DEPARTMENT OF MECHANICAL ENGINEERING

Validation Report
Power Take-Off Gear Reduction Unit

Mechanical Design B (04 22964)
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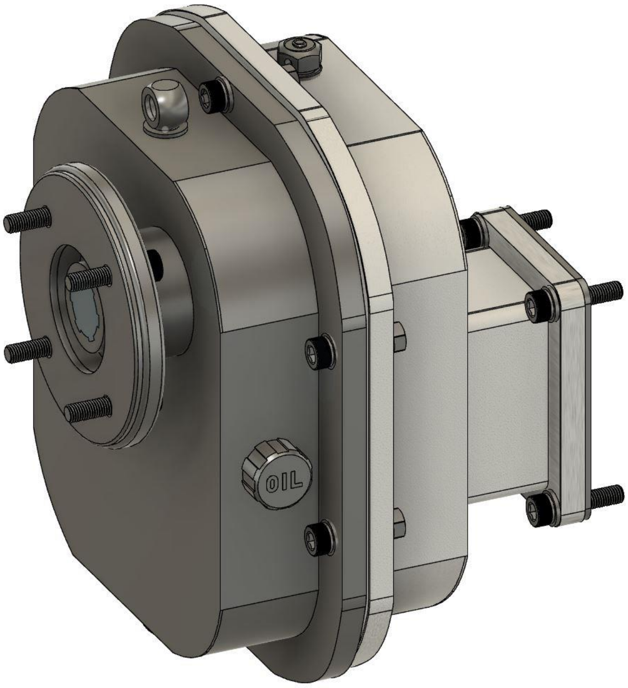


Part 1: Validation Calculations

1.0 / DESIGN SPECIFICATION

1.1 / Performance

A power take-off (PTO) gear reduction unit is required to facilitate power transfer from an industrial diesel generator to a hydraulic pump. The unit must process the following characteristics:

- **Compatibility:** The unit should seamlessly interface with a 45kW industrial diesel engine that operates at a crankshaft speed of 2800 revolutions per minute (rpm).
- **Hydraulic Pump Connection:** Designed to connect to a hydraulic pump system with the capability to deliver a flow rate of 25 liters per minute of hydraulic oil at a pressure of 150 bar.
- **Shaft Orientation:** Should be configured to integrate with two shaft ends, both rotating in the same anticlockwise direction.

1.2 / Maintenance and Lifetime

- All parts should have a lifetime of 12,000 hours except for gears which have a 20,000 hour lifetime and shafts which should have an infinite life.
- Lubricants will be selected according to manufacturer guidelines for viscosity to reduce leakages and extend component lifespan.
- Lubrication will be easy, requiring no disassembly.
- Maintenance and assembly will be user-friendly, without the need for special tools.
- A comprehensive maintenance schedule will be provided to prevent neglect-related failures.
- Gears will undergo regular maintenance based on operating conditions.
- Periodic inspections, including visual checks, photos and operation feedback, will occur every 2500 hours to ensure optimal performance and safety.

1.3 / Materials

- Components exhibit high wear, fatigue and vibration resistance for an extended lifespan, including infinite fatigue resistance.
- Materials with low thermal expansion prevent size changes during operation.
- Gears are made from EN8 for increased hardness.
- Shafts, driven by gears, use low alloy steel for withstanding dynamic loads and minimising deflection.
- Cast iron casing provides strength while being lightweight, with high thermal capacity for heat dissipation.
- Inner components are machined for smooth surfaces, reducing friction and wear.
- Shaft materials handle dynamic loads and maintain stiffness for precision performance.

1.4 / Quantity and Cost

- Initial annual production of 100,000 units with cost-efficient high performance.
- Low production costs but having a high level of performance.

1.5 / Aesthetics and Ergonomics

- Design should facilitate convenient maintenance or adjustments.
- The gearbox should be easily installable with a user-friendly assembly and disassembly process.
- Ensure that the design prioritises user safety by eliminating sharp edges throughout.

2.0 / GEAR DESIGN

To calculate the gear ratio, the input speed and the desired flow was used. This analysis resulted in an output speed of 1300 rpm and a transmitted power of 5.5 kW, as depicted in Figure 1 below.

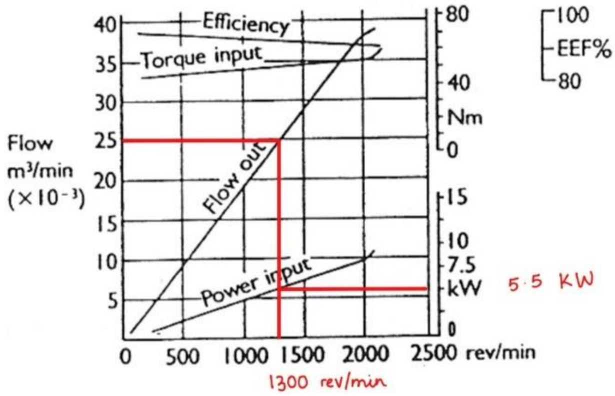


Figure 1: Graph showing the gear pump characteristics [1]

From the values found on the graph, a gear ratio was calculated using Equation 1:

$$\text{Gear Ratio} = \frac{\text{Pinion Speed}}{\text{Wheel Speed}} = \frac{2800}{1300} = \mathbf{2.15} \qquad (1)$$

The process of designing the gear was carried out using GP100 software, and this involved entering the data outlined in Table 1, as presented below.

Table 1: Gear design parameters entered into GP100

Parameter	Data Inputted	Justification
Minimum number of pinion teeth	20	Ensures that the gear system operates effectively and maintains its longevity
Approximate life hours	20,000	Ensures suitability over a long lifespan
Ground vs Cut Gears	Ground	High precision, reduced heat generation and high surface finish
Pinion and Wheel Materials	EN24 Steel and EN8 Steel	Excellent wear resistance as pinion performs more revolutions
Tooth size preferences	Non-standard	Provides optimised performance and reduced weight
Gear Ratio	2.15	Calculated from requirements
Helical vs Spur gear	Spur	Higher precision and efficiency
Transmitted Power (kW)	5.5	From specification
Pinion Speed (revs/min)	2800	From specification

Figures 2 and 3 displays the design results and details obtained from the GP100 software.

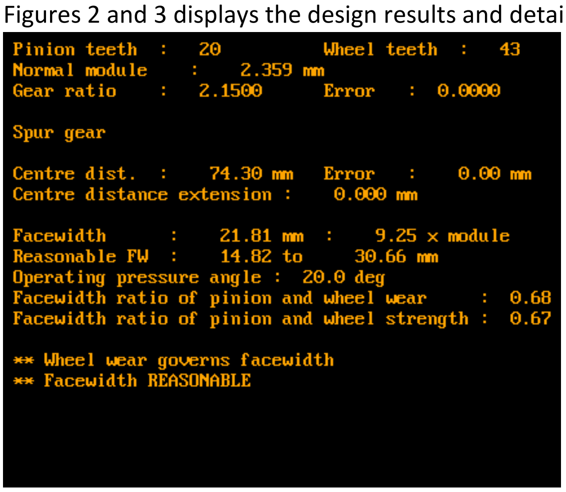


Figure 2: Design results from GP100

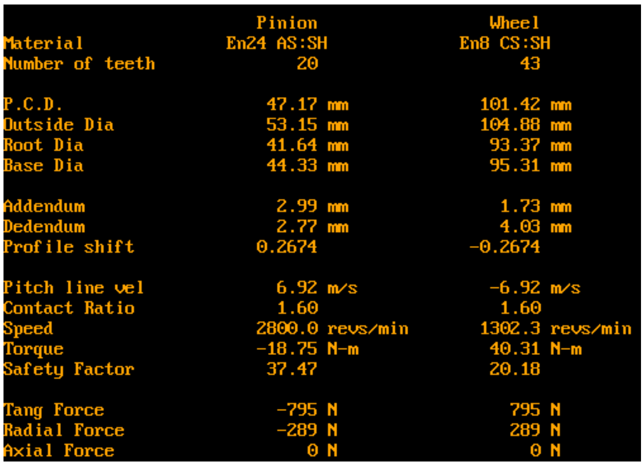


Figure 3: Design details from GP100

[1] https://canvas.bham.ac.uk/courses/71238/files/15580085?module_item_id=3439840c

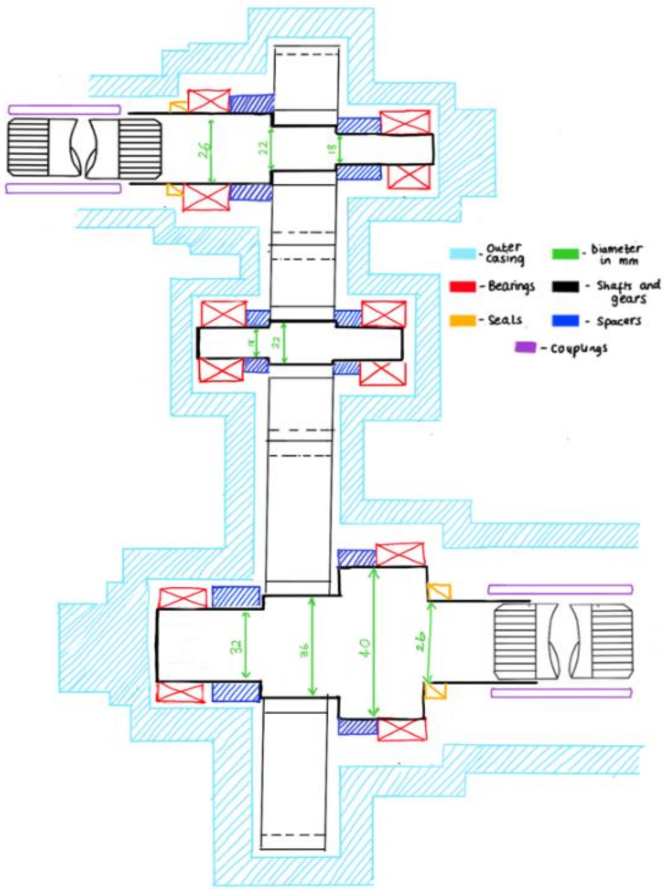


Figure 4: Sketch of casing and internal components

Figure 4 illustrates a sketch of the outer casing for the PTO gear reduction unit. The drawing displays the location and number of shafts, gears, bearings, seals, nuts, bolts and spacers. The arrangement of the gears ensures optimal torque transmission and speed reduction. Bearings are strategically placed to support the shafts and reduce friction, guaranteeing a smooth and efficient operation. The seals are integrated to prevent the entry of contaminants and maintain the system's integrity. The sketch also provides a comprehensive view of the numerous nuts, bolts and spacers meticulously distributed throughout the assembly, ensuring structural stability and alignment. This detailed visual representation serves as a valuable reference for engineers and technicians, offering a clear roadmap for the alignment of the internal components.

3.0 / BEARINGS

The bearings serve to reduce friction and allow for smoother rotation as well as avoiding any oscillations parallel to the direction it develops, which cuts down on the amount of energy consumption. The bearing has to meet a minimum lifetime requirement of 12,000 hours, minimum shaft diameters calculated in section 4.3 in the shaft analysis, and minimum required load rating which are calculated using Equations 2, 3 and 4. Table 2 shows the parameters for the shafts used to calculate the basic life rating, basic load rating and equivalent bearing load. These were used to determine the bearings that should be selected and whether they are able to withstand the loads applied to them.

