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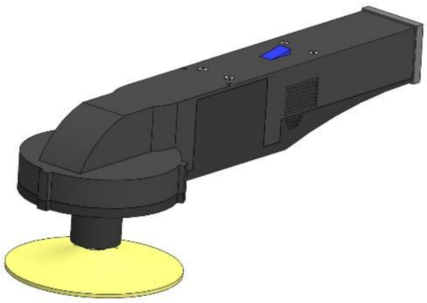

Assignment 3 – Detail Design

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2. Introduction

This report describes the detail design of a cordless car/vehicle polisher and is the capstone of the design process. This phase succeeds the concept and basic design phase. As stated in the previous assignment, a car polisher uses rotating motion of a polishing head, combined with a specialised polishing fluid, to remove oxidation, light imperfections and scratches. Electric energy stored in a battery pack is turned into rotational-mechanical energy of the motor shaft, which is then translated to the output shaft via means of a bevel gearset.

The final concept sketch, an output from basic design, is produced in Figure 1 below. The concept sketch shows the use of a bevel gearset to transmit power from the horizontal plane to the vertical plane. The motor sits in a housing parallel to that of the battery pack while an adequate housing design has been developed in the detail design. No substantial deviations from the final concept sketch have been made; the means of power transmission, layout of the device, packaging and components have remained the same. However, more machine element components (such as extra bearings and a means of connecting the motor shaft to an intermediate shaft) have been added to the final design. These were generally added after engineering calculations and detailed sizing of components was completed.

