Aero-Engine Speed Reducer

Mechanical Design B (04-22964)

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Group No: 26

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Product Design Specification

Table 1: Product Design Specification

N.o.	Parameter	Spec
1.1	Operation	Speed reducer transmission to reduce Wankel engine speed from 6500 rpm (cruise) and 7000 rpm (max power) to 2000 rpm at the propeller.
1.2		Must handle maximum power output of 50 horsepower, whilst maintaining 45 horsepower at cruise settings.
1.3		The system must accommodate two operational modes: maximum cruise and maximum power, adjusting to varying conditions during flight.
2.1	Dimensions	Gearbox output shaft to extend no more than 90 mm from the engine axis.
2.2		The centre distance between gears must be 90 mm or less, adhering to the limitations.
3.1	Safety	*1* Should be designed to prevent failure during engine power surges or extreme conditions.
3.2		Ensure the system can withstand vibrational stress.
4.1	Aesthetics and Finish	Should be designed to fit the engine casing and the propeller seamlessly.
4.2		Smooth surface finish between components to reduce friction and minimise wear.
4.3		The external surface should be corrosion-resistant to be able to withstand corrosion and environmental effects.
5.1	Materials	Gears should be high quality for a long-life expectancy.
5.2		Gears material should differentiate depending on expected life expectancy/size.
5.3		Shaft material should be tough and fatigue resistant.
5.4		High-performance bearings and seals for durability and long service life.
6.1	Power Considerations	Reduction ratio of 3.25:1 to reduce propeller speed to 2000 rpm.
7.1	Maintenance	Easy access to lubricant ports.
7.2		Regular inspections should be conducted on critical components for wear and tear.
7.3		Key Components should be easily replaceable, minimising downtime.
8.1	Life Expectancy	It must be designed for a minimum of 2000 hours of run time and maximum cruise power.
8.2		Shafts and gears should be designed for infinite fatigue life under normal operating conditions
9.1	Manufacture	Production volume of 200 units per annum.
9.2		Should be designed in mind of manufacture and assembly.

Gear Design

To calculate the gear ratio, we used the specification stating we had to go from the engine speed of 6500 to 2000 rpm at the propeller. A gear ratio was calculated using Equation 1:

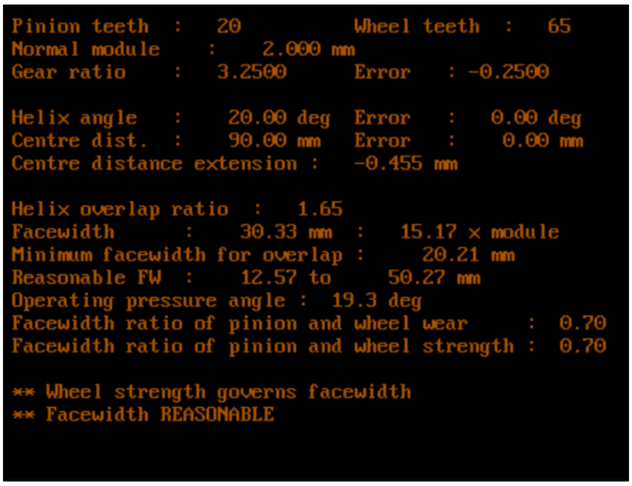
$$Gear\ Ratio = \frac{Pinion\ Speed}{Wheel\ Speed} = \frac{6500}{2000} = 3.25\ [1]$$

The gear was designed using the GP100 software; the inputted data is shown in Table 1. The outputs of these parameters are show in Figure 1 and 2.

Table 2: GP100 Inputted data

Parameter	Data Inputted	Justification
Gear Ratio	3.5+-0.25	This allowed us to get the max power of 7000 rpm with a tolerance of 0.25, and the 6500 rpm cruise speed.
Minimum Pinion Teeth	20	This was through trial and error after choosing the other parameters. This should be over 17 to prevent interference.
Centre Distance	90+0-5	The maximum centre distance, as specified in the brief.
Helix Angle	20+-0.5	Adding a helix angle was chosen due to the benefits it has over a spur gear. It has a longer life and works smoother compared to spur gears. 20 degrees was chosen after some trial and error as it's a good balance between axial and radial load distribution.
Power	37.99	The power the engine provides is 50 bhp, which is then converted into kWh.
Pinion Speed	6500	This is the cruise speed that the engine will be going the majority of the time.
Total Life	2000 hours	As specified in the brief.
Ground vs Cut	Ground	Ground provides a better surface finish allowing less friction between the gears, allowing a longer life.
Pinion: Wheel Material	En34 and En24 SH	The pinion material had to be stronger than the wheel, and this was also through trial and error until we got a reasonable face width.
Preferred Tooth Size	1st Preferred	Aligns with manufacturing standards and machining tools would be easy to find to manufacture these gears.

Figures 1 and 2 show the outputted results from GP100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Pinion teeth   :   20           Wheel teeth   :   65
Normal module     :    2.000 mm
Gear ratio     :   3.2500       Error    : -0.2500

Helix angle    :    20.00 deg   Error    :    0.00 deg
Centre dist.   :    90.00 mm    Error    :     0.00 mm
Centre distance extension :     -0.455 mm

Helix overlap ratio   :    1.65
Facewidth        :    30.33 mm   :    15.17 x module
Minimum facewidth for overlap :     20.21 mm
Reasonable FW   :    12.57 to      50.27 mm
Operating pressure angle :  19.3 deg
Facewidth ratio of pinion and wheel wear      :  0.70
Facewidth ratio of pinion and wheel strength :  0.70

** Wheel strength governs facewidth
** Facewidth REASONABLE
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Figure 1: Design Results

	Pinion	Wheel
Material	En34 S:CH	En24 AS:SH
Number of teeth	20	65
P.C.D.	42.57 mm	138.34 mm
Outside Dia	47.74 mm	140.26 mm
Root Dia	37.98 mm	130.50 mm
Base Dia	39.69 mm	129.00 mm
Addendum	2.59 mm	0.96 mm
Dedendum	2.29 mm	3.92 mm
Profile shift	0.2926	-0.5202
Pitch line vel	14.49 m/s	-14.49 m/s
Contact Ratio	1.69	1.69
Speed	6500.0 revs/min	2000.0 revs/min
Torque	-55.31 N-m	179.74 N-m
Safety Factor	12.63	13.00
Tang Force	-2598 N	2598 N
Radial Force	-968 N	968 N
Axial Force	-946 N	946 N

Figure 2: Design Details

The pressure angle works as it is very close to 19.3 degrees. The facewidth is reasonable, and the gear ratio and speeds are perfect to meet to the specification.

Bearing Specifications

The dynamic bearing load P is needed as it is the load which controls the contact stresses. It is calculated using Equation 2.

$$P = XF_r + YF_a \ [2]$$

Where F_r is the actual radial bearing load (N), F_a is the actual axial bearing load (N), and X and Y are the respective radial and axial load factors for the bearing.

The expected bearing life is needed to determine its running time in hours. The following equation is used to obtain the bearing life:

$$L_{10} = \left(\frac{C}{P}\right)^n \ [3]$$

Where L_{10} is the number of revolutions (in millions) that a group of identical bearings will last before 10% of bearings fail, while at a constant speed. C represents the basic bearing load rating, and P is the applied load on the bearing. The value of n is assumed to be 3 for ball bearings and 10/3 for roller bearings.

The bearing's lifespan is calculated in hours using Equation 4:

$$L_{10h} = L_{10} \times \left(\frac{10^6}{60N}\right) \ [4]$$

Where L_{10h} is the lifespan in hours and N is the rotational speed in rpm.

Tapered roller bearings were chosen for the input shaft as they can support high radial and axial loads which is shown in the GP100 results in Figure 2. For the output shaft, deep groove ball bearings were chosen as they can mainly withstand radial loads, and the resultant axial force is very small compared to the input shaft, which has too big of an axial force for the ball bearings to handle. If they were used, they would fail very quickly due to the shear stress. Although deep groove ball bearings are cost effective, tapered roller bearings are necessary.

The Schaeffler catalogue was used to source the bearings. Product information and the calculated results are included in Table 2.

Table 3: Bearing Specifications

Type of Bearing	Tapered Roller		Deep Groove Ball	
Location	Input 1	Input 2	Output 1	Output 2
Inner Diameter d (mm)	35	25	30	40
Outer Diameter D (mm)	72	52	55	68
Width B (mm)	18.25	16.25	13	15
Dynamic Axial Load Factor Y	1.6	1.6	0	0
Dynamic Radial Load Factor X	0.56	0.56	1	1
Radial Dynamic Load Rating C (N)	54,000	32,500	13,500	17,800
Radial Bearing Load F_r (N)	81.288	886.712	563.437	404.563
Axial Bearing Load F_a (N)	473	473	31	31
Equivalent Bearing Load P (N)	802.321	1253.359	563.437	404.563
Lifespan L_{10h} (Hours)	1.908×10^8	7.939×10^6	1.146×10^5	7.010×10^5
Product	30207-A	30205-A-P5	6006-RSR	6008-RSR

After calculations, all the bearings surpass the minimum requirement of 2000 hours.