Table 2: Parameters for shafts

	Input	Middle	Output
Life Index, n (Ball Bearing)	3	3	3
Radial Load, F_r (N)	289	1030	289
Axial Load, F_a (N)	0	0	0
Lifetime, $L_{10,h}$ (hrs)	12,000	12,000	12,000
Rotational Speed, N (rpm)	2800	2800	1300

Equation 2 was used to calculate the basic life rating (L_{10}) which depends on its hours of operation and its required rotational speed.

$$L_{10} = \frac{L_{10,h} \times N \times 60}{10^6} = \mathbf{2016\ revs} \;\&\; \mathbf{936\ revs} \qquad (2)$$

Where:

$L_{10,h}$ is life in hours (12,000 hours), **N** is speed in rpm (Input/Middle: 2800 revs/mins, Output: 1300 revs/min)

The input and middle shaft have the same rotational speed so will have the same basic life rating of 2016 revs. The output shaft has a lower rotational speed so has a basic life rating of 936 revs.

Equation 3 was used to calculate the equivalent bearing load:

$$P = XF_r + YF_a \tag{3}$$

Where:
F_r = Actual radial bearing load (N), **F_a** = Actual axial bearing load (N), **X and Y** = Defined by the bearing manufacturers for a type and size of bearing (X=1, Y=0)

Note: X=1 and Y=0 when $\frac{F_a}{F_r} \leq$ n.

The above note is applicable as there are zero axial loads and n is equal to 3.

$$P_{input} = (1)(289) + (0)(0) = \mathbf{289\ N}$$
$$P_{middle} = (1)(1030) + (0)(0) = \mathbf{1030\ N}$$
$$P_{output} = (1)(289) + (0)(0) = \mathbf{289\ N}$$

The bearing's life expectancy, based on the actual load, comes from Equation 4, which was reworked to find the basic load rating using the previously determined bearing life and load from Equations 2 and 3.

$$L_{10} = \left(\frac{C}{P}\right)^n \quad \rightarrow \quad C = P + L_{10}^{\frac{1}{n}} \tag{4}$$

Where:
C = Basic Loading Rate (kN), **P** = Equivalent Bearing Load (N), **L_{10}** = Life Rating (revs)

$$C_{input} = 289 + 2016^{\frac{1}{3}} = \mathbf{3.65\ kN}$$
$$C_{middle} = 1030 + 2016^{\frac{1}{3}} = \mathbf{13\ kN}$$
$$C_{output} = 289 + 936^{\frac{1}{3}} = \mathbf{2.83\ kN}$$

The calculated values were used to find the appropriate bearings for each shaft from the SKF catalogue by considering minimum required loading rate, inner/outer diameters and bore width as shown in Table 3.

Table 3: Parameters for shafts

Shaft	Location	Minimum Required Load Rating, C (kN)	Dynamic Load Rating, C (kN)	Static Load Rating, C (kN)	Inner/ Outer Diameter (mm)	Width (mm)	Limiting Speed (revs/min)	Product Code from SKF
Input	Engine Side	3.65	8.06	4.75	25/47	8	20,000	16005[2]
	Pump Side		19.5	13.2	17/47	19	13,000	4303 ATN9[3]
Idler	Both Sides	13	19.5	13.2	17/47	19	13,000	4303 ATN9[3]
Output	Engine Side	2.83	11.9	7.35	30/55	9	17,000	16006[4]
	Pump Side		8.06	4.75	25/47	8	20,000	16005[2]

[2] https://www.skf.com/uk/products/rolling-bearings/ball-bearings/deep-groove-ball-bearings/productid-16005
[3]https://www.skf.com/uk/products/rolling-bearings/ball-bearings/deep-groove-ball-bearings/productid-4303%20ATN9
[4] https://www.skf.com/uk/products/rolling-bearings/ball-bearings/deep-groove-ball-bearings/productid-16006

4.0 / SHAFT DESIGN

4.1 / Material Selection - Medium Carbon Steel ASTM A29 4140

Research was conducted to identify the best material for a mechanical shaft. Granta EduPack was used, due to its comprehensive materials database, to identify Medium Carbon Steel ASTM A29 4140 as the most suitable material, as it has a high tensile strength and good machinability. Its properties listed in Table 4.

Table 4: Properties of material (Medium Carbon Steel ASTM A29 4140)

Density	Young's Modulus	Yield Strength	Ultimate Tensile Strength	Shear Modulus	Fracture Toughness
7750 kg.m^{-3}	200 GPa	376 MPa	565 MPa	77 GPa	32 MPa.m$^{1/2}$

4.2 / Shear Force and Bending Moments

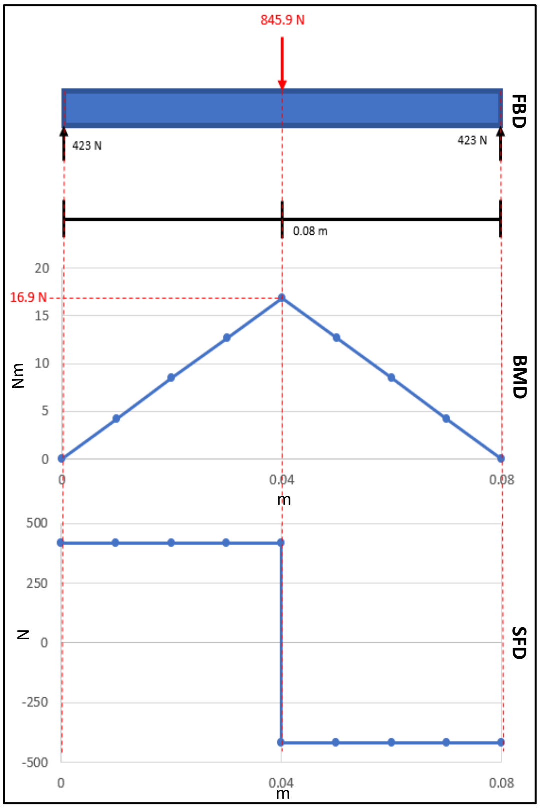


Figure 5: Free body diagram, bending moment and shear force diagram for input and output shaft

The parameters obtained from GP100 were used to solve for the resultant forces on the shaft. These were used to calculate the bending moments and shear forces, as seen in Figure 5. Equation 5 was used to solve for the resultant force, using the tangential and radial force, and had the same value for both the input and output shaft.

$$F_{res} = \sqrt{F_{tang}{}^2 + F_{rad}{}^2} = \mathbf{845.9\ N} \qquad (5)$$

Figure 5 illustrates a free body diagram showing the resultant and reaction forces. **An assumed value of 0.08 m was used for the length of the shafts since it requires less space and it makes the PTO unit more compact.**

$$\sum F_{res} = 0 \therefore F_A + F_B = 3387\ N \qquad (6)$$

$$\sum M_A = 0 \therefore 0.04F_{res} = 0.08F_B \qquad (7)$$

$$F_A = F_B = \mathbf{422.95\ N}$$

The shear force and bending moment diagrams in Figure 5 show the variation in shear force across the beam as well as the section of the beam that will be subject to maximum bending moment.

The Romax Concept 2022[5] software was used for calculating the minimum distance between the pinion and wheel, alongside determining an angle between them. The angle chosen was 90° since it was easier for both design and manufacturing while still keeping it compact. This facilitated the drawing of the free body diagram as shown in Figure 6. Given that the middle shaft is subject to both the driving force from the pinion and the reaction force from the wheel, the total resultant force can be calculated by considering the tangential (F_t) and radial (F_r) forces. The forces on the idler can be calculated in the x and y direction using Equations 8 and 9, then combined to calculate resultant force using Equation 10.

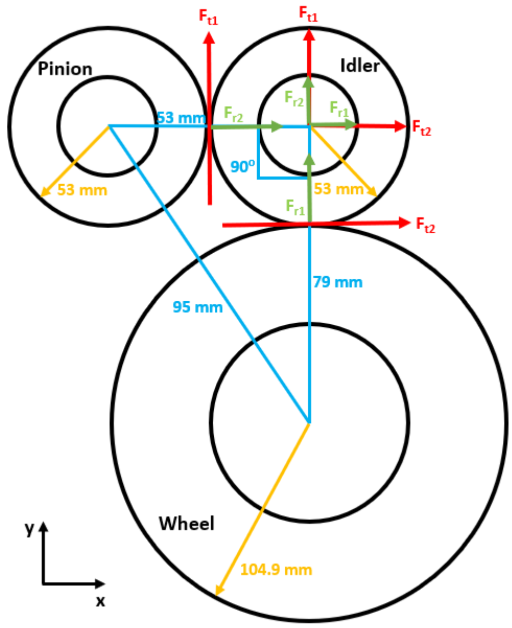


Figure 6: Free body diagram of forces acting on Idler

$$F_x = F_{r1} + F_{t2} = F_{rp} + F_{tw} \qquad (8)$$
$$F_x = 289 + 795 = \mathbf{1084\ N}$$

$$F_y = F_{r2} + F_{t1} = F_{rw} + F_{tp} \qquad (9)$$
$$F_y = 289 + 795 = \mathbf{1084\ N}$$

$$F_{res} = \sqrt{F_x{}^2 + F_y{}^2} = \mathbf{1533\ N} \qquad (10)$$

[5]Romax Concept 2022

4.3 / Minimum Shaft Diameter

The minimum shaft diameter was calculated using Equation 11.

$$d_{min} \geq \left[\frac{32 f_s}{\pi S_y}\sqrt{M^2 + T^2}\right]^{\frac{1}{3}} \qquad (11)$$

Table 5 shows the properties needed to work out the minimum diameter for the shaft.

Table 5: Parameters for shafts

	Input	Middle	Output
Yield Strength, S_y, (MPa)	376	376	376
Safety Factor	3	3	3
Resulting Bending Moment, M, (Nm)	16.9	30.2	16.9
Torque, T (Nm)	±18.75	±18.75	40.31
Minimum Diameter	**0.0127 m**	**0.0146 m**	**0.0153 m**

Input shaft: Diameter range of 0.013 m - 0.03 m. The shaft diameter at the pinion gear is 0.022 m.
Middle shaft: Diameter range of 0.015 m - 0.022 m. The shaft diameter at the idler gear is 0.022 m.
Output shaft: Diameter range of 0.022 m - 0.044 m. The shaft diameter at the wheel gear is 0.036

4.4 / Stress Concentration

To evaluate if the chosen diameter can withstand both reversed and mean stresses without failing, calculations were performed using the parameters from Table 6. Equations 12 and 13 were applied to compute bending shear stress (σ_x) and torsional shear stress (τ_x) for the shafts, and the results are presented in Table 6.

$$\sigma_x = \frac{My}{I} \quad (12) \qquad \tau_x = \frac{Tr}{J} \quad (13)$$

The stress values derived from Table 6 were subsequently used to calculate the normal principal stresses ($\sigma_{max,min}$) and the reverse bending stress (σ_r) using Equations 14 and 15 below. These values are shown in Table 7. Notably, σ_y was presumed to be zero as there were no axial forces.

Table 6: Parameters for calculating normal principal stresses and principal shear stresses.