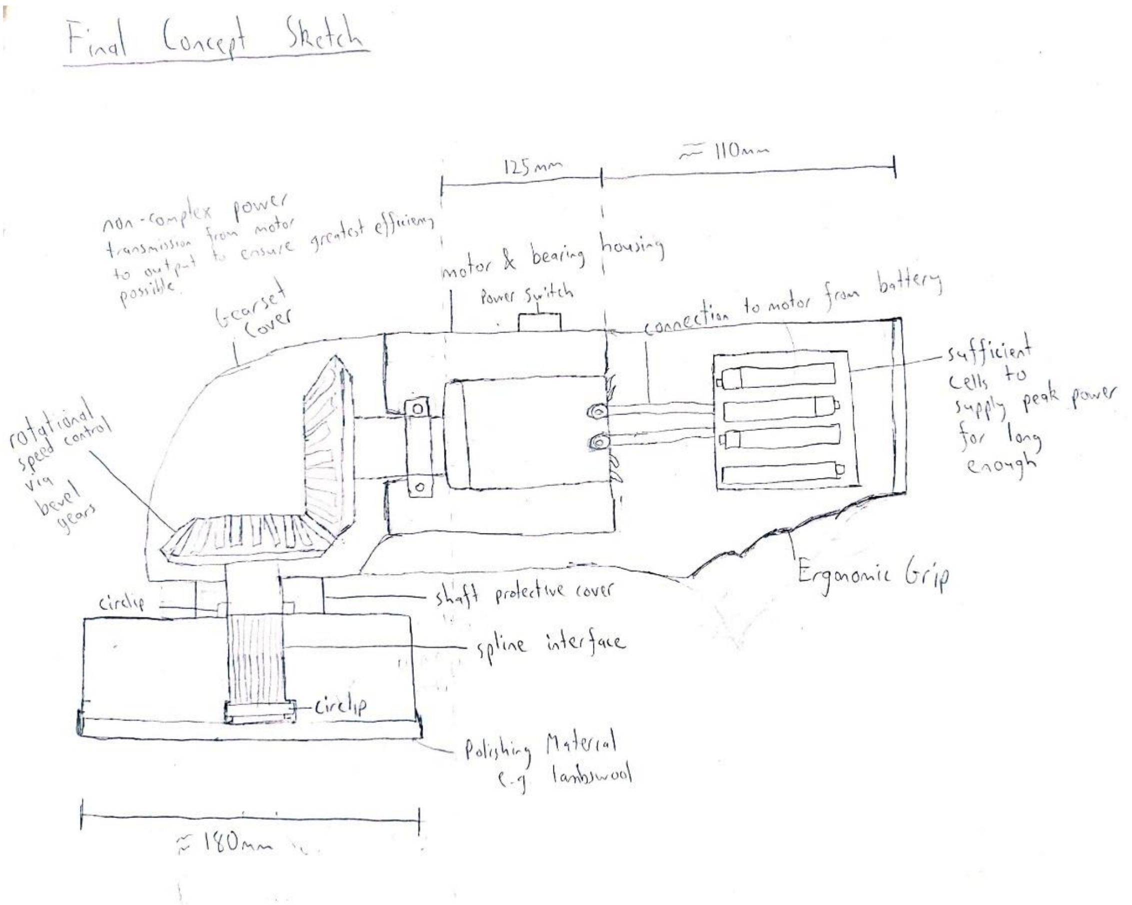

Figure 1 - Basic Design Final Concept Sketch

The detail design phase involves performing detailed sizing and engineering calculations based on operating and maximum load cases, determining shaft characteristics (shaft strength and thus minimum allowable diameters and maximum allowable lengths), calculating gear attributes, calculating various operating scenario to ensure adequate performance, as well as bearing life assessments. These calculations ensure the product will meet the specifications set and can be found in Appendix A.

A detailed CAD model is then developed (using the outputs of the basic design phase as well as the sizing calculations). Proper motor locating features, load path design, sensible layout and packaging, external casing design, design for manufacture and assembly, conceptual electronic design, design for simplicity and elegance and proportional design are all considerations for the CAD model. Descriptions of the above are found throughout the report in relevant design description sections.

A BOM costing is then included in order to determine a cost for the initial prototype. Cost-reduction of the mass-produced parts is considered, and a value-engineered costing is determined. The total cost is compared with the initial target cost and a profit-margin is added. Overheads, development and ramp-up costs, tooling costs and other expenses are then motivated and stated. Following this, a development plan and cash-flow schedule with an indication of break-even point is displayed. This includes a roll-out plan with various adjustments made to show profitability within 2 years from the prototype manufacture.

This detail design report includes all information to show that the product will perform as designed (and required), so that no major surprises will amount during the prototype and testing stage.

