Design Report

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Group 13

This report details the full design procedure with appropriate references to the added respective appendices. The methods with design choices and assumptions are contained within this report.

Work Completed:	Description:	Member(s) Responsible:	Checked by:
CAD	This includes all CAD drawings, with annotations, parts list, orthographic projections, and dimensions	*Emeale *Matthew	*Michael *Will-Mari
Keyways, Retaining Rings, Seals, Machining (locations values)	The design, research, calculations, and appropriate selection of the keyways, retaining rings and seals. Research on machining applications, location for design and respective values	*Will-Mari	*Matthew *Michael
Bearings,	Design, research, calculations, and selection from the bearings catalogue and research on correct welding dimensions and symbols	*Michael *Matthew	*Cross-Checked
Shafts	The calculations, dimensions, material selection, shoulders, fillets, bending and deflection	*Matthew *Michael	*Cross-Checked
Iteration (Shafts)	Checking, correcting, and adjusting for perfection	*Matthew *Will-Mari	
Gears	The calculations of the pitting/contact and bending stresses using the AGMA equations and research into the material selection	*Matthew *Michael *Will-Mari *Emeale	*Cross-Checked
Lubrication	Research on best lubrication method, application and implementation within the design and lubrication type	*Matthew	*Group (Meeting)
Corrosion Protection	Protecting casing from corrosion by finding correct protection method and application to design	*Emeale	*Matthew
Casing	Designing and iteration of design for the most optimum design and practicality	*Matthew *Will-Mari	*Emeale
Compilation	The compiling of each of the reports and ordering of the SMATH documents into correct format	*Michael *Matthew	*Group before each hand-in
Minutes	Keeping minutes	*Michael	*Matthew

Signatures

MA Blackwell: MABlackwell

W Bezuidenhoudt:

MS De Lange:

EP Maritz

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MS De Lange:

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Team charter

Code of conduct:
All team members have all agreed upon the following:

Teamwork

Treat all members of the team with respect. Do not tolerate intimidation or harassment. Treat all members fairly and do not tolerate discrimination. Include all team members in important decisions. Compromising on different ideas.

Commitment

Each team member is expected to be on time for scheduled meetings. All members need to complete their assigned work by the agreed date. Each member has a responsibility to the team to perform their tasks to their fullest ability. Be present and engaged.

Diversity

Respect the different cultures and languages in the team. Do not discriminate against gender. Decide not to give or take offence.

Integrity

Honesty amongst each other. Do not commit plagiarism.

Member agreement:

All members agree to be contactable to between working hours.
All members agree to attend all group meetings.
In the event of a member missing a meeting notice has to be given at least 1hr prior to the meeting.
Meeting minutes must be disseminated at least 1 day after the meeting. The final version of the minutes will be sent out in pdf format to avoid alterations.
All meeting minutes must be signed. This will take the form of a physical signature or agreement via email that the attached minutes are correct.
All members will wash their hands before an in-person meeting.

Table of Contents

List of Figures

List of Tables

Abstract

This report details the design of a two-stage marine gearbox with a speed reduction ratio of 2.5:1 to be fitted to a light cargo marine vessel. It starts by defining the engineering requirements and specifications to be adhered to in the designing process. This is followed by a brief explanation how an initial design was chosen. Thereafter, each component/aspect of the gearbox is discussed, and explanation is given to design choices and methodology behind the selection process of items such as material and sizing of components. The report concludes with a systematic comparison between the engineering requirements and the final design to prove that the design meets and exceeds all requirements and expectations.

1. Introduction

Due to the latest CO_2 emissions legislation, a light cargo vessel with 32 feet in length requires a new custom 2.5:1 reduction gearbox to run with the recently fitted low emissions diesel engine. The design process is represented below with consideration taken to every detail. The requirements of the customer were used as a basis for the core design. This report evaluates the extent to which the final design satisfies these requirements. The report also serves as sufficient proof that the concept will function as desired with all the necessary calculations, specifications, and drawings to back it up. In addition, it provides evidence of the suitability of the product and its reliability. The numerous features included in the gearbox design sets the design apart from its competitors with added improvements to maintainability, ease of service and safety.

2. Engineering Requirements

Below is a table containing a collection of all the engineering requirements that the gearbox design must adhere to.

Requirement description	**Required Value**	**Research (Typical Values)**	**References**
Basic gearbox requirements			
Shaft alignment	input and output shafts co-linear	Design dependent (DD)	-
Rotation direction	input same as output	DD	-
Gear type	helical gears	DD	(Bredell, 2020) (Boyce, 2020)
Geometry	symmetrical about vertical longitudinal plane	DD	-
Gear ratio	2.5:1	3:1 or higher	Neeson, J. D. 2009, December 11. (J.D.Neeson, 2009)
Construction & layout	two-stage	DD	-
Torque fluctuation	5%	Case Specific	(J.B.Woodward, 1973)
Duty cycle and design life	2 hours per day, 7 days a week for 10 years.	DD	-
Gear fabrication parameters			
Normal pressure angle	20°	14.5°, 20°, 25°	(G.Budynas & Nisbett, 2015)
Normal modules	2, 3, 4, 5 (mm)	2, 3, 4, 5 (mm)	Project (Bredell, 2020)
Helix angles	20°, 25°, 30°	5° - 45°	(Boyce, 2020)

Hardness	300-350 (HB)	24-40 (Rc)	Appendix C, Table 11
AGMA Quality	7	DD	-
Casing			
Assembly process	welded assembly	DD	-
Material	laser cut S355JR steel plates.	Cast iron FT25	(Masson Marine, 2005) (Bloch, 2007)
Corrosion protection	Specified by designer	DD	-
Interface surface	rigid inboard steel mounting rails	DD	-
Interfaces			
Input shaft			
Shaft alignment	Input Shaft must align with the Engine output shaft	DD	-
Shaft connection	coupling	coupling	-
Output shaft			
Shaft alignment	gearbox output shaft interfaces with propeller shaft	DD	-
Diameter	propeller shaft and gearbox output shaft have same diameter	25 – 100 mm	(Bloch, 2007) (Engineering ToolBox, 2004)
Load capacity (forward)	13.7 kN		-
Load capacity (reverse)	13.7kN x 10%		-
Reliability and safety			-
Fatigue design reliability	95%		-
Safety factor	1.5	1.5 - 2	(Engineering ToolBox, 2004)

Table 1: List of Engineering requirements, with values and references.

3. Concept generation and selection

Methodology and Procedure

As a point of departure, it was decided that each member would come up with his/her design concept and present this concept to the group in the following meeting (Appendix – G, meeting minutes). After the concept presentations and a detailed discussion of benefits and drawbacks of each design, the most suitable design was chosen. The main factors were the simplicity of the design and adherence to all geometrical constraints. Each design is presented below with a brief explanation of why that design was or was not selected as well as the Pugh chart used in the initial selection process.

Concept Designs

Design one – Emeale

The above design was not selected due to the input and output shafts not rotating in the same direction, the middle gear and shaft are at a right angle to the other shafts which adds complexity to the design. These angles would pose structural concerns in terms of strength and support.

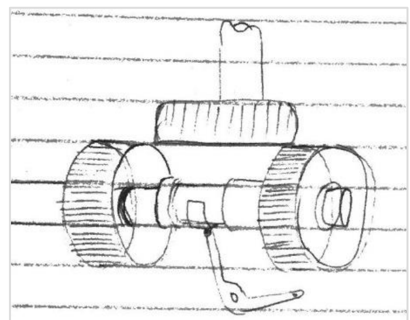

Figure 1: Concept Design One

Design two – Matthew

This design is over complicated due to the addition of additional gears and shafts. This would not only increase the cost but would make the gearbox in its entirety much larger and heavier. The purpose was to achieve symmetry with a horizontal layout, but it was decided that a vertical layout would be more suitable. The design however is much stronger as each of the intermediate shafts would only be transmitting half of the torque and could therefore be smaller.

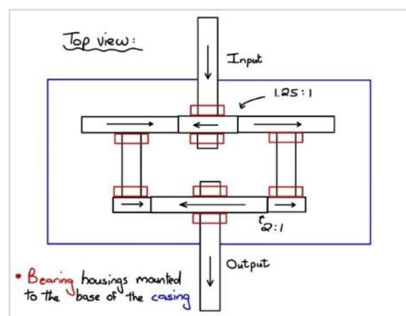

Figure 2: Concept Design Two

Design Three – Michael

This design would cost more than any other design due to the use of long helical gears and the manufacture of such gears would be tedious, expensive and error prone. The design however uses only three gears and is more simplistic than the other designs. The required face width would be large and impractical and so this design was not chosen.

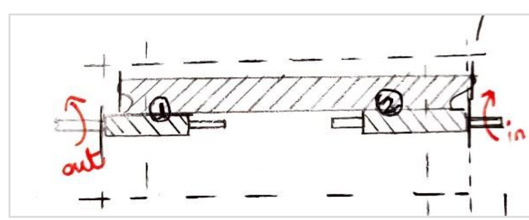

Figure 3: Concept Design Three

Design Four – Will-Mari

This was the best design due to the practicality of the layout and the use of less components. It adhered to all the criteria and would be the easiest and most cost effective to manufacture and design. This design will perform reliably and be simple to maintain and was therefore chosen as the design that would be developed further.

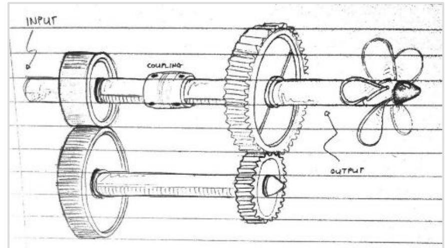

Figure 4: Concept Design Four

Comparison Table

A comparison table is used to further aid in the choice of the best concept design. The table below shows that concept four ticks all the correct boxes and hence was chosen to be the best design.

Concept Criterion	Emeale's Concept (1)	Matthew's Concept (2)	Michael's Concept (3)	Will-Mari's Concept (4)
Helical Gears	☒	✓	✓	✓
Input direction = Output direction	☒	✓	✓	✓
2 Stage Gearbox	✓	✓	✓	✓
Symmetry (about Y-axis)	✓	✓	✓	✓
Cost	☒	☒	☒	✓
Simplicity (Design)	☒	☒	✓	✓
Originality	✓	✓	✓	✓

Table 2: Comparison Table Concept Design

Pugh Selection Chart

To ensure that the best design was chosen, the fourth (highest rated) concept was used as the datum to compare the rest of designs to and prove the design is indeed the best.

