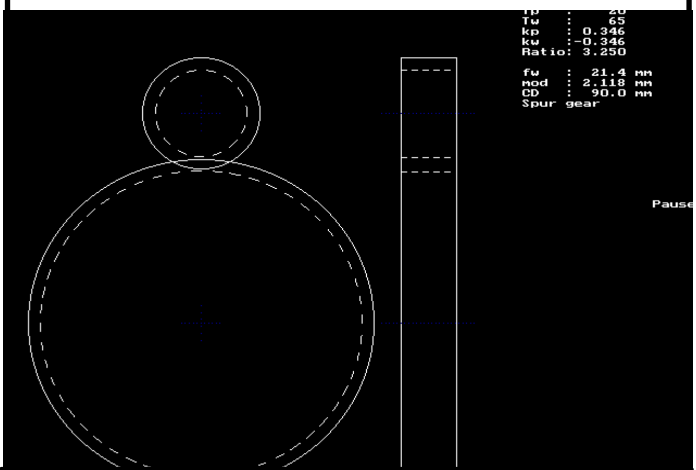


Figure 3: Sketch of the wheel and pinion positioning.

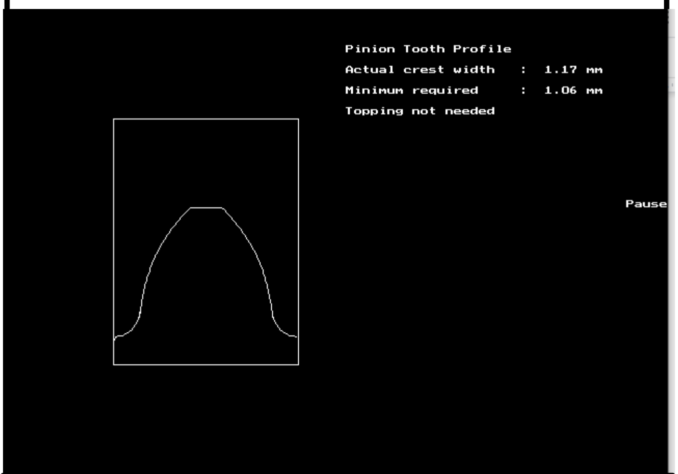


Figure 4: Image of the pinion tooth profile

Bearings Selection

All bearing load and specifications were calculated using the SFK bearing catalogue.

Selected Bearing Specifications:

Bearing Type	Dimensions			Load Ratings		Mass (kg)	Designation (SKF)
	d(mm)	D(mm)	B(mm)	C (kN)	C_0 (kN)		
Deep Grove Ball Bearing	28	58	16	16.8	9.5	0.17	62/28
Deep Grove Ball Bearing	30	90	23	43.6	23.6	0.75	6406
Tapered Roller Bearing	32	58	17	45.1	46.5	0.19	320/32 X

Dynamic Load, C:

Ball Bearing	Tapered Roller Bearing:
$C = P(L_{10})^{\frac{1}{3}}$	$C = P(L_{10})^{0.3}$

Equivalent Bearing Load, P:

$P = XF_r + YF_a$	$X, Y = radial, axial\ load\ factor$ $F_r, F_a = radial, axial\ loads$

Load Life Relationship, L_{10}:

$L_{10} = \frac{60nL_{10h}}{10^6}$ $L_{10} = 4200\ million\ revs$	L_{10h} = 10,000hrs n = 7000rpm

Input Shaft Bearings:

Both ball bearings located on the input shaft experience no axial loads and radial loads are equally distributed between the two.

Input shaft bearings	
Bearing type: Single Row, Deep Grove Ball Bearing	$F_r = 438N$ $F_a = 0N$ $\therefore P = F_r$ $C = 438 \times 4200^{\frac{1}{3}}$ $= 7.07kN$ 7.07kN < 16.8kN ∴Bearing (62/28) is suitable for this application
$C = 16.8kN$ $C_0 = 9.5kN$ $d = 28mm$	
SKF Designation: 62/28	

Output shaft Bearings:
The output shaft has a ball bearing behind the spur gear, as well as a tapered roller bearing behind the propeller to absorb thrust loads. Both bearings experience thrust forces from the propeller.