The description section of this report begins with motor location, shaft connection and cooling provisions, while force transfer and shaft support (namely bearing design) follow this. Gearbox design and tool interface are described next, with component packaging (and ergonomic design), following this. This section is concluded with a description of primary material selection and manufacturing considerations.

The final specification table acquired from the basic design phase is commented on to show the success of the current design's compliance to the specifications while reasonable adjustments are motivated.

A description of the risks, drawbacks and future modifications of the current design are announced before a closing paragraph on the success of the design process is presented.

3. Detail Design Description

Detailed descriptions of particular aspects of the detail design process are included in this section, with focus placed on the engineering logic followed and reasoning used, to create the final design.

3.1 Motor Location, Shaft connection and Cooling provision

3.1.1 Motor Location and Shaft Connection

The motor is the heart of this product and it was initially requested that a product be created around it. Thus, many of the features were designed around the motor, or bearing the motor in mind (namely cooling of the motor, connection to the shaft, ensuring no limits motor limits were exceeded, and ensuring no force was transmitted to the motor).

Proper location of the motor is critical in ensuring that the drive shaft interacts correctly with the intermediate shaft and that no extra forces are created in the machine (ensuring optimal efficiency of power transmission). Location of the shaft is reliant on the housing it is connected to, as well as the power transmission device (cylindrical sleeve of the intermediate shaft in this case) that it is connected to.