	Concepts			
Selection Criteria	4	1	2	3
Input direction = Output direction	Datum	-	+	+
Helical Gears		+	+	+
2 Stage Gearbox		-	-	+
Cost		+	+	-
Symmetry (about Y-axis)		+	+	+
Originality		+	+	+
Number of gears ⩽ 4		-	-	+
Coupling connection		+	+	+
Overall Gear ratio 2,5:1		-	+	-
Number of +	9	5	7	7
Number of -	0	4	2	2
Total	**9**	**1**	**5**	**5**

Table 3: Pugh Selection Chart

From these two selection methods, it can be seen that concept 4 is indeed the best design.

4. Final Design

4.1 Gear Design

The process of gear design is an iterative process. This process started with estimating appropriate values for parameters such as normal module, number of teeth, Helix angle (Appendix A, A3). The preliminary material chosen was AISI 4130 Q&T at 540^{o}C (Appendix A, A1). Torque and force calculations were done and the rest of the AGMA process was followed. It was then to be determined whether the selected parameters resulted in an acceptable safety factor. This safety factor has been specified as 1.5.

The initial design and associated gear ratios proved to be impractical to implement as it results in a large gearbox due to the unequal gear ratios and resulted in very large stresses for the smaller gears. Therefore, it was decided to use equal gear ratios for each stage of the gearbox. This also solves the problem of collinearity between the input and output shafts. Iteration of possible gear ratios and tooth numbers resulted in a final gear ratio of 1.575 per stage and tooth numbers of 40 for the pinions and 63 for the gears. This choice allows the pinions to be large enough to withstand the stresses while ensuring that the larger gears are not unnecessarily large. The process of nitriding the gears would have proven extremely beneficial but due to hardness limitations from the manufacturer, this method could not be exploited.

A final face width of 131.25mm for the first stage and 183.71mm for the second stage was chosen. This provided safety factors ranging from 1.51 for contact stress to 3.53 for bending. These large face widths are due to the choice of using a Brinell hardness of 300. Although the manufacturer has said that they can provide hardness of between 300 and 350 on the Brinell scale, it is not possible for them to guarantee a specific value within that range. Therefore, a choice of hardness above 300 in the design process would have resulted in a large uncertainty of cycle life for the gears and there would have been a high likelihood of premature failure of gears.

Factors

Symbol	Gear Specifications	Gear 1-pinion	Gear 2-gear	Gear 3-pinion	Gear 4-gear
N	Number of teeth (N)	40.00	63.00	40.00	63.00
M	Module (mm)	4.00	4.00	4.00	4.00
D	Pitch Diameters (mm)	170.30	268.20	170.30	268.20
B	Face width (mm)	131.25	131.24	183.71	183.71
T	Torque (Nm)	617.03	971.82	971.82	1530.62
W	Angular Velocity (rad/s)	272.27	172.87	172.87	109.76
Q	Quality (Q)	7.00	7.00	7.00	7.00
(σ_c)	Contact Stress (MPa)	440.78	351.63	453.62	361.88
($\sigma_{c,all}$)	Allowable Contact Stress (MPa)	664.28	681.39	681.39	698.95
(SH)	Safety Factor	1.51	1.94	1.50	1.93
(σ_b)	Bending Stress (MPa)	62.66	61.18	66.36	64.79
($\sigma_{b,all}$)	Allowable Bending Stress (MPa)	142.00	144.10	144.10	146.23
(SF)	Safety Factor	3.40	3.53	3.26	3.39

Table 4: Fundamental Gear Factors

Full table of gear factors is available in Appendix A (A2).

Starting assumptions and iteration of factors

During first iterations, some assumptions were made for specific factors. The over-load factor was initially set to 2, however it was discovered later that this was unnecessary since a factor of 1.05 was specified in the project description. The temperature factor was assumed to be 1 since the operating temperature of the gearbox would never reach 120°C. The surface condition was set to the standard 1 which is reasonable with the design of the current gearbox. All the gears are made of the same material hence the hardness ratio of one gear to another will always be 1. The module was initially set to 3, but after the calculations were performed it was discovered this value was too small and the gears did not meet the necessary requirements. The process was repeated with a value of 4mm, this brought the safety factors to an acceptable level.

CAD design and projections

The final CAD drawings of each gear with respective dimensions and specifications is shown Appendix A (A4, Figure 6).

4.2 Shaft Design

Shaft design was started by the formation of a naming convention for the forces which would be carried forward through to the bearing calculations (Appendix B, B2). Preliminary drawings of each of the shafts (Appendix B, B1) were drawn to obtain rough estimates of applied forces which could then be used to select a first iteration material. Research into the most commonly used materials for shafts along with the respective heat treatments was then conducted. This research led to the selection of AISI 4130 which is heat treated (Q&T) at 540 degrees Celsius.

Preliminary estimates of stress concentrations were made to allow for the use of the DE-Soderberg criterion. From this criterion, a minimum diameter was determined, and the design was adjusted accordingly. MATLAB was used to determine and plot the graphs of shear stresses, bending moments, as well as slopes and deflections (Calculations report, Appendix B, B4, B5, B6). These graphs were used in the selection of the appropriate seals, bearings, circlips, and keyways. They were also used in the deflection calculations to ensure that the gears would continue to mesh correctly despite a slight deflection of the shafts. As the drawings progressed to a more complete state, to included keyways, retaining ring grooves and shoulders with fillets, the calculations were updated to account for the new and more accurate stress concentrations. Through this development process, it was found that the material could not withstand the stresses that developed in the shafts. It was decided that the same material would suffice if it were to be heat treated at 425 degrees Celsius to increase the ultimate and yield strengths (Shigley, Table A-21, page 1050).

Resulting safety factors were subsequently calculated and measured against the requirement of 1.5. The process was iterated multiple times and final changes to the shaft length and gear face widths were made for optimum efficiency, weight, and practicality. The result of these iterations can be seen below along with the final safety factors for the shafts.

Full table of gear factors is available in Appendix A (A2).

Shaft Design Factors Table				
Shaft Factors Name	Symbol	Unit	Value	Reasoning/Assumptions
Material		AISI	4130	Q&T at 425^0C
Minimum tensile strength	S_{ut}	(MPa)	1280	Material Property
Yield strength	S_y	(MPa)	1190	Material Property
Final Design Factors (Using DE-Soderberg criterion)				
Safety factor (Input Shaft)	S_{f1}		3.33	d = 58mm at critical point (65mm - 7mm for keyway)
Safety factor (Intermediate Shaft)	S_{f2}		1.63	d = 58mm at critical point (65mm - 7mm for keyway)
Safety factor (Output Shaft)	S_{f3}		1.53	d = 58mm at critical point (65mm - 7mm for keyway)

Table 5: Fundamental Shaft Factors

4.3 Bearing selection

After the completion of the gear and shaft design, efforts were shifted to determining the bearings that would be required to support the shafts. The design started by using the gear forces to calculate the bearing reaction forces. These forces were then used in further calculations (Calculations Report - Appendix C). Once the reaction forces on the bearings were determined, (‘Calculations report’, Appendix B) an SMATHS document was compiled to calculate the equivalent and dynamic load of the each of the six bearings (Appendix C, C2). One of the bearings is required to withstand the thrust of the propellor and is therefore larger than the other bearings. All bearings could have been the same size, but this would have resulted in severely overdesigned bearings, therefore, the bearing that experienced the second largest force was used and a safety factor of 1.5 was applied to determine the necessary bearing ratings since this ensured that the rest of the bearing safety factors would be higher than 1.5.

Once the minimum dynamic load C was determined, the appropriate bearing was then selected. The bearings were selected from an SKF catalog, (SKF , 2018). It must be noted that the TIMKEN bearings catalog, (The TIMKEN Company, 2017) was consulted as well. However, the TIMKEN tapered roller bearings did not meet the criteria for the specified forces needed to be withstood by the bearings in our design process. Nonetheless, the SKF bearings proved to be more than adequate and could withstand larger forces.

The bearing’s primary requirements were that it had to withstand both the radial forces due to the gears as well as axial forces placed on it by the helical gears and the propeller thrust. The factor a_1, which is used in the SKF document to adjust the basic life to account for reliability at 99.9% (the highest available), is 0.25 (shown in this report’s appendix C, C3). This was used in conjunction with a SKF life modification factor of 2.5. This life modification factor (graph shown in Appendix C, C6) was chosen due to the lack of contaminant particles expected within the gearbox as well as the high level of lubrication and constant load (described in project brief (Bredell, 2020)) this represent area C (Appendix C, C5).

The full bearing dimensions along with its respective properties and factors are shown in more detail in appendix C of this document (C1).

Bearing Design Factors Table				
Bearing Factors Name	Symbol		Value	Reasoning/Description
Thust factor (safety factor radial)	Y		1.6	Catalog rating
Radial factor (safety factor axial)	X		0.37	Catalog rating
Required dynamic load rating	C	(KN)	62.12	Max force for bearings (1,2,3,4,6) (excluding SF = 1.5)
Required dynamic load rating	C	(KN)	130.00	Max force for bearing 5 (excluding SF = 1.5)
Dynamic load rating(choosen bearing)	C	(KN)	104.00	Bearings 1,2,3,4,6
Dynamic load rating(choosen bearing)	C	(KN)	196.00	Bearing 5
Final safety factor	S_f		1.67	Bearings 1,2,3,4,6
Final safety factor	S_f		1.51	Bearing 5

Table 6: Bearing Design Factors

There was concern about the bearing misalignment due to the shaft deflection calculated in the calculations report. For the tapered roller bearing, Shigley gives approximate misalignments as shown in Appendix C (C9), of between 0.0005 and 0.0012 radians. The chosen bearings shown in Appendix C, C7 is of an explorer type, since SKF suggest that if there is any deflection the explorer bearings are recommended. The tolerance for misalignment for this bearing is between 2 to 4 arcminutes (Appendix C, C8), which corresponds to between 0.00058 and 0.00116 radians. Very similar to Shigley. Maximum shaft angle at the bearings is $1.26*10^{-5}$ (Appendix C, C5) which is much smaller than the allowable angle. The recommended life in hours is of a marine propulsion system is shown in Appendix C, C4 at between 60 000 and 100 000 hours, even though the hours shown in calculations is less this will suffice due to the unique conditions the gearbox will operate in.

4.4 Keyway Design

The keyway design started by selecting an appropriate material and then using the formulas for keyways to determine if this material will hold against the forces. Analysis of the keyways were performed on all three shafts. Once the shear was determined as shown in the calculations report (Appendix D), the length of the keyway could be determined. This was checked against the maximum length of the keyway to ensure the design was within the correct parameters. To simplify the manufacturing process a nominal keyway length was selected for all the shafts and gears. The final dimensions calculated were a width of 18mm, a height of 11mm, groove depth of 7mm and length of 21.8mm.

