University of Birmingham

Department of Mechanical Engineering

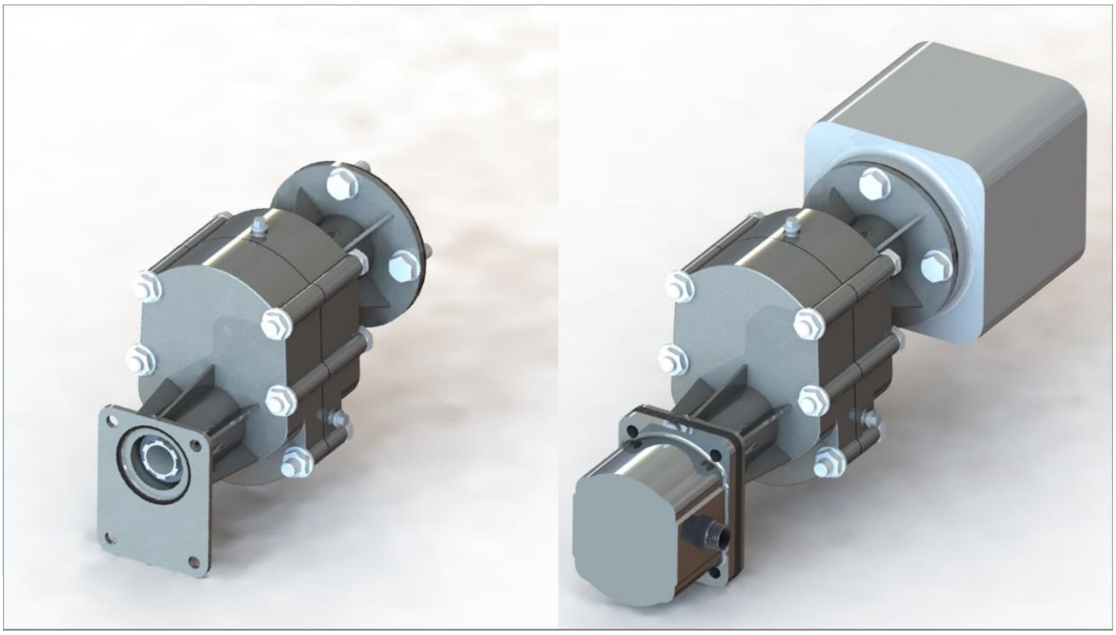
Power Take-Off Rear Reduction Unit

Mechanical Design B (04-22964)

Module Coordinator: Professor K D Dearn

Transmission Group 2

Student Name (ID): Vinush Vigneswaran (1690302), Yang Zhang (1726733), Weidong Liang (1975439)



1. AIMS & OBJECTIVES

Aim: The predominant aim of this project is to design a Power-Take Off (PTO) reduction unit which can be easily fitted to a PTO shaft of a 45kW industrial diesel engine, and interfaced to drive a hydraulic pump, to produce 25 l/min of oil.

Objectives:

- To produce a gearbox unit to decrease output shaft speed.
- To consider power transmission by implementing couplings.
- To produce a PTO gear reduction unit which can be easily assembled, sufficiently light to be manually handled, with low maintenance (lubrication).
- To consider manufacturing costs and time constraints for a batch production of 10,000 units per year.

***NOTE:** Assumptions and Key Results from Calculations are highlighted in RED*

2. PRODUCT DESIGN SPECIFICATION (PDS)

***Table 1:** Simplified Product Design Specification*

Parameters	Values
Gearbox speed reduction	2800 to 1300 RPM (293.2 to 136.1 rad/s)
Gear pump power	5 kW
Product Life	Gears: 20,000 hours, Shaft: infinite life and other components: 12,000 hours
Gear Ratio	2:1
Gear Type	External Spur Gears
Pinion & Idler Material	Carbon steel, AISI 1022, normalized (EN32)
Wheel Material	Low alloy steel, AISI 4340, oil quenched & tempered at 650°C (EN24 HARDING)
Overall Dimensions (LWH)	204 x 117 x 194 mm
Weight	Light weight (< 16 kg)
Lubrication	Splash lubrication
Coupling	Female splined coupling for input & output shaft. Additional male-splined coupling required for gear pump tapered & threaded end.
Casing	Casing must cover all dynamic components
Manufacturability	Standard parts and minimise secondary processes. Chamfers for safety of manual handling.
Batch Production Units	10,000 per year
Standards	British Standards (BS 8888 and BS 5686:1986)
Cost	Low cost manufacturing and assembly procedure
Maintenance	Limited maintenance required
Assembly	Precise location for each connecting part. Dynamic components are located and protected. Easy sub-assemblies to reduce assembly procedure.