As seen in Figure 3 below, the motor is attached to a housing bracket (pink) by means of three M4 countersunk screws. These do not provide location of the motor but ensure the motor is attached properly to the bracket. The housing bracket is attached to the top casing (transparent grey in Figure 4) by means of two nuts and countersunk bolts. The conical heads of the countersunk bolts help in providing location of the bracket (and thus motor) relative to the top casing. The holes for the bolts on the top casing are located precisely on the top casing and ensure that the motor is located in the horizontal plane. Slots on the bracket (to fit M4 bolts) have been created so that the bracket can be attached to top casing with ease and so that the motor is not over-located (as this allows the bracket to be shifted to account for manufacturing that may not be perfectly machined). This can be examined well in Figure 4 below.

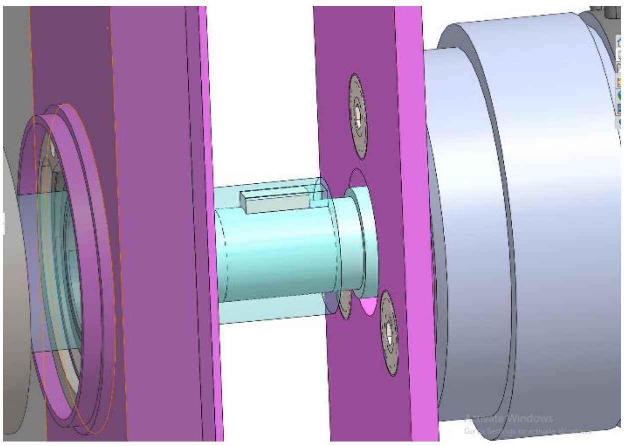

Figure 2 - Close-Up of Motor to Bracket Attachment

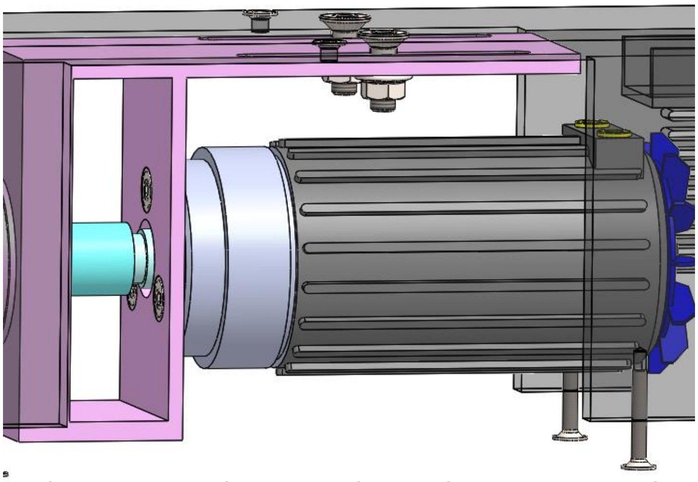

Figure 3 - Bracket to Casing and Motor Connection

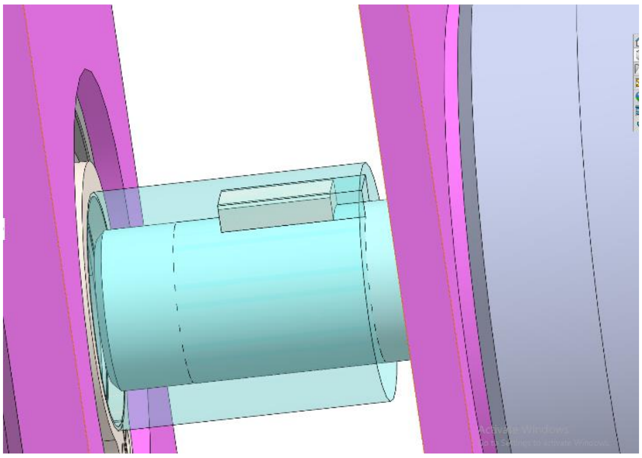

Figure 5 - Shaft Connection Close-Up

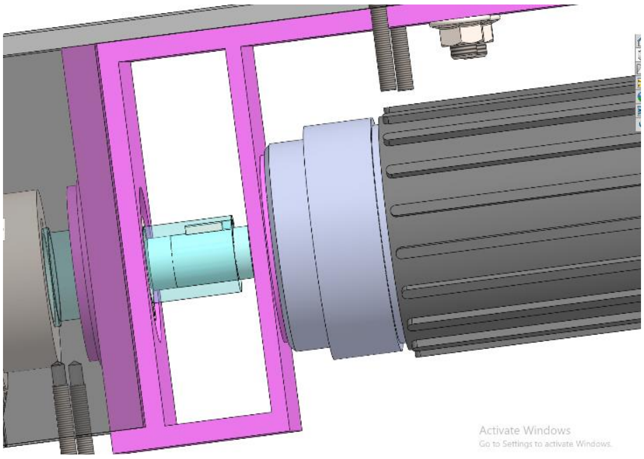

Figure 4 - Shaft Connection Zoomed Out

The connection between the motor and intermediate shaft can be examined in Figures 4 and 5 above. The motor's shaft is connected to the intermediate shaft by means of a key. The intermediate shaft contains a bored hole with a keyway suitable for the size of the motor's keyway (the bored hole is the same as that of the motor shaft's outer diameter). A key is then inserted onto the motor shaft and the cylindrical sleeve of the intermediate shaft is slide over the key and shaft of the motor. The description of bearing locations can be found in the next section that regards force transfer and shaft support.

The key ensures proper torque transfer and the bearings ensure proper axial and radial location of both the motor shaft and intermediate shaft. Figure 6 below shows a full view of the motor connection to intermediate shaft and intermediate shaft connection to output shaft. The locating features of the aluminium bracket are shown too. Shaft strength calculations and preliminary shaft sketches can be found in Appendix A – Section 11.5 Shaft Calculations. Both the input and output shaft were concluded to have more than enough fatigue strength, a diameter much greater than that of the minimum required diameter, and a length much smaller than that of the maximum, thereby deeming the shafts fit for use in the design.