Keyway Design Factors Table				
Keyway Factors Name	Symbol		Value	Reasoning/Description
Gear Interface keys				
Width	b	(mm)	18.00	All keys have same width
Height	h	(mm)	11.00	All keys have same height
Length	L	(mm)	21.80	All keys have same length
Safety factor	S_f		1.50	Designed into all keys

Table 7: Keyway Design Factors

4.5 Retaining Ring Selection

Force values calculated in the shaft and bearing selection were carried forward to the selection of the retaining rings. These forces were used in determining the moments, torques and forces at the respective shoulders, therefore providing the stresses which the rings would need to endure (Calculations report, Appendix E). Using the stresses, a ring was then selected based on its ability to hold up to the required forces. Using the modified Goodman criteria in conjunction with the catalogue groove specifications a factor of safety could then be calculated. The safety factors must have a minimum value of 1.5 to adhere to the design requirements.

The Smalley catalogue (Smalley, 2020) was used to obtain the required widths, heights and tolerances of the retaining rings used to secure the position of both the bearings and the gears on the shafts. The dimensions were used with figures A-15-16 and A-15-17 (G.Budynas & Nisbett, 2015) to calculate the stress concentrations. If the factor of safety for the chosen ring did not meet the minimum requirement, another was chosen. This iterative process continued until the final selected retaining rings met the specifications and were deemed satisfactory. Different rings were selected for different forces as the forces at the gears and bearings differed substantially. The chosen rings to be used at the bearings are Spirolox DNS-45 rings, and for the gears, Spirolox DNS-65. The retaining ring used at the bigger bearing was DNS-80. These rings can be customized to withstand greater forces due to their spiral design. It was calculated that two additional turns of the Spirolox DNS-45 at the bearings would need to be added, therefore increasing the width, and reducing the stress concentration. An additional 4 turns would need to be added to the Spirolox-DNS-65 at the gears due to the higher forces.

Retaining Ring Design Factors Table				
Shaft Factors Name	Symbol	Unit	Value	Reasoning/Assumptions
Retaining Ring Factors				
Safety factors $_{(bearing\ A\&B)}$	S_f		3.95	Based on groove depth
Safety factors $_{(bearing\ C\&D)}$	S_f		2.22	Based on groove depth
Safety factor $_{(bearing\ E\&F)}$	S_f		1.54	Based on groove depth
Safety factors $_{(gear\ 1)}$	S_f		3.34	Based on groove depth
Safety factors $_{(gear\ 2\&3)}$	S_f		1.6	Based on groove depth
Safety factors $_{(gear\ 4)}$	S_f		1.55	Based on groove depth

Table 8: Retaining Ring Design Factors

4.6 Seal selection

The seals were selected based on their design application, speed of the shafts, lubrication type and orientation. Among these, the predominant factors are the interface speed and application use. A very secure and tight seal needed to be selected to ensure no water leakage into the gearbox and no lubrication leakage out of the gearbox. The shaft speed of 2700 rpm was designed for to account for fluctuations in speed (Appendix E).

According to the SKF catalogue (SKF Group, 2019), HMS5 seals provide the optimal and most secure design. It is both compatible with the selected lubricant as well as being able to operate at a speed of 3000 rpm. There are also favorable design characteristics such as the improved sealing ability, lubrication retention, high misalignment tolerances and good wear resistance. The seals will be slid onto the shafts and pressed into the bearing bore. The bore dimension is matched to that of the bearings to ensure a secure fit.

Lip Seal Design Factors Table				
Radial Lip Seal Factors	Symbol		Value	Reasoning/Description
Nominal seal width	w	(mm)	10.00	
Design and lip material			HMS5	
Allowable velocity	V_a	(m/s)	9.40	
Max Operating velocity	V_n	(m/s)	5.42	
Safety factor	S_f		1.73	Based on velocities

Table 9: Lip Seal Design Factors

4.7 Coupling Selection

Both the input and output shafts must be able to transmit torque and the output shaft must be able to transmit the axial thrust developed by the propellor. There are a few designs that can accommodate this requirement. Some of which include keyways for transmitting torques and pins for transmitting axial forces. These solutions, however, cause major stress concentrations within the shaft that would require a very thick shaft to withstand the stresses. The goal of the design is to optimize the gearbox size and weight and therefore a cone coupling is used. This coupling design uses a cone that is fitted over the shaft ends and then uses clamping screws and pressure rings to compress the cone onto both shafts. This method is very strong and can withstand torques of 1700 Nm and axial forces of 61 KN which is far more than the required 1608 Nm of torque and 13700 N axial force. This coupling allows to for minimal material removal of the shaft. The only machining operations required are filleting the ends and ensuring a tolerance of 10 – 25 microns for the clamped section of the shaft. The structure of the coupling can be seen in (Appendix D, Figure 11). All relevant information can be found in Appendix D.

4.8 Lubrication Selection:

To improve maintainability and ensure a reliable gearbox of superior quality is delivered, the lubrication method must be chosen with care and due diligence as to increase the life of the gearbox and the individual components within the casing.

Grease and oils are the two most used lubricants for bearings. Due to the need of having to lubricate both the bearings and the gears, it was decided that a common lubricant would be used for both. This allows for smooth operation without the cross contamination of either lubricant. High speed gears require oil as a lubricant; therefore, it was decided to use oil for the lubrication of all components within the gearbox. Synthetic oil was chosen over mineral oil due to its superior lubrication ability, longevity, and its ability to maintain its viscosity at a wide range of temperatures.

The most common methods of distributing lubricant to the desired locations are splash lubrication, drip lubrication and forced feed lubrication. The simplest and most cost-effective option is splash lubrication. The bearings closest to the gears are lubricated directly by the gears while others are lubricated using a lubricant throwing device (CAD 2020 – 1113 – Sheet 2). To ensure complete lubrication, each bearing is fitted with a lubricant collector (as seen below) that collects unused lubricant that is splashed past the bearings. This oil is then fed to the bearing through a channel as seen in (Reference to CAD). This is a widely used method in the gearbox industry and is extremely effective at ensuring comprehensive lubrication.

Initially, a collector bath and tube were used to transfer lubricant from the upper bearings to the lower bearings, however this caused interference with the assembly of the gearbox. Instead, every second blade of the lubricant splasher was twisted to splash lubricant directly at the lower bearing. The properties and recommended oil are shown in the CAD drawing (CAD 2020 – 2213 – Sheet 1)

The gearbox requires 15.5L of lubricant during operation.

4.9 Casing Design

After Completion of the geartrain design, an initial casing concept was developed. This concept was a basic shell concept that did not contain any structural supports and would require significant optimization. A few iterations were made in which different support locations where explored and the process of the assembly was considered.

The final design makes use of a hexagonal shape at the bottom of the casing (CAD 2020 – 2213 – Sheet 1) to make the casing lighter, more compact and stronger than a rectangular shape. The outside walls are welded together to form a shell and a permanent ribcage is welded to the inside to add structural rigidity. A wall has been placed between the input and output shaft to add strength and minimize the likelihood of lubricant moving from one bath to the other since it is important to keep lubricants at an optimal level.

Consideration was given to assembly such that the bearing seats are large enough to allow the input and output shafts to be inserted from the sides of the casing. The Input and output gears and inner bearings will be lowered into position from the top and the input and output shafts will be inserted from the sides of the casing. Thereafter the outer bearings and seals will be inserted from the sides. The last step is to put the intermediate shaft bearing on the input side in place and then lower the intermediate shaft (with gears pre-installed) at a slight angle into the casing. Once the shaft is in place, the remaining bearing is inserted from the side and the end cap bolted in position. A removable lid is bolted onto the casing after assembly to allow for easy access during assembly as well as streamlined maintenance. The lid is simple to save cost but has a groove that allows for the use of a ring seal to ensure that water does not seep between the casing and the lid. An endcap is used to close off the hole where the intermediate shaft bearing is inserted. A basket structure is used to effectively transfer the forces from the ribcage to the rails (CAD 2020 – 1113 – Sheet 3). A total of 8 M24 High-Strength-Friction-Grip bolts are used to fasten the gearbox to the rails.

Misalignment Considerations:

Since slight misalignment of shafts is common, this was accounted for in the design. To account for vertical misalignments, shims will be used to ensure correct height. The rail system was designed to allow for a vertical alignment adjustment of +/- 2.5mm. To account for horizontal adjustment, the bolt holes for the HSFG Bolts are made 1mm larger than required to allow the for a +/- 0.5mm horizontal and axial adjustment. This procedure of bolt hole adjustment is common for applications such as gearbox or motor shaft misalignment and does not compromise strength since the bolts are designed to be tightened to high levels of tension which allows the horizontal forces to be transferred by friction between the plates, washers and bolts rather than by shear through the bolt itself. Jacking bolts will be used for micro adjustments of the alignment.

Corrosion Protection:

Due to the wet and salty environment that the gearbox will operate in, corrosion can easily occur. Therefore, all exposed outer surfaces will be coated by PPG SigmaCover 380 universal epoxy anticorrosive primer (PPG, 2019) to seal the casing and protect it from corrosion resulting in a longer lasting casing. In the event that the protective coating is damaged by personnel or equipment, the primer can be reapplied to the exposed location after thorough cleaning of the metal. To ensure complete coverage, airless spray will be used. A thinner (Thinner 91 – 92) of roughly 0 – 10% of the volume must be added. The coating can also be applied by making use of a brush after repairs were made. A film thickness of 200µm will be required. A nozzle pressure of 20-25MPa with a 4 m^2/L spreading rate should be used to apply the coat (PPG, 2019).

Gearbox Deflection:

Due to the combination of the ribcage along with the reinforced custom rail mounts, deflection of the casing is minimal. This claim is supported by the results of a static stress analysis performed in inventor. The total gearbox deflection is less than 11 Microns (well within the required tolerance).

Total Gearbox Mass:

The complete gearbox mass is 508.607 kg

4.11 Reliability

The overall required reliability is to be above 95% as stipulated in the design report. The gearbox has a total of 13 components in series and during the design of each component, the reliability had to be carefully selected to adhere to this requirement. After iteration, the final reliability of the bearings was chosen to be 99.9% and the shafts were designed to have a reliability of 99.99%. This allowed the gears to have a reliability of 99%. The use of 99% reliability for the gears instead of 99.9% allowed for a major decrease in size and weight. Using the formula for reliability of components in series (START, 2016), the total reliability of the system is $(0.99^4)*(0.9999^3)*(99.9^6)*100 = 95.456\%$

4.12 Final Design Assessment:

Below can be seen the final assessment and comparison between the final design and the engineering requirements.