Location	Torque, T (Nm)	Radius, R (m)	Polar Moment of Inertia, J (m^4)	Maximum Bending Moment, M (Nm)	Distance from neutral axis, y (m)	Moment of Inertia, I (m^4)	Bending Shear Stress, σ_x (MPa)	Torsional Shear Stress, τ_x (MPa)
Output	40.31	0.018	1.6×10^{-7}	16.9	0.018	8.2×10^{-8}	3.71	4.53
Middle	18.75	0.011	2.3×10^{-8}	30.2	0.011	1.1×10^{-8}	30.2	8.97
Input	18.75	0.011	2.3×10^{-8}	16.9	0.011	1.1×10^{-8}	16.9	8.97

$$\sigma_{max,min} = \frac{\sigma_x + \sigma_y}{2} \pm \left(\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \left(\tau_{xy}\right)^2\right)^{\frac{1}{2}} \quad (14) \qquad \sigma_r = \frac{\sigma_{max} - \sigma_{min}}{2} \quad (15)$$

Principal shear stresses ($\tau_{max,min}$) were calculated using data from Table 7 and applied in Equation 16. Equation 17 was used to determine the mean shear stress, and these results are documented in Table 7.

$$\tau_{max,min} = \pm\left(\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \left(\tau_{xy}\right)^2\right)^{\frac{1}{2}} \quad (16) \qquad \tau_m = \frac{\tau_{max} - \tau_{min}}{2} \quad (17)$$

Table 7: Values for stresses and shear stresses

Location	σ_{max}	σ_{min}	σ_r	τ_{max}	τ_{min}	τ_m
Output	6.59	-2.88	4.74	4.90	-4.90	4.90
Middle	32.7	-2.46	17.6	17.6	-17.6	17.6
Input	20.7	-3.87	12.3	12.3	-12.3	12.3

The stresses were used to assess the safety of the diameters in accordance with the Soderberg Criterion.

4.5 / Fatigue Analysis

The Revised Endurance Strength (S'_e) was calculated to determine the safety factors for the corresponding shafts, shown in Equation 18. The parameters and justifications are listed in Table 8.

$$S'e = K_a \, K_b \, K_c \, K_d \, K_e \, K_m \, S_e \qquad (18)$$

Table 8: Parameters for calculating the Revised Endurance Strength

Stress Factors	Symbol	Value			Justification
		Input	Middle	Output	
Surface Finish	K_a	0.9	0.9	0.9	From surface factor/tensile strength graph (Figure 7)

Size	K_b	0.88	0.88	0.84	$1.1289d^{-0.097}$ for $8 < d \leq 250$ mm
Reliability	K_c	0.81	0.81	0.81	From 99% reliability factor table where $K_c=1-0.08Z_r$
Temp	K_d	1	1	1	$K_d = 1$ For $T \leq 450°C$
Notch Sensitivity	K_e	0.65	0.65	0.56	$K_e = 1/K_f$ where $K_f = K_t$. Values for K_t taken from Figure 8
Miscellaneous Effects	K_m	1	1	1	Zero miscellaneous factors
Endurance Strength (MPa)	S_e	282.5	282.5	282.5	$S_e = 0.5 S_u$ where S_u= 565 MPa for the UTS of carbon steel ASTM A29 4140
Revised Endurance Strength (MPa)	S'_e	117.5	177.5	96.66	Using $S'_e = K_a K_b K_c K_d K_e K_m S_e$

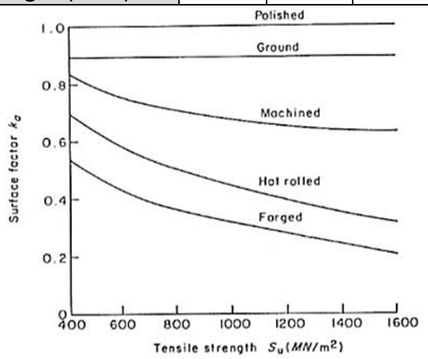


Figure 7: Graph used to obtain the value for K_a [6]

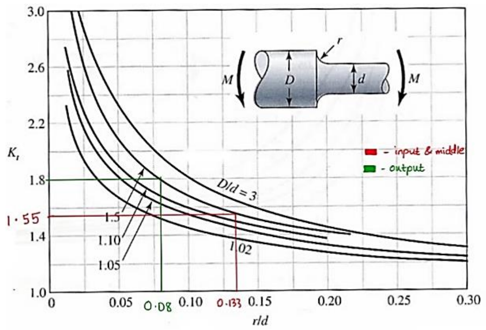


Figure 8: Graph used to obtain the value for K_t [7]

$$f_s = \frac{S_{sy}}{\left[\left(\frac{1}{4}\left(\frac{S_y}{S_e}\right)\sigma_r\right)^2 + (\tau_m)^2\right]^{\frac{1}{2}}} \tag{19}$$

Using the Soderberg Criterion shown in Equation 19, safety factors of 28 for the output shaft, 8.34 for the middle shaft, and 11.9 for the input shaft were calculated. These values were found to be below the GP100 guidelines, indicating that the chosen shaft diameters are safe for operation and were not altered.

4.6 / Linear Deflection and Twisting Loads

The deflection of the shaft is important in shaft analysis. Too much linear or torsional deflection can affect the shaft's performance and cause it to fail. Deflection was calculated by employing Equations 20 and 21, using the parameters shown in Table 9.

$$\delta = \frac{PL^3}{48EI} \quad (20) \qquad \Theta = \frac{TL}{GJ} \quad (21)$$

Table 9: Parameters for shafts

Shaft	Resultant Force, P (N)	Torque, T (Nm)	Distance Between Bearings, L (m)	Young's Modulus, E (Pa)	Shear Modulus, G (Pa)	Moment of Inertia, I (m⁴)	Polar Moment of Inertia, J (m⁴)	Linear Deflection δ (m)	Torsional Deflection Θ (deg)
Input	845.9	18.75	0.08	200×10^9	77×10^9	1.1×10^{-8}	2.3×10^{-8}	**4.1×10^{-6}**	**0.05 °**
Middle	1533	18.75	0.08	200×10^9	77×10^9	1.1×10^{-8}	2.3×10^{-8}	**7.3×10^{-6}**	**0.05 °**
Output	845.9	40.31	0.08	200×10^9	77×10^9	8.2×10^{-8}	1.6×10^{-7}	**5.5×10^{-7}**	**0.01°**

4.7 / Critical Shaft Speed

The critical shaft speed was calculated using the following equation:

$$N_c = \frac{30}{\pi}\sqrt{\frac{g}{\delta_{max}}} \tag{22}$$

Where: **N_C** = critical speed (rpm), **g** = gravitational constant (9.81 ms^{-2}) and **δ** = maximum deflection (m).

Table 10: Critical speed values for each shaft

Shaft	Maximum Deflection	Critical Shaft Speed
Input	4.1×10^{-6} m	**14771 rpm**
Middle	7.3×10^{-6} m	**11070 rpm**
Output	5.5×10^{-7} m	**40330 rpm**

[6] https://canvas.bham.ac.uk/courses/71238/pages/variable-loading-designing-against-fatigue
[7] https://canvas.bham.ac.uk/courses/71238/pages/variable-loading-designing-against-fatigue

5.0 / TORQUE TRANSMISSION COUPLINGS

Splines were employed for torque transmission, with the essential condition that they must withstand shear and compressive stress without experiencing failure. The fatigue life factor (L_f) was calculated to be 0.5 as the number of torque cycles was selected to be 100,000. The parameters in Table 11 were used to calculate Equations 23, 24 and 25.

$$\tau = \frac{16TK_s}{\pi {D_i}^3} \tag{23}$$

$$S_s^a = FOS_{shear} S_s \frac{K_a}{L_f} \tag{24}$$

$$\sigma_c = \frac{TK_s}{ndL_e r} \tag{25}$$

Table 11: Calculations for spline

Parameters	Description	Input to gear	Middle to gear	Output to gear	Input coupling	Output coupling
Spline designation	ISO standard	6 x 20 x 22	6 x 20 x 22	6 x 31 x 36	6 x 24 x 26	6 x 24 x 26
Torque transmitted, T (Nm)	-	18.75	18.75	40.31	18.75	40.31
Number of teeth, n	-	6	6	6	6	6
Reduced diameter, D_i (mm)	-	20	20	31	24	24
Depth of spline, d (mm)	-	1	1	2.5	1	1
Effective length, L_e (m)	-	0.02	0.02	0.02	0.05	0.05
Mean radius of spline, r (mm)	$r = \frac{D_i + D}{4}$	10.5	10.5	16.75	12.5	12.5
Service factor, K_s	For fixed spline, $\frac{K_{application} \times K_{design}}{K_{fatigue}}$ $K_{\text{application}}$ = 1.0, K_{design} = 1.0, K_{fatigue} = 0.5	2	2	2	2	2
Shear stress, τ (MPa)	Calculated from Equation 23	23.87	23.87	9.57	8.7	5.54
Factor of Safety	$FOS_{shear} = \frac{S_{sy}}{\tau}$	7.8	7.8	19.64	21.6	33.93
Allowable shaft stress, S_s^a (MPa)	Calculated from Equation 24	372.3	372.3	375.9	375.8	375.9
Compressible stress, σ_c (MPa)	Calculated from Equation 25	29.76	29.76	14.33	8.62	12.49
Factor of Safety	$FOS_c = \frac{S_y}{\sigma_c}$	12.63	12.63	26.23	43.61	30.1

Table 11 indicates that the safety factors exceed 2, ensuring the spline's reliability. The calculation of allowable shaft stress aimed to assess whether shear stress values remained within acceptable limits. The table demonstrates that the allowable shaft stress values significantly exceed the shear stress values.

6.0 / SEALING AND LUBRICATION

6.1 / Sealing Specification

Oil seepage is avoided by using seals. The shafts extending out of the casing for the gear pump and the PTO require dynamic seals as there is relative rotating motion present at the shaft. Areas where there is no motion a static seal is used. Based on their capacity to retain oil, operating conditions and shaft speeds, the seals in Table 12 were chosen.

Table 12: Seals specification

Seal	Location	Material	Operating Conditions
Lip Seal (Dynamic)	Engine/Pinion shaft	Nitrile Rubber	-40°C to 100°C Max shaft speed of 17,000 revs/min
	Wheel shaft/Pump		
Gasket (Static)	Between casing halves	Fluoropolymer	-25°C to 200°C
	Gearbox/Engine interface	Nitrile Rubber	
	Gearbox/Pump interface	Nitrile Rubber	
O-ring (Static)	Oil inlet opening	Nitrile Rubber	-30°C to 100°C
	Oil level opening		
	Oil drain opening		

The chosen oil for the engine is the Special Tec AA 5W-20[8], which is well-matched with the seals, thus justifying the selection of O-rings for the oil opening. These seals are budget-friendly and have good resilience under high pressure.

6.2 / Lubrication Specification

Splash lubrication was used to ensure an oil depth of twice the tooth depth, provide a good splash for pinions and bearings, and to keep churning losses to a minimum. The pitch line velocity of 14.13 m/s was obtained from GP100 and used in Figure 9, then the dynamic viscosity from the graph was used in Equation 26. The ISO Grade of the lubrication was found to be 100 from using Figure 10.

$$v = \frac{\mu}{\rho} = \frac{0.09}{880} = \mathbf{102\ cSt} \quad (26)$$

Where:

$\boldsymbol{v}$ = Kinematic Viscosity, $\boldsymbol{\mu}$ = Dynamic Viscosity (Pa s) (From Figure 9), $\boldsymbol{\rho}$ = Density of oil (kg/m^3) (88% that of water)

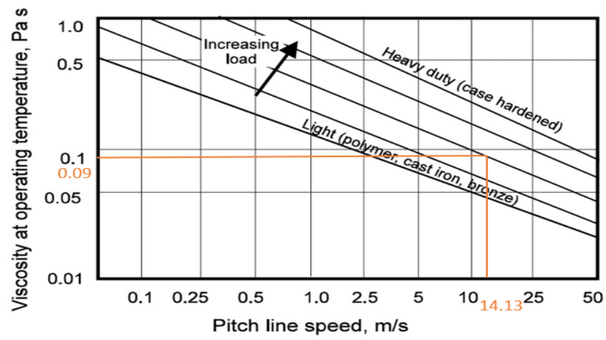


Figure 9: Pitch line viscosity graph[9]

ISO Viscosity grade	Kinematic viscosity center value 10^{-6}m²/s (cSt) (40°C)	Kinematic viscosity range 10^{-6}m²/s (cSt) (40°C)
ISO VG 2	2.2	More than 1.98 and less than 2.42
ISO VG 3	3.2	More than 2.88 and less than 3.52
ISO VG 5	4.6	More than 4.14 and less than 5.06
ISO VG 100	100	More than 90.0 and less than 110

Figure 10: ISO Grade table

2158K961 Hydraulic 100[10] was chosen as it is able to operate between temperatures of 70°C - 90°C and the specified ISO Grade. The properties are shown in Table 13.