Bearing behind Spur Gear	
Bearing type: Single Row, Deep Grove Ball Bearing	$F_r = 310N$ $F_a = 880N$ $\frac{F_a}{F_r} = 2.84$ $\frac{F_a}{C_0} = 0.037$ $2.84 > e$ X=0.56, Y=2.25 $\therefore P = 0.56(310) + 2.25(880)$ $P = 2408.8N$ $C = 2408.8(4200)^{\frac{1}{3}}$ $C = 38.86kN$ 38.86kN<43.6kN ∴Bearing (6406) is suitable for this application
$C = 43.6kN$ $C_0 = 23.6kN$ $d = 30mm$	
SKF Designation: 6406	

Bearing behind Propeller	
Bearing type: Single Row, Tapered Roller Bearing	$F_r = 563.87N$ $F_a = 880N$ $\frac{F_a}{F_r} = 1.56$ $2.84 > e$ X=0.4, Y=1.3 $\therefore P = 0.4(563.87) + 1.3(880)$ $P = 1369.55N$ $C = 1369.55(4200)^{0.3}$ $C = 16.73kN$ 16.73kN<45.1kN ∴Bearing (329/32 X) is suitable for this application
$C = 45.1kN$ $C_0 = 46.5kN$ $d = 32mm$	
SKF Designation: 320/32 X	

Mounting Bearings: Mechanical method

1. Ensure that the bearings, shaft and mounting area is clean from dust and other contaminants.
2. Ensure that all parts in contact with the bearings abide to their dimensional and geometrical tolerances.
3. The shaft and housing have to be aligned, and the bearings must be perpendicular to the shaft.
4. Using a bearing fitting tool kit SKF TMFT 36, apply forces to both the inner and outer rings.
5. Push the bearing along the to its required location between the housing and the shaft.

Shaft design

6. Shaft material properties	
Young's Modulus = 200 GPa	BS EN 10277-3 Carbon steel
Yield strength = 430 MPa	Chosen for enhanced machinability and reasonable strength
Tensile strength = 590 MPa	
Fatigue strength at 10^7 cycles = 295 MPa	Low end value from CES EduPack chosen to be conservative
Shear modulus = 77 GPa	
Compressive stress = 430 MPa	
Elongation at failure = 22%	

Freebody diagram:

Figure 5: Freebody diagram of the Pinion shaft

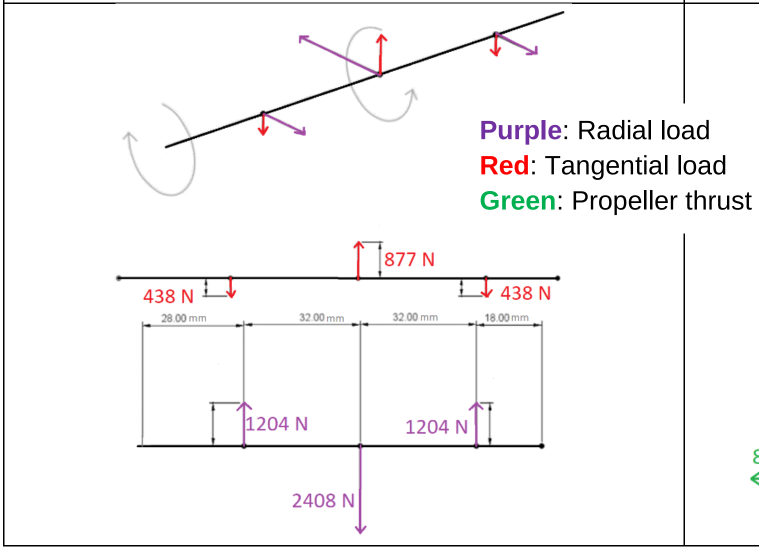
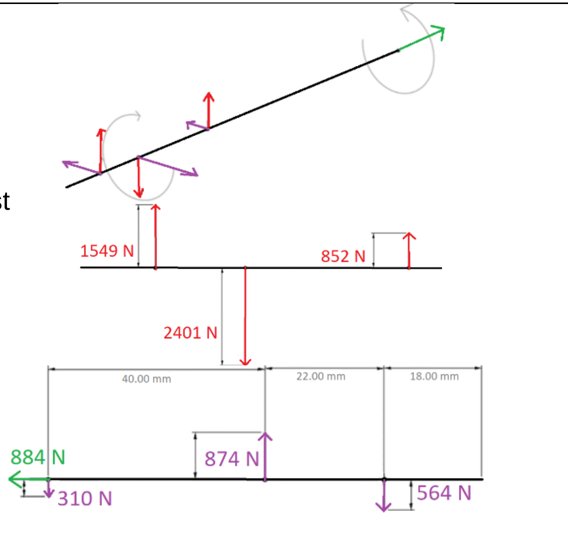
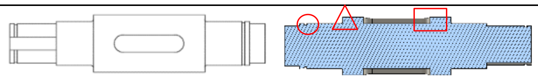