3. GEAR DESIGN & SPECIFICATION

Assumption: Calculation were conducted at a constant pump speed of 1300 rpm to obtain a flow rate at 25 bar. Hence any deviation from the input engine speed, means the stated flow rate cannot be achieved. Tolerance was set to 0 for GP100 calculations.

3.1. Gear Ratio: The gears are designed to reduce the speed from the engine to the pump by a factor of 2.15, as shown by equation 1. The pump speed of 136.1 rad/s will provide a flow rate of 25 liters/min of oil at 150 bar pressure.

$$(\boldsymbol{Eqn\ 1}) \qquad G_R = \frac{\omega_{engine}}{\omega_{pump}} = \frac{293.2}{136.1} = \mathbf{2.15\ (2\ s.f.)}$$

3.2. Gear Type & Configuration: Spur gears were used for this product due to the ease of manufacturing, assembly and high-power transmission efficiency [1]. They are also compact and easy to install/replace. The most common disadvantage of spur gears, are that they are noisy at higher speeds [1], this can be reduced by considering secondary processes such as grinding, to provide a “fine” surface finish [2].The output and input shaft rotate anticlockwise (three gear configuration).

3.3. GP100 Results: GP100 is a software that outputs design results for external gears based on constraints and objective parameters, from Table 1. The following results were obtained:

Pinion teeth : 20 Wheel teeth : 43
Normal module : 2.000 mm
Gear ratio : 2.1500 Error : 0.0000

Spur gear

Centre dist. : 63.00 mm Error : 0.00 mm
Centre distance extension : 0.000 mm

Facewidth : 24.90 mm : 12.45 x module
Reasonable FW : 12.57 to 26.00 mm
Operating pressure angle : 20.0 deg
Facewidth ratio of pinion and wheel wear : 0.40
Facewidth ratio of pinion and wheel strength : 0.88

** Wheel wear governs facewidth
** Facewidth REASONABLE

	Pinion	Wheel
Material	En32 mild:	En24 AS:H&
Number of teeth	20	43
P.C.D.	40.00 mm	86.00 mm
Outside Dia	45.07 mm	88.93 mm
Root Dia	36.07 mm	79.93 mm
Base Dia	37.59 mm	80.81 mm
Addendum	2.53 mm	1.47 mm
Dedendum	1.97 mm	3.03 mm
Profile shift	0.2674	-0.2674
Pitch line vel	5.86 m/s	-5.86 m/s
Contact Ratio	1.60	1.60
Speed	2800.0 revs/min	1302.3 revs/min
Torque	-17.05 N-m	36.65 N-m
Safety Factor	19.69	28.64
Tang Force	-852 N	852 N
Radial Force	-310 N	310 N
Axial Force	0 N	0 N

Figure 1: *GP100 external gear design results.*

---

[1] *Petry-Johnson, T.T., Kahraman, A., Anderson, N.E. and Chase, D.R., 2008. An experimental investigation of spur gear efficiency. Journal of Mechanical Design, 130(6).*
[2] *Borner, J. and Houser, D.R., 1996. Friction and bending moments as gear noise excitations. SAE transactions, 105, pp.1669-1676.*

4. BEARING RATINGS *(FBD and force calculations in Section 5.4)*

The bearing (**SKF Bearings**) has to meet the lifetime requirement (**12,000 hrs**), geometric restrictions of the shaft (**min diameter and length of shafts**), as well as the loading conditions, factor of safety and rotation speed, as shown in the table below.

Assumptions: The weight of sleeves and seals were not included in the weight of the bearing load. Rotational friction were neglected, and assumed to be perfectly lubricated. The maximum rotational speed was taken to be 2800 RPM. The elastic deformation and deflection (due to moments) in the bearing will be neglected.

***NOTE:** An example is shown (in grey) for Input Shaft Bearing (refer to table 3).*

***Table 2:** Parameters required for bearing calculations.*

Parameter	Input (pinon)	Middle (idler)	Output (wheel)
F_r, Radial Loads (KN)	0.453	0.785	0.453
N, Rotation speeds (rpm)	2800	2800	1300
L_{10h}, Minimum lifetime (hrs)	12,000	12,000	12,000

Equation 2 and 3 were used to calculate the required minimum dynamic load rating of the bearings. Equation 2 calculates the static equivalent load (P_{or}). Refer to table 3 for summarised bearing calculation results.

$$(\boldsymbol{Eqn\ 2}) \qquad P_{or} = X \cdot F_r + Y \cdot F_a, \qquad\qquad P_{or} = (1)(453.3) + 0 = 453.3\ N$$