Table 13: Lubricant specification

Lubricant	Type	ISO Grade	Viscosity, cSt at 40°C	Viscosity Index	Pour point (°C)	Flash Point (°C)
2158K961 Hydraulic 100	Paraffinic base oil with multiple additives (anti-foam, corrosion, rust)	100	100	97	-9	240

[8] https://www.liqui-moly.com/en/gb/special-tec-aa-5w-20-p000335.html#20792
[9] https://canvas.bham.ac.uk/courses/71238/pages/gear-lubrication
[10] https://www.mcmaster.com/products/iso-100-oil/

7.0 / MAINTENANCE SCHEDULE

Table 14: Maintenance Schedule

Area of Maintenance	Maintenance Procedure	Tools Required	Frequency	Operational hours
Inspect Oil Levels	Top up oil to ensure it is at the recommended levels to prevent friction and wear.	Oil gauge, cleaning cloth	Daily	8
Visual Inspection	Inspect for signs of leaks, loose fasteners or obvious damage (specifically housing and all connections).	Inspection light, magnifying glass	Daily	8
Grease Fittings	Lubricate grease fittings to prevent wear.	Grease gun, lubricant	Weekly	40
Inspect Breather Nut	Clean breather nut to maintain optimal temperature and avoid clogging.	Wrench	Weekly	40
Inspect Seals	Replace seals if there are signs of cracks, leaks or wear.	-	Weekly	40
Tighten Fasteners	Tighten fasteners, bolts and nuts to the manufacturing specifications to avoid misalignment.	Torque wrench, screwdrivers	Weekly	40
Change Oil	Replace lubricating oil and follow disposal procedure for old oil.	Oil drain pan, replacement oil	Monthly	160
Inspect Gears	Inspect for wear, pitting or damage and replace if necessary.	-	Monthly	160
Check Alignment	Verify alignment of PTO gear reduction unit with connected equipment and adjust where necessary.	Level	Quarterly	480
Inspect Ventilation	Ensure ventilation system is operating correctly, clean fan if there is debris present.	Cleaning brush	Quarterly	480
Replace Seals and Gaskets	Replace all seals and gaskets in case of wear.	Gasket scraper, wrench	Annually	1920
Replace Bearings	Replace bearings if there are signs of wear, cracking or if the life rating has been met.	Pliers	Annually	12000

Part 2: Design for Assembly

8.0 / BILL OF MATERIALS AND ASSEMBLED COMPONENTS

8.1 / Bills of Materials

The components for both designs are shown in Tables 15 and 16. The part numbers are presented in the exploded views in section 8.2, while the quantity is used to implement the Lucas Method in section 10. The cost and manufacturers of the components are in section 10, while the materials are shown in Figures A1 and A2 in the Appendix.

Table 15: Bills of Materials for Design 1

Part	Qty	Component Name
01	1	Case - Engine Half
02	1	Case - Pump Half
03	1	Case - Gasket Seal
04	2	Lip Seal
05	10	Casing Bolt
06	10	Casing Nut
07	20	Casing Washer
08	1	Pinion Shaft
09	1	Idler Shaft
10	1	Wheel Shaft
11	4	Shaft Nut
12	4	Shaft Washer
13	2	Pinion/Idler Gear
14	1	Wheel Gear
15	2	Pinion Bearing
16	2	Idler Bearing
17	2	Wheel Bearing
18	4	Pinion Spacer
19	4	Idler Spacer
20	4	Wheel Spacer
21	2	Female Coupling Sleeve
22	12	Mounting Bolt - Engine Side
23	16	Mounting Washer - Engine Side
24	4	Mounting Nut - Engine Side
25	12	Mounting Bolt - Pump Side
26	16	Mounting Washer - Pump Side
27	4	Mounting Nut - Pump Side
28	2	Mounting Interface - Engine Side
29	2	Mounting Interface - Pump Side
30	3	Oil Plug
31	3	Oil Plug Washer
32	1	Output Coupling Adapter
33	1	Pump Nut
34	1	Pump Washer

Table 16: Bills of Materials for Design 2

Part	Qty	Component Name
01	1	Case - Engine Half
02	1	Case - Pump Half
03	1	Case - Gasket Seal
04	1	Gasket Seal – Engine Side
05	1	Gasket Seal – Pump Side
06	2	Lip Seal
07	3	O-ring Seal
08	6	Casing Bolt
09	6	Casing Nut
10	12	Casing Washer
11	1	Pinion Shaft
12	1	Idler Shaft
13	1	Wheel Shaft
14	2	Pinion/Idler Gear
15	1	Wheel Gear
16	1	Pinion Bearing - Engine Side
17	1	Pinion Bearing - Pump Side
18	2	Idler Bearing
19	1	Wheel Bearing - Engine Side
20	1	Wheel Bearing - Pump Side
21	1	Pinion Spacer - Engine Side
22	1	Pinion Spacer - Pump Side
23	2	Idler Spacer
24	1	Wheel Spacer - Engine Side
25	1	Wheel Spacer - Pump Side
26	2	Female Coupling Sleeve
27	4	Mounting Bolt - Engine Side
28	4	Mounting Washer - Engine Side
29	4	Mounting Bolt - Pump Side
30	4	Mounting Washer - Pump Side
31	3	Oil Plug
32	3	Oil Plug Washer
33	1	Breather Nut
34	1	Breather Nut Washer
35	1	Output Coupling Adapter
36	1	Pump Nut
37	1	Pump Washer

8.2 / Exploded View

Figures 11 and 12 demonstrate how all the components come together within the PTO through an exploded view representation. The balloon numbers are in accordance with Tables 15 and 16 in section 8.1.

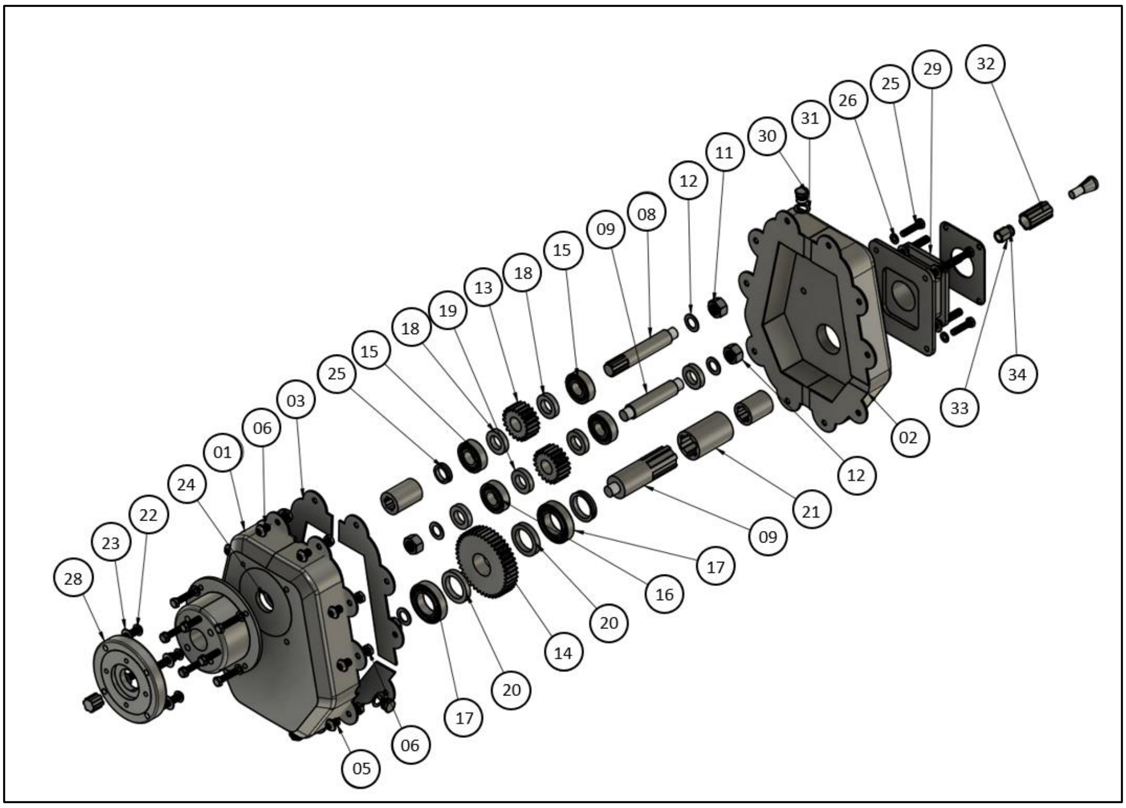


Figure 11: Exploded view of design 1

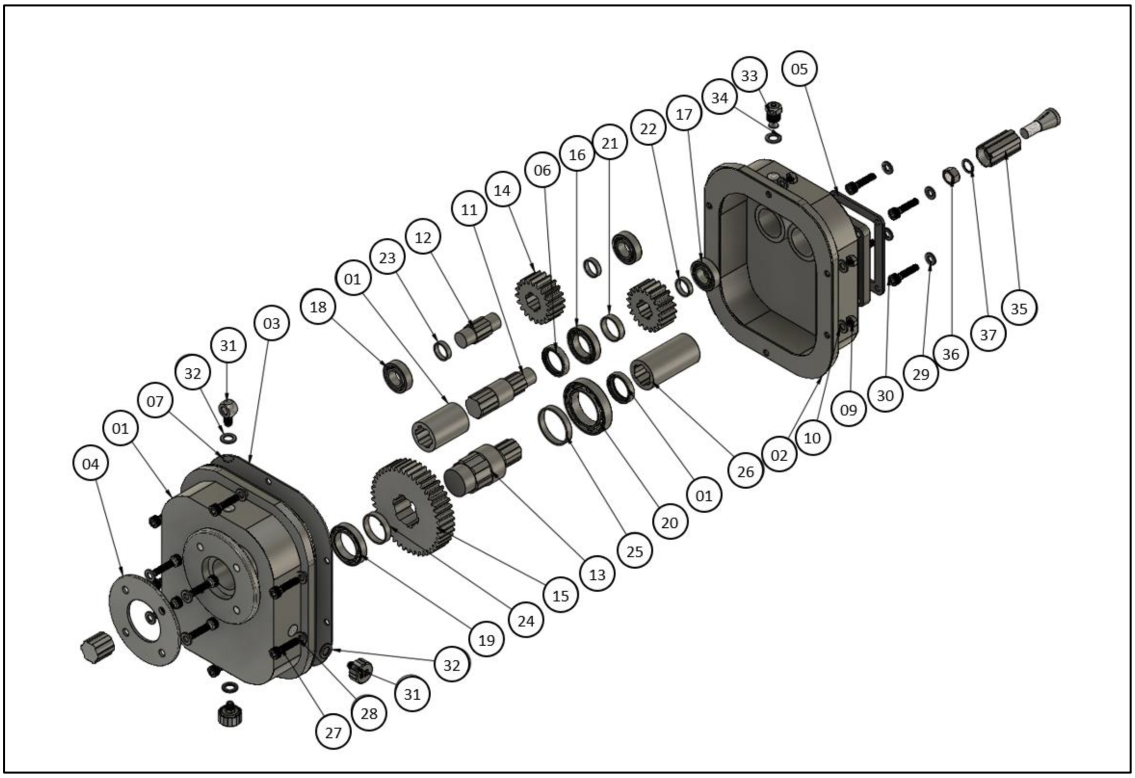
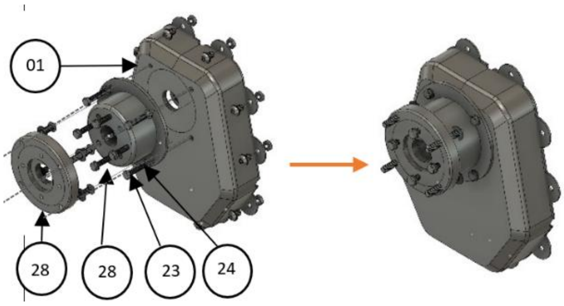


Figure 12: Exploded view of design 2

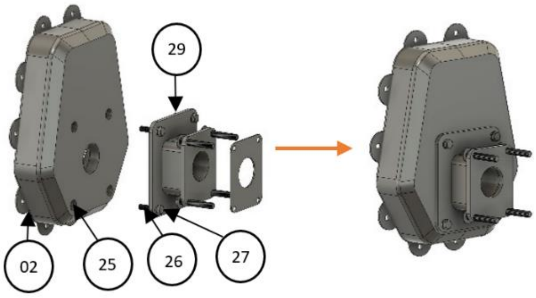
8.3 / Assembly Instructions

All parts can be referenced to using Tables 15 and 16.