Shaft Design

Shear Force and Bending Moments

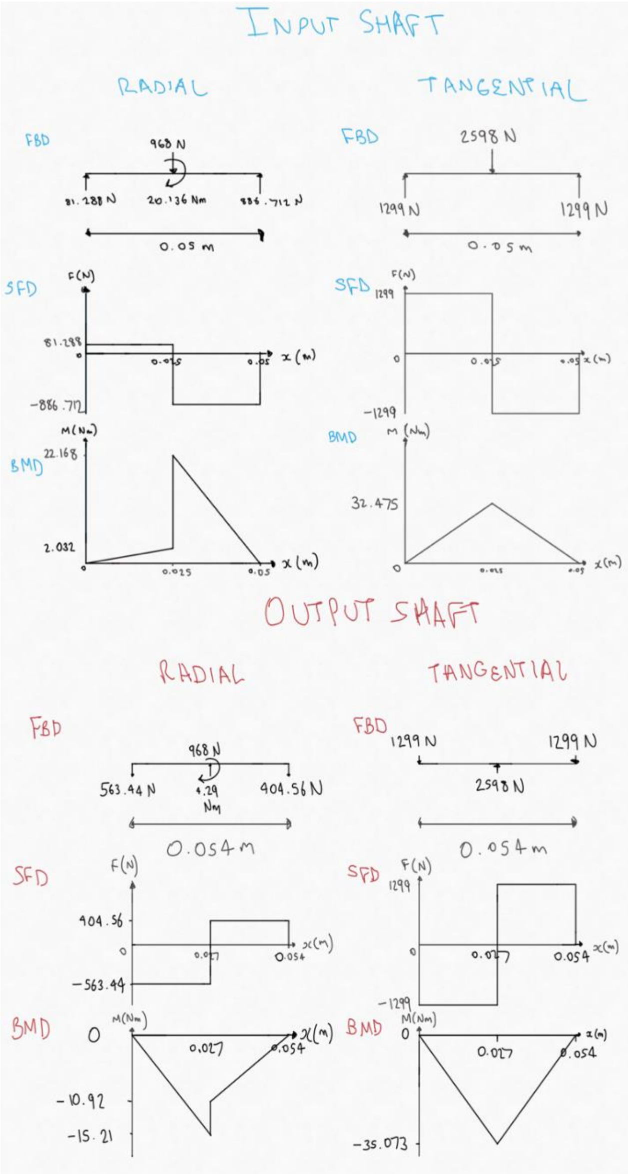


Figure 3: Free Body, Shear Force and Bending Moment diagrams of the shafts

The values for the radial and tangential forces of the gears were acquired from GP100. Shaft lengths of 0.05 m and 0.054 m were chosen for the input and output respectively to allow the other components to fit and adhere with the specification. The axial moment on the radial free body diagram is calculated using Equation 5. Figure 3 shows the shear force and bending moment diagrams. The resultant force and moment are calculated for each section using Equations 6 and 7, and the results are shown in Table 4.

$$M_a = F_{a,pinion} \times Pitch\ Circle\ Radius\ [5]$$

$$F_{res} = \sqrt{(F_t)^2 + (F_r)^2}\ [6]$$

$$M_{res} = \sqrt{(M_t)^2 + (M_r)^2}[7]$$

Table 4: Resultant shear force and bending moments

	Input Shaft	Output Shaft
Resultant force from Bearing 1 to Gear (N)	1301.54	1415.93
Resultant force from Gear to Bearing 2 (N)	1572.79	1360.54
Resultant moment from bearing 1 to gear (Nm)	39.32	36.73
Resultant moment from Gear to Bearing 2 (Nm)	0	0

AISI 4140 was chosen for the input shaft material as it is tough and strong, as well as possessing high wear resistance, which is necessary as the pinion will undergo more revolutions than the wheel. For the output shaft, AISI 4340 was selected as its tensile strength and toughness are superior to AISI 4140 which is needed to withstand the higher loads exerted by the wheel. The properties are outlined in Table 5.

Table 5: Mechanical properties of AISI 4140 and AISI 4340

Symbol	Mechanical Properties	Input Shaft (AISI 4140)	Output Shaft (AISI 4340)
E	Young's Modulus (GPa)	200	200
S_u	Ultimate Tensile Strength (MPa)	655	745
S_y	Yield Strength (MPa)	415	470
G	Shear Modulus (GPa)	80	80

Twisting Loads

Shafts can experience too much deflection or fail if the stresses exceed the material's mechanical properties. Evaluating these stresses is crucial to assess whether the shaft can safely operate under these conditions. The factors in Table 6 were used to calculate the bending stress, axial stress and torsional shear stress using Equations 8, 9 and 10 respectively.

$$\sigma_b = \frac{My}{I}\ [8] \qquad \sigma_a = \frac{4F_{a,gear}}{\pi d^2}\ [9] \qquad \tau = \frac{Ty}{J}\ [10]$$

Table 6: Factors for calculating different types of stresses

Symbol	Factor	Input	Output	Justification
T	Torque (Nm)	-55.31	179.74	Values are taken from GP100.
Y	Shaft radius (m)	0.0125	0.015	These values were adjusted through trial and error to obtain safety factors that lie between 5 and 6, which is suitable in aerospace applications.
M	Resultant Bending Moment (Nm)	39.32	36.73	Taken from the resultant of the radial and tangential bending moments.
I	Moment of Inertia (m^4)	1.92×10^{-8}	1.26×10^{-7}	Using Equation $I = \pi d^4/64$.
J	Polar Moment of Inertia (m^4)	3.83×10^{-8}	2.51×10^{-7}	Using Equation $I = \pi d^4/32$.
τ	Torsional Shear Stress (Pa)	-1.80×10^{7}	1.43×10^{7}	Using Equation 10.
σ_b	Bending Stress (Pa)	2.56×10^{7}	5.85×10^{6}	Using Equation 8.
σ_a	Axial Stress (Pa)	1.93×10^{6}	7.53×10^{5}	Using Equation 9.
σ_c	Combined Stress (Pa)	2.76×10^{7}	6.60×10^{6}	The sum of the bending and axial stress.

Fatigue Analysis

The revised endurance strength of each shaft is outlined in Table 7, where various factors were chosen or calculated. The revised endurance strength is calculated using Equation 11.

Table 7: Factors for determining the revised endurance strength

Symbol	Factor	Input Shaft	Output Shaft	Justification
K_a	Surface Finish	0.9	0.9	Obtained from Figure 5, assuming the finish is ground.
K_b	Size	0.87	0.83	Estimated using $1.189d^{-0.097}$ for $8<d\leq250$mm.
K_c	Reliability	0.81	0.81	Estimated using table where $K_c=1-0.08Z_r$ and 99% reliability.
K_d	Temperature	1	1	Value is 1 for T≤450°C.
K_e	Stress Concentration Modifying	0.61	0.63	$K_e=1/K_f$ where it is assumed that $K_f=K_t$. K_t is obtained from Figure 6.
K_m	Miscellaneous effects	1	1	There are no miscellaneous effects.
S_e	Endurance Strength (MPa)	328	373	$S_e=0.5S_u$ where S_u is the Ultimate Tensile Strength of AISI 4140 (Input) and AISI 4340 (Output).
S'_e	Revised Endurance Strength (MPa)	127	142	Using Equation 11.

$$Revised\ Endurance\ Strength\ (S'_e) = K_aK_bK_cK_dK_eK_mS_e\ [11]$$

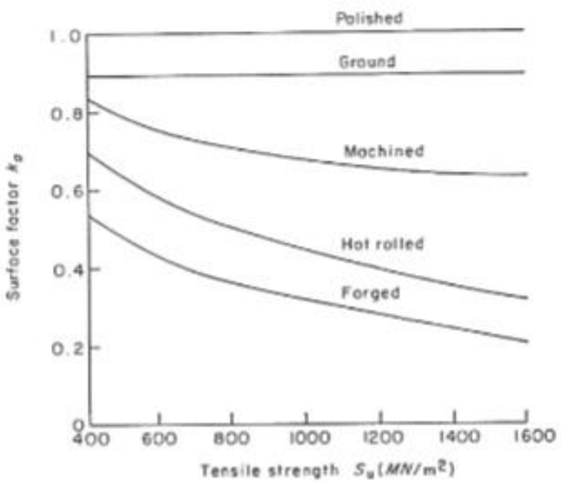


Figure 4: Graph for surface finish factor

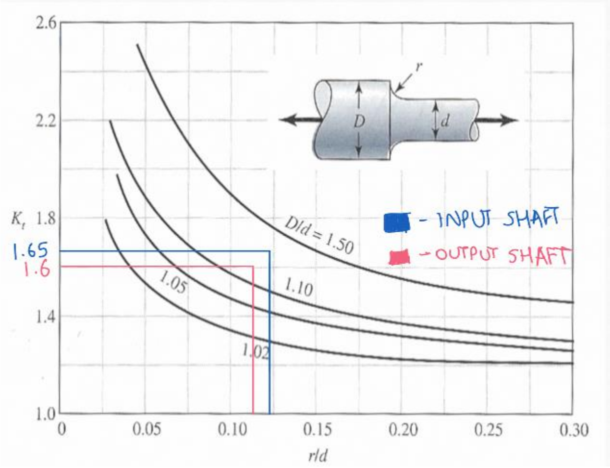


Figure 5: Graph for stress concentration factor

Since the shafts are subjected to combined loading, Equation 12 is used to calculate the safety factor resulting in 4.26 for the input shaft and 5.56 for the output shaft, which are reasonable values.