Requirement description	**Required Value**	**Was this Achieved?**	**Comments**
Basic gearbox requirements			
Shaft alignment	Input and output shafts co-linear	YES	
Rotation direction	Input same as output	YES	
Gear type	Helical gears	YES	
Geometry	Symmetrical about vertical longitudinal plane	YES	
Gear ratio	Within 1% of 2.5:1	YES	2 Stages @ 1.575:1 = 2.48 (99.2%)
Construction & layout	Two-stage	YES	
Duty cycle and design life	2 hours per day, 7 days a week for 10 years.	YES	This was considered in the stress and reliability calculations
Power rating	Engine 168kW	YES	These values were considered during stress an force calculations.
Speed rating	2600rpm	YES	
Standardized Components	-	YES	Ball bearings, seals, retaining rings, keys and couplings were selected from standardized catalogues
Gearbox weight	Less than engine weight (578Kgs)	YES	Total gearbox mass is 508.607kgs
Gear fabrication parameters			
Normal pressure angle	20°	YES	
Normal modules	2, 3, 4, 5 (mm)	YES	A Normal module of 4mm was used
Helix angles	20°, 25°, 30°	YES	A helix angle of 20˚ was used
Hardness	300-350 (HB)	YES	All calculations were performed using 300 HB
AGMA Quality	7	YES	This was used for stress calculations
Casing			
Assembly process	Welded assembly	YES	All welds are specified in the drawing report

Material	Laser cut S355JR steel plates.	YES	Specified in the drawing report
Corrosion protection	Specified by Designer	YES	PPG SigmaCover 380 universal epoxy anticorrosive primer
Interface surface	Direct mount to inboard rails	YES	Rail mount system is welded as part of the casing
Shaft positioning	100mm	YES	Shaft centre positioned 100mm above the rail mount
Maintainability	-	YES	Removable top lid to ensure easy access to components
Interfaces			
Input shaft			
Shaft deflection	Less than 0.038mm	YES	All deflections were well under 0.038mm
Shaft connection	Coupling (capable of thrust and torque transmission)	YES	The coupling can support torques and axial forces much larger than required
Shaft material	Wear resistant	YES	Fatigue analysis was performed
Output shaft			
Shaft alignment	Gearbox output shaft interfaces with propeller shaft	YES	
Diameter	Propeller shaft and gearbox output shaft have same diameter	YES	
Load capacity (forward)	13.7 kN	YES	This was considered during bearing selection
Load capacity (reverse)	13.7kN x 10%	YES	Bearings are supported to ensure that a reverse load can be sustained.
Reliability and safety			
Fatigue design reliability	95%	YES	An Overall Reliability of 95.456% was Achieved
Safety factor	1.5	YES	All critical components were designed to have a minimum safety factor of 1.5

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6. Appendices

Appendix A – Gears

A1. Material Selection Process

Background information - The most common material used for gears is Steel or Cast Iron. Steel is used because of its desirable characteristics; high strength to weight ratio, high resistance to wear, enhanced physical properties when heat treated and cost (Gear Motions, 05/12/2017). However Cast Iron is also commonly used due to its good wear properties, machinability and ease of cast manufacture (Gonzalez, 2015)

Hardness - Considering two gear sets each of a different Brinell Hardness Number (BHN), the gear set with the higher number will be more compact and smaller as well as lighter. High hardness also indicates surface durability or wear strength. This means there is greater tooth resistance to failure. Often in the process of surface hardening a case-hardening process is used, producing a hard-outer surface but leaving the core soft, this is done because increased hardness causes materials to become brittle (Gonzalez, 2015).

In the figure of material specifications and heat treatments, (Engineersegde, 2020) one can see various AISI materials, there related hardness values and how they were heated treated to achieve the various properties. This treatment is important as this will affect the cost of manufacture. For our application in a gear box, it seems the most suited would be the AISI 1020, which has a Hardness of 111, Tensile strength of 380 Mpa and yield of 210 Mpa, if hot rolled. If cold rolled it will have a hardness number of 131, Tensile strength of 470 Mpa and yield of 390 Mpa (G.Budynas & Nisbett, 2015). Most materials used specifically for helical gears are various types of steel, aluminum, bronze, nylon, etc.

Safety Factor - Typical safety factors needed for gears are around 1.15 to 1.5. However, the choice is ultimately decided by the designers of the gears. Cost as well as over-designed must be considered to prevent heavy gears and gearbox, with an increase in gearbox size. The safety factor needs to be large enough that the gears will not fail unexpectedly and must have a reasonable life expectancy.

Requirements - The material needs to be strong enough to resist the forces of propulsion as well as the required torques that is placed on the teeth. Since more calculations needed to be provided before those are determined. The safest bet it to stick to steel, especially of an AISI grade, given in the prescribed textbook. This would also limit the range down to one of the 40xx steels since anti-corrosion would need to be considered due to the gears being placed inside a gearbox design and mostly covered in lubrication oil

Final selection - AISI 4340 Q&T @ 540˚C (Through Hardened) (Table A-21, Shigley p1050). Locally Available. Satisfies hardness requirements (with a heat treatment condition V or W). Satisfies relationship between hardness and tensile strength. Heat treatment was based on figure showing hardness for different heat treatments.

Table 6 showing different materials and heat treatments. This was used for the material selection initially.

Material	Hardness		Typical Heat Treatment,
	Case	Core	
Specification	Rc	Bhn	Characteristics, and Uses
Case-Hardening Steels			
AISI 1020 **AISI 1116**	55–60	160–230	Carburize, harden, temper at 350°F. For gears that must be wear-resistant. Normalizedmaterial is easily machined. Core is ductile but has little strength.
AISI 4130 **AISI 4140**	50–55	270–370	Harden, temper at 900°F, Nitride. For parts requiring greater wear resistance than that of through-hardened steels but cannot tolerate the distortion of carburizing. Case is shallow, core is tough.
AISI 4615 **AISI 4620** **AISI 8615** **AISI 8620**	} 55–60 } 55–60	170–260 200–300	Carburize, harden, temper at 350°F. For gears requiring high fatigue resistance and strength. The 86xx series has better machinability. The 20 point steels are used for coarser teeth.
			Carburize, harden, temper at 300°F.
AISI 9310	58–63	250–350	Primarily for aerospace gears that are highly loaded and operate at high pitch line velocity and for other gears requiring high reliability under extreme operating conditions. This material is not used at high temperatures.
			Harden, temper at 1200°F, Nitride.
Nitralloy N and Type 135 Mod. (15-N)	90–94	300–370	For gears requiring high strength and wear resistance that cannot tolerate the distortion of thecarburizing process or that operate at high temperatures. Gear teeth are usually finished before nitriding. Care must be exercised in running nitrided gears together to avoid crazing of case-hardened surfaces.
Through-Hardening Steels			
AISI 1045 **AISI 1140**	24–40	...	Harden and temper to required hardness. Oil quench for lower hardness and water quench for higher hardness. For gears of medium and large size requiring moderate strength and wear resistance. Gears that must have consistent, solid sections to withstand quenching.
			Harden (oil quench), temper to required hardness.
AISI 4140 **AISI 4340**	24–40	...	For gears requiring high strength and wear resistance, and high shock loading resistance. Use 41xx series for moderate sections and 43xx series for heavy sections. Gears must have consistent, solid sections to withstand quenching.

Table 10: Hardness and Heat treatment for various metals (Engineersegde, 2020)

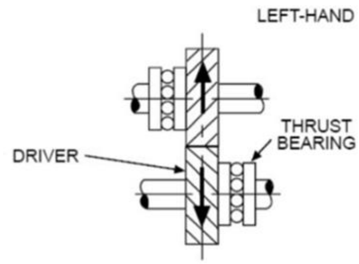
A2. Full List of Gear Factors:

Symbol	Gear Specifications	Gear 1- pinion	Gear 2- gear	Gear 3- pinion	Gear 4- gear
N	Number of teeth (N)	40.00	63.00	40.00	63.00
M	Module (mm)	4.00	4.00	4.00	4.00
D	Pitch Diameters (mm)	170.30	268.20	170.30	268.20
B	Face width (mm)	131.25	131.24	183.71	183.71
T	Torque (Nm)	617.03	971.82	971.82	1530.62
W	Angular Velocity (rad/s)	272.27	172.87	172.87	109.76
Q	Quality (Q)	7.00	7.00	7.00	7.00
Used For:	**AGMA Factors**				
(σc)	Elastic co-efficient (ZE)	186420.00	186420.00	186420.00	186420.00
(σc),(σb)	Dynamic (Kv)	1.56	1.56	1.56	1.56
(σc),(σb)	Size (Ks)	1.15	1.17	1.15	1.17
(σc),(σb)	Overload (Ko)	1.05	1.05	1.05	1.05
(σc),(σb)	Load Distribution (Kh)	1.29	1.29	1.40	1.40
(σc)	Surface Condition (Zr)	1.00	1.00	1.00	1.00
(σc)	Geometry (Zi)	0.17	0.17	0.17	0.17
(σc,all),(σb,all)	Load Cycle (Zn)	0.77	0.79	0.79	0.81
(σc,all)	Hardness Ratio (Zw)	1.00	1.00	1.00	1.00
(σc,all),(σb,all)	Reliability (Yz)	1.00	1.00	1.00	1.00
(σc,all),(σb,all)	Temperature (Yt)	1.00	1.00	1.00	1.00
(σb)	Bending Geometry (YJ)	0.59	0.61	0.59	0.61
(σb,all)	Allow Bending Stress No. (Mpa)(St)	248.20	248.20	248.20	248.20
(σc),(σb)	Rim Thickness (Kb)	1.00	1.00	1.00	1.00
(σc)	Contact Stress (Mpa)	440.78	351.63	453.62	361.88
(σc,all)	Allowable Contact Stress (Mpa)	664.28	681.39	681.39	698.95
(SH)	Saftey Factor	1.51	1.94	1.50	1.93
(σb)	Bending Stress (Mpa)	62.66	61.18	66.36	64.79
(σb,all)	Allowable Bending Stress (Mpa)	142.00	144.10	144.10	146.23
(SF)	Saftey Factor	3.40	3.53	3.26	3.39

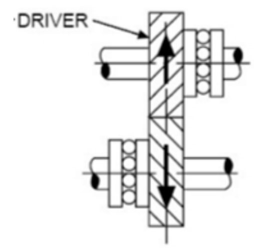
Table 11: Full List of Gear Factors

A3. Hand of Helix

During forward Operation (anticlockwise motion as seen from behind), we want the induced thrust force to counteract the propeller thrust. Therefore, by choosing driven gear to be left-handed as follows.



This means that the driver gear will right-handed. The induced thrust of gear 2 should counter that of gear 3 so that the axial forces do not add, but rather, subtract from one another leaving a smaller force needing to be designed for. Choosing the second gear set to be right-handed was follows:



And combining the two sets together, the final gear train is set up in such a way that the forces from the trust and the forces due the gears at output cancel one another. This occurs at the top shaft as well where the gear forces from the different sets (input and output) oppose one another as well.