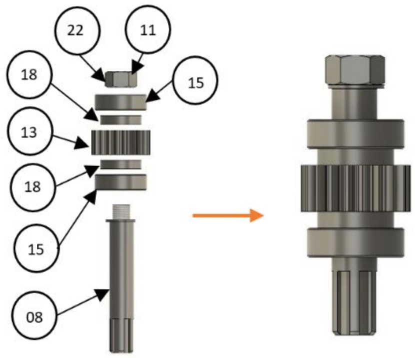
1 Attach part 28 to part 01 using parts 22, 23 and 24.



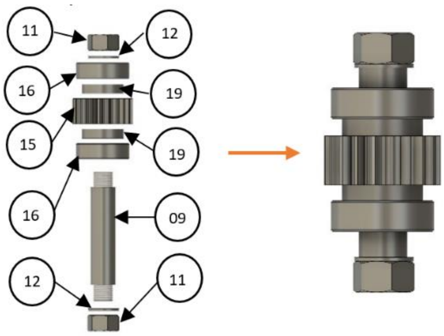
2 Attach part 29 to part 02 using parts 25, 26 and 27.



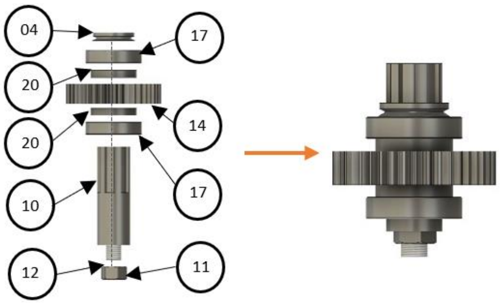
3 Slide parts 15, 18 and 13 onto the pinion shaft in the shown order. Slide on part 22 then screw on part 11.



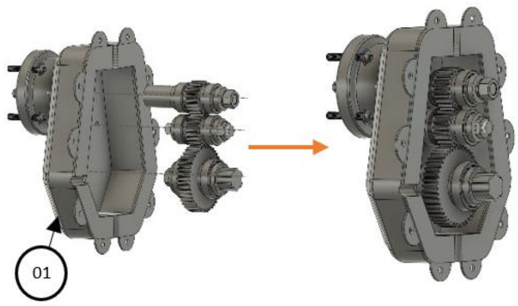
4 Slide parts 16, 19 and 15 onto the idler shaft in the shown order. Slide on both parts 12 then screw on parts 11 either side.



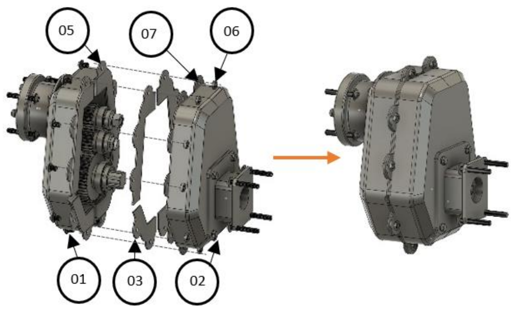
5 Slide parts 17, 20, 14 and 04 onto the wheel shaft in the shown order. Slide on part 12 then screw on part 11.



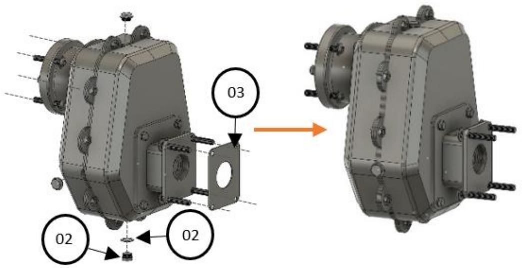
6 Insert the pinion shaft, idler shaft and wheel shaft into the engine case (part 01) in order respectively.



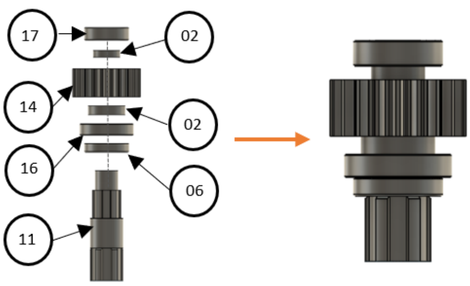
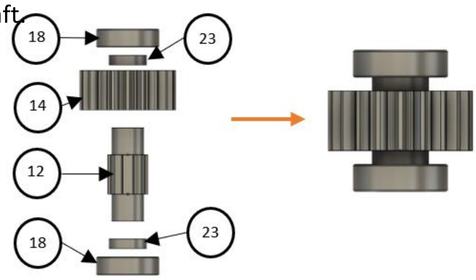
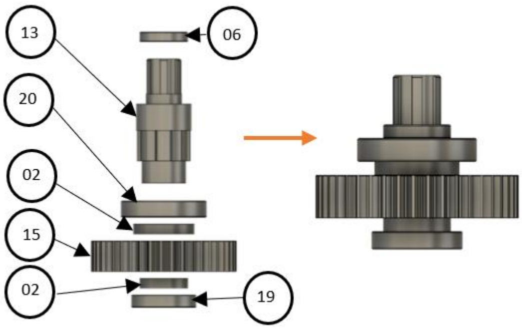
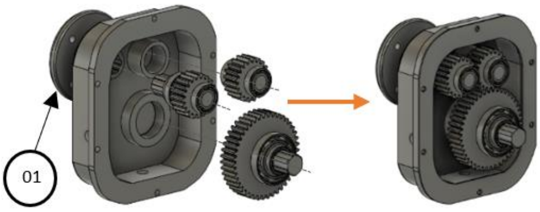
7 Lay the case gasket over the edges of the open casing and cover with the pump side of the case. Secure the case using parts 05, 06 and 07.



8 Final assembly is inserting parts 30 and 31 into the casing. Connect the PTO to the engine and pump either side using parts 22, 23 and 24. Mount the female coupling sleeves on the pump and engine interface.



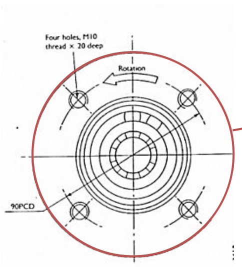
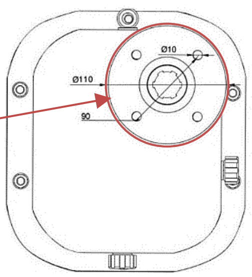
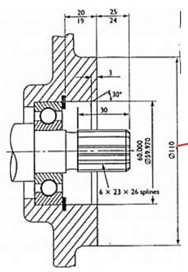
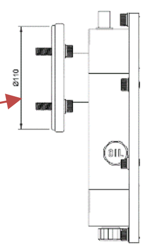
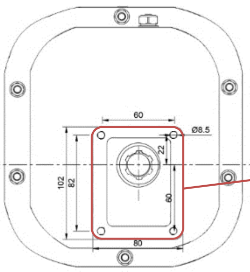
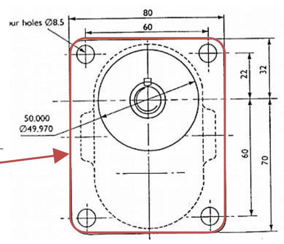
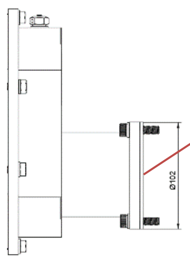
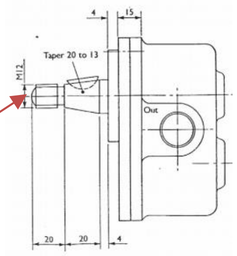
The following assembly instructions demonstrate the changes in assembly from design 1 to design 2.

1 Slide parts 06, 16, 02, 14, 02 and 17 onto the pinion shaft, part 11, in the shown order. 	**2** Slide part 14 onto the centre of part 12. Slide parts 23 and 18 either side of the idler shaft. 
3 Slide parts 13, 20, 02, 15 and 19 onto the wheel shaft, part 13, in the shown order and part 06 from the other side. 	**4** Insert the pinion shaft, idler shaft and wheel shaft into the engine case (part 01). The pinion being connected to the engine output coupling, the wheel connected at the bottom and the idler in between. 

Engine and Pump Interface Connection:
Table 17 details the interface connections between the engine, PTO unit and pump. The chosen interfaces are suitable for the systems' integrations for the following reasons:

1. **Standardisation and compatibility**: Dimensions and bolt threads match standard sizes which allows for easy assembly and replacement.
2. **Maintenance and serviceability:** Design allows for easy access to bolts and connection points.
3. **Alignment and tolerance:** Precise dimensions aid in proper alignment which increases longevity.

Table 17: Interface connections

Engine to PTO Interface Dimensions (Front View)	Engine to PTO Interface Dimensions (Side View)
 *Figure 13: Engine interface*  *Figure 14: PTO interface[11]*	 *Figure 15: Engine interface [11]*  *Figure 16: PTO interface*
PTO to Pump Interface Dimensions (Back view)	**PTO to Pump Interface Dimensions (Side View)**
 *Figure 17: PTO interface*  *Figure 18: Pump interface[11]*	 *Figure 19: PTO interface*  *Figure 20: Pump interface side[11]*

[11] https://canvas.bham.ac.uk/courses/71238/files/15580072?module_item_id=3439835

9.0 / DFA CONSIDERATIONS

9.1 / Sub-assembly

Table 18: Sub-assembly considerations

Component	Consideration
Shaft Arrangement	**1. Straight shaft** • Has reduced manufacturing costs and simplicity • Limited flexibility **2. Step down** • Improved power distribution • Speed reduction and torque increase
Bearings	**1. Plain bearing** • Low cost and easy to install • May not absorb shocks and vibrations as effectively as other types of bearings **2. Ball bearing** • Has increased efficiency and longer service life

Component	Consideration
Seals	**1. Circular gasket** • Expensive and are more prone to leaking **2. Flat seal** • Cost effective and can withstand higher mechanical stress, ensuring durability
Pump Mounting	**1. No Supports** • Increased vibration and reduced stability will occur which will shorten the equipment lifespan **2. Supports** • Supports can extend the life of the pump by reducing stress and strain on its components, resulting in a longer operational lifespan.