$$Safety\ Factor\ from\ Combined\ Loading\ (f_s) = \frac{S_{sy}}{\left[\frac{1}{4}\left[\left(\frac{S_y}{S_{e'}}\right)\sigma_r\right]^2 + [\tau_m]^2\right]^{\frac{1}{2}}}\ [12]$$

Deflection and Critical Speed

Calculating the total deflection and critical speed are essential to prevent the misalignment of components and increased wear, using Equations 15 and 16. Results are shown in Table 8.

$$Horizontal\ Deflection\ (\delta_H) = \frac{F_{t,pinion} \times L^3}{48EI}\ [13] \qquad Vertical\ Deflection\ (\delta_V) = \frac{F_{r,pinion} \times L^3}{48EI}\ [14]$$

$$Total\ Deflection\ (\delta_{Tot}) = \sqrt{{\delta_H}^2 + {\delta_V}^2}\ [15] \qquad Critical\ Speed\ (N_c) = \frac{\sqrt{\frac{48EI}{mL^3}}}{2\pi} \times 60\ [16]$$

Table 8: Total Deflection and Critical Speed of each shaft

	Horizontal Deflection (m)	Vertical Deflection (m)	Total Deflection (m)	Critical Speed (rpm)
Input Shaft	1.76 x 10^{-6}	6.57 x 10^{-7}	1.88 x 10^{-6}	9.51 x 10^{5}
Output Shaft	1.07 x 10^{-6}	3.99 x 10^{-7}	1.14 x 10^{-6}	2.64 x 10^{5}

Torque Transmission Couplings

Table 9: Coupling Specification

Type of Coupling	Key-way
Product Name	High-Grip Clamping Shaft Couplings
Brand	McMaster-Carr
Function	Carries the torque from the engine into the input shaft.
Material	Stainless Steel
Key Features	This coupling is particularly useful in high torque scenarios.
Is a spacer required?	No
Cost per piece (£)	188.36
Method to attach the coupling to the shaft	Slide it on, use bolts to secure it.
Inside Diameter (mm)	40
Outside Diameter (mm)	60
Length (mm)	90
Max Speed (rpm)	Unrated
Max Torque (Nm)	54.75 (Equation from McMaster-Carr based on horsepower)

- The maximum speed is not rated, and while a standardized part with all parameters would be ideal, the decision was made to exclude the unrated max speed due to limited part availability. Based on similar parts from McMaster-Carr, an estimated max speed of 6000-7000 rpm is a reasonable assumption.
- Initially, we considered using rubber flexible couplings for their ability to correct misalignment, which is beneficial in high-vibration machines. However, we opted for a clamped keyway coupling instead, as it offers better stability, durability, and is a pre-made part, making sourcing easier during manufacturing.
- The shaft features a spline, where the key is directly attached to the shaft itself. As a result, there are no shear stress calculations needed for the key, since it is part of the shaft rather than being a separate component.

Input Shaft Coupling Calculations:

There are only calculations for the input shaft as there is only one coupling. There are no couplings on the output shaft

Table 10: Input Shaft Coupling Final Values

Factor	Symbol	Value	Parameters	Unit
Failure due to Shear	F	4424.8	Where T=Max Torque	N
Shear stress	τ_{design}	7.87×10^6	-	Pa
Key Area	A_s	1.2×10^{-4}	Where the key width is 0.25 times the shaft diameter	m^2
Bearing Area of the Key	A_b	2.81×10^{-4}	-	m^2
Bearing Stress	σ_{design}	1.57×10^7	-	Pa

Calculations:

$F = \frac{2T}{D}$ [17] $\tau_{design} = \frac{F}{A_s} = \frac{2T}{D\omega L}$, $A_s = \omega L$ [18] $To\ avoid\ failure\ due\ to\ shear:\ \tau_{design} \leq \frac{S_{sy}}{f_s}$ [19]

$$7.87 \times 10^6 \leq 4.47 \times 10^7 \;\therefore\; True$$

$$\sigma_{design} = \frac{F}{A_b} = \frac{4T}{DHL}, \; A_b = \frac{LH}{2} \; [20]$$

The inequality above ensures that the shear stresses on the couplings won't cause any cracks or fatigue damage on the part.

$$Bearing\ load\ failures\ can\ be\ avoided\ if\colon\ \sigma_{design} \leq \frac{S_y}{f_s} \; [21]$$

$$1.57 \times 10^7\ Pa \;\leq\; 8.94 \times 10^7 \;\therefore\; True$$

ISO Straight Sided Spline Capacity Calculations:

These calculations are used to calculate the compressive and shear stresses of the spline on the input shaft in design 1. These are important for understanding how the shaft and coupling will interact and how they will react to forces put on them.

$$\sigma_c = \frac{T \times K_s}{n \times d \times L_e \times r} \; [22] \; K_s = K_a \times \frac{K_d}{K_f} \; [23] \; \tau = \frac{16 \times T \times K_s}{(\pi \times {D_i}^3)} \; [24]$$

No. of Cycles is based on 2000 total flight hours, which we approximated to be 2 hours per flight, therefore 1000 stop/start cycles.

Table 11: Spline Calculations

Factor	Symbol	Value	Parameters	Unit
Compressive Stress	σ_c	2.44×10^7	-	Pa
Service Factor	K_s	1.11	For a fixed/guided spline.	-
Design Factor	K_d	1	For a fixed/ close fit loaded shaft.	-
Spline Application Factor	K_a	2	For a medium shock power source and uniform load.	-
Fatigue Life Factors for Splines	K_f	1.8	At ~ 1000 stop/start cycles.	-
No. of Teeth	n	1	-	-
No. of Cycles	-	~1000	At 2000 flight hours.	-
Wear Life Factor for Spline	K_w	~6.152	At 6500 rpm.	-
Spline Distribution Factor	K_m	~1	At 0.001 mm/mm misalignment and a 40 mm facewidth.	-
Shear Stress in the Shaft	τ	4.836×10^6	Where T = Max Torque	Pa

Sealing Specification

Table 12 shows the seals that we used in our design to stop oil leaking out of the transmission. Bore-sealing seals worked best because…

Table 12: Sealing Specification

Seal Type	Bore-Sealing Spring-Loaded Rotary Shaft Seals with Wiper Lip	Bore-Sealing Spring-Loaded Rotary Shaft Seals with Wiper Lip	Bore-Sealing Spring-Loaded Rotary Shaft Seals with Wiper Lip	Bore-Sealing Spring-Loaded Rotary Shaft Seals with Wiper Lip	Gasket
Location	1	2	3	4	Between Casings
Manufacturer	McMaster-Carr				Custom
Motion	Rotating				Still
Max Speed (rpm)	7600	10,500	8,700	6,600	17,000
Type of Use	Transmission Shaft				Casing
Material	Buna-N Rubber (Nitrile Rubber) with Steel Case and Spring				Fluoropolymer
Function	Keeps the oil on the shaft in the transmission and stops leaks.				Seal casing halves
Retaining Ring Required?	Yes				No
Inner Diameter (mm)	35	25	30	40	-
Temperature Range (°C)	-40 to 210				-40 to 210
Hardness	Durometer 75A (Hard)				-
Specifications Met	DIN 3760, ISO 6194				-
Cost (per piece in £)	4.89	5.04	4.89	6.37	-

Lubrication Specification

Table 13: Lubrication Specification

Factor	Unit	Value
Oil Type	-	Mineral Oil: Paraffinic
Oil Viscosity Range	Pa s	0.075 to 0.2
Temperature Range	°C	70 to 90
Viscosity Index (VI)	-	80 to 100
Pour Point	°C	-40 to -10
Flash Point	°C	< 250
Oil Density	$\frac{kg}{m^3}$	~ 877.36
Additives	-	Dispersants, Extreme Pressure (EP), Anti-oxidants, Foam inhibitors, Pour point depressants.
Lubricant to Additive Ratio	-	90:10
Lubrication Method	-	Oil is spread using splash and a trough of oil at the bottom of the casing, due to a higher pitch line velocity.
Pitch Line Velocity (ms^{-1})	ms^{-1}	Pinion: 14.49/ Wheel: -14.49
Baffle Required? (Velocity>12.5 ms−1)	-	Yes
Products	-	Rye Oil: Industrial Gear Oil 150
Cost	£ per Litre	Rye oil: 2.82

Pitch line velocity calculations are required to find the viscosity of the oil and whether a baffle is needed:

$$Pitch\ Line\ Velocity = \frac{\pi \times D \times N}{60},\ where\ D = Pitch\ Diameter\ and\ N = Gear\ speed\ in\ rpm\ [25]$$

Table 14: Pitch Line Velocity/Diameter

Factor	Symbol	Shaft 1	Shaft 2	Parameters	Unit
Pitch Line Velocity	PLV	14.49	47.08	-	ms^{-1}
Pitch Circle Diameter	D	42.57	138.34	-	mm

The value of pitch line velocity for the second shaft is -14.49 ms^{-1} on GP 100, however when I calculate it using the equation I get 47.08 ms^{-1}. This is because the PLV for the input shaft has the same value as the output shaft, because they are in contact but just in the opposite direction.