Where,

- $\boldsymbol{P_{or}}$ *is the static equivalent load (KN),*
- $\boldsymbol{F_a}$ *is the axial loads (KN)*
- ***e** is the life index (3 for ball bearing)*
- $\boldsymbol{F_r}$ *is the radial loads (KN),*
- ***X** and **Y** are the calculations factors, when* $\frac{F_a}{F_r} \le e,\ X = 1, Y = 0.$

Equation 3 calculates the bearing life rating [3] in hours (L_{10h}).

$$(\boldsymbol{Eqn\ 3}) \quad L_{10h} = \frac{\left(C/P_{or}\right)^e \times 10^6}{60 \times N} \quad [3]$$

Where,

- $\boldsymbol{L_{10h}}$ *is rating life (hours),*
- $\boldsymbol{P_{or}}$ *is the static equivalent load rating of the bearing*
- $\boldsymbol{C}$ *is the dynamic load rating of the bearing*
- $\boldsymbol{N}$ *is the rotational speed (rpm)*
- ***e** is the life index (3 for ball bearing)*

The example below shows the rearranged formula for C. The bearing's dynamic load rating must be greater than the minimum dynamic load calculated, C (i.e. **> 5.74 KN**).

$$(using\ Eqn\ 3)\ C_{min} = P_{or} \times \left(\frac{L_{10h} \times 60 \times N \times 10^6}{10^6}\right)^{\frac{1}{3}}$$

$$= (453.3) \cdot \left(\frac{(12000) \times 60 \times (2800) \times 10^6}{10^6}\right)^{\frac{1}{3}} = \boldsymbol{5.74\ KN}$$

---

[3] 2009. Ball And Roller Bearings. 2nd ed. [ebook] US: NTN Corporation, pp.14-24. Available at: <http://www.ntnamericas.com/en/website/documents/brochures-and-literature/catalogs/ntn_2202-ixe.pdf> [Accessed 11 March 2020]

The validation process of the bearing selections was also conducted. The safety factors (***FoS***) calculations were using the ratios between bearings' static load ratings divided by the static equivalent loads [4]:

$$(\boldsymbol{Eqn\ 4}) \qquad Fos = \frac{P}{P_{or}}\ [4], \quad Fos = \frac{(2{,}850)}{453} = \mathbf{6.3}$$

Where,

- $\boldsymbol{P}$ *is the static load rating of the selected product.*
- $\boldsymbol{P_{or}}$ *is the static equivalent load of the requirement*.

Comments: The safety factor is high due to the stepped shaft design, whereby larger bearings are required to satisfy the bore diameter. Based on the dynamic load ratings and minimum diameter of different shafts, the bearings were selected from the SKF data sheet and tabulated below:

Table 3: *Summary of Bearing selection and calculations*

Shafts	Minimum Dynamic load ratings, C_{min} (KN)	Static equivalent loads, $\boldsymbol{P_{or}}$ (KN)	Bearing Part Number	Dynamic load rating, $\boldsymbol{C}$ (KN)	Static load rating, $\boldsymbol{P}$ (KN)	*FoS*	Product Code from SKF
Input shaft	**> 5.726**	0.453	9	**5.85**	2.85	**6.3**	*SKF-16002*
			8	**6.37**	3.65	**8.1**	*61904-2RS1*
Middle shaft	**> 9.917**	0.785	10	**10.6**	6.2	**7.9**	*4201 ATN9*
			11	**14.8**	9.5	**12.1**	*4203 ATN9*
Output shaft	**> 4.434**	0.453	9	**5.85**	2.85	**6.3**	*SKF-16002*
			8	**6.37**	3.65	**8.1**	*61904-2RS1*

5. SHAFT DESIGN & ANALYSIS

5.1. Analysis & Calculation Assumptions:

- Ignore the weight of gear, shaft, bearing, collar and coupling.
- The shaft is simply support by two bearings at each end (pure bending).
- The shaft is under steady static load and can be assume as point load.
- Any mechanical losses is negligible.

NOTE: *An example (in grey) is shown for the input shaft.*

5.2. Material Selection

Comments: The AISI 1137 carbon steel was chosen for the shaft, which is widely considered as a practical material is used in shaft [5]. The main reason of using this material is its relatively low density and price, also it is

Table 4: *Properties of AISI 1137 Carbon Steel*

Property	Value
Young's modulus, *E*	205 *GPa*
Yield strength, $\boldsymbol{S_y}$	535 *MPa*
Shear modulus, *G*	84 *GPa*
Tensile strength, $\boldsymbol{S_u}$	720 *MPa*
Fatigue strength @ $\mathbf{10^7}$ cycles	*MPa*

[4] 2005. Shaft And Bearing Calculation. FLYGT ITT Industries.
[5] Borcherding, G., Newberg, B. and Dohogne, L., Emerson Electric Co, 2005. Motor having a high carbon shaft and powder metal bearing and method of using and manufacturing thereof. U.S. Patent Application 10/832,706.

suitable for high tensile application and cyclic loading [5]. The property of material is shown below in Table 4.