9.2 / General Assembly

Table 19: General assembly considerations

Component	Considerations
Gears	**1. Straight alignment** • Takes up more space so more material used **2. Triangular arrangement** • Compact for space optimisation

Component	Considerations
Housing	**1. Main Housing** • Protects internal components **2. Attach Seals and Gaskets** • Prevents leaks and contaminations

10.0 / DFA ANALYSIS

A DFA quantitative evaluation was performed using the Lucus method to provide a relative measure of assembly difficulty, involving four sequential steps.

Step 1: Design Efficiency (EI)

Components are categorized into Group A (essential) or Group B (non-essential) as shown in Table 20. The design efficiency percentage is calculated using Equation 27.

$$EI = \frac{A}{(A+B)} \times 100\% = \frac{52}{(52+102)} \times 100\% = 33\% \tag{27}$$

Step 2: Handling/Feeding Ratio

An assessment of component handling and insertion time considered size/ weight (A), handling complexity (B), orientation (C), and rotational symmetry(D), summarised in Table 20 and shown in Table A1 (Appendix). The casing and pump nut required more effort due to size and accessibility, respectively. Most parts had no handling issues, although some had challenging orientations. All parts had a rotational symmetry index of 0. The handling index for each part was calculated and used to determine the overall handling ratio.

$$Handling\ Ratio = \frac{Total\ Handling\ Index\ (HI)}{Number\ of\ Essential\ Components} = \frac{82.4}{52} = 1.5 \tag{28}$$

Step 3: Fitting Ratio

To calculate the fitting ratio, each component was scored on ease of fitting across six categories: placement and fastening (A), process direction (B), insertion (C), access (D), alignment (E), and insertion force(F). The fitting index (FI) totals these scores, presented in Table 20, and the fitting ratio is computed with Equation 29 using Table A1 data. Some parts require additional holding or multiple insertions, affecting their index. Most components score 0, indicating no difficulty, except for specific parts like the casing and pump nut that scored higher due to complexity in insertion and restricted access. Alignment and insertion resistance were generally not issues.

$$Fitting\ Ratio = \frac{Total\ Fitting\ Index\ (FI)}{Number\ of\ Essential\ Components} = \frac{245.5}{52} = 4.7 \quad (29)$$

Step 4: Calculating the cost of manufacturing each component

To calculate the Manufacturing Cost Index (M_i) first the material cost (M_c) and the relative cost (R_c) are calculated using Equation 30 and 31 with the values in Figure A1 in the Appendix. The results are then used in Equation 32 and recorded in Table 21.

$$M_c = V\ C_{mt}\ W_c \quad (30)$$
$$R_c = C_c\ C_{mp}\ C_s\ (C_t\ or\ C_f) \quad (31)$$
$$M_i = R_c\ P_c + M_c \quad (32)$$

Where:

$\mathbf{C_c}$ = complexity factor, $\mathbf{C_{mp}}$ = Material factor, $\mathbf{C_s}$ = Minimum section, $\mathbf{C_t}$ = tolerance factor or $\mathbf{C_f}$ = finish factor (whichever is greater), $\mathbf{P_c}$ = processing cost, **V** = volume (mm^3), $\mathbf{C_{mt}}$ = material cost and $\mathbf{W_c}$ = waste coefficient

The total cost of manufacturing these parts are **33743.48 cents.** The same approach was used for the second design to assess its level of improvement. The results were calculated below using Equations 27, 28 and 29.

Step 1: Design Efficiency (EI):

$$EI = \frac{49}{(49+32)} \times 100\% = 60\%$$

Step 2: Handling Ratio:

$$Handling\ Ratio = \frac{91}{49} = 1.85$$

Step 3: Fitting Ratio:

$$Fitting\ Ratio = \frac{219.1}{49} = 4.4$$

Step 4 is calculating cost of manufacturing each component. The total cost is **24813.09 cents.**

Table 20: Component indexes for designs 1 and 2

Design 1					Design 2				
Part	Qty	Grp.	HI	FI	Part	Qty	Grp.	HI	FI
Case - Engine Half	1	A	1.6	5.7	Case - Engine Half	1	A	1.6	5.7
Case - Pump Half	1	A	1.6	5.7	Case - Pump Half	1	A	1.6	5.7
Case - Gasket Seal	1	A	1.5	6.7	Case Gasket Seal	1	A	1.5	6.7
Gasket Seal – Engine Side	1	A	1.5	6.7	Gasket Seal – Engine Side	1	A	1.5	6.7
Gasket Seal – Pump Side	1	A	1.5	6.7	Gasket Seal – Pump Side	1	A	1.5	6.7
Lip Seal	2	A	1.5^2	2.3^2	Lip Seal	2	A	1.5^2	2.3^2
O-ring Seal	3	A	1.5^3	2.3^3	O-ring Seal	3	A	1.5^3	2.3^3
Casing Bolt	10	A	1.1^6	5^6	Casing Bolt	6	A	1.1^6	4.1^6
Casing Nut	10	A	1.1^6	5.1^6	Casing Nut	6	A	1.1^6	4.2^6
Casing Washer	20	B	1^{12}	3.4^{12}	Casing Washer	12	B	1^{12}	2.1^{12}
Pinion Shaft	1	A	1.1	4	Pinion Shaft	1	A	1.1	4
Idler Shaft	1	A	1.1	4	Idler Shaft	1	A	1.1	4
Wheel Shaft	1	A	1.1	4	Wheel Shaft	1	A	1.1	4
Pinion/Idler Gear	2	A	1^2	3.3^2	Pinion Gear	1	A	1	3.3
Wheel Gear	1	A	1	3.3	Idler Gear	1	A	1	3.3
Pinion Bearing - Engine Side	1	A	1	3.3	Wheel Gear	1	A	1	3.3
Pinion Bearing - Pump Side	1	A	1	1	Pinion Bearing – Engine Side	1	A	1	3.3
Idler Bearing	2	A	1^2	1^2	Pinion Bearing – Pump Side	1	A	1	1
Wheel Bearing - Engine Side	1	A	1	1	Idler Bearing	2	A	1^2	1^2
Wheel Bearing - Pump Side	1	A	1	1	Wheel Bearing – Engine Side	1	A	1	1
Pinion Spacer - Engine Side	1	B	1^2	2^2	Wheel Bearing – Pump Side	1	A	1	1
Pinion Spacer - Pump Side	1	B	1^2	2^2	Pinion Spacer – Engine Side	1	B	1	2
Idler Spacer	2	B	1^4	2^4	Pinion Spacer – Pump Side	1	B	1	2
Wheel Spacer - Engine Side	1	B	1	2	Idler Spacer	2	B	1^2	2^2
Wheel Spacer - Pump Side	1	B	1^2	2^2	Wheel Spacer –Engine Side	1	B	1	2
Female Splined Sleeve	2	A	1^2	1.7^2	Wheel Spacer – Pump Side	1	B	1	2
Mounting Bolt - Engine Side	4	A	1^4	2.1^4	Female Splined Sleeve	2	A	1^2	1.7^2

Mounting Washer – Engine Side	4	B	1^4	2.1^4	Mounting Bolt – Engine Side	4	A	1^4	2.1^4
Mounting Bolt - Pump Side	4	A	1^4	2.1^4	Mounting Washer - Engine Side	4	B	1^4	2.1^4
Mounting Washer - Pump Side	4	B	1^4	2.1^4	Mounting Bolt – Pump Side	4	A	1^4	2.1^4
Oil Plug	3	A	1.3^3	2.1^3	Mounting Washer - Pump Side	4	B	1^4	2.1^4
Oil Plug Washer	3	B	1^3	2.1^3	Oil Plug	3	A	1.3^3	2.1^3
Output Coupling Adapter	1	A	1.5	1.7	Oil Plug Washer	3	B	1^3	2.1^3
Pump Nut	1	A	2.2	4.1	Breather Nut	1	B	1	2
Pump Washer	1	B	1.7	3.5	Breather Nut Washer	1	B	1	2
-	-	-	-	-	Output Coupling Adapter	1	A	1.5	1.7
-	-	-	-	-	Pump Nut	1	A	2.2	4.1
Total	**-**	**-**	**82.4**	**245.5**	**Total**			**91**	**219.1**

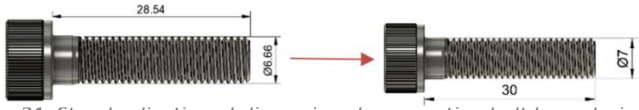
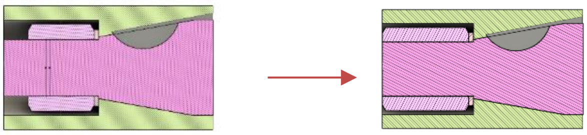
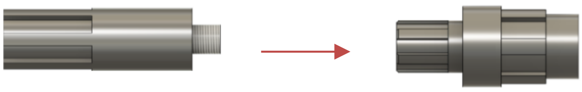
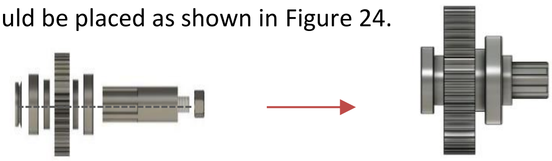
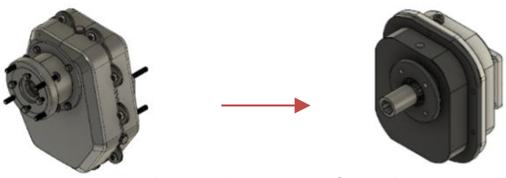
Table 21: Cost of manufacturing components

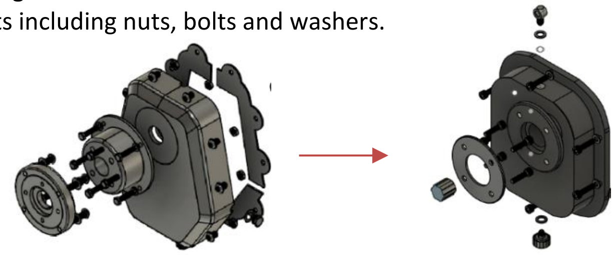
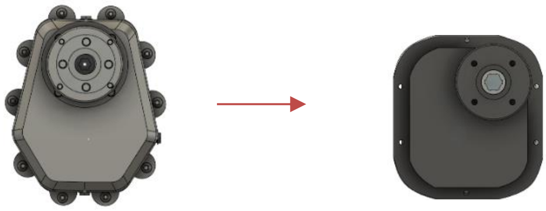
Design 1				Design 2			
Part	Material	Manufacturer	M_i	Part	Material	Manufacturer	M_i
Input shaft	Low C steel	-	109.2854	Input shaft	Low C steel	-	192.0581
Pinion gear	Alloy steel	-	168.4607	Pinion gear	Alloy steel	-	168.998
Idler shaft	Low C steel	-	95.12988	Idler shaft	Low C steel	-	145.7807
Idler gear	Alloy steel	-	168.4607	Idler gear	Alloy steel	-	171.75
Output shaft	Low C steel	-	219.3149	Output shaft	Low C steel	-	74.08288
Wheel gear	Alloy steel	-	783.192	Wheel gear	Alloy steel	-	780.44
Casing	Cast iron	-	13716	Casing	Cast iron	-	2357.08
Gasket	Rubber	-	33.29765	Gasket-case	Rubber	-	13.15375
Coupling sleeve	Al alloy	-	108.3085	Gasket-Engine side	Rubber	-	8.91945
Mounting Interface E	Cast iron	-	904.6847	Gasket- Pump side	Rubber	-	13.15375
Mounting Interface P	Cast iron	-	437.796	Coupling sleeve	Al alloy	-	108.7213
Output coupling	Al alloy	-	43.5441	Output coupling	Al alloy	-	43.9569
Lip Seal	Rubber	9562K76	298	Lip Seal	Rubber	1199N14	557
Casing Bolt	Stainless S	92095A319	129	O-ring Seal	Rubber	9262K105	10
Casing Nut	Stainless S	90592A024	16	Casing Bolt	Stainless S	92095A319	129
Casing Washer	c	98689A257	38	Casing Nut	Stainless S	90592A024	16
Shaft Nut	Stainless S	90592A024	16	Casing Washer	Stainless S	98689A257	38
Shaft Washer	Stainless S	98689A257	38	Pinion Bearing E	Aluminium	5972K246	1781
Pinion Bearing	Aluminium	5972K246	1781	Pinion Bearing P	Aluminium	5972k311	928
Idler Bearing	Aluminium	5972K246	1781	Idler Bearing	Aluminium	5972k311	928
Wheel Bearing	Aluminium	5972K248	2769	Wheel Bearing E	Aluminium	5972k217	2480
Pinion Spacer	Aluminium	93852A139	679	Wheel Bearing P	Aluminium	5972k249	3503
Idler Spacer	Aluminium	93852A139	0	Pinion Spacer E	Aluminium	96505A127	943
Wheel Spacer	Aluminium	93852A145	999	Pinion Spacer P	Aluminium	96505A122	800
Mounting Bolt E	Stainless S	94500A323	847	Idler Spacer	Aluminium	96505A122	800
Mounting Washer E	Stainless S	93475A260	10	Wheel Spacer E	Aluminium	96505A131	400
Mounting Nut E	Stainless S	90592A022	967	Wheel Spacer P	Aluminium	93475A370	503
Mounting Bolt P	Stainless S	94500A323	847	Mounting Bolt E	Stainless S	94500A323	847
Mounting Washer P	Stainless S	93475A260	10	Mounting Washer E	Stainless S	93475A260	10
Mounting Nut P	Stainless S	90592A022	967	Mounting Bolt P	Stainless S	94500A323	847
Oil Plug	Stainless S	7739K189	2351	Mounting Washer	Stainless S	93475A260	10
Oil Plug Washer	Stainless S	7739K189	2351	Oil Plug	Stainless S	7739K189	2351
Pump Nut	Stainless S	91423A160	47	Oil Plug Washer	Stainless S	7739K189	2351
Pump Washer	Stainless S	98689A118	15	Breather Nut	Stainless S	8631T11	441
-	-	-	-	Breather Nut Washer	Stainless S	93475A260	2351
Total Cost			**33743.48**				**24813.09**