Maintenance Schedules

In the case of a Wankel engine, regular maintenance is required to maintain engine efficiency, performance and safety. Thermal management as suggested by research done by Wu, W., Lin, Y., and Chow, L (2014) is one of the main maintenance considerations when considering Wankel rotary engines due to the thermal stress concentrations, this can be reduced however using heat pipes to spread the heat out evenly. The key factors that need to be regularly checked are shown in Table 12.

Table 15: Maintenance Schedule

System Checks	Schedule	Comments
Visual Check	Every Flight	Check for visual damage, any leakages, and make sure everything's in operating order.
Thermal Management System	Every 100 to 200 flight hours	Check gearbox operating temperatures.
Seals and Bearings	Every 50 to 100 flight hours	Check for wear, leakages and misalignment.
Oil and Lubrication System	Every 50 to 100 flight hours	Check for oil discolouration or particulates.
Cooling System and Heat Pipe System	Every 50 to 100 flight hours	Check pipes for leakages and wear.
Gear System	Every 100 to 200 flight hours	Check for wear, fractures and misalignment.
Gearbox Casing	Every 50 to 100 flight hours	Check for wear, debris and fractures.
Gearbox Replacement	After 2000 flight hours	

Bill of Materials

Table 16: Bill of Materials for the Initial and Final Design.

Initial Design				Final Design			
Component	**Qty**	**Part No.**	**Material**	**Component**	**Qty**	**Part No.**	**Material**
Case – Engine Half	1	1	Cast Iron	Case – Engine Half	1	1	Cast Iron
Case – Pump Half	1	2	Cast Iron	Case – Pump Half	1	2	Cast Iron
Case – Gasket Seal	1	3	Rubber	Case – Gasket Seal	1	3	Rubber
Lip Seal 1	1	4	Stainless Steel/Nitrile Rubber	Lip Seal 1	1	4	Stainless Steel/Nitrile Rubber
Lip Seal 2	1	5	Stainless Steel/Nitrile Rubber	Lip Seal 2	1	5	Stainless Steel/Nitrile Rubber
Lip Seal 3	1	6	Stainless Steel/Nitrile Rubber	Casing Bolt	17	6	Stainless Steel
Lip Seal 4	1	7	Stainless Steel/Nitrile Rubber	Pump Casing Nut	9	7	Stainless Steel
Pump Casing Bolt	9	8	Stainless Steel	Bearing 1	1	8	Stainless Steel
Pump Casing Nut	9	9	Stainless Steel	Bearing 2	1	9	Stainless Steel
Engine Casing Bolt	8	10	Stainless Steel	Bearing 3	1	10	Stainless Steel
Bearing 1	1	11	Stainless Steel	Bearing 4	1	11	Stainless Steel
Bearing 2	1	12	Stainless Steel	Pinion	1	12	EN34
Bearing 3	1	13	Stainless Steel	Wheel	1	13	EN24
Bearing 4	1	14	Stainless Steel	Input Shaft	1	14	AISI 4140
Pinion	1	15	EN34	Output Shaft	1	15	AISI 4340
Wheel	1	16	EN24	Oil Plug	2	16	Stainless Steel
Input Shaft	1	17	AISI 4140	Oil Plug Washer	2	17	Stainless Steel
Output Shaft	1	18	AISI 4340	Engine Casing Washer	17	18	Stainless Steel
Keyway Coupling	1	19	Stainless Steel	Spline Coupling	1	19	Stainless Steel
Coupling Bolt	4	20	Stainless Steel	-	-	-	-
Oil Plug	2	21	Stainless Steel	-	-	-	-
Oil Plug Washer	2	22	Stainless Steel	-	-	-	-
Engine Casing Washer	17	23	Stainless Steel	-	-	-	-

Exploded View

Figures 7 and 8 show the exploded views, which show where each component is in the design. The numbers in the balloons correspond to the respective part numbers from Table 12.

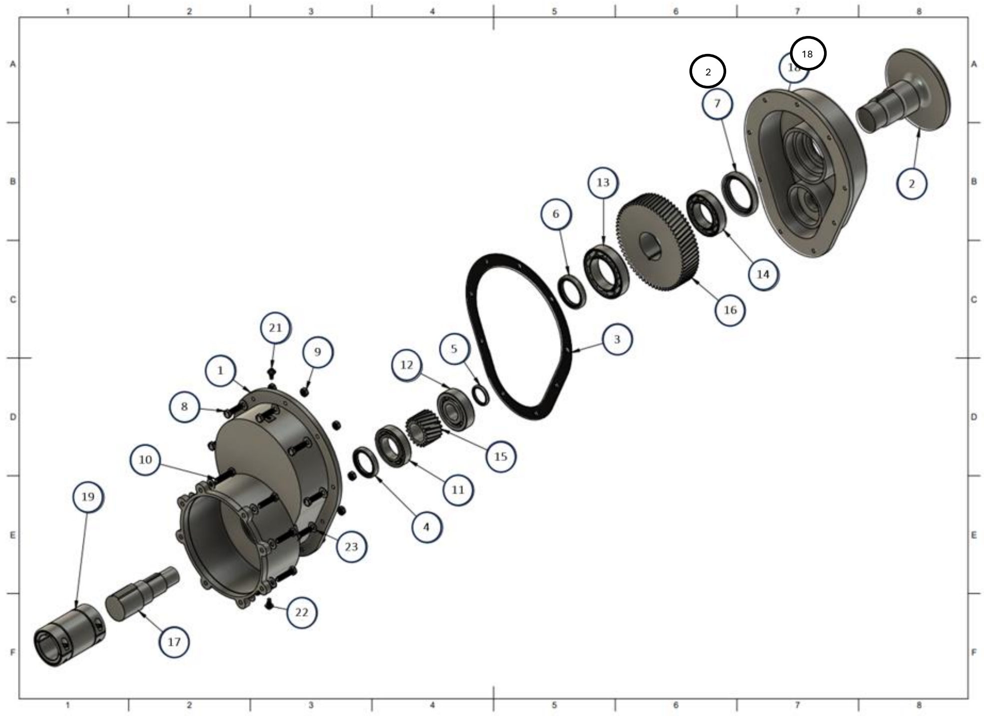


Figure 7: Exploded view of Initial Design.

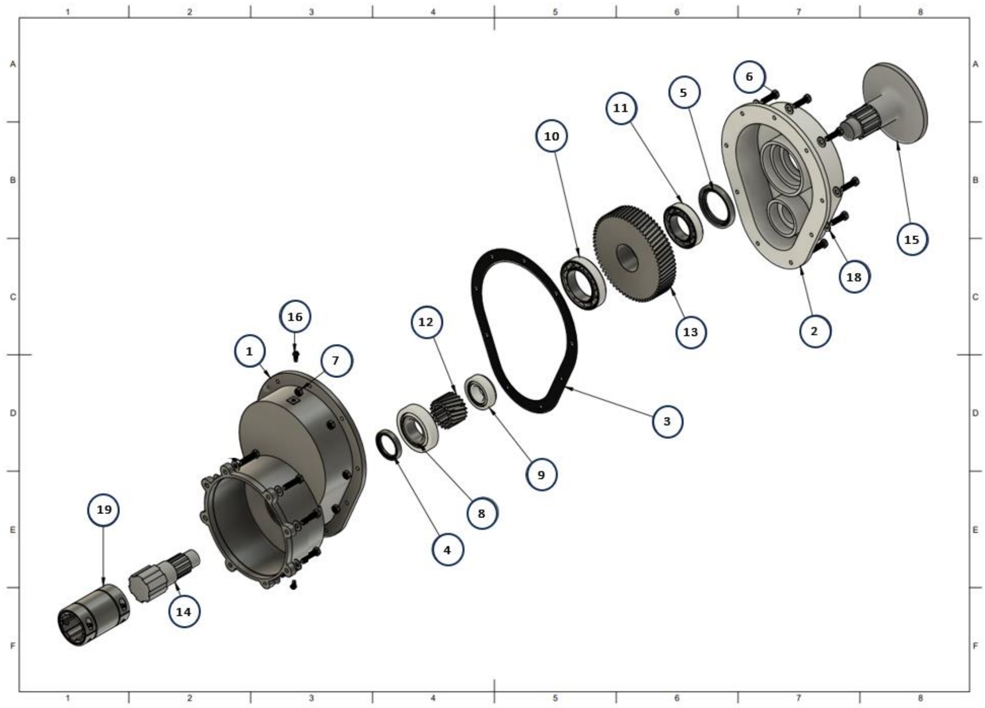
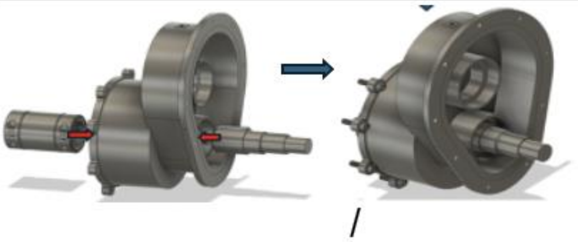
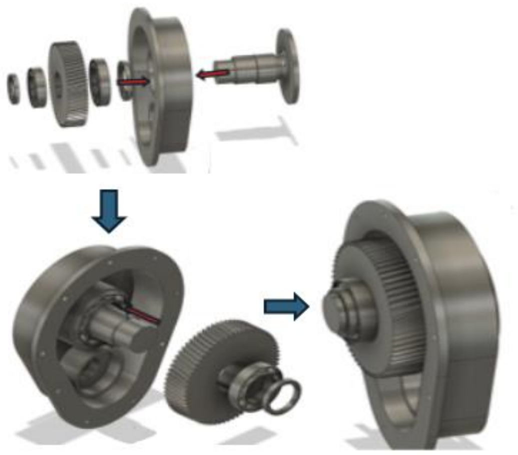
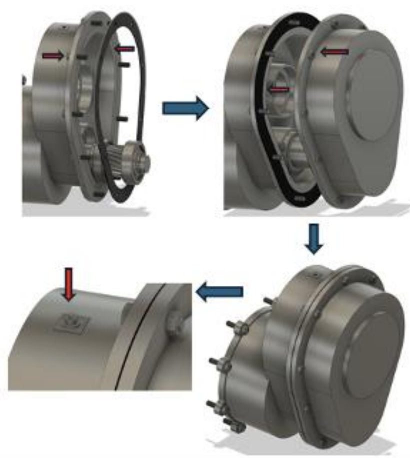