5.3. Free Body Diagram of Gear Load

Figure 2 shows that the pinion and idler are subjected to **tangential** (F_t) and **radial** (F_r) forces, calculated using equation 5 and 6 [6]. The idler will experience both driving force of the pinion and reaction force of the wheel, therefore the total force in each axis can be calculated using equation 7 and 8. Resultant force can be calculated using equation 9.

Where,
T is the torque be transmitted
D is the diameter of gear
$\emptyset$ is the pressure angle equals 20°

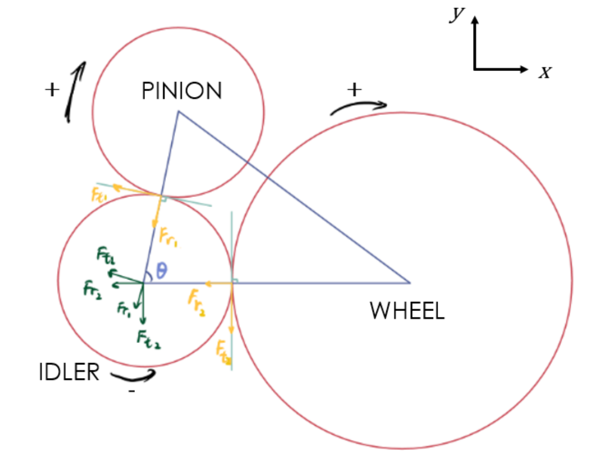


Figure 2: Tangential and radial forces

$$(\boldsymbol{Eqn\ 5})\quad F_t = \frac{2 \cdot T}{D}[6]\,, \qquad (\boldsymbol{Eqn\ 6})\ F_r = F_t \cdot tan\emptyset\ \ [6]$$

The x and y-direction forces acting on the shaft can be found using equation 7 and 8[6].

$$(\boldsymbol{Eqn\ 7})\quad F_x = F_{rw} + F_{rp}cos\theta + F_{tp}sin\theta \qquad (\boldsymbol{Eqn\ 8})\quad F_y = F_{tw} + F_{rp}sin\theta - F_{tp}cos\theta$$

Where, $\boldsymbol{F_{rw}}$ is wheel radial force, $\boldsymbol{F_{rp}}$ is pinion radial force, $\boldsymbol{F_{tw}}$ is wheel tangential force, $\boldsymbol{F_{tp}}$ is pinion tangential force, θ is the angle between pinion and wheel.

The total resultant force acting on the shafts can be calculated using equation 5, summarised in table 5. Where the tangential and radial forces were obtained from GP100 results – figure 1.

$$(\boldsymbol{Eqn\ 9})\quad F_{resultant} = \sqrt{{F_x}^2 + {F_y}^2}, \quad F_{resultant} = \sqrt{(852.68)^2 + (310.35)^2} = 907\ N$$

Table 5: Summary of forces acting on the shaft due to gears

Parameter	Input shaft	Middle shaft	Output shaft
F_t , Tangential force (N)	852.68	1202.89	852.68
F_r , Radial force (N)	310.35	1009.34	310.35
$\boldsymbol{F_{resultant}}$, Total resultant force (N)	906.64	1570.26	906.64
Torque (N.m)	**17.05**	**17.05**	**36.73**

5.4. Shaft FBD & Bearing Force Calculation

Based on the free body diagram shown in Figure 2, the bearing load can be calculated using static equilibrium theory, in which the sum of all forces equal to 0 (equation 10), and the moment about a point end equals to 0 (equation 11). The reaction forces of the shaft can be obtained using equation 10 and 11.