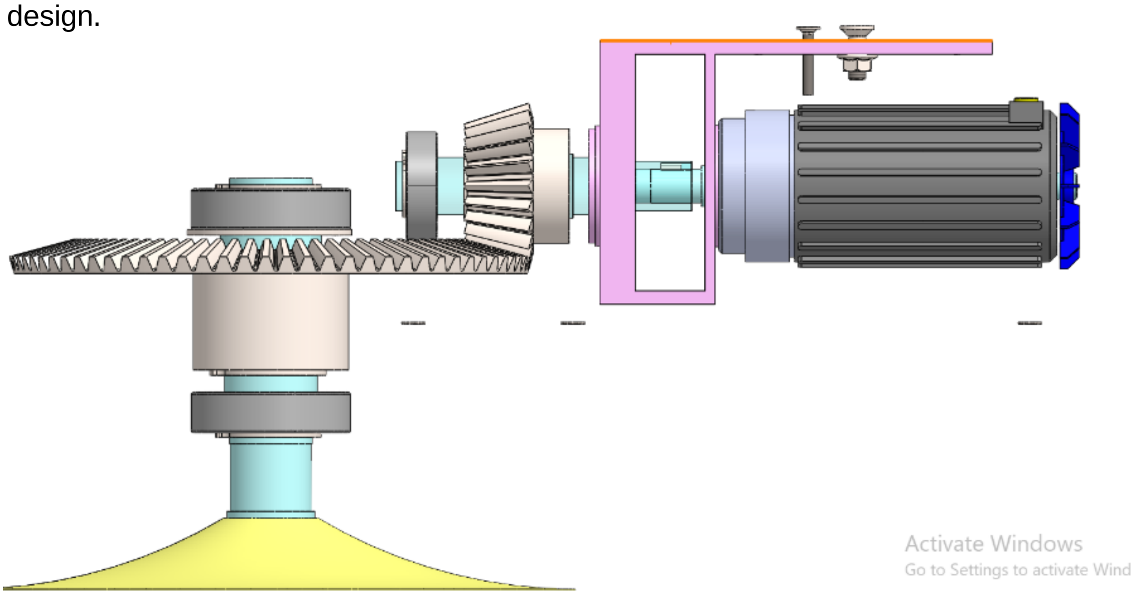

Figure 6 - Full View of Mechanism

3.1.2 Cooling Provisions

Although no detail design calculations have been done regarding cooling of the motor, many provisions have been made. In designing the cooling provisions, inspiration was taken from the work of Huang et al. [1], who created an air flow housing for cooling of motors. The air flow pattern allows the fan to pull air through the impeller as well as through the holes created for the blades. The fan connected to the motor is that of a centrifugal fan. Air enters a centrifugal fan axially and passes through the impeller radially before being collected in the volute casing and being discharged at right angles to the inlet flow [2].

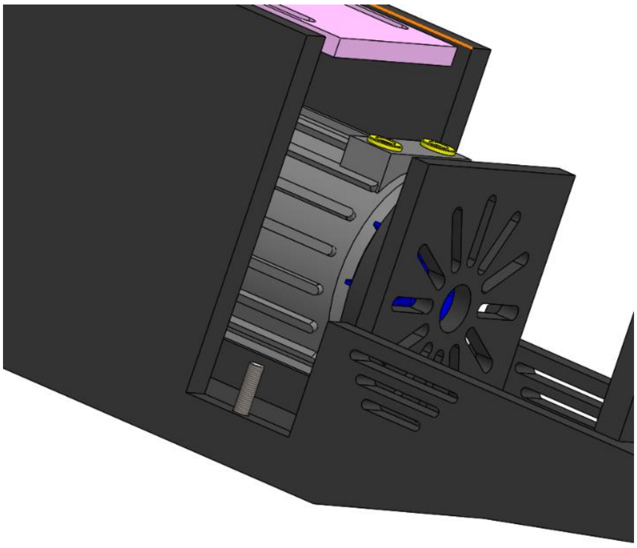

Figure 8 - Air Flow (Top Casing Hidden)

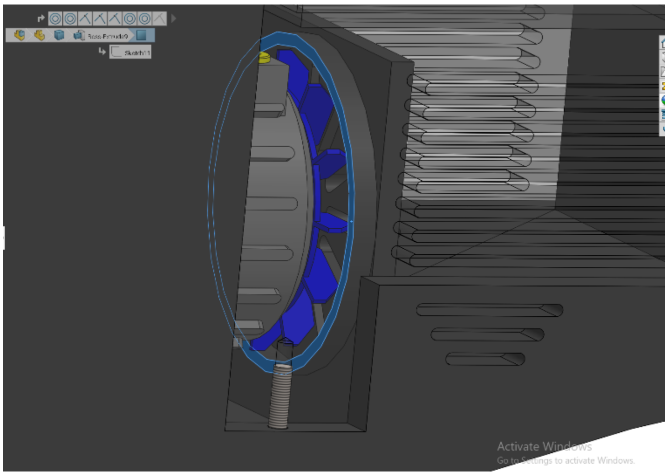

Figure 7 - Air Flow (Back View)

An air flow pattern (Figure 8) was created as part of the bottom casing and has a hole in the middle for the impeller to pull in air as well as holes for the blades to do a similar job. A small circular casing (seen and highlighted in Figure 7 above) ensures all flow is directed towards the vanes/blades, which direct the airflow towards the fins. A gap between the air flow extrude and the battery has been created so that air can be pulled towards the motor (through the ventilation holes that have been created in both the top and bottom casings). This creates a small air chamber where air can flow from the outside, through the ventilation holes, through the air flow extrude, around the blades and finally, through the fins, to provide cooling to the motor. The air chamber can be examined well in Figure 11 on the following page. The aluminium housing bracket (as described previously), sits above the motor and provides a form of convection cooling as well. The following page shows Figures 9 to 11, which all give visual representations of the cooling provisions.