Analysis of design 1:
The initial design aimed for a 60% efficiency but only achieved 33% using the Lucas method, signalling significant room for improvement. The feeding analysis showed parts handling and insertion times fell below the ideal threshold of 1.5, necessitating redesign. While aiming for a fitting ratio of 2.5, the outcome exceeded this target, indicating a need for adjustment. Simplifying the number of parts and reducing complexity is critical for enhancing manufacturability and cost-effectiveness.
Analysis of design 2:
For the second design, it achieved a design efficiency of 60%, which was a significant improvement compared to the initial design, indicating a successful reduction in part count. The feeding ratio increased, getting closer to the target of 2.5, and the fitting ratio decreased, also approaching the 2.5 target. These results demonstrated that the DFA implementations in the second design were successful. Furthermore, the total manufacturing cost for all components decreased in the second design, which was advantageous for improving the design overall.

11.0 / IMPROVEMENTS USING DFA

Table 22: DFA improvement techniques

DFA technique	Example of evidence in design
Reduction of parts	Reduced number of parts results in fewer steps required to assemble product. Parts have been reduced from a total of 154 to 81 from design 1 to 2.
Standardised Parts	Standard dimensions of parts are used to simplify inventory management and for ease of replacement.  *Figure 21: Standardisation of dimensions for mounting bolt from design 1 to 2*
Designing for tool access	Parts are easily accessible which enables ease of disassembly for maintenance and repair. Figure 22 shows the nut depth being increased to make it accessible without special tools.  *Figure 22: Difference in nut depth from design 1 to design 2*
Designing asymmetrical components	Ensures that parts can only be assembled in the correct way which increases the ease of assembly. Figure 23 demonstrates the shaft being changed from cylindrical to a step-down shaft for this purpose and to prevent components slipping off.  *Figure 23: Cylindrical to step-down shaft from design 1 to 2*
Modular house design	Sub-assemblies allow components to be categorised together to make smaller assemblies that can later be combined. The shaft was changed to have self-locating features such as varying diameters to make it easier to determine where each component should be placed as shown in Figure 24.  *Figure 24: Self-locating features on shaft and sub-assembly of components*
Colour coded components	Assists with reducing errors in assembly as parts are easy to identify. Figure 25 shows the casing colour being changed to make the input and output sides easily recognisable.  *Figure 25: Colour coding casing from design 1 to 2*

Welded components	Reduces the number of components that require assembly. Figure 26 illustrates how the mounting interfaces have been welded onto the casing to reduce the number of components including nuts, bolts and washers.  *Figure 26: Interfacing welded from design 1 to 2*
Design for ease of machinability	Reduces manufacturing cost and facilitates automation of product. The casing was altered to remove unique shapes and excessive cutting as shown in Figure 27.  *Figure 27: Simplified casing from design 1 to 2* *Figure x: Simplified casing*

12.0 / DESIGN FOR AUTOMATED ASSEMBLY

Table 23: Modifications for automated assembly

Modification	Location	Features	Benefit
Simplify Component Designs	Throughout design	Use standardised shapes and sizes for easy robot handling. Avoid custom tools or manual changes.	Faster assembly, enhanced robustness and improved quality.
Self-locating features	Casing, seals and shafts	Include components like snap-fit connectors, which automatically align and secure themselves, or peg-and-hole arrangements, where parts slot into place without manual intervention.	Reduced human error, faster assembly and improved accuracy.
Magnetic latching	Shafts, bearings and gears	Assist in aligning and securing gearbox gears, shafts, and bearings during assembly, ensuring precise positioning without manual adjustments.	Reduced error risk, no external power required and reversible.
Consistent tolerances	Throughout design	Gear Tooth Tolerances: Ensuring that the gear teeth have consistent shapes and dimensions. Shaft Diameters: Maintaining uniform diameters for input and output shafts.	Reduced rejection rates, higher assembly speed and lower material waste.
Error Detection and Correction	Throughout design	Sensors: Sensors monitor torque, pressure, and temperature. If values go outside the expected range, it signals a problem and prompts corrective actions. Force Feedback: Force sensors detect excessive force or resistance, indicating potential issues during assembly.	Improved product quality, enhanced reliability and reduced scarp.

Part 3: Feedback

Based on meetings with the professors and in-person tutorial sessions, feedback was received on the designs and calculations.

1. **Use a uniform thickness for the casing to ensure even loads are acting.**

Feedback: A casing with uniform thickness can provide a consistent structural integrity, ensuring that it can withstand loads without deformation. Even distribution of material can help prevent weak spots and stress concentrations that could lead to cracks or breaks under load.

2. **Avoid a blind spline for the shafts for ease of manufacturing.**

Feedback: A blind spline means that the grooves for the splines do not run through the entire length of the shaft and are closed at one end. This can make manufacturing harder because it can be difficult to machine.

3. **Enclose bearings of all shafts into the casing to reduce the number of parts.**

Feedback: Integrating the bearings directly into the casing can help in reducing the total number of parts required for the assembly. This can simplify the design, make the assembly process quicker, reduce potential alignment issues and make the overall system more compact.

4. **Use Pythagoras' Theorem to calculate the minimum distance between the Pinion and Wheel gears to optimise space.**

Feedback: Pythagoras' Theorem can be used in the design phase to accurately calculate the distances between gears in a right-angled triangular arrangement. This ensures that gears are neither too close (to avoid interference and excessive wear) nor too far apart (to avoid unnecessary use of space and potential misalignment), optimising the use of space within the unit.

5. **Avoid external rib supports as they are likely to break.**

Feedback: External rib supports, while they can add strength, may also become points of weakness, especially under dynamic loads or impact. They can act as stress risers where cracks initiate and propagate. Avoiding them in favour of internal reinforcements or a more robust casing design can lead to a smoother external surface.

Part 4: Appendix

Table A1: Design 1 indexes

Part	Qty	Group	A	B	C	D	**HI**	A	B	C	D	E	F	**FI**
Case - Engine Half	1	A	1.5	0	0.1	0	1.6	5	0	0.7	0	0	0	5.7
Case - Pump Half	1	A	1.5	0	0.1	0	1.6	5	0	0.7	0	0	0	5.7
Case - Gasket Seal	1	A	1	0	0.5	0	1.5	6	0	0.7	0	0	0	6.7
Gasket Seal – Engine Side	1	A	1	0	0.5	0	1.5	6	0	0.7	0	0	0	6.7
Gasket Seal – Pump Side	1	A	1	0	0.5	0	1.5	6	0	0.7	0	0	0	6.7
Lip Seal	2	A	1	0	0.5	0	1.5^{2}	2.3	0	0	0	0	0	2.3^{2}
O-ring Seal	3	A	1	0	0.5	0	1.5^{3}	2.3	0	0	0	0	0	2.3^{3}
Casing Bolt	10	A	1	0	0.1	0	1.1^{6}	5	0	0	0	0	0	5^{6}
Casing Nut	10	A	1	0	0.1	0	1.1^{6}	5	0.1	0	0	0	0	5.1^{6}
Casing Washer	20	B	1	0	0	0	1^{12}	3.3	0.1	0	0	0	0	3.4^{12}
Pinion Shaft	1	A	1	0	0.1	0	1.1	3.3	0	0.7	0	0	0	4
Idler Shaft	1	A	1	0	0.1	0	1.1	3.3	0	0.7	0	0	0	4
Wheel Shaft	1	A	1	0	0.1	0	1.1	3.3	0	0.7	0	0	0	4
Pinion/Idler Gear	2	A	1	0	0	0	1^{2}	3.3	0	0	0	0	0	3.3^{2}
Wheel Gear	1	A	1	0	0	0	1	3.3	0	0	0	0	0	3.3
Pinion Bearing - Engine Side	1	A	1	0	0	0	1	3.3	0	0	0	0	0	3.3
Pinion Bearing - Pump Side	1	A	1	0	0	0	1	1	0	0	0	0	0	1
Idler Bearing	2	A	1	0	0	0	1^{2}	1	0	0	0	0	0	1^{2}
Wheel Bearing - Engine Side	1	A	1	0	0	0	1	1	0	0	0	0	0	1
Wheel Bearing - Pump Side	1	A	1	0	0	0	1	1	0	0	0	0	0	1
Pinion Spacer - Engine Side	1	B	1	0	0	0	1^{2}	2	0	0	0	0	0	2^{2}
Pinion Spacer - Pump Side	1	B	1	0	0	0	1^{2}	2	0	0	0	0	0	2^{2}
Idler Spacer	2	B	1	0	0	0	1^{4}	2	0	0	0	0	0	2^{4}
Wheel Spacer - Engine Side	1	B	1	0	0	0	1	2	0	0	0	0	0	2
Wheel Spacer - Pump Side	1	B	1	0	0	0	1^{2}	2	0	0	0	0	0	2^{2}
Female Splined Sleeve	2	A	1	0	0	0	1^{2}	1	0	0.7	0	0	0	1.7^{2}
Mounting Bolt - Engine Side	4	A	1	0	0	0	1^{4}	2	0.1	0	0	0	0	2.1^{4}
Mounting Washer - Engine Side	4	B	1	0	0	0	1^{4}	2	0.1	0	0	0	0	2.1^{4}