---

[6] *Dearn. 2018. Learning Pack: Shaft Design And Associated Technology. 1st ed. pp.1-17.*

$$(\boldsymbol{Eqn\ 10})\quad \sum F_y = 0\ , \qquad\qquad (\boldsymbol{Eqn\ 11})\quad \sum M = 0,$$

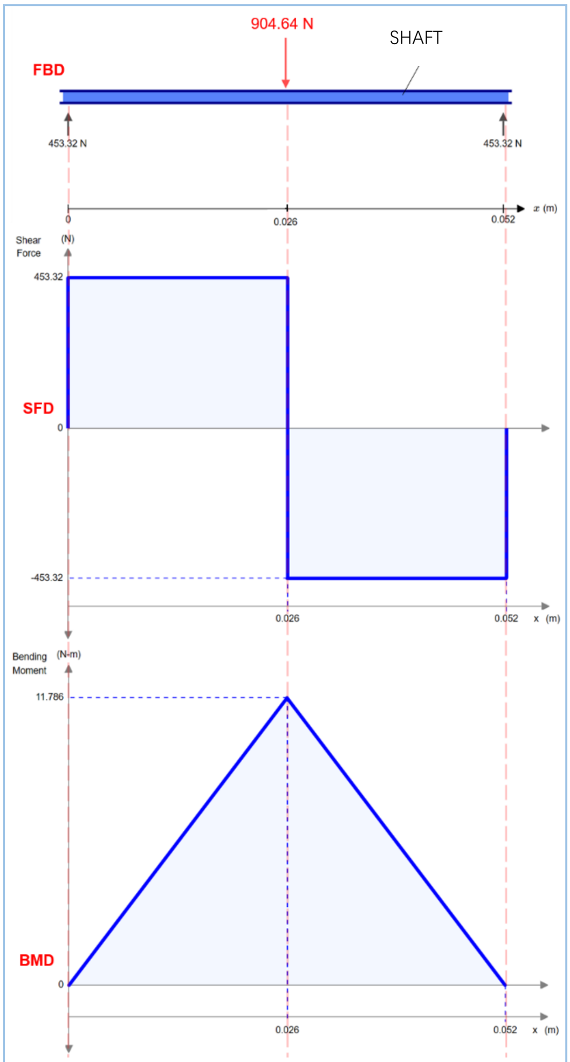


Figure 3: *Free body diagram, shear force diagram and bending moment diagram for the input shaft.*

Input: $\begin{cases} B_1 + B_2 + F_{resultant} = 0 \\ B_1(X_1 + X_2) + B_2X_2 = 0 \end{cases} \begin{cases} B_1 = 453.32\ N \\ B_2 = 453.32\ N \end{cases}$

Middle: $\begin{cases} B_1 + B_2 + F_{resultant} = 0 \\ B_1(X_1 + X_2) + B_2X_2 = 0 \end{cases} \begin{cases} B_1 = 785.13\ N \\ B_2 = 785.13\ N \end{cases}$

Output: $\begin{cases} B_1 + B_2 + F_{resultant} = 0 \\ B_1(X_1 + X_2) + B_2X_2 = 0 \end{cases} \begin{cases} B_1 = 453.32\ N \\ B_2 = 453.32\ N \end{cases}$

B_1 and B_2 are the reaction forces (bearing positions).

The shear force along the shaft can be obtained as shown in figure 3. The bending moment along the beam was calculated using equation 12. Figure 3 shows the bending moment diagram. The **maximum moment** along the shaft was found (for input shaft).

$$(\boldsymbol{Eqn\ 11})\qquad M = F_{Shear} \times d$$

$$M = (453.32) \times (0.026) = \boldsymbol{11.79\ N.m}$$

Where, F_{Shear} is the shear force, d is the distance.

Table 5*: Summary of maximum shear force and maximum bending moment.*

	Input Shaft	Middle Shaft	Output shaft
Maximum Shear Force (N)	453.32	785.13	453.32
Maximum Bending Moment (N.m)	11.79	20.41	11.79

5.5. Shaft Minimum Diameter (MSST)

The Maximum shear stress theory (MSST)[7] was used to predict the failure of the shaft. The factor of safety was 2 (general condition). The formula for calculating the minimal shaft diameter was given by equation 12. The results are summarised in Table 6.

$$(\boldsymbol{Eqn\ 12})\qquad d_{min} = \left(\frac{32f_s}{\pi\sigma_Y}\sqrt{M^2 + T^2}\right)^{\frac{1}{3}}\quad [7]$$

Where, $\boldsymbol{f_s}$ is the factor of safety, $\boldsymbol{\sigma_Y}$ is the yield stress of material (see table 5), $\boldsymbol{M}$ is the maximum moment along the beam, $\boldsymbol{T}$ is the torque be transmitted by the shaft.