Figure 6: Freebody diagram of the Wheel shaft



Critical sections for failure:



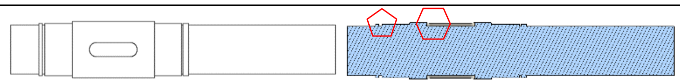


Figure 7: Shaft diagram showing sections of high stress concentration. (left: pinion shaft, right: wheel shaft)

Shaft is most likely to fail at stress concentration points. The points are outlined with shapes. Stress concentrates usually at shoulders and grooves where there is a sudden change in diameter.

○	Groove for bearing retainer ring
△	Shoulder to axially locate bearing
□	Keyway groove, pinion gear
⬠	Shoulder to axially locate bearing
⬡	Groove for keyway, wheel gear

Steady loading minimum diameter:

MSST chosen ductile material, assume safety factor of 3

Minimum shaft diameter= $\left(\frac{32 \cdot f_s}{\pi \cdot S_{yield}} \cdot \sqrt{M^2 + T^2}\right)^{\frac{1}{3}}$

$d = \left(\frac{32 \cdot 3}{\pi \cdot 430 \cdot 10^6}\sqrt{41^2 + 51^2}\right)^{\frac{1}{3}}$ $d = 16.7\ mm \ \approx 17\ mm$	$D = \left(\frac{32 \cdot 3}{\pi \cdot 430 \cdot 10^6}\sqrt{36.2^2 + 165^2}\right)^{\frac{1}{3}}$ $D = 22.9\ mm \ \approx 23\ mm$

This is the minimum diameter from the steady state loads. Maximum Shear Stress Theory is used to calculate the shaft diameter as steel is a ductile material. It assumes when maximum shear stress exceeds shear strength, the material yields. This diameter value will act as a baseline for our design.

Bending Rigidity:

Pinion shaft	Wheel shaft
$\delta_{max} = \frac{PL^3}{48EI}$	$\delta_{MAX} = \frac{Pb \cdot (3L^2 - 4b^2)}{48EI}$
$\delta_{max} = \frac{2563 \cdot 0.064^3}{48 \cdot 200 \cdot 10^9 \cdot \left(\frac{\pi \cdot 0.017^4}{32}\right)}$ $y = 8.54 \times 10^{-3}\ mm$	$\delta_{max} = \frac{2563 \cdot 0.022 \cdot (3 \cdot 0.062^2 - 4 \cdot 0.022^2)}{48 \cdot 200 \cdot 10^9 \cdot \left(\frac{\pi \cdot 0.023^4}{32}\right)}$ $y = 2.05 \times 10^{-3} mm$

Torsional Rigidity $\emptyset = \frac{TL}{GJ}$

Pinion shaft	Wheel shaft	The maximum of twist due to the applied torque is 0.23 degrees, which makes it capable to handle the torque load.
$\emptyset_A = \frac{51 \cdot 0.06}{77 \cdot 10^9 \cdot (\frac{\pi \cdot 0.017^4}{32})}$ $\emptyset_A = 4.85 \times 10^{-3}\ rad$	$\emptyset_B = \frac{165 \cdot 0.150}{77 \cdot 10^9 \cdot (\frac{\pi \cdot 0.03^4}{32})}$ $\emptyset_B = 4.03 \times 10^{-3}\ rad$	