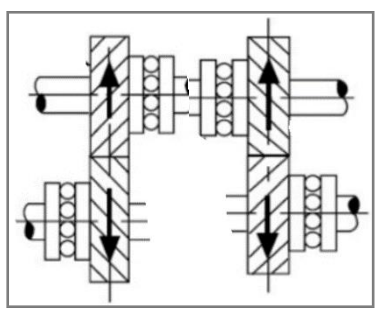


Figure 5: Final Helix Angle Selection

A4. CAD Drawings

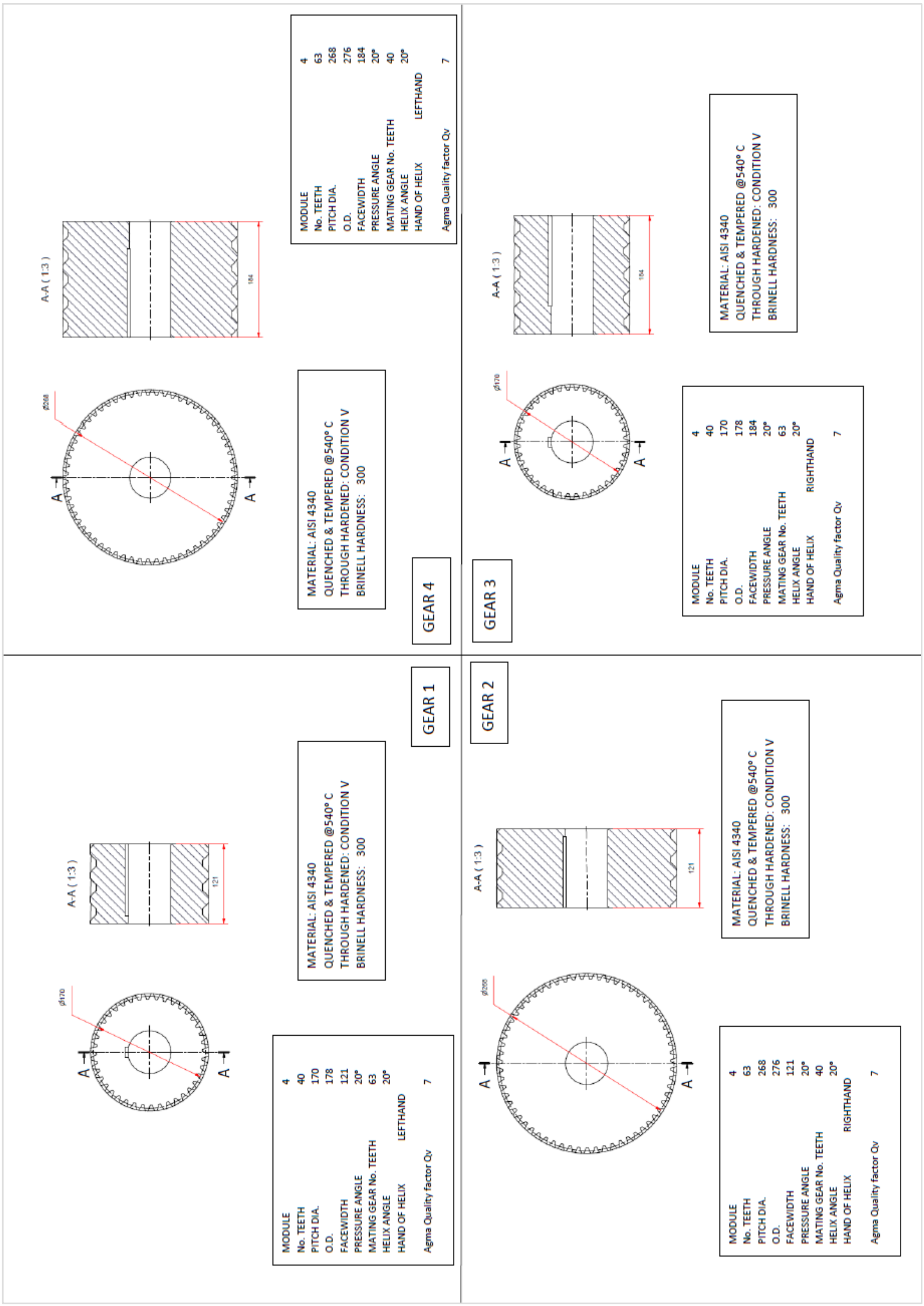
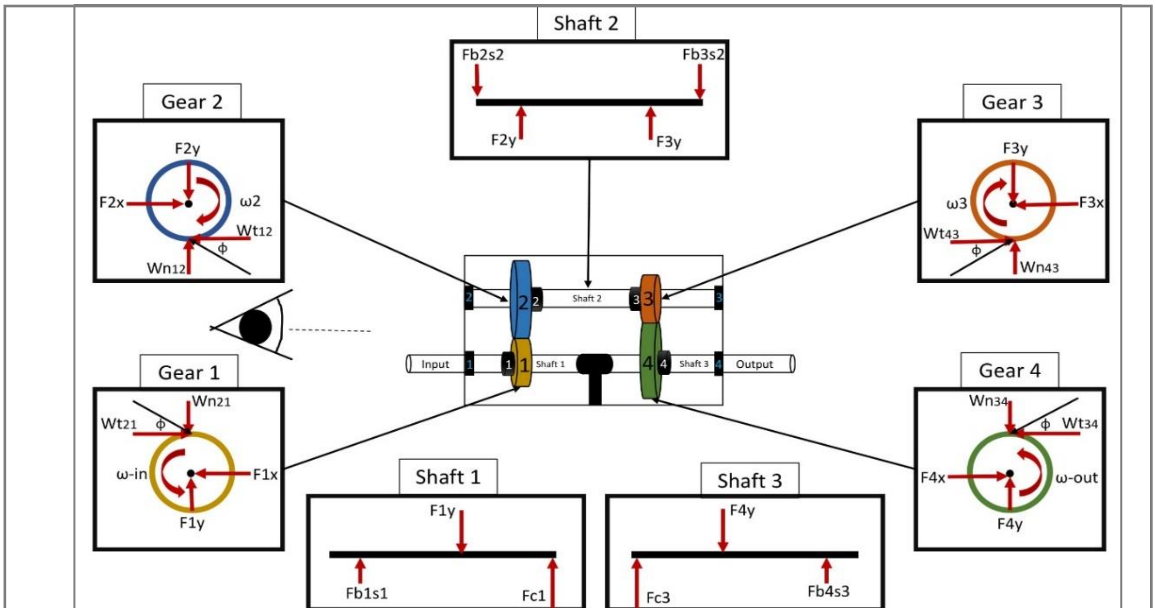


Figure 6. CAD orthographic drawing of gears

Appendix B – Shafts

B1. Preliminary Force Diagrams



Nomenclature

- Fb2s2 – "Force of bearing 2 on shaft 2"
- Fc1 – "Force of coupling on shaft 1"
- Wt21 – "Tangential force gear 2 exerts on gear 1"
- Wn32 – "Normal force gear 3 exerts on gear 2"
- F1y – "Reaction force on gear 1 in y-direction"

Equations

- Wt21=Wt12
- Wt43=Wt34
- Wn21=Wn12
- Wn43=Wn34
- F1x=Wt21
- F2x=Wt12
- F3x=Wt43
- F4x=Wt43
- F1y=Wn21
- F2y=Wn12
- F3y=Wn43
- F4y=Wn34
- W-in=W2
- W2=W3
- W3=W-out
- Fb2s2=F2y=F3y=Fb3s2

Figure 7. Basic Preliminary shaft force analysis

B2. Final Drawing References

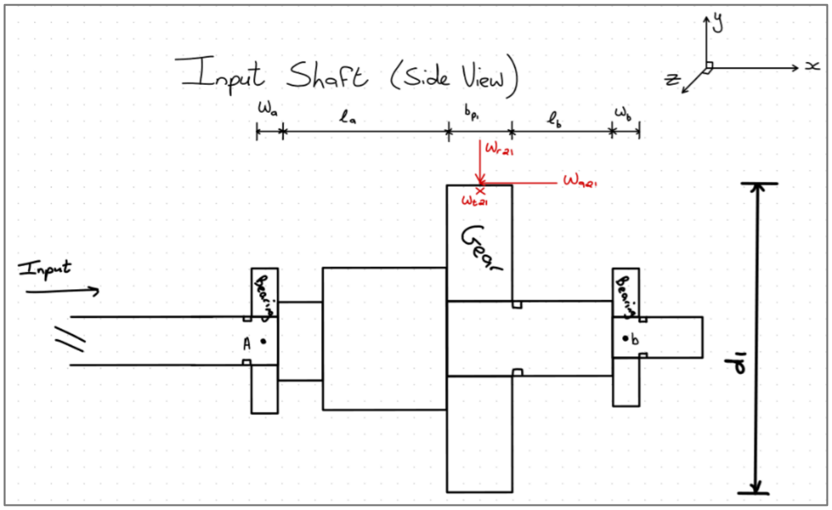


Figure 8: Input shaft (hand) drawing used for final calculations

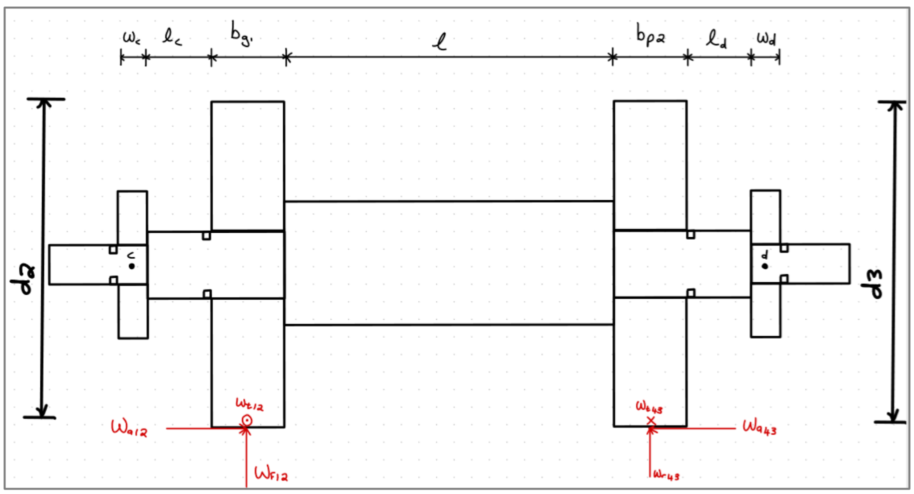


Figure 9: Intermediate shaft (hand) drawing

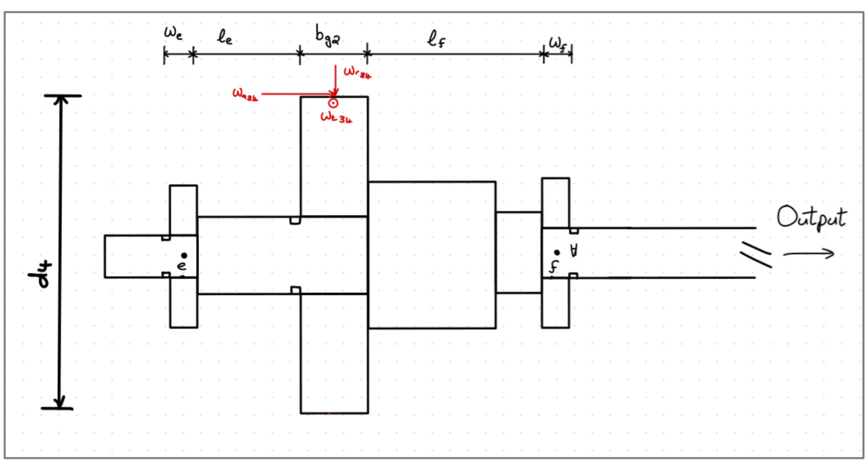


Figure 10: Output shaft (hand) drawing

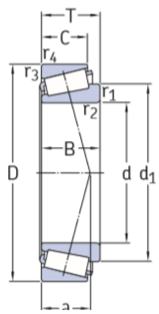
B3. Full List of Gear Factors:

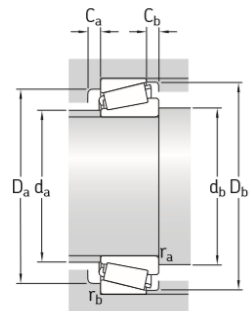
Shaft Design Factors Table				
Shaft Factors Name	Symbol	Unit	Value	Reasoning/Assumptions
Material		AISI	4130	Q&T at 425^{0}C
Minimum tensile strength	S_{ut}	(MPa)	1280	Material Property
Yield strength	S_y	(MPa)	1190	Material Property
Surface factor	K_a		0.68	Machined
Size factor	K_b		0.79	51mm < d < 254mm
Loading factor	K_c		1.00	Bending + Torsion + Axial
Temperature factor	k_d		1.02	60^{o}c Operating Temperature
Reliability factor	k_e		0.70	99.99%
Miscellaneous factor	K_f		1.00	No Adverse Conditions
Marin equation	S_e	(MPa)	245.26	
Notch sensitivities	q		0.90	r = 2mm
	q_{shear}		0.91	r = 2mm
Stress concentration	K_t		2.14	Keyway induced stress concentration
	K_{ts}		3.00	Keyway induced stress concentration
	K_f		2.03	Keyway induced stress concentration
	K_{fs}		2.82	Keyway induced stress concentration
First Design Iteration				
Minimum diameter	D_1	(mm)	44.30	Saftey factor of 1,5
	D_2	(mm)	56.20	Saftey factor of 1,5
	D_3	(mm)	57.50	Saftey factor of 1,5
Final Design Factors (Using DE-Soderberg criterion)				
Safety factor (Input Shaft)	S_{f1}		3.33	d = 58mm at critical point (65mm - 7mm for keyway)
Safety factor (Intermediate Shaft)	S_{f2}		1.63	d = 58mm at critical point (65mm - 7mm for keyway)
Safety factor (Output Shaft)	S_{f3}		1.53	d = 58mm at critical point (65mm - 7mm for keyway)

Appendix C- Bearings

C1. Selection for catalogue, (SKF , 2018)

d 35 – 45 mm





Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation	Dimension series to ISO 355 (ABMA)
d	D	T	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	–
45	75	20	71,7	80	8,8	7 000	8 500	0,34	► 32009 X	3CC
	80	26	104	114	12,9	6 700	8 000	0,55	► 33109	3CE
	85	20,75	81,6	76,5	8,65	6 300	8 000	0,47	► 30209	3DB

Dimensions							Abutment and fillet dimensions									Calculation factors		
d	d_1 ≈	B	C	$r_{1,2}$ min.	$r_{3,4}$ min.	a	d_a max.	d_b min.	D_a min.	D_a max.	D_b min.	C_a min.	C_b min.	r_a max.	r_b max.	e	Y	Y_0
mm							mm									–		
45	60,7	20	15,5	1	1	16	52	52,5	67	68	72	4	4,5	1	1	0,4	1,5	0,8
	63	26	20,5	1,5	1,5	18	52	53,5	69	72	77	4	5,5	1,5	1,5	0,37	1,6	0,9
	63,1	19	16	1,5	1,5	17	54	53,5	74	77	80	3	4,5	1,5	1,5	0,4	1,5	0,8

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation	Dimension series to ISO 355 (ABMA)
d	D	T	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	–
50	110	42,25	196	216	24,5	4 500	6 000	1,95	32310 B	5FD
	110	42,25	211	212	24	4 800	6 300	1,85	► 32310	2FD

Dimensions							Abutment and fillet dimensions									Calculation factors		
d	d_1 ≈	B	C	$r_{1,2}$ min.	$r_{3,4}$ min.	a	d_a max.	d_b min.	D_a min.	D_a max.	D_b min.	C_a min.	C_b min.	r_a max.	r_b max.	e	Y	Y_0
mm							mm									–		
50	83,1	40	33	2,5	2	33	62	61,5	83	101	103	5	9	2,5	2	0,54	1,1	0,6
	77,7	40	33	2,5	2	27	63	61	90	101	102	5	9	2,5	2	0,35	1,7	0,9

C2. Formulas used, (SKF , 2018)

$$L_{nmh} = \left(\frac{10^6}{60\,n}\right) L_{nm}$$

where

L_{nm} = SKF rating life (at 100 – n[1])% reliability) [millions of revolutions]

L_{nmh} = SKF rating life (at 100 – n[1])% reliability) [operating hours]

L_{10} = basic rating life (at 90% reliability) [millions of revolutions]

a_1 = life adjustment factor for reliability (table 3, page 90, values in accordance with ISO 281)

a_{SKF} = life modification factor

C = basic dynamic load rating [kN]

P = equivalent dynamic bearing load [kN]

n = rotational speed [r/min]

p = exponent of the life equation
= 3 for ball bearings
= 10/3 for roller bearings

$$P = X F_r + Y F_a$$

where

P = equivalent dynamic bearing load [kN]

F_r = actual radial bearing load [kN]

F_a = actual axial bearing load [kN]

X = radial load factor for the bearing

Y = axial load factor for the bearing

$$L_{nm} = a_1\, a_{SKF}\, L_{10} = a_1\, a_{SKF} \left(\frac{C}{P}\right)^p$$

C3. Guideline for bearing hours (SKF , 2018)

Values for life adjustment factor a_1

Reliability	Failure probability n	SKF rating life L_{nm}	Factor a_1
%	%	million revolutions	–
99,9	1	L_{1m}	0,25

C4. Adjustment factor a_1 (SKF , 2018)

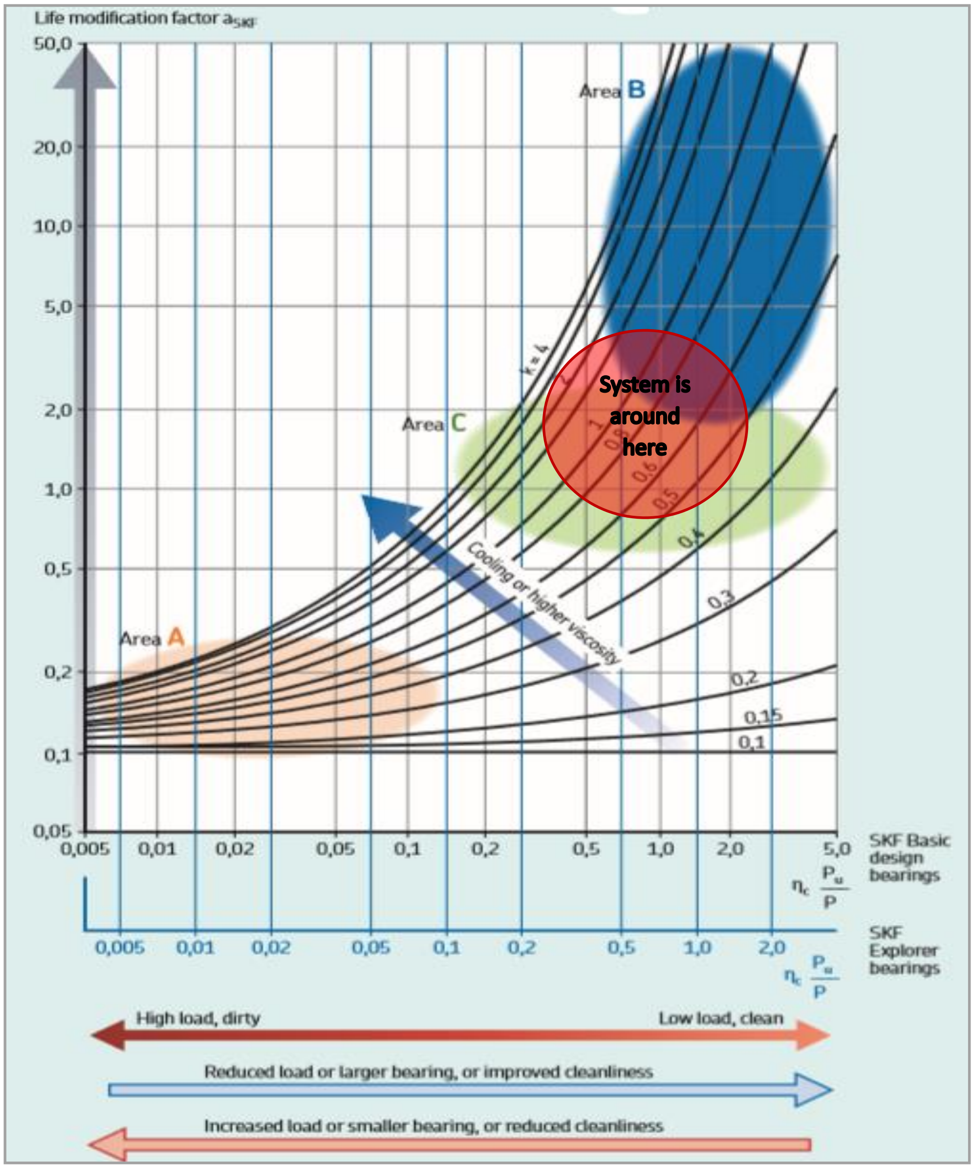
Guideline values of specification life for different machine types

Machine type	Specification life Operating hours
Water works machinery, rotary furnaces, cable stranding machines, propulsion machinery for ocean-going vessels	60 000 ... 100 000

C5. Graph Area Explanations

- **Area A** is dominated by very high load and/or severe indentations.
  The lubricating conditions in this domain can only marginally improve the expected fatigue life, so a potential life improvement depends on what dominates the relationship between the contamination level and the load level P_u/P. To achieve a greater SKF rating life, either the load must be reduced, or the cleanliness must be improved, or both.
- **Area C** is where the life modification factor is less sensitive to changes.
  Deviations from estimated load level, cleanliness factor and lubrication conditions (for example, from uncertainties in temperature) will not substantially affect the value of a_{SKF}, which means the resulting SKF rating life is more robust.
  In the load level domain, area C has the ranges:
  - $P_u \le P \le 0{,}5$ C for ball bearings
  - $P_u \le P \le 0{,}33$ C for roller bearings

C6. Life modification factor



C7. 33109 Bearing

33109

Popular item
SKF Explorer
Tapered roller bearings, single row

BEARING DATA

- Tolerances:
  metric bearings: Normal and CL7C, CLN
  inch bearings: Normal and CL, deviating width

BEARING INTERFACES

- Seat tolerances for standard conditions
- Tolerances and resultant fit

C8. Permissible Misalignment (SKF)

Permissible misalignment	SKF Explorer bearings: ≈ 2 to 4 minutes of arc Where misalignment cannot be avoided, SKF recommends using only SKF Explorer bearings. The permissible angular misalignment between the inner and outer rings depends on the size and internal design of the bearing, the radial internal clearance in operation and the forces and moments acting on the bearing. As a result, only approximate values are listed here. Any misalignment increases bearing noise and reduces bearing service life.