---

[7] *Dearn. 2018. Learning Pack: Variable Loading - Designing Against Fatigue. 1st ed. pp.1-23.*

Table 6: Calculation using equation 12 of minimum shaft diameter using MSST

Input shaft	Middle shaft	Output shaft
$\boldsymbol{d_{min}} = \left(\frac{\mathbf{32 \times 2}}{\boldsymbol{\pi * 535 * 10^6}}\sqrt{\mathbf{11.79^2 + 17.05^2}}$ $\boldsymbol{= 9.24\ mm}$	$d_{min} = \left(\frac{32 \times 2}{\pi * 535 * 10^6}\sqrt{20.41^2 + 17.05^2}\right)^{\frac{1}{3}}$ $\boldsymbol{= 10.04\ mm}$	$d_{min} = \left(\frac{32 \times 2}{\pi * 535 * 10^6}\sqrt{11.79^2 + 36.73^2}\right)^{\frac{1}{3}}$ $\boldsymbol{= 11.37\ mm}$

5.6. Bending and Torsional Deflection

The bending deflection of the shaft is essential for shaft analysis, an excessive deflection can hinder the shaft's function. The deflection of the shaft is subject to the position of the gear and the Young's modulus of the materials. In this design situation, the bending deflection can be calculated using Equation 13. The result are summarized in Table 7.

$$(\boldsymbol{Eqn\ 13}) \qquad \delta_{max} = \frac{PL^3}{48EI} \qquad\qquad (\boldsymbol{Eqn\ 14}) \qquad \theta_{max} = \frac{TL}{GJ}$$

Where for equation 13, P is the load on shaft, L is the length of the shaft, E is the Young's modulus, $I = \frac{\pi d^4}{64}$ is the second moment of area. For equation 14, T is the torque applied on shaft, G is the shear modulus, $J = \frac{\pi d^4}{32}$.

Table 7: Calculation using equation 13 of vertical deflection on shaft

Input shaft	Middle shaft	Output shaft
$\boldsymbol{\delta_{max}} = \frac{\mathbf{906.64 * 0.052^3}}{\mathbf{48 * 205 * 10^9} * \frac{\boldsymbol{\pi * 0.00924^4}}{\mathbf{64}}}$ $\boldsymbol{= 3.62 \times 10^{-2}\ mm}$	$\delta_{max} = \frac{1570.26 * 0.052^3}{48 * 205 * 10^9 * \frac{\pi * 0.01004^4}{64}}$ $\boldsymbol{= 4.49 \times 10^{-2}\ mm}$	$\delta_{max} = \frac{906.64 * 0.052^3}{48 * 205 * 10^9 * \frac{\pi * 0.011374^4}{64}}$ $\boldsymbol{= 1.58 \times 10^{-2}\ mm}$

The torsional deflection of the shaft can be calculated using eqn 14, shown in table 8.
Assumption: Constant torsion was applied on a uniform circular cross section with minimum diameter.

Table 8: Calculation using equation 14 of torsional deflection on shaft

Input shaft	Middle shaft	Output shaft
$\boldsymbol{\theta_{max}} = \frac{\mathbf{17.05}}{\mathbf{84 * 10^9} * \frac{\boldsymbol{\pi * 0.015^4}}{\mathbf{32}}} \boldsymbol{= 0.041 rad}$ $\boldsymbol{= 2.35°}$	$\theta_{max} = \frac{17.05}{84 * 10^9 * \frac{\pi * 0.015^4}{32}} = 0.041\ rad$ $\boldsymbol{= 2.35°}$	$\theta_{max} = \frac{36.73}{84 * 10^9 * \frac{\pi * 0.0165^4}{32}} = 0.052\ rad$ $\boldsymbol{= 2.97°}$

Comment(s): From the calculation result, the angle of twist of each shaft is < 3° per meter, which reasonable for general applications [7].

5.7. Critical Speed

The critical speed for single mass load shaft can be calculated by Equation 15. Calculations shown in table 9.

$$(\boldsymbol{Eqn\ 15}) \qquad n_c = \frac{30}{\pi}\sqrt{\frac{g}{\delta_{max}}}$$

***Table 9**: Calculation using equation 15 of shaft critical speed*

Input shaft	Middle shaft	Output shaft
$n_c = \frac{30}{\pi}\sqrt{\frac{9.81}{3.62\times10^{-2}}}$ $= \mathbf{4972\ rpm}$	$n_c = \frac{30}{\pi}\sqrt{\frac{9.81}{4.49\times10^{-2}}}$ $= \mathbf{4461\ rpm}$	$n_c = \frac{30}{\pi}\sqrt{\frac{9.81}{1.58\times10^{-2}}}$ $= \mathbf{7523\ rpm}$

Comment(s): According to Ritz [8], the maximum operational speed of the shaft must remain under 75% of the theoretical critical speed. The maximum operational speed of pinion and middle shaft '2800 rpm', and the wheel shaft '1600 rpm' were 56.3% and 21.3 % of the critical speed. Both were under 75% limit hence no change was made to diameter.