Figure 8: Exploded view of Final Design.

Assembly Instructions

Design 1:

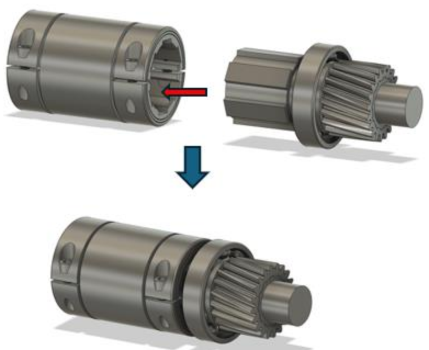
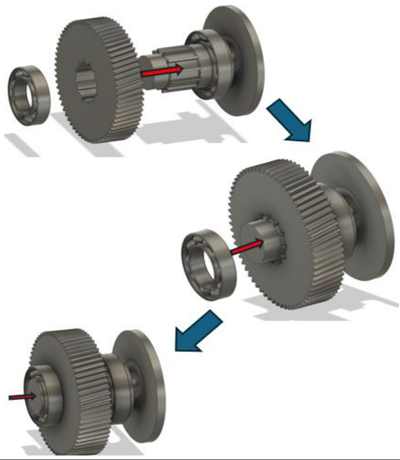
Table 17: Initial Design Assembly Sequence

No.	Instruction	Diagram
1	• Clamp the engine casing to a workbench to secure it in position. • Insert the input shaft along with the keyway coupling. The key on the shaft slots into the keyway on the coupling ensuring a tight fit. Secure the coupling bolts using an Allen key. • Simultaneously, fasten the coupling to the drive shaft using coupling bolts. • Attach the engine casing to the engine block with screws and washers before securing. • Remove the clamps once stable.	
2	• Afterwards, attach components in the following order: lip seal 1, bearing 1, pinion, bearing 2 and lip seal 2, using a handheld press for precise fits. • Align the pinion's keyway with the input shaft's key to ensure correct alignment.	
3	**Sub - Assembly** • Assemble the output shaft by attaching it to the wheel along with bearings 3 and 4, and lip seals 3 and 4. • Clamp the pump casing to the workbench, using a cloth in between them to prevent scratches. • Insert the output shaft through the upper hole in casing and then press fit parts in order of lip seal 4, bearing 4, wheel, bearing 3, lip seal 3.	
4	• Place engine casing washers on the exterior of the casing and screw the bolts for gasket alignment. Align and secure it with the engine casing by screwing the nuts onto the bolts. • Pour oil into the oil plug before screwing the plug into position.	

Design 2:

The assembly process for the final design is the same as the initial design, other than the process of attaching the input shaft to the wheel and the output shaft to the coupling. This is due to the change in coupling design which uses a spline design.

Table 18: Final Design Assembly Sequence

Instruction Number	Instruction	Diagram
1	• Press fit the output shaft into the spline coupling, aligning the shaft's keys with the keyways in the coupling. (For clarity, the casing is removed in the image to demonstrate the process)	
2	• Press fit the input shaft into lip seal 2, bearing 4 then the wheel gear, ensuring that the keys are aligned with the keyways. • Press fit bearing 3 onto the end of the shaft.	

DFA Considerations

Sub-assembly

Table 19: Sub-assembly considerations

Component	Consideration
Shaft	**Non-stepped Shaft -** Increases simplicity, driving manufacturing costs down, but is harder to align other components on the shaft.
Bearing	**Plain Bearing -** Low maintenance and cost, however, there is more friction between surfaces.
Seal	**Non-contacting Seal –** Doesn't require lubrication and can operate at higher speeds than lip seals.
Pump Casing	**No Supports –** More alignment and stability issues which may cause surface indentation.

General Assembly

Table 20: General Assembly Considerations

Component	Consideration
Gasket Seal	**Snap-Fit connection** – Use a snap-fit for easier alignment, and eliminate the use of nuts and bolts, lowering costs
Casing	**Chamfered Edges –** Greater stress distribution but may increase manufacturing costs. **Latching Mechanism –** Integrate a latch between the engine and pump casing to prevent separation during operation.

DFA Analysis

A design evaluation was carried out to reduce the number of parts, cost and time of designing the assembly. The Lucas method was implemented as it gives a relative measure of assembly difficulty which is carried out in four steps. The calculations below are for the initial design.

Step 1: Design Efficiency (Functional Analysis)

This step involves sorting components into group A (essential part) and group B (non-essential part) and calculating the number of essential parts relative to the total number of parts. This is done to reduce non-essential parts for the modified assembly.

$$Design\ Efficiency\ (EI) = \frac{A}{A+B} \times 100\% = \frac{16}{16+7} \times 100\% = 69.57\%\ [26]$$

Step 2: Feeding Ratio (Feeding Analysis)

Tables are used to provide scores of the insertion and handling of each part, known as a Feeding Index. A target index of 1.5 is required for each part, and the part should be modified if it exceeds 1.5. These indices are needed to calculate the Feeding Ratio of the assembly:

$$Feeding\ Ratio\ = \frac{Total\ Feeding\ Index}{Number\ of\ Essential\ Components\ (A)} = \frac{28.6}{16} = 1.79\ [27]$$

Where the Total Feeding Index represents the sum of the indices of each part and the number of essential components is obtained in step 1. A Feeding Ratio of 2.5 is an ideal value.

Step 3: Fitting Ratio (Fitting Analysis)

Like the Feeding Analysis, appropriate tables are used to score the Fitting Index for each part. It is desirable to have a Fitting Index of 1.5 for each part and 2.5 for the Fitting Ratio.

$$Fitting\ Ratio\ = \frac{Total\ Fitting\ Index}{Number\ of\ Essential\ Components\ (A)} = \frac{99.2}{16} = 6.2\ [28]$$

Step 4: Cost of Manufacturing

This step serves to measure the manufacturing cost rather than calculate the part and assembly's actual cost and is essential for material selection and the part-making processes.

$$Manufacturing\ Cost\ Index\ (M_i) = R_c P_c + M_c\ [29]$$

Where R_c is the Relative Cost, P_c is the Processing Cost and M_c is the Material Cost.

$$M_c = VC_{mt}W_c \; [30]$$

Where V is the Volume, C_{mt} is the Material Cost per Unit Volume and W_c is the Waste Coefficient.

$$R_c = C_c C_{mp} C_s \cdot (C_t \; or \; C_f) \; [31]$$

Where C_c is the Complexity Factor, C_{mp} is the Material Factor, C_s is the Minimum Section, C_t is the Tolerance Factor and C_f is the Finish Factor. Whether C_t or C_f is chosen depends on which value is greater.

To evaluate the improvement in the final design, Equations 26, 27 and 28 were used:

$$Design\ Efficiency\ (EI) = \frac{16}{16+3} \times 100\% = 84.21\%$$

$$Feeding\ Ratio = \frac{23.3}{16} = 1.46$$

$$Fitting\ Ratio = \frac{78.9}{16} = 4.93$$

The justifications for the feeding index and fitting index, and the factors used to calculate the manufacturing cost index are found in the Appendix.