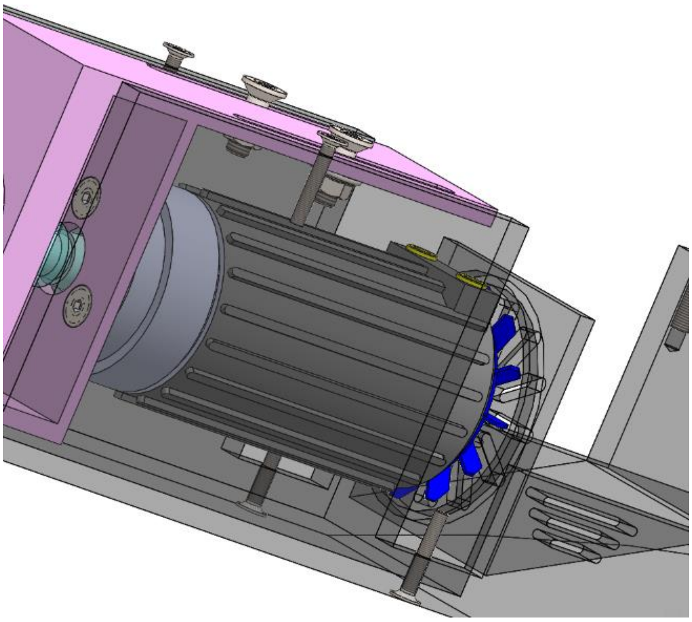

Figure 9 - Zoomed Out and Transparent Cooling Provision

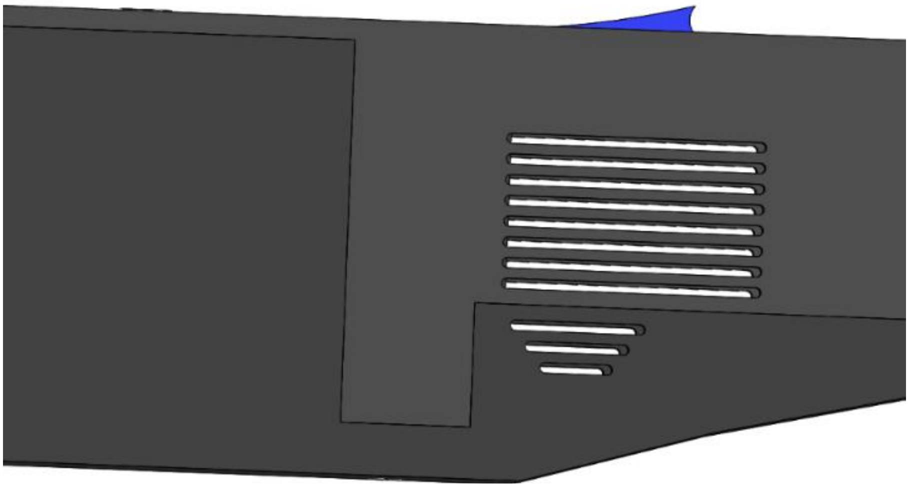

Figure 10 - Ventilation Holes

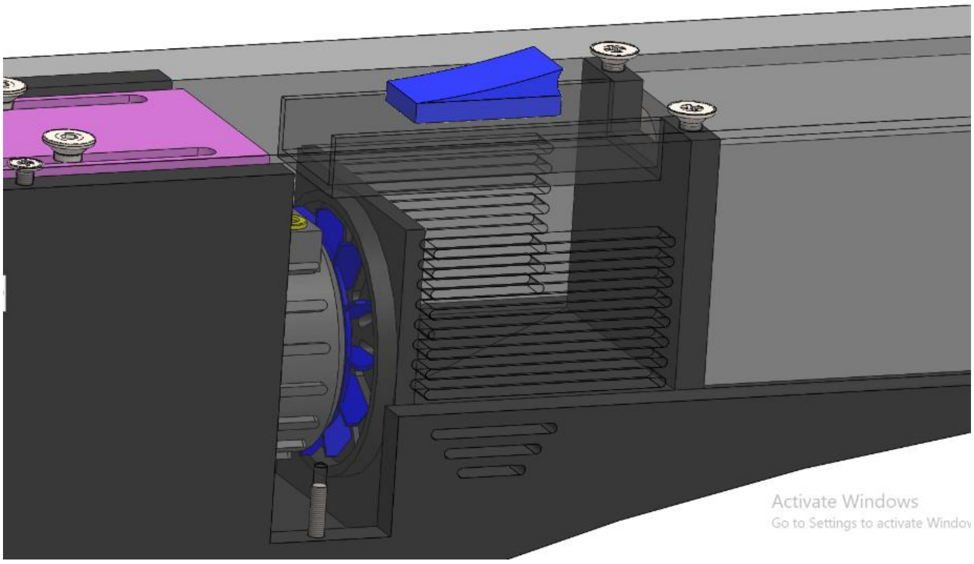

Figure 11 - Overview of Cooling Provisions

3.2 Force Transfer and Shaft Support

The main contributor to force creation in the design is that of the bevel gearset. The gearset contributes radial, axial and tangential forces, which need to be diminished by some means. The calculations for the magnitudes of these forces at maximum load and operating loads can be seen in Appendix A: Section 11.3: Gear and Bearings Calculations.

I made use of bearings on the intermediate shaft and output shaft to transfer forces away from the gears and shafts and towards the top and bottom casings (examine Figure 12 below). Calculations with regards to bearings can be found in Appendix A: Section 11.3. The bearings chosen were 6005 for the output shaft and 6002 for the intermediate shaft. The required load ratings for the intermittent and output shafts are 1292N and 894N respectively (at maximum operating conditions). The 6002 and 6005 bearings have dynamic