5.8. Fatigue Analysis

Fatigue causes majority of failure in the mechanical field, it is necessary to evaluate the component's fatigue life. The Soderberg criterion [7] (Equation 15) was used to calculate the factor of safety under a completely reversed bending moment and a steady torque.

$$(\mathbf{Eqn\ 15}) \qquad f_s = \frac{s_{sy}}{\left[\frac{1}{4}\left[\left(\frac{S_y}{S_e}\right)\sigma_r\right]^2 + [\tau_m]^2\right]^{\frac{1}{2}}} [6] \qquad (\mathbf{Eqn\ 16}) \qquad S_e{'} = K_a K_b K_c K_d K_e K_m S_e$$

Where, s_{s_y} is the yield strength in shear, S_y is the yield strength, S_e is the endurance strength, σ_r is the reverse stress, τ_m is the mean shear stress.

Comment(s): In order to get the more accurate analysis result, the revised endurance strength was introduced, which take other external effect factor into consideration, the formula is equation 16. The parameters for calculation is summarised in Table 10.

***Table 10**: Parameters for Factor of Safety Calculation*

Parameters	Description	Input	Middle	Output
Yield strength in shear, s_{s_y}	Generally for ductile material, $s_{s_y} = 0.5S_y$	267.5 MPa	267.5 MPa	267.5 MPa
Mean Shear Stress, τ_m	$\tau_m = \frac{\tau_{max} + \tau_{min}}{2}$	12.86 MPa	12.86 MPa	34.09 MPa
Reverse Stress, σ_r	$\sigma_r = \frac{\sigma_{max} - \sigma_{min}}{2}$	17.79 MPa	30.80 MPa	21.88 MPa
Surface Factor, K_a	The shaft is machined, and tensile strength is 590.	0.75	0.75	0.75
Size Factor, K_b	$K_b = 1.189d^{-0.097}$ for 8 < d ≤250mm	1.79	1.79	1.80
Reliability, K_c	$K_c = 1 - 0.08Z_r$, for 99% reliability, $Z_r = 2.326$	0.814	0.814	0.814
Temperature factor, K_d	The shaft generally run in room temperature, $K_d = 1$, for $T \leq 450°$	1	1	1
Stress concentration modifying factor, K_e	$K_e = 1/K_f$, K_f is stress concentration factor, which is obtained base on the geometry of shaft	0.625	0.625	0.625
Miscellaneous Effects, K_m	The shaft is running in gear oil, where the corrosion will reduced around 30% lifetime [1].	0.7	0.7	0.7
endurance strength, S_e	$S_e = 0.5S_u$, for steel which tensile strength< 1400MPa	300 MPa	300 MPa	300 MPa
Revised endurance strength, $S_e{'}$	Using equation 16 to calculated	**143.4 MPa**	**143.4 MPa**	**144.2 MPa**
Factor of safety, f_s		**7.52**	**4.54**	**5.05**

[8] *Maday, C.J., 1974. A class of minimum weight shafts.*

Comment(s): The factor of safety is more than 1, which means all shafts meet the requirement of infinite life.

5.9. Coupling Torque Transmission: Spline Validation

The splines were used to transmit the torque, and it was required do not failure due to the shear and compressible stress, the calculation of factor of safety is shown below.

$$(\boldsymbol{Eqn\ 17}) \qquad \tau = \frac{16 \cdot T \cdot K_s}{\pi \cdot {D_i}^3} \qquad\qquad (\boldsymbol{Eqn\ 18}) \qquad \sigma_c = \frac{T \cdot K_s}{n \cdot d \cdot L_e \cdot r}$$

Table 11: Specification and Calculation for Spline

Parameters	Description	Input to gear	Output to gear	Middle to gear	Input to hub	Output to hub
Spline Designation	ISO standard straight sided spline, sourced from RoyMech[9]	6×16×20	6×16×20	6×13×16	6×23×26	6×23×26
Torque transmitted, T		17.05	36.73	17.05	17.05	36.73
Number of teeth, n		6	6	6	6	6
Reduced diameter, D_i		16	16	13	23	23
Depth of spline, d	$d = (D - D_i)/2 - 0.6$	1.2	1.2	0.9	0.9	0.9
Effective length, L_e		0.02	0.02	0.02	0.052	0.052
Mean radius of spline, r	$r = (D_i + D)/4$	9	9	7.25	12.25	12.25
Service factor, K_s	For fixed spline, $K_s = \frac{K_{application} \times K_{design}}{K_{fatigue}}$. The value of each facto was determined based on Roymech. $K_{application} = 2.2$, $K_{design} = 1.0$, $K_{fatigue} = 0.4$	5.5	5.5	5.5	5.5	5.5
Shear stress, τ	Calculate from equation 13.	61.05	131.52	113.82	20.55	44.28
Factor of Safety	$FOS_{shear} = s_{s_y}/\tau$	**4.38**	**2.03**	**2.35**	**13.02**	**6.04**
compressible stress , σ_c	Calculate from equation 14.	72.36	155.88	119.76	27.26	58.73
Factor of Safety	$FOS_c = s_y/\sigma_c$	**7.39**	**3.43**	**4.47**	**19.62**	**9.11**

Comment(s): Through the calculation, the factor of safety is bigger than 2, which mean the spine will not fail.