Table 21: Initial and Final Design Component Indices

Initial Design					**Final Design**				
Component	**Grp.**	**Part No.**	**Feeding Index**	**Fitting Index**	**Component**	**Grp.**	**Part No.**	**Feeding Index**	**Fitting Index**
Case – Engine Half	A	1	1.3	6.1	Case – Engine Half	A	1	1.3	6.1
Case – Pump Half	A	2	1.3	6.1	Case – Pump Half	A	2	1.3	6.1
Case – Gasket Seal	B	3	1.9	3.4	Case – Gasket Seal	B	3	1.9	3.4
Lip Seal 1	A	4	1.5	3.4	Lip Seal 1	A	4	1.5	3.4
Lip Seal 2	A	5	1.5	3.4	Lip Seal 2	A	5	1.5	3.4
Lip Seal 3	B	6	1.5	3.4	Casing Bolt	A	6	1	6.8
Lip Seal 4	B	7	1.5	3.4	Pump Casing Nut	A	7	1.5	4.1
Pump Casing Bolt	A	8	1	6.8	Bearing 1	A	8	1	3.4
Pump Casing Nut	A	9	1.5	4.1	Bearing 2	A	9	1	3.4
Engine Casing Bolt	A	10	1	6.8	Bearing 3	A	10	1	3.4
Bearing 1	A	11	1	3.4	Bearing 4	A	11	1	3.4
Bearing 2	A	12	1	3.4	Pinion	A	12	1	3.4
Bearing 3	A	13	1	3.4	Wheel	A	13	1	3.4
Bearing 4	A	14	1	3.4	Input Shaft	A	14	1.3	3.4
Pinion	A	15	1	3.4	Output Shaft	A	15	1.3	3.4
Wheel	A	16	1	3.4	Oil Plug	A	16	1.7	6.8
Input Shaft	A	17	1.3	3.4	Oil Plug Washer	B	17	1	4.1
Output Shaft	A	18	1.3	3.4	Engine Casing Washer	B	18	1	4.1

Keyway Coupling	B	19	1.3	3.4	Spline Coupling	A	19	1	3.4
Coupling Bolt	B	20	1	6.7	-	-	-	-	-
Oil Plug	A	21	1.7	6.8	-	-	-	-	-
Oil Plug Washer	B	22	1	4.1	-	-	-	-	-
Engine Casing Washer	B	23	1	4.1	-	-	-	-	-
Total	-	-	**28.6**	**99.2**	**Total**	-	-	**23.3**	**78.9**

Table 22: Estimate of cost of manufacturing each component

Initial Design				Final Design			
Component	**Manufacturer**	**Process To Make Part**	**M_i**	**Component**	**Manufacturer**	**Process To Make Part**	**M_i**
Case – Engine Half	-	Sand Cast	1631.7	Case – Engine Half	-	Sand Cast	1671.5
Case – Pump Half	-	Sand Cast	1336.9	Case – Pump Half	-	Sand Cast	1331.7
Case – Gasket Seal	-	Plastic Mold	1429.9	Case – Gasket Seal	-	Plastic Mold	1429.9
Lip Seal 1	McMaster-Carr	Machine	4667.2	Lip Seal 1	McMaster-Carr	Machine	4654.6
Lip Seal 2	McMaster-Carr	Machine	4654.6	Lip Seal 2	McMaster-Carr	Machine	4667.2
Lip Seal 3	McMaster-Carr	Machine	4692.6	Casing Bolt	McMaster-Carr	Forge	122.6
Lip Seal 4	McMaster-Carr	Machine	4669.2	Pump Casing Nut	McMaster-Carr	Forge	685.5
Pump Casing Bolt	McMaster-Carr	Forge	1441.3	Bearing 1	Schaeffler	Forge	16688.1
Pump Casing Nut	McMaster-Carr	Forge	685.5	Bearing 2	Schaeffler	Forge	16633.5
Engine Casing Bolt	McMaster-Carr	Forge	1441.9	Bearing 3	Schaeffler	Forge	16639.8
Bearing 1	Schaeffler	Forge	16633.5	Bearing 4	Schaeffler	Forge	16668.1
Bearing 2	Schaeffler	Forge	16688.1	Pinion	-	Machine	1167.5
Bearing 3	Schaeffler	Forge	16639.8	Wheel	-	Machine	3544.5
Bearing 4	Schaeffler	Forge	16668.1	Input Shaft	-	Machine	4821.9
Pinion	-	Machine	1167.5	Output Shaft	-	Machine	6013.1
Wheel	-	Machine	3562.5	Oil Plug	-	Forge	4423.1
Input Shaft	-	Machine	4821.9	Oil Plug Washer	McMaster-Carr	Press	1140.2
Output Shaft	-	Machine	6065.9	Engine Casing Washer	McMaster-Carr	Press	14.2
Keyway Coupling	McMaster-Carr	Machine	11495.7	Spline Coupling	McMaster-Carr	Machine	11318.7
Coupling Bolt	McMaster-Carr	Forge	5488.3	-	-	-	-
Oil Plug	-	Forge	4423.1	-	-	-	-
Oil Plug Washer	McMaster-Carr	Press	1140.2	-	-	-	-
Engine Casing Washer	McMaster-Carr	Press	14.2	-	-	-	-
Total	-	-	**131459.5**	**Total**	-	-	**113635.7**

Analysis of the initial design:

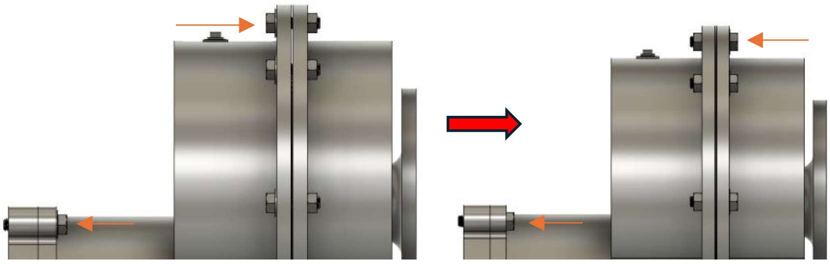
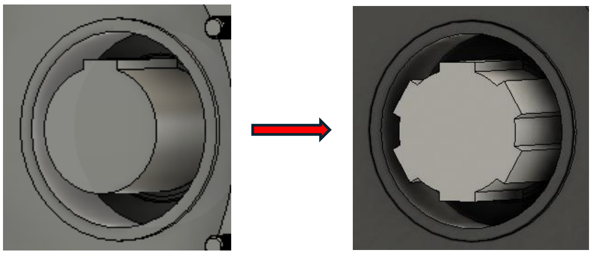
- Design Efficiency of 69.57% was achieved, surpassing the target of 60% for initial designs. Further improvement remained through eliminating non-essential components.
- Feeding Ratio was 1.79, which was close to the ideal ratio of 2.5.
- Fitting Ratio was 6.2, which was very far from the desirable fitting ratio of 2.5, indicating misalignments in component integration.
- Manufacturing Cost Index is 131459.5, indicating a need for cost reduction through simplifying the design.

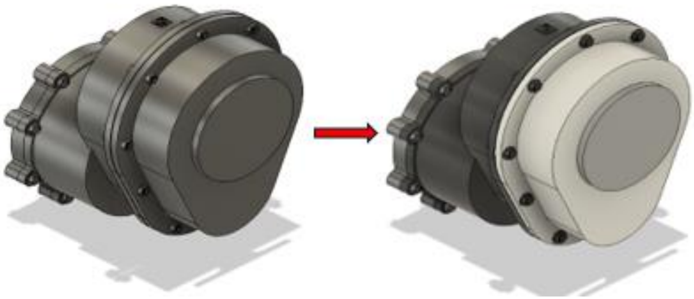
Analysis of the final design:

- Design Efficiency of 84.21%, displaying significant improvement by reducing the complexity such as removing the two lip seals and making components essential such as replacing the keyway coupling with a spline coupling.
- The Feeding Ratio decreased to 1.46, as the non-essential component indices were removed. While the ratio moved further away from the ideal value, it allowed for improvements in the other stages.
- Fitting Ratio decreased to 4.93, moving closer to the ideal fitting ratio. This demonstrated a better alignment of components.
- Manufacturing Cost Index decreased to 113635.7, reflecting success in the design improvement by making it more cost-effective.

Improvements made using DFA

Table 23: DFA Improvements

DFA Improvement	Justification
Reduction of Parts	The overall number of parts were reduced. The lip seals were reduced by 2 as the other 2 lip seals weren't essential for the new design.
All bolts are the same	The bolts for the engine casing and the transmission casing were initially M7 and M8. Afterwards, they were made the same size to improve issues having to find which bolt is M7 and M8 during assembly.
Direction of Assembly	The bolts are made to be screwed in from the same direction for every part. The figures show the difference between the clarity of changing the direction of the bolts. 
Keyway to Spline	The shaft designs were changed from keyways to splines. This meant it was symmetrical and the load can be distributed more evenly on the shaft when its rotating. 

Colour Coding Parts	Makes it easier to recognise parts when assembling. Makes the assembly sequencing a lot quicker. 