C9. Permissible Misalignment (Shigley)

Slopes	
Tapered roller	0.0005–0.0012 rad
Cylindrical roller	0.0008–0.0012 rad
Deep-groove ball	0.001–0.003 rad
Spherical ball	0.026–0.052 rad
Self-align ball	0.026–0.052 rad
Uncrowned spur gear	<0.0005 rad

Appendix D – Coupling

The coupling must be able to transfer a torque of 1608 Nm and a 13700 N axial force. The cheapest and most effective option is shown below. Coupling: Ringspann Cone Clamping Coupling RWK EEO

Coupling Mechanism:

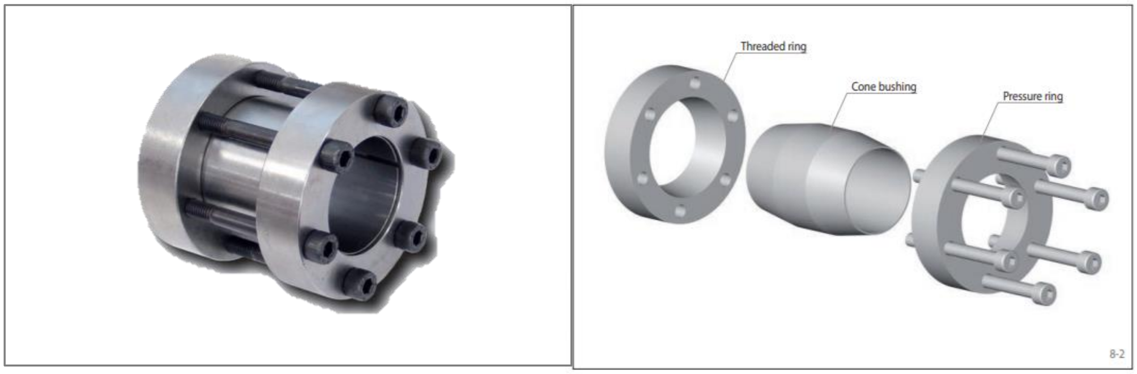
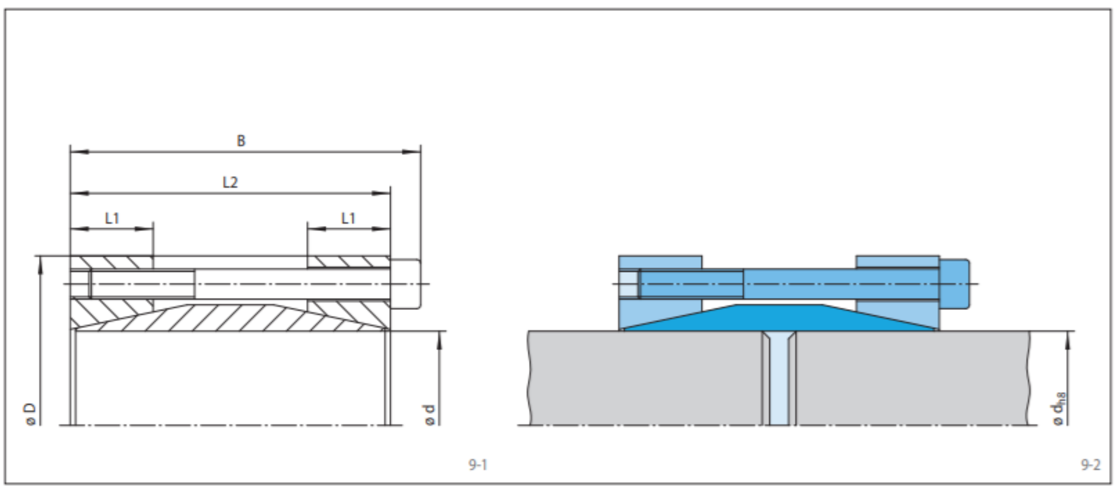


Figure 9: Coupling Mechanism

Extract from Datasheet:

Reference: Ringspann Catalogue (Ringspann, 2019).



Coupling size	Max. transmissible torque or axial force		Clamping screws				D	B	L1	L2	Weight
d	$T_{K\,max}$	$F_{ax\,max}$	Tightening torque M_S	Number	Size	Length					
mm	Nm	kN	Nm				mm	mm	mm	mm	kg
0045	1 700	61	37	8	M 8	80	95	93	22	85	2,00

Appendix E – Seal Selection

The Seal Selected was the HMS5 seals.

<u>Selection criterion</u>

1. Minimum interface speed of 2700 rpm
2. Must be compatible with the selected lubrication
3. Must have the ability to retain liquid
4. Must have the ability to resists water penetration
5. Bore diameter = 80 mm
6. Must have the ability to withstand shaft misalignment

The HMS5 seal meets all the requirements specified above. This seal is compatible with most oil and grease lubrications, can withstand interfacial speeds of 4500 rpm and has good lubrication retention and water penetration resistance properties. According to the SKF catalogue (2019, p106) additional benefits include, good resistance to ageing and wear. The outer diameter is lined with beads to improve the sealing ability. The seal design allows for high dynamic run out as well as shaft to bore misalignment. Below, figure 1 and table 1 present the layout of seal along with the required dimensions.

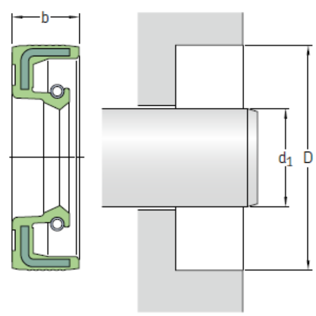


Figure 10: Seal Shaft and Bore Dimensions (SKF Group, 2019)

Dimensions Shaft d_1	Bore D	Nominal seal width b	Designation[1)]	ISO / DIN
mm			–	–
45	62	7	45×62×7	
cont.	62	8	45×62×8	•
	62	10	45×62×10	
	65	8	45×65×8	•
	65	10	45×65×10	
	68	7	45×68×7	
	68	10	45×68×10	
	68	12	45×68×12	
	72	8	45×72×8	
	72	10	45×72×10	
	75	8	45×75×8	
	75	10	45×75×10	
	80	10	45×80×10	

Figure 11: Seal Shaft and Bore Dimensions (SKF Group, 2019)

Pumping ability

Speed Rotating	Circumferential	Pumping time Standard NBR	SKF compound RG
r/min	m/s	s	
1 000	3,1	–	117
1 500	4,7	280	69
2 000	6,3	186	50
2 500	7,9	130	40
3 000	9,4	102	31
3 500	11,0	82	25
4 000	12,6	68	21
4 500	14,1	57	18

Figure 12: Speed Requirements (SKF Group, 2019)

table - Counterface tolerances for metric shafts

Nominal d_1		Deviation	
over	incl.	high	low
mm		µm	
6	10	0	-90
10	18	0	-110
18	30	0	-130
30	50	0	-160

Figure 13: Tolerances Required (SKF Group, 2019)

Appendix F – Miscellaneous

Meeting Minutes

<u>Meeting minutes for group 13 - 1</u>

Date: 2020/08/02
Duration: 25 Minutes
Meeting number: 1
People attended: 4
Next Meeting: 2020/08/06

This was the first meeting of team. Main agendas where to get to know one another and sort out the basis.

Agenda	Main Points	Main Results
1	Get to know team members	Each person introduced themselves
2	Nominate and vote on team leader	Matthew was selected
3	Assign sub-project team leaders	*Drawings-Emeale *Compilation-Michael *Calculations-Will-Mari *Project Manager-Matthew
4	Set intermediate dates for individual concepts	*Rough Concepts – due 6 Aug *Finial Design – due 8 Aug
5	Schedule next meeting	6 Aug 2020
6	Discuss days that team members are unavailable	*Will-Mari – Unavailable on Tuesdays and Wednesday evenings *Matthew – Unavailable on Wednesday evenings
7	Team member to take meeting minutes	Michael takes minutes
8	Agree on general font size and style for consistency	Calibri (Body) 12
9	Assign member for compilation of document	Michael will do Compiling
10	Past experiences of project	*Set personal due dates before actual due dates to allow for complications and checking. *Use constructive criticism. *Be able to understand what it is that you are doing and also be able to explain it to team members. *Use correct reliable sources *Get preferential time so all team members are accommodated for

Meeting minutes for group 13 - 2

Date: 2020/08/06
Duration: 25 Minutes
Meeting number: 2
People attended: 3
Next Meeting: 2020/08/08

This meeting is to discuss the various designs and list the positives and negatives of each design
Excuses: Michael – Attending to personal matter

Agenda	Main Points	Main Results
1	Go through Will-Mari's Design	*Similar to Michaels first design *No bearing positions *Very simple *Re-design casting *Need support for free gear
2	Go through Emeale's Design	*Good design *Might be expensive *Reverse switch not needed *Input/Output do not rotate in same direction
3	Go through Matthew's Design	*Good Design *Might be to complex *Increase in cost
4	Go through Michael's Design	1* Good simple design 1*Gar ratio will need to be corrected 2*Good design 2*Gear ratios need to be fixed 2*Gear lengths need to be adapted.
5	Find the specifics of the concept that will be chosen	*Find relative cost of the gear box *Will thrust requirements be met
6	Finial design vote and drawings	*Vote will be done on WhatsApp and one person will do final drawing *Team gives feedback about annotations