Mounting Bolt - Pump Side	4	A	1	0	0	0	1^4	2	0.1	0	0	0	0	2.1^4
Mounting Washer - Pump Side	4	B	1	0	0	0	1^4	2	0.1	0	0	0	0	2.1^4
Oil Plug	3	A	1	0.2	0.1	0	1.3^3	2	0.1	0	0	0	0	2.1^3
Oil Plug Washer	3	B	1	0	0	0	1^3	2	0.1	0	0	0	0	2.1^3
Output Coupling Adapter	1	A	1	0	0.1	0	1.5	1	0	0.7	0	0	0	1.7
Pump Nut	1	A	1.5	0.7	0	0	2.2	2	0	0	1.5	0	0.6	4.1
Pump Washer	1	B	1	0.7	0	0	1.7	2	0	0	1.5	0	0	3.5
Total							**82.4**							**245.5**

Table A2: Design 2 indexes

Part	Qty	Group	A	B	C	D	**HI**	A	B	C	D	E	F	**FI**
Case - Engine Half	1	A	1.5	0	0.1	0	1.6	5	0	0.7	0	0	0	5.7
Case - Pump Half	1	A	1.5	0	0.1	0	1.6	5	0	0.7	0	0	0	5.7
Case Gasket Seal	1	A	1	0	0.5	0	1.5	6	0	0.7	0	0	0	6.7
Gasket Seal – Engine Side	1	A	1	0	0.5	0	1.5	6	0	0.7	0	0	0	6.7
Gasket Seal – Pump Side	1	A	1	0	0.5	0	1.5	6	0	0.7	0	0	0	6.7
Lip Seal	2	A	1	0	0.5	0	1.5^2	2.3	0	0	0	0	0	2.3^2
O-ring Seal	3	A	1	0	0.5	0	1.5^3	2.3	0	0	0	0	0	2.3^3
Casing Bolt	6	A	1	0	0.1	0	1.1^6	5	0	0	0	0	0	5^6
Casing Nut	6	A	1	0	0.1	0	1.1^6	5	0.1	0	0	0	0	5.1^6
Casing Washer	12	B	1	0	0	0	1^{12}	3.3	0.1	0	0	0	0	3.4^{12}
Pinion Shaft	1	A	1	0	0.1	0	1.1	3.3	0	0.7	0	0	0	4
Idler Shaft	1	A	1	0	0.1	0	1.1	3.3	0	0.7	0	0	0	4
Wheel Shaft	1	A	1	0	0.1	0	1.1	3.3	0	0.7	0	0	0	4
Pinion Gear	1	A	1	0	0	0	1	3.3	0	0	0	0	0	3.3
Idler Gear	1	A	1	0	0	0	1	3.3	0	0	0	0	0	3.3
Wheel Gear	1	A	1	0	0	0	1	3.3	0	0	0	0	0	3.3
Pinion Bearing – Engine Side	1	A	1	0	0	0	1	3.3	0	0	0	0	0	3.3
Pinion Bearing – Pump Side	1	A	1	0	0	0	1	1	0	0	0	0	0	1
Idler Bearing	2	A	1	0	0	0	1^2	1	0	0	0	0	0	1^2
Wheel Bearing – Engine Side	1	A	1	0	0	0	1	1	0	0	0	0	0	1
Wheel Bearing – Pump Side	1	A	1	0	0	0	1	1	0	0	0	0	0	1
Pinion Spacer – Engine Side	1	B	1	0	0	0	1	2	0	0	0	0	0	2

Pinion Spacer – Pump Side	1	B	1	0	0	0	1	2	0	0	0	0	0	2
Idler Spacer	2	B	1	0	0	0	1^2	2	0	0	0	0	0	2^2
Wheel Spacer – Engine Side	1	B	1	0	0	0	1	2	0	0	0	0	0	2
Wheel Spacer – Pump Side	1	B	1	0	0	0	1	2	0	0	0	0	0	2
Female Splined Sleeve	2	A	1	0	0	0	1^2	1	0	0.7	0	0	0	1.7^2
Mounting Bolt – Engine Side	4	A	1	0	0	0	1^4	2	0.1	0	0	0	0	2.1^4
Mounting Washer - Engine Side	4	B	1	0	0	0	1^4	2	0.1	0	0	0	0	2.1^4
Mounting Bolt – Pump Side	4	A	1	0	0	0	1^4	2	0.1	0	0	0	0	2.1^4
Mounting Washer - Pump Side	4	B	1	0	0	0	1^4	2	0.1	0	0	0	0	2.1^4
Oil Plug	3	A	1	0.2	0.1	0	1.3^3	2	0.1	0	0	0	0	2.1^3
Oil Plug Washer	3	B	1	0	0	0	1^3	2	0.1	0	0	0	0	2.1^3
Breather Nut	1	B	1	0	0	0	1	2	0	0	0	0	0	2
Breather Nut Washer	1	B	1	0	0	0	1	2	0	0	0	0	0	2
Output Coupling Adapter	1	A	1	0	0.1	0	1.5	1	0	0.7	0	0	0	1.7
Pump Nut	1	A	1.5	0.7	0	0	2.2	2	0	0	1.5	0	0.6	4.1
Pump Washer	1	B	1	0.7	0	0	1.7	2	0	0	1.5	0	0	3.5
Total							**91**							**245.5**

Part	Group	Material	Process	C_c	C_{mp}	C_S	C_t	C_f	R_c	V	C_{mt}	W_c	M_c	P_c	Manufacturer	Mi
Input shaft	A4	Low carbon steel	Machine	5.3	1.4	1	2.2	1.2	8.904	46064	0.00068	3	93.97056	1.72		109.2854
Pinion gear	A2	Alloy steel	Machine	1.2	2.5	1	1.2	1.2	3.6	31326	0.00259	2	162.26868	1.72		168.4607
Idler shaft	A4	Low carbon steel	Machine	5.3	1.4	1	2.2	1.2	8.904	39125	0.00068	3	79.815	1.72		95.12988
Idler gear	A2	Alloy steel	Machine	1.2	2.5	1	1.2	1.2	3.6	31326	0.00259	2	162.26868	1.72		168.4607
Output shaft	A4	Low carbon steel	Machine	5.3	1.4	1	2.2	1.2	8.904	100000	0.00068	3	204	1.72		219.3149
Wheel gear	A2	Alloy steel	Machine	1.2	2.5	1	1.2	1.2	3.6	150000	0.00259	2	777	1.72		783.192
Casing	B3	Cast iron	Sand Cast	2.6	1.2	1	1.9	2.3	7.176	4633333	0.00105	2.8	13621.99902	13.1		13716
Gasket	A1	Rubber	Plastic mould	1	1.1	1	1	1	1.1	89479	0.00035	1	31.31765	1.8		33.29765
Female coupling sleeve	A2	aluminum alloy	Machine	1.2	1	1	1.2	1	1.2	21861	0.00243	2	106.24446	1.72		108.3085
Mounting interface- Engine side	A4	Cast iron	Sand cast	1.8	1.2	1	1.9	1	2.16	642043	0.00105	1.3	876.388695	13.1		904.6847
Mounting interface- Pump side	A4	Cast iron	Sand cast	1.8	1.2	1	1.9	1	2.16	300000	0.00105	1.3	409.5	13.1		437.796
Output coupling adapter	A2	Aluminum alloy	Machine	1.2	1	1	1.2	1	1.2	8535	0.00243	2	41.4801	1.72		43.5441
Lip Seal									0				0		9562K76	298
Casing Bolt									0				0		92095A319	129
Casing Nut									0				0		90592A024	16
Casing Washer									0				0		98689A257	38
Shaft Nut									0				0		90592A024	16
Shaft Washer									0				0		98689A257	38
Pinion Bearing									0				0		5972K246	1781
Idler Bearing									0				0		5972K246	1781
Wheel Bearing									0				0		5972K248	2769
Pinion Spacer									0				0		93852A139	679
Idler Spacer									0				0		93852A139	0
Wheel Spacer									0				0		93852A145	999
Mounting Bolt - Engine Side									0				0		94500A323	847
Mounting Washer - Engine Side									0				0		93475A260	10
Mounting Nut - Engine Side									0				0		90592A022	967
Mounting Bolt - Pump Side									0				0		94500A323	847
Mounting Washer - Pump Side									0				0		93475A260	10
Mounting Nut - Pump Side									0				0		90592A022	967
Oil Plug									0				0		7739K189	2351
Oil Plug Washer									0				0		7739K189	2351
Pump Nut									0				0		91423A160	47
Pump Washer									0				0		98689A118	15
																33743.48

Figure A1: Cost of manufacturing of design 1

Part	Group	Material	Process	C_c	C_{mp}	C_S	C_t	C_f	R_c	V	C_{mt}	W_c	M_c	P_c	Manufacturer	Mi
Input shaft	A4	Low carbon steel	Machine	5.3	1.4	1	2.2	1.2	16.3	80403	0.00068	3	164.02212	1.72		192.0581
Pinion gear	A2	Alloy steel	Machine	1.2	2.5	1	1.2	1.2	2	31961	0.00259	2	165.55798	1.72		168.998
Idler shaft	A4	Low carbon steel	Machine	5.3	1.4	1	2.2	1.2	16.3	57718	0.00068	3	117.74472	1.72		145.7807
Idler gear	A2	Alloy steel	Machine	1.2	2.5	1	1.2	1.2	3.6	31961	0.00259	2	165.55798	1.72		171.75
Output shaft	A4	Low carbon steel	Machine	5.3	1.4	1	2.2	1.2	16.3	22572	0.00068	3	46.04688	1.72		74.08288
Wheel gear	A2	Alloy steel	Machine	1.2	2.5	1	1.2	1.2	2	150000	0.00259	2	777	1.72		780.44
Casing	A3	Cast iron	Sand Cast	1.3	1.2	1	1.9	2.3	6.8	1800000	0.00105	1.2	2268	13.1		2357.08
Gasket-case	A1	Rubber	Plastic mould	1	1.1	1	1	1	1.1	31925	0.00035	1	11.17375	1.8		13.15375
Gasket- engine side	A1	Rubber	Plastic mould	1	1.1	1	1	1	1.1	19827	0.00035	1	6.93945	1.8		8.91945
Gasket- Pump side	A1	Rubber	Plastic mould	1	1.1	1	1	1	1.1	31925	0.00035	1	11.17375	1.8		13.15375
Female coupling sleeve	A2	Aluminum alloy	Machine	1.2	1	1	1.2	1	1.44	21861	0.00243	2	106.24446	1.72		108.7213
output coupling adapter	A2	Aluminum alloy	Machine	1.2	1	1	1.2	1	1.44	8535	0.00243	2	41.4801	1.72		43.9569
Lip Seal															1199N14	557
O-ring Seal															9262K105	10
Casing Bolt															McMaster 92095	129
Casing Nut															Mc Master 9059:	16
Casing Washer															98689A257	38
Pinion Bearing - Engine Side															5972K246	1781
Pinion Bearing - Pump Side															5972k311	928
Idler Bearing															5972k311	928
Wheel Bearing - Engine Side															5972k217	2480
Wheel Bearing - Pump Side															5972k249	3503
Pinion Spacer - Engine Side															96505A127	943
Pinion Spacer - Pump Side															96505A122	800
Idler Spacer															96505A122	800
Wheel Spacer - Engine Side															96505A131	400
Wheel Spacer - Pump Side															93475A370	503
Mounting Bolt - Engine Side															94500A323	847
Mounting Washer - Engine Side															93475A260	10
Mounting Bolt - Pump Side															94500A323	847
Mounting Washer - Pump Side															93475A260	10
Oil Plug															7739K189	2351
Oil Plug Washer															7739K189	2351
Breather Nut															8631T11	441
Breather Nut Washer																
Pump Nut															91423A160	47
Pump Washer															98689A118	15
Total																**24813.09**

Figure A2: Cost of manufacturing of design 2