Automated Assembly

Table 24: Automated Assembly

Modification	Location	Justification
Use standardised parts	All	Using standardised parts reduce tool customisation, saving money and requires less programming to be done on the robots due to fewer variables.
Reduction in the type of screws and nuts	All	This allows there to be less tools in the factory, which saves money, space and time. This is because the robots only need fewer tool changes and less space will be taken up by tools.
Simplified Design	All	This will make it easier for the robots to access the parts and manipulate them in space. Along with simplified includes a symmetrical design, this will improve alignment issues.
Orientation of assembly	All	Fewer manufacturing orientations yields fewer computations for the robot to do to connect the parts.
Reduction of small parts and additional grip.	Washers, small screws	Larger parts are a lot easier to handle, as robots don't have the same dexterity or grip as humans. Robots will need grippy appendages to be able to move parts from place to place.
Easy access to parts	All	Robots often have large arms, and aren't as flexible, so ensuring easy access allows for decreased assembly time.
Reduce the number of materials and finishes	All	A constant material and finish will reduce the strength and grip variation requirements in the robots.
Decrease the number of assembly steps	All	Fewer assembly steps allow for the robot to be as efficient as possible by reducing the complexity in programming.
Chamfers	All	Chamfers improve alignment on press fit parts, allow for larger tolerances and decrease the manufacturing times.

Other Information

The module team were able to help our group to critically analyse what were the best decisions for the transmission design, particularly when it came to deciding spur or helix gear, helix angle and gear size. It was most beneficial having the team on hand during the GP100 training sessions, the software was confusing at first but after some dedication and questions asked, it became clearer.

Table 25: Feedback

Week	Feedback Given	Action
1	Understanding of the difference between a pinion and wheel.	Sketched out gear designs.
2	Helped narrow down the input values for GP100, particularly for material selection.	Used trial and error in GP100 to choose the optimal number of teeth and materials for the shafts.
3	Explained the key equations needed to ensure the safety and longevity of the shaft.	Refined the shaft calculations, especially the critical speed and stress concentrations.
4	Explained keyway seals.	Completed the seal and lubrication specification.
5	Helped our group understand the types of keyways and couplings. Suggested adding more diagrams and comments	Completed the torque transmission coupling specification.
6	Asked about how to improve the layout of the report and suggested improving the safety factor value in the shaft specification as it was too high.	Ensured that the information provided was concise and removed unnecessary diagrams, and output shaft diameter was adjusted from 40 to 30 mm to allow safety factor of shafts to fall within 4-6.
7	Ensure the project is completed before project due date to prevent any technical problems in uploading the document to Canvas, effecting	

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Appendix

Initial Design	Feeding Index				
Part Number	A	B	C	D	Total
1	1	0	0.1	0.2	1.3
2	1	0	0.1	0.2	1.3
3	1	0.6	0.1	0.2	1.9
4	1.5	0	0	0	1.5
5	1.5	0	0	0	1.5
6	1.5	0	0	0	1.5
7	1.5	0	0	0	1.5
8	1	0	0	0	1
9	1.5	0	0	0	1.5
10	1	0	0	0	1
11	1	0	0	0	1
12	1	0	0	0	1
13	1	0	0	0	1
14	1	0	0	0	1
15	1	0	0	0	1
16	1	0	0	0	1
17	1	0	0.1	0.2	1.3
18	1	0	0.1	0.2	1.3
19	1	0	0.1	0.2	1.3
20	1	0	0	0	1
21	1	0.2+0.5	0	0	1.7
22	1	0	0	0	1
23	1	0	0	0	1

Final Design	Feeding Index				
Part Number	A	B	C	D	Total
1	1	0	0.1	0.2	1.3
2	1	0	0.1	0.2	1.3
3	1	0.6	0.1	0.2	1.9
4	1.5	0	0	0	1.5
5	1.5	0	0	0	1.5
6	1	0	0	0	1
7	1.5	0	0	0	1.5
8	1	0	0	0	1
9	1	0	0	0	1
10	1	0	0	0	1
11	1	0	0	0	1
12	1	0	0	0	1
13	1	0	0	0	1
14	1	0	0.1	0.2	1.3
15	1	0	0.1	0.2	1.3
16	1	0.5+0.2	0	0	1.7
17	1	0	0	0	1
18	1	0	0	0	1
19	1	0	0	0	1

Figure A1: Factors for calculating the Feeding Index for the initial and final design.

Initial Design	Fitting Index						
Part Number	A	B	C	D	E	F	Total
1	2+4	0.1	0	0	0	0	6.1
2	2+4	0.1	0	0	0	0	6.1
3	2+1.3	0.1	0	0	0	0	3.4
4	2+1.3	0.1	0	0	0	0	3.4
5	2+1.3	0.1	0	0	0	0	3.4
6	2+1.3	0.1	0	0	0	0	3.4
7	2+1.3	0.1	0	0	0	0	3.4
8	2+4	0.1	0.7	0	0	0	6.8
9	2+1.3	0.1	0.7	0	0	0	4.1
10	2+4	0.1	0.7	0	0	0	6.8
11	2+1.3	0.1	0	0	0	0	3.4
12	2+1.3	0.1	0	0	0	0	3.4
13	2+1.3	0.1	0	0	0	0	3.4
14	2+1.3	0.1	0	0	0	0	3.4
15	2+1.3	0.1	0	0	0	0	3.4
16	2+1.3	0.1	0	0	0	0	3.4
17	2+1.3	0.1	0	0	0	0	3.4
18	2+1.3	0.1	0	0	0	0	3.4
19	2+1.3	0.1	0	0	0	0	3.4
20	2+4	0	0.7	0	0	0	6.7
21	2+4	0.1	0.7	0	0	0	6.8
22	2+1.3	0.1	0.7	0	0	0	4.1
23	2+1.3	0.1	0.7	0	0	0	4.1

Final Design	Fitting Index						
Part Number	A	B	C	D	E	F	Total
1	2+4	0.1	0	0	0	0	6.1
2	2+4	0.1	0	0	0	0	6.1
3	2+1.3	0.1	0	0	0	0	3.4
4	2+1.3	0.1	0	0	0	0	3.4
5	2+1.3	0.1	0	0	0	0	3.4
6	2+4	0.1	0.7	0	0	0	6.8
7	2+1.3	0.1	0.7	0	0	0	4.1
8	2+1.3	0.1	0	0	0	0	3.4
9	2+1.3	0.1	0	0	0	0	3.4
10	2+1.3	0.1	0	0	0	0	3.4
11	2+1.3	0.1	0	0	0	0	3.4
12	2+1.3	0.1	0	0	0	0	3.4
13	2+1.3	0.1	0	0	0	0	3.4
14	2+1.3	0.1	0	0	0	0	3.4
15	2+1.3	0.1	0	0	0	0	3.4
16	2+4	0.1	0.7	0	0	0	6.8
17	2+1.3	0.1	0.7	0	0	0	4.1
18	2+1.3	0.1	0.7	0	0	0	4.1
19	2+1.3	0.1	0	0	0	0	3.4

Figure A2: Factors for calculating the Fitting Index for the initial and final design