6. SEALING AND LUBRICATION

6.1. Lip-seals selection:

Table 12: *Lip-seals selected from SKF.*

Counter-faces	Lip Seal Config	Designation	Material	Supl.
Input Shaft lip-seal	Straight	22×32×7 HMS5 RG	Nitrile Rubber	***SKF***
Output Shaft lip-seal	Straight	22×32×7 HMS5 RG	Nitrile Rubber	***SKF***

Table 13: *Dynamic sealing (lip-seals) operation & conditions.*

Conditions	Statements
Shafts surface-finish	Well surface finish (Ra)
Shafts diameters (mm)	20 (Input & output)
(Eqn 19) Circumferential operation-speed (m/s), V	**2.930** (Input) **1.360** (Output)
Operating pressure (MPa)	Very low (≈0)
Lubrication type	Splash lubrication
Orientation	Horizontal

---

[9] *Roy, B., n.d. ISO Straight Sided Spline. [online] Roymech.org. Available at: <https://roymech.org/Useful_Tables/Keyways/Spline.html> [Accessed 8 February 2020].*

Table 12 shows the selected lip-seals. The lip-seals were selected to provide sealing in both ends of the gearbox (input end & output end) which rotate at a speed of 2800 rpm for input and 1300 rpm for the output. Based on the designs of the shafts as well as casing, the operating conditions of the dynamic lip-seals are shown in table 13. The circumferential operation-speeds (V) [10] was used to select the lip-seal (eqn 19).

$$(\boldsymbol{Eqn\ 19}) \qquad V = \frac{N \times 2\pi}{60} \times R$$

Where, N, the rotational speed of the shaft (input-2800 rpm; output-1300 rpm) and R, shafts radius (meter).

6.2. Lubrication:

Three lubrication methods were considered: splash lubrication, grease lubrication and forced oil circulation. The pitch line velocity of the gears is 5.86 m/s (see Figure 1). This is within the tangential velocity range for the splash lubrication. For a horizontal shaft the lubricant level must be between 1h and 3h [11], where h is the tooth depth. The static level of the lubricant is shown below, where L is the lubrication level.

$$Tooth\ depth,\ h = 4.5\ mm\ \therefore\ 4.5\ mm\ < L\ < 13.5\ mm$$

Comment(s): For a pinion speed of 2800 RPM at 5 kW, the lubricant must have an ISO viscosity grade of ISO VG 68. Hence Mulplus DX68 from Eneos was chosen [11].

7. FEEDBACK

Some example of the formative feedback provided by Professor K Dearn and PGTAs provided has been summarised in the table 14.

Table 14: *Feedback and Implementation.*

Feedback Provided	Implementation of Feedback
Bearing holder shoulder should not come in contact with the dynamic part of the bearing.	Used SKF data sheet to find the recommended shoulder height for the specific bearing to implement in design.
Material for the pinion and wheel must be different due to lifetime specification.	Ensured the pinion material specification allows for twice the lifetime of wheel material.
Use BS8888 standards when drawing bolts.	Altered drawing to show thread as specified on BS8888 (solid line).
Change external rib structure for manufacture.	Changed angled ribs to straight ribs on one side of the casing.
Ensure static and dynamic components have clearance.	Changed washer size on pump hub to ensure it does not interfere with static components.
Sharp edges on the casing.	Removed/ design change (e.g. fillets) were implemented on the casing corners.
How can you make the assembly simple and symmetric?	Used same shape for main casing geometry. Exact same shaft and hub used for input and output.
Coupling is too complicated to manufacture.	Used sub-components, internally splined main hub and externally splined pump interface.
Gaskets do not need to be manufactured.	Changed from in-house manufacturing to purchasing from suppliers (cheaper option).

[10] *SKF, 2018. Rolling Bearings.*
[11] *KHK Gears. 2015. Lubrication Of Gears | KHK Gears. [online] Available at: <https://khkgears.net/new/gear_knowledge/gear_technical_reference/lubrication-of-gears.html> [Accessed 2 April 2020].*