load ratings of 5850N and 11900N respectively. Bearings with less load capacity were considered but were eliminated due to their much higher costs (the 6005 and 6002 bearings from SKF are some of their most produced bearings). The much higher load ratings also mean that any excess forces that amount due to the polishing head in operation will be transferred safely to the casing without compromising safety and life of the product. The bearings ensure that a life of 14 000 hours with a 90% reliability can be maintained. This life cycle time and reliability was chosen as it is typical of machines with intermittent use but where reliability is of great importance [3]. Calculations that affirm this can be found in Appendix A: Section 11.3.

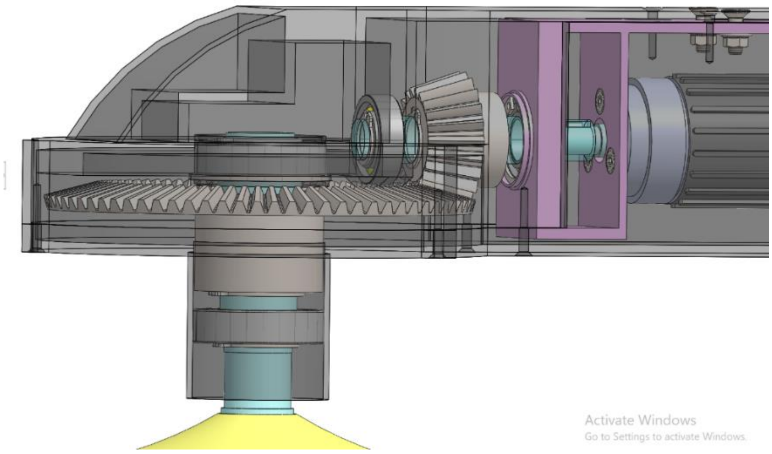

Figure 12 - Overview of Locating Features and Force Transfer

The bearings ensure that no force is transmitted to the motor shaft and a brief overview of where all the bearings are located relative to the motor's shaft can be seen in Figure 13 on the following page. This Figure also shows which bearings are locating, non-locating and the way each bearing has been located.