Meeting minutes for group 13 - 3

Date: 2020/08/08
Duration: 59 Minutes
Meeting number: 3
People attended: 3
Next Meeting: TBA

Meeting was to discuss the chosen design and what needs to change as well as decide on which team members does what. Excuses: Emeale

Agenda	Main Points	Main Results
1	Final Chosen design	Will-Mari's design was chosen
2	Positives about design	*Simple and neat *Few components to keep it cheap *Easy to manufacture and get a case
3	Situation of the bearings	*Need to decide on where the bearings would be placed on the shaft *Decided to place turn the design 180 degrees so driver gears where at the bottom *Thrust bearing would take both the propeller and the axial force of the helical gears
4	Support at centre of the shaft	*Could have been a coupling *Decided that supports attached to the metal case compartment would be better *Could possibly use bearings for support *Decided to create own supports
5	Thrust Bearings	*Need to decide on what thrust bearings to be added to the shafts
6	Gear ratios	*Matthew used his concept design to show us how the calculations would work *Matthew will try various modules and pressure angles to find the best gearing ratio
7	Drawing of the finial design	*Matthew will do a rough sketch and pass on to Emeale *Emeale will do the finial Concept drawing for the team of the chosen design
8	Gear standards	*Michael will do basic research into how the standard for gears is set out
9	Dates	*Finish files by Wednesday and finish Compiling on Thursday
10	Concerns	*Thrust bearing might not be strong enough to take the full load of the propeller as well as the force due to the bearings

<u>Meeting minutes for group 13 - 4</u>

Date: 2020/08/03
Duration: 50 Minutes
Meeting number: 4
People attended: 4
Next Meeting: 2020/08/15

Meeting was to discuss the final hand-in of the report and to double check that everything was in order and all members were happy with the final layout.

Agenda	Main Points	Main Results
1	What was discussed in MDB-344 meeting just before Groups meeting	*Needed lubrication for the bearings and oil bath might be best option
2	Keyways or machined shaft	*We decided it would be better if the gears where separate and held on by keys and keyways rather than machined onto shaft itself as this would waste a lot of material and split casing would be required to fix gears inside gearbox
3	Bearings and Couplings	*Talked about bearings and about placement *Coupling is mainly for support and to make sure that the input and the output shaft are perfectly aligned
4	Gear ratios	*Gear ratios will have to be added to this hand-in, Matthew will send quick summary of those on group to be added to the task
5	Drawings	*Emeale needs to draw casing with the gears in 2D, showing all respective couplings and bearing placements
6	Layout	*Task layout should have annotated drawings, not as currently is with all different bullets *Concept pictures to be in middle of the page and the annotations out to the side.
7	Annotations	*Each member must annotate their own design and add it to the group to be added to the task *These include good and bad things about concepts
8	Circlips	*All gears must have circlips add to them for support and for alignment

<u>Meeting minutes for group 13 - 5</u>

Date: 2020/08/15
Duration: 25 Minutes
Meeting number: 5
People attended: 4
Next Meeting: 2020/08/17

Meeting was to discuss what everyone should prepare for the next task and what needs to be done and has to be handed in.

Agenda	Main Points	Main Results
1	Due date of next task	*The due date for this task is the 28 August in two weeks' time

2	Matthew listed all the things to address on the agenda	*Force analysis of gears, we have to keep it symbolic as we are not yet at the point of calculating the forces *Torque, power and thrust *Get alternative gear ratios *Naming conventions for the bearings, gears and shaft *Start with engineering requirements and house of qualities.
3	View of gearbox and gears	*The input should be on the left of the with the output drawn on the right of the casing for future designs, the main drive shaft will be located at the bottom and the secondary shaft will be located on the top, Matthew will provide this on the group
4	Person doing the force analysis and free body diagrams of the gears and shaft	*Michael
5	Person responsible for alternative gear ratios	*Will-Mari
6	Person responsible for the shaft design	*Emeale
7	Future meetings	*2020/08/17 at 19:00 *2020/08/22 at 16:00

Meeting minutes for group 13 – 1 (w)

(WhatsApp Meeting)
Date: 2020/08/17
Meeting number: 1
Next Meeting: 2020/08/21

First WhatsApp meeting, just a text message session where we stated the work, we would each do from the given list that Matthew supplied

Agenda	Main Points	Main Results
1	Work Distribution	Here's the list of things to do: * Engineering requirements (Will-Mari volunteered) * Find the easiest/most suitable comparison method (house of qualities, etc.) *Determine all parts of design that will affect the reliability. And determine how easy it is to increase the reliability of each

		component. (effect on required safety factor and effect on cost) *Double check all values of torque and duty cycle calculations as well as the gear design procedure. (someone other than Matthew) *Choose a preliminary gear material (research must be done as to the available materials. Must comply with requirements stated in the project document) *Double check force diagrams (someone other than Michael) *Do force and torque calculations using 1st iteration values for module, diameter, etc. (list of values posted on MS Teams under Calculation folder)
2	Timeline	To be done by the following Thursday
3	Michael's Section	*Preliminary gear material
4	Will-Mari's Section	*Engineering requirements as well as the most suitable method for comparison.
5	Emeale & Matthew	*They will split the remaining work up between themselves. Still to be decided.
6	"Meeting minutes"	Meeting minutes done on WhatsApp to be done by Michael

<u>Meeting minutes for group 13 - 7</u>

Date: 2020/08/20
Duration: 25 Minutes
Meeting number: 7
People attended: 4
Next Meeting: 2020/08/22

Meeting minutes:

This was weekly meeting where we decided to who will do what and for when. Spoke about what was discussed in previous machine design meeting.

Agenda	Main Points	Main Results
1	Preliminary points	*Matthew provided group with a preliminary document all that was needed to be discussed in this meeting

		*This involved all the work required for the upcoming task *These included all factors required for the stress and strength calculations as well as the tangential and normal forces, Getting the face width and teeth, etc
2	Stress Calculations	*This will be shared between Will-Mari and Michael *How these will be split is still to be decided.
3	Strength Calculations	*This is to be done by Emeale
4	Wt, Wn and Face width, teeth, etc	*This is to be done by Matthew
5	Due date	*This is all t be done before the next meeting at 4pm on Saturday.
6	Buddy ratings	*Reminder to do buddy ratings before tomorrow
7	Reading	*Everyone to read through comparison methods in the document as well as engineering requirements, and to decide on methods of comparison on day of meeting.

Meeting minutes for group 13 - 8

Date: 2020/08/26
Duration: 40 Minutes
Meeting number: 8
People attended: 4
Next Meeting: 2020/08/22

This meeting was organized to discuss the value and calculations required for Task 2, as well as setting out all the other required work not yet completed for this week's hand-in.

Agenda	**Main Points**	**Main Results**
1	CAD drawings of the gears and shaft	*Emeale will start the CAD drawing once all values and factors have been finalized. *Emeale will also start working on the shaft design however this is only needed for next semester
2	Errors in Smath Code	*Matthew and Will-Mari will go through the calculations in Smaths and find out the errors that are causing the allowable and current stresses to be equal to one other.
3	Getting factors to get appropriate safety factors	*Matthew will go through the equations and fiddle with the different factors to see what can be done to raise the safety factor above one. One of the values mentioned to change is the modulus from 3 to 4 or 5mm. *Can also change the overload factor from 2 to 1.75 *Can change the cycle factor from 0.9 to 0.95.
4	Seals	*Will-Mari looked at all the different seals that we can add to the design and found out that radial lip seals would the best option and so these would

		need to be included in the design drawing of the gearbox
5	Casing	*Members need to look at the casing and see where the seals will fit
6	Fixing gears on shaft	*Research needs to be done to find out how one can secure the gears on the shaft without using circlips to prevent them sliding up and down on the shafts.
7	*Minutes and compiling	*must compile and do minutes for this week's hand-in

<u>Meeting minutes for group 13 - 9</u>

Date: 2020/09/25
Duration: 28 Minutes
Meeting number: 9
People attended: 4
Next Meeting: 2020/08/22

Meeting agenda was to figure out the work needed for the upcoming hand-in, as well as how the work would be split between the members

Agenda	**Main Points**	**Main Results**
1	CAD drawing of the cast, shafts, couplings, retaining rings and bearings	*Emeale will create the final document with all the necessary detail in CAD
2	Double checking shaft analysis and bearing calculations	*Michael and Emeale will double check these equations
3	MATLAB, AGMA and shaft analysis	*Matthew has thoroughly created and modified the smaths files for shaft analysis, co-complied by Michael. Matthew also created the MATLAB files which plot the graphs for shear, bending, defection and torsion *Michael coded the equations used for shaft analysis in the smath document
4	Keyways and retaining rings	*Will-Mari will do this section; members will have to help where they can and double check. Will-Mari will do the research as well as all the calculations needed for this section of the report
5	Dates	*On the date of the meeting the shaft analysis and graphs with necessary shaft diameters t be completed as this is needed for other sections
6	Fillet	*Since the final shaft hasn't yet been established the fillet radius at the shoulder will be set to 2 mm

<u>Meeting minutes for group 13 - 10</u>

Date: 2020/10/4
Duration: 58 Minutes
Meeting number: 10
People attended: 4
Next Meeting: 2020/10/13

Meeting was to discuss previous marks and how to improve going forward. Discussion on what to do for upcoming days for next hand-in.

Agenda	Main Points	Main Results
1	Previous Marks	*Group did not do to well, discussion on how to improve going forward, what went wrong and why
2	Best Lubrication method	*Matthew showed basic ideas on how to lubricate gears with baths, same oil was decided to be used. Grease nipple not feasible, baths will need supports to side (EDIT: design was later changed)
3	Corrosion protection, material selection for casing and CAD	*Emeale volunteered to do these tasks for the next upcoming hand in
4	Casing design	*Will-Mari and Matthew will design the casing and hand this to Emeale to do the CAD
5	Welding	*Michael will do the welding for the next hand in, get the symbols and minimum width of the weld.

<u>Meeting minutes for group 13 - 11</u>

Date: 2020/10/13
Duration: 20 Minutes
Meeting number: 11
People attended: 4
Next Meeting: TBA

Design iteration and final report hand in discussion. What still needs to be done and what needs to change

Agenda	Main Points	Main Results
1	Iteration	*Changes in the gear width need to be made in order to reduce the weight of the design and since the reliability can be lowered, this will be done to lower costs.
2	Because of iteration	*Changes will now need to be made to the keyways, retaining rings and gears. CAD will need to now include shoulders and retaining rings.
3	Pugh chart	*This still needs to be added so it can be placed in the final report
4	Due date	Finish the respective sections by 26 October