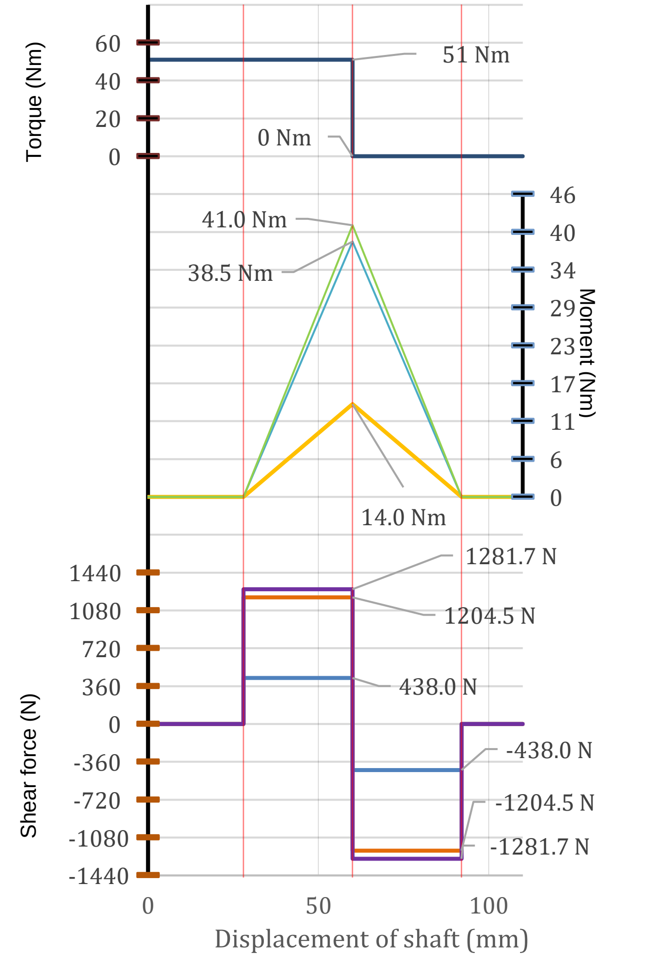
Critical Speed $n_c = \frac{30}{\pi}\sqrt{\frac{g}{\delta_{max}}}$

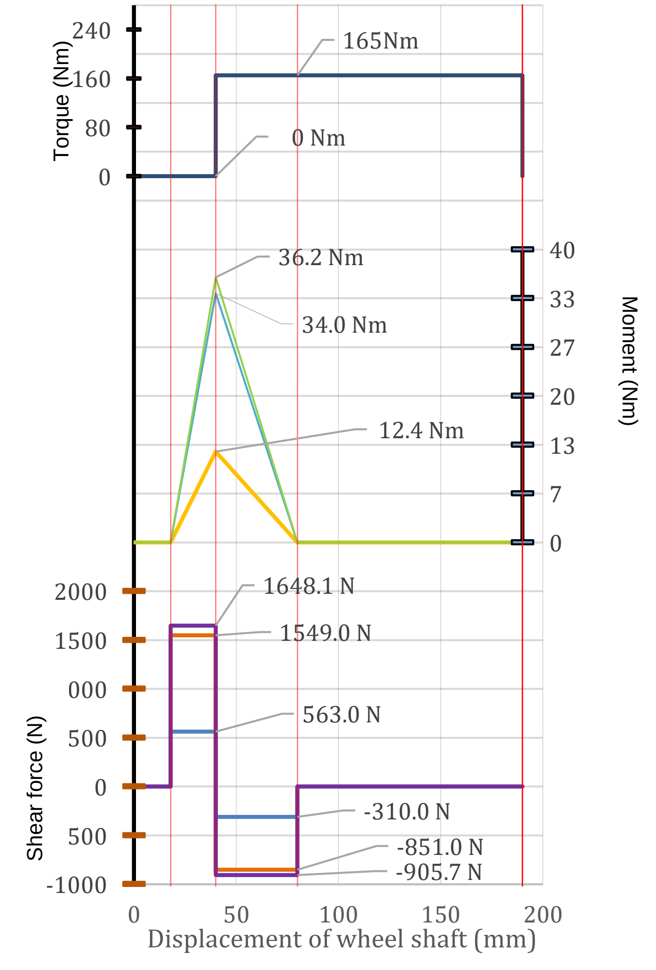
Pinion shaft	Wheel shaft	Pinion shaft's critical speed is close to its operating speed. n_c can be increased with a thicker shaft with lower deflection. Wheel shaft critical speed is suitable
$n_c = \frac{30}{\pi}\sqrt{\frac{9.81}{8.54 \times 10^{-6}}}$ $= 10235\ RPM > 7000\ RPM$	$N_c = \frac{30}{\pi}\sqrt{\frac{9.81}{7.09 \times 10^{-7}}}$ $= 20900\ \text{RPM} > 2000\ \text{RPM}$	