Part	Group	Material	Process	Cc	Cmp	Cs	Ct	Cf	Rc	V (mm^3)	Cmt (cents/mm^3)	Wc	Mc	Pc	Manufacturer	Mi
Engine Casing	B4	Cast Iron	Sand cast	1.8	1	1	2.8	3.5	6.3	1020000	0.00105	1.3	1392.3	38		1631.7
Pump Casing	B4	Cast Iron	Sand cast	1.8	1	1	2.8	3.5	6.3	804000	0.00105	1.3	1097.46	38		1336.86
Gasket Seal (Casing)	C1	Rubber	Plastic Mold	1	1.5	1	1.9	1	2.85	13985.587	0.00035	1	4.89495545	500		1429.89496
Pinion	C2	EN34 Steel	Machine	1.4	2.5	1	1.2	1.1	4.2	18899	0.00259	2.4	117.476184	250		1167.47618
Wheel	C2	EN24 Steel	Machine	1.4	2.5	1	1.2	1.1	4.2	404200	0.00259	2.4	2512.5072	250		3562.5072
Input Shaft	A4	AISI 4140 Steel	Machine	5.3	2.5	1	1.2	1.1	15.9	109000	0.00259	3	846.93	250		4821.93
Output Shaft	A4	AISI 4340 Steel	Machine	5.3	2.5	1	1.2	1.1	15.9	269100	0.00259	3	2090.907	250		6065.907
Lip Seal 1	C3	Stainless Steel/Rubber	Machine	3.1	1.5	1	2.5	1.5	11.625	3594	0.00341	1.4	17.157756	400	McMaster-Carr	4667.15776
Lip Seal 2	C3	Stainless Steel/Rubber	Machine	3.1	1.5	1	2.5	1.5	11.625	956	0.00341	1.4	4.563944	400	McMaster-Carr	4654.56394
Lip Seal 3	C3	Stainless Steel/Rubber	Machine	3.1	1.5	1	2.5	1.5	11.625	8930	0.00341	1.4	42.63182	400	McMaster-Carr	4692.63182
Lip Seal 4	C3	Stainless Steel/Rubber	Machine	3.1	1.5	1	2.5	1.5	11.625	4023	0.00341	1.4	19.205802	400	McMaster-Carr	4669.2058
Pump Casing Bolt	A2	Stainless Steel	Forge	2.1	2	1.5	2.5	3	18.9	1300	0.00341	1.1	4.8763	76	McMaster-Carr	1441.2763
Pump Casing Nut	C1	Stainless Steel	Forge	1	2	1.5	3	3	9	394.494	0.00341	1.1	1.47974699	76	McMaster-Carr	685.479747
Engine Casing Bolt	A2	Stainless Steel	Forge	2.1	2	1.5	2.5	3	18.9	1457	0.00341	1.1	5.465207	76	McMaster-Carr	1441.86521
Bearing 1	A3	Stainless Steel	Forge	2.3	2	1.5	3.2	2.6	22.08	17974	0.00341	1.2	73.549608	750	Schaeffler	16633.5496
Bearing 2	A3	Stainless Steel	Forge	2.3	2	1.5	3.2	2.6	22.08	31304	0.00341	1.2	128.095968	750	Schaeffler	16688.096
Bearing 3	A3	Stainless Steel	Forge	2.3	2	1.5	3.2	2.6	22.08	19503	0.00341	1.2	79.806276	750	Schaeffler	16639.8063
Bearing 4	A3	Stainless Steel	Forge	2.3	2	1.5	3.2	2.6	22.08	26426	0.00341	1.2	108.135192	750	Schaeffler	16668.1352
Keyway Coupling	A4	Stainless Steel	Machine	5.3	4	1.6	1.2	1.1	40.704	1.29E+05	0.00341	3	1319.67	250	McMaster-Carr	11495.67
Coupling Bolt	A2	Stainless Steel	Forge	2.1	2	1.5	4.6	3	28.98	2958	0.00341	1.1	11.095458	189	McMaster-Carr	5488.31546
Oil Plug	A2	Stainless Steel	Forge	2.1	2	1	2.4	2.8	11.76	356.753	0.00341	1.1	1.3381805	376		4423.09818
Oil Plug Washer	C1	Stainless Steel	Press	1	1.5	1	3.8	1.2	5.7	40.883	0.00341	1.2	0.16729324	200	McMaster-Carr	1140.16729
Engine Casing Washer	C1	Stainless Steel	Press	1	1.5	1	3.8	1.2	5.7	128.278	0.00341	1.2	0.52491358	2.4	McMaster-Carr	14.2049136
Total																**131459.499**

Figure A3: Factors for Manufacturing Stage in initial design

Part	Group	Material	Process	Cc	Cmp	Cs	Ct	Cf	Rc	V (mm^3)	Cmt (cents/mm^3)	Wc	Mc	Pc	Manufacturer	Mi
Engine Casing	B4	Cast Iron	Sand cast	1.8	1	1	2.8	3.5	6.3	1049133.266	0.00105	1.3	1432.06691	38		1671.46691
Pump Casing	B4	Cast Iron	Sand cast	1.8	1	1	2.8	3.5	6.3	800248.678	0.00105	1.3	1092.33945	38		1331.73945
Gasket Seal (Casing)	C1	Rubber	Plastic Mold	1	1.5	1	1.9	1	2.85	13985.587	0.00035	1	4.89495545	500		1429.89496
Pinion	C2	EN34 Steel	Machine	1.4	2.5	1	1.2	1.1	4.2	18898.982	0.00259	2.4	117.476072	250		1167.47607
Wheel	C2	EN24 Steel	Machine	1.4	2.5	1	1.2	1.1	4.2	4.01E+05	0.00259	2.4	2494.4808	250		3544.4808
Input Shaft	A4	AISI 4140 Steel	Machine	5.3	2.5	1	1.2	1.1	15.9	109000	0.00259	3	846.93	250		4821.93
Output Shaft	A4	AISI 4340 Steel	Machine	5.3	2.5	1	1.2	1.1	15.9	2.62E+05	0.00259	3	2038.071	250		6013.071
Lip Seal 1	C3	Stainless Steel/Rubber	Machine	3.1	1.5	1	2.5	1.5	11.625	956.732	0.00341	1.4	4.56743857	400	McMaster-Carr	4654.56744
Lip Seal 2	C3	Stainless Steel/Rubber	Machine	3.1	1.5	1	2.5	1.5	11.625	3594.141	0.00341	1.4	17.1584291	400	McMaster-Carr	4667.15843
Pump Casing Nut	C1	Stainless Steel	Forge	1	2	1.5	3	3	9	394.494	0.00341	1.1	1.47974699	76	McMaster-Carr	685.479747
Casing Bolt	A2	Stainless Steel	Forge	2.1	2	1.5	2.5	3	18.9	1457.268	0.00341	1.1	5.46621227	6.2	McMaster-Carr	122.646212
Bearing 1	A3	Stainless Steel	Forge	2.3	2	1.5	3.2	2.6	22.08	31304.82	0.00341	1.2	128.099323	750	Schaeffler	16688.0993
Bearing 2	A3	Stainless Steel	Forge	2.3	2	1.5	3.2	2.6	22.08	17974	0.00341	1.2	73.549608	750	Schaeffler	16633.5496
Bearing 3	A3	Stainless Steel	Forge	2.3	2	1.5	3.2	2.6	22.08	19503.774	0.00341	1.2	79.8094432	750	Schaeffler	16639.8094
Bearing 4	A3	Stainless Steel	Forge	2.3	2	1.5	3.2	2.6	22.08	26425.876	0.00341	1.2	108.134685	750	Schaeffler	16668.1347
Oil Plug	A2	Stainless Steel	Forge	2.1	2	1	2.4	2.8	11.76	356.753	0.00341	1.1	1.3381805	376		4423.09818
Oil Plug Washer	C1	Stainless Steel	Press	1	1.5	1	3.8	1.2	5.7	40.883	0.00341	1.2	0.16729324	200	McMaster-Carr	1140.16729
Engine Casing Washer	C1	Stainless Steel	Press	1	1.5	1	3.8	1.2	5.7	128.278	0.00341	1.2	0.52491358	2.4	McMaster-Carr	14.2049136
Spline Coupling	A4	Stainless Steel	Machine	5.3	4	1.6	1.2	1.1	40.704	1.12E+05	0.00341	3	1142.691	250	McMaster-Carr	11318.691
Total																**113635.665**

Figure A4: Factors for Manufacturing Stage in final design

Max Torque on McMaster-Carr Coupling Calculation:

$$Max\,Torque\ on\ the\ Coupling = \frac{hp\times 63000}{rpm} = \frac{50\times 63000}{6500} = 484.62\ (lb.in) = 54.75\ Nm$$

Shear Stress on Key-way Calculations:

$$Failure\ Due\ to\ Shear:\ F = \frac{2T}{D} = \frac{2\times 55.31}{0.025} = 4424.8\ N]$$

$$\tau_{design} = \frac{F}{A_s} = \frac{2T}{D\omega L},\ where\ A_s = \omega L$$

$$A_s = (0.012\times 0.25)\times 0.04\ =\ 1.2\times 10^{-4}\ m^2$$

$$\tau_{design}(Shear\ Stress) = \frac{4424.8}{1.2\times 10^{-4}} = 7.87\times 10^6\ Pa$$

$$To\ avoid\ failure\ due\ to\ shear:\ \tau_{design} \le \frac{S_{sy}}{f_s}\ :\ 7.87\times 10^6 \le \frac{2.075\times 10^8}{4.64} = 4.47\times 10^7$$

$$\therefore\ True\ Bearing\ Stress:\ \sigma_{design} = \frac{F}{A_b} = \frac{4T}{DHL},\ where\ Bearing\ Area:\ A_b = \frac{LH}{2}$$

$$A_b = \frac{0.09\times(0.025\times 0.25)}{2} = 2.81\times 10^{-4}\ m^2$$

$$\sigma_{design} = \frac{4424.8}{2.81\times 10^{-4}} = 1.57\times 10^7\ Pa$$

$$Bearing\ load\ failures\ can\ be\ avoided\ if:\ \sigma_{design} \le \frac{S_y}{f_s}$$

$$1.57\times 10^7\ Pa\ \le\ \frac{4.15\times 10^8}{4.64} = 8.94\times 10^7\ \therefore\ True$$

Pitch Line Velocity Equations:

$$Shaft\ 1: PLV = \frac{\pi.42.57\times 10^{-3}.6500}{60} = 14.49\ ms^{-1}$$

$$Shaft\ 2: PLV = \frac{\pi.138.34\times 10^{-3}.6500}{60} = 47.08\ ms^{-1}$$

Spline Calculations:

$$K_s = 2\times\frac{1}{1.8} = 1.11$$

$$Radius = \frac{Outside\ Diameter + Inside\ Diameter}{4} = \frac{43\ mm + 40\ mm}{4} = 20.75\ mm$$

$$\sigma_c = \frac{54.75\times 1.11}{1\times 3\times 10^{-3}\times 40\times 10^{-3}\times 20.75\times 10^{-3}} = 24406626.51 = 2.44\times 10^7 Pa$$