Spline stresses:	**Keyway stresses:**
Analysing spline coupling from engine output shaft to gearbox input shaft (pinion shaft) P.D. = Pitch Diameter (m) <u>Compressive stress</u> (S_c): $S_c = \frac{2T}{n(P.D.)Lh} = \frac{2 \times 51}{6 \cdot 0.038 \cdot 0.004 \cdot 0.002}$ $= 5.59\ MPa < 430\ MPa$ The S_c value is only 1.3% of the material rating <u>Shear stress</u> (S_s): $S_s = \frac{4T}{\pi(P.D.)^2 L} = \frac{4 \times 51}{\pi \cdot 0.038^2 \cdot 0.04} = 1.12\ MPa$ <u>Strain</u> γ_{xymax} (in xy plane) $\gamma_{xymax} = \frac{1.12\ MPa}{77\ GPa} = 0.00146\% < 22\%$ Significantly less strain than the limit before failure <u>Bending stress</u> $S_b = \frac{2T}{(P.D.)^2 L} = \frac{2 \times 51}{0.038^2 \cdot 0.04} = 1.77\ MPa$ 0.4% of yield stress	Key used to locate gear radially n=number of keys/splines Keyway dimensions chosen according to BS46 1958 converted to metric, 8mm width, 3.3mm depth, 30mm length <u>Compressive stress</u> $S_c = \frac{4T}{hLDn} = \frac{4 \times 165}{0.008 \cdot 0.03 \cdot 0.032 \cdot 2}$ = 43MPa < 295MPa 15% of fatigue stress unliikely to fail due to fatigue <u>Shear stress</u> $\tau = \frac{2T}{WLDn} = \frac{2 \cdot 165}{0.008 \cdot 0.022 \cdot 0.032 \cdot 2} =$ 29.3 MPa <u>Strain</u> $\gamma_{max} = \frac{29.3 \times 10^6}{77 \times 10^9} = 0.038\% < 22\%$ Will not fail at this level of strain

Figure 8: Torque, BMD and SFD of Pinion shaft. (see below)

Figure 9: Torque, BMD and SFD of Wheel shaft. (see below)





SFD R | SFD T | SFD TR
BMD R | BMD T | BMD TR
Torque

R= Radial forces
T= Tangential forces
TR= Resultant force from radial and tangential forces

Real endurance Limit ($S_{e'}$) *Modifying Factors*

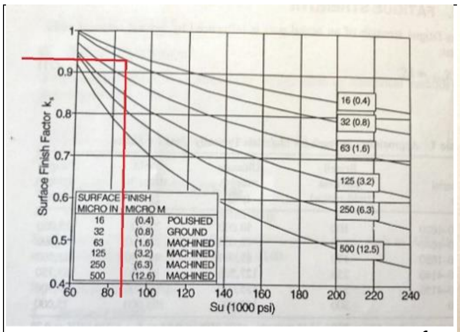


Figure 10: $k_s$surface finish factor[1]

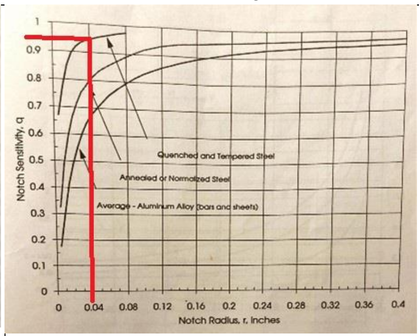


Figure 11: q_n Average notch fatigue sensitivity factor[1]

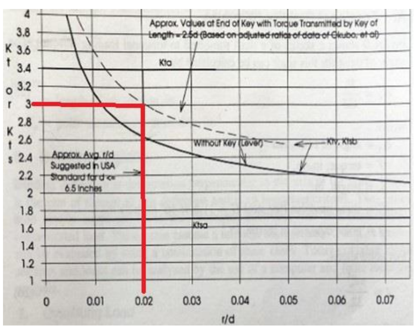


Figure 12: k_t stress concentration factor for torsion of a shaft with keyway[1]

Thrust from propeller

$$F = \frac{P}{v} = \frac{37285}{42.2} = 884N$$

[1] *South, D. and Mancuso, J., 1994. Mechanical Power Transmission Components. New York: M. Dekker, pp.1-16.*

Real endurance strength $S_{e'}= k_f k_s S_e$

Endurance strength $S_e \approx 0.5\,UTS = $ 295 MPa

q_n : Average fatigue sensitivity factor	Theoretical stress concentration factor	$k_s = 0.93$
$q_n = \frac{1}{q-\frac{a}{r}} = \frac{1}{0.95-\frac{0.015}{0.04}}$ q_n =1.038 'a'value given from UTS of steel[1,] refer to fig 11	k_t= 3 see fig 12 Occurs at pinion shaft with keyway under torsion load. Assume 1mm notch radius (0.04 inch)	0.8 microns surface, grounded, refer to fig.10

<u>k_f Stress concentration factor at pinion keyway</u>

$$k_f = \frac{1}{1+q_n(k_t-1)} = \frac{1}{1+1.038(3-1)} = 0.325$$

$$S_{e'} = 0.325 \cdot 0.93 \cdot 295 \cdot 10^6 = 89.2\ MPa$$

Cyclic loading should not exceed 89.2 MPa as fatigue failures may start occurring at that stress level

Other values of k_t	Critical locations
1.58	Pinion shaft shoulder, torsion load r/d = 0.02
2.05	Pinion shaft shoulder, bending load r/d=0.02
1.2	0.65mm radius groove pinion shaft, torsion load
1.3	0.65mm radius groove pinion shaft, bending load
2.18	Bending of pinion shaft with keyway 0.2mm fillet radius

Dynamic loading based on MSST and Soderberg criteria for cyclic loading

$$\text{minimum shaft diameter} = \{\frac{32 f_s}{\pi}[\left(\frac{M}{S_{e'}}\right)^2 + \left(\frac{T}{S_{yield}}\right)^2]^{\frac{1}{2}}\}^{\frac{1}{3}}$$

Pinion shaft	Wheel shaft
$d = \{\frac{32\cdot 3}{\pi}[\left(\frac{41}{89.2\cdot 10^6}\right)^2 + \left(\frac{51}{430\cdot 10^6}\right)^2]^{\frac{1}{2}}\}^{\frac{1}{3}}$ $d = 24.4mm$	$D = \{\frac{32\cdot 3}{\pi}[\left(\frac{36.2}{89.2\cdot 10^6}\right)^2 + \left(\frac{165}{430\cdot 10^6}\right)^2]^{\frac{1}{2}}\}^{\frac{1}{3}}$ $D = 25.7mm$

Soderberg criteria used instead of Goodman and Gerber parabola as it is more conservative. The pinion diameter increased by 7mm to accommodate the geometry of shaft with shoulders and keyways. The same $S_{e'}$ was applied to wheel shaft to be conservative. Safety factor of 3 is assumed. Higher diameter values can be used to recalculate lower values of stresses and deflection.

<u>Summary/limitations</u>

This shaft analysis assumed a safety factor of 3 to calculate minimum dimensions, as this gearbox is required to be highly reliable as per product design specification. In theory, as all stresses on the shaft are under the real endurance limit of 89.2MPa, failure due to fatigue is unlikely, except in cases of manufacture error with pre-existing cracks. However even so, as the estimated endurance limit is very conservative, paired with the ductility of metal, the failure should not be catastrophic.

Thrust is ignored in calculations, as the direction of thrust is orthogonal to both radial and axial and tangential load (from the gears).True resultant force acting on bearings can be calculated by using 3D Pythagoras theorem, however the magnitude is insignificant compare to torsion and bending forces. The point where most stress concentrates is at the pinion keyway with 43 MPa of compressive stress. This can be improved by using a spline or increasing overall diameter. Another way would be surface treatment such as shoot peening to prestress shaft to improve endurance strength. The analysis did not take shaft and gear weight into account. It can be modelled as a uniform distributed beam to be more accurate. The pinion shaft weighs around 360g, and the pinion and wheel gear weighs around 0.2 and 2.3 kg respectively, which would increase shaft bending stress. This however only represents 3-8%% of forces acting on the shafts, which can be covered by the safety factor of 3. When calculating real endurance limit, many factors have not been considered, such as temperature, reliability, size, and load factors. These factors should be considered for a more accurate estimate. This shaft is shown to be reliable under designed load and fulfils the product design specification points **8.1** and **8.3.**

Lubrication: The transmission fluid is doubled as the gearbox lubricant. Due to the layout of the gears, the oil spray method would offer an even distribution of lubricant to all parts(Collins, 2017). Nozzles are installed where the moving parts make contact; gear teeth and bearings. This is coupled with the oil splash method to further lubricate and remove heat. The oil bath collects the residual lubricant and a pump would recycle the lubricant back to the container to maintain oil bath levels. The transmission fluid would then be filtered and reused.

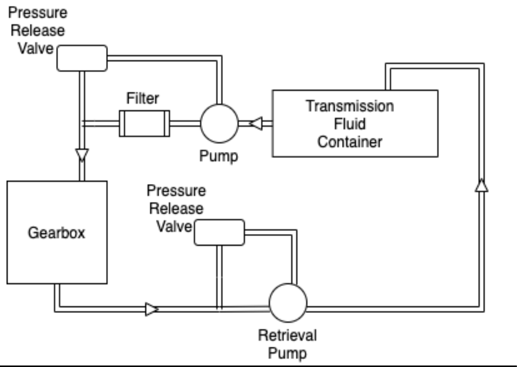


Figure 13: Diagram of Lubrication system

The lubrication system can be shown in *fig.13.*

Baffled Sump Pan: Lubricant in the sump may lean during flight, leading to parts being insufficiently dipped into the bath, or oil pick up issues(BMW E36 325i engine M50 Y M52 sump oil pan baffle kit, n.d.). This could lead to increased oil levels, and result in churning. To avoid this, a sump pan oil baffle kit will be installed. As shown in *fig.14,* the kit allows for single directional flow into the main collection space, where the oil pump can easily pick up and recycle the lubricant. This system would minimize lubrication issues as well as decrease maintenance requirements.

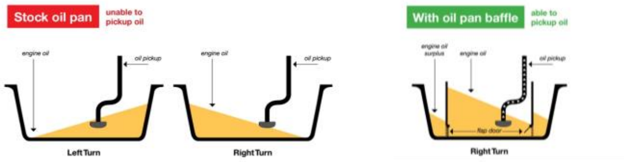


Figure 14: Diagram of Baffled sump pan

Lubricant name	Fabrication	Part Number	Viscosity			Characteristics
			Index	40˚C	100 ˚C	
Ravenol ATF Mercon LV Fluid	Synthetic	1211137	150	29.2	5.9	Ravenol ATF Mercon LV Fluid meets gear oil specifications. It also has a high oxidation stability and protects against abrasion, wear, corrosion, foam formation.

(RAVENOL ATF Mercon LV Fluid, n.d.)

Seals: To prevent lubricant leakage, all openings of the gearbox case have to be appropriately sealed. This includes the input and output shaft openings, as well as two ports for the lubrication system. As the oil ports have no moving parts, a static seal would be sufficient, standard O-ring boss seals can be used on both oil inlet and retrieval lines. Due to the rotational movement between the shaft and the housing bores at shaft openings, radial shaft seals were used instead. SKF's CRWA1 Wave Seal serves its purpose as a low friction seal (CRW1, CRWA1, CRWH1 and CRWHA1, n.d.). Its sealing lip creates a dynamic and static seal against the shaft, while the outer surface forms a static seal against the housing bearing (Radial shaft seals: static and dynamic performance, n.d.). There is also a second seal lip facing outwards to prevent dust particles from entering and damaging the bearings(CRW1, CRWA1, CRWH1 and CRWHA1, n.d.).

SKF CRWA1 Wave Seal Specifications:

Shaft Diameter (mm)	Housing Bore (mm)	Seal width (mm)	Lip Material	Seal Design	Designation
20	35	7	Nitrile Rubber (NBR)	HMSA10	20x35x7HMSA10 RG
30	45	7	Nitrile Rubber (NBR)	HMSA10	30x45x7HMSA10 RG

2020. *Billy Mcnally's Pump And Seal Manual*. McNally Institute.
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Maintenance Schedule

Part		Action	Initial Inspection	Repeat every* (flight hrs)
Gearbox	Gearbox	Listen for noise/vibration and leaks	After first use	Each operation
	Gearbox	Alignment of gearbox to engine	After first 1,000 flight hrs	20,000 hrs
	Gearbox	Dismantle and inspect all parts of gearbox		40,000 hrs
	Bolts	Check and tighten all bolt connections	After first 1,000 flight hrs	10,000 hrs
	Case	Visual check for cracks or discolorations	After first use	Weekly
Gears	Gears	Visual inspection for any signs of pitting or wear	After first 1,000 flight hrs	10,000 hrs
	Gears	Check for misalignment of gear teeth (blue ink test)	After first 1,000 flight hrs	20,000 hrs
Shaft	Shaft	Check for shaft alignment. Inspect for cracks, bent shaft, and wear	After first 1,000 flight hrs	10,000 hrs
Bearings	Rollers, Raceway, Cage	Visual inspection for flaking, scoring, cracks, indents, wear, discoloration, rust	After first 1,000 flight hrs	10,000 hrs
Lubrication System	All system parts	Inspect for leakages and sludging. Clean system.	After first 1,000 flight hrs	20,000 hrs
	Oil Filter	Clean and replace filter	After first 1,000 flight hrs	40,000 hrs
	Transmission Oil/Lubricant	Check for oxidation, discolouration and viscosity of fluid. Ensure proper fluid levels.	Before first use	100 hrs
	Pump	Check pressure of pump	Before first use	100 hrs
	Seals	Check for leakages. Replace when necessary	After first use	40,000 hrs

Feedback

Feedback	Development made/resolutions formed
Using Non-standardised gears would cause complications in manufacturing when using a non-integer module or non-standard pressure angle.	We found a manufacturing company to source gears from that are at a standardised size and that suit our design criteria at the same time.
To counteract the thrust force, it was recommended to use helical gears or use thrust bearings that would absorb this.	In our design we decided to add a thrust bearing to absorb this thrust force.
The epicyclic formation leads to more complications as it involves more components, greater mass and involves greater costs.	We decided to re-evaluate our original design and choose a 2-step reduction, as the less-complicated design will be more sophisticated and reliable.
The face width is too large, and this governs the pinion strength.	We chose a design where the amount of pinion teeth was not too large which led to a reduction in the face width, we also improved the materials quality by using a high-carbon steel with a greater EN value.
Consider a way to recycle oil throughout the transmission.	We have incorporated a baffled sump meaning oil can be scavenged.
Make the report more concise	We ensured to not repeat equations when used more than once so we just reference them if they have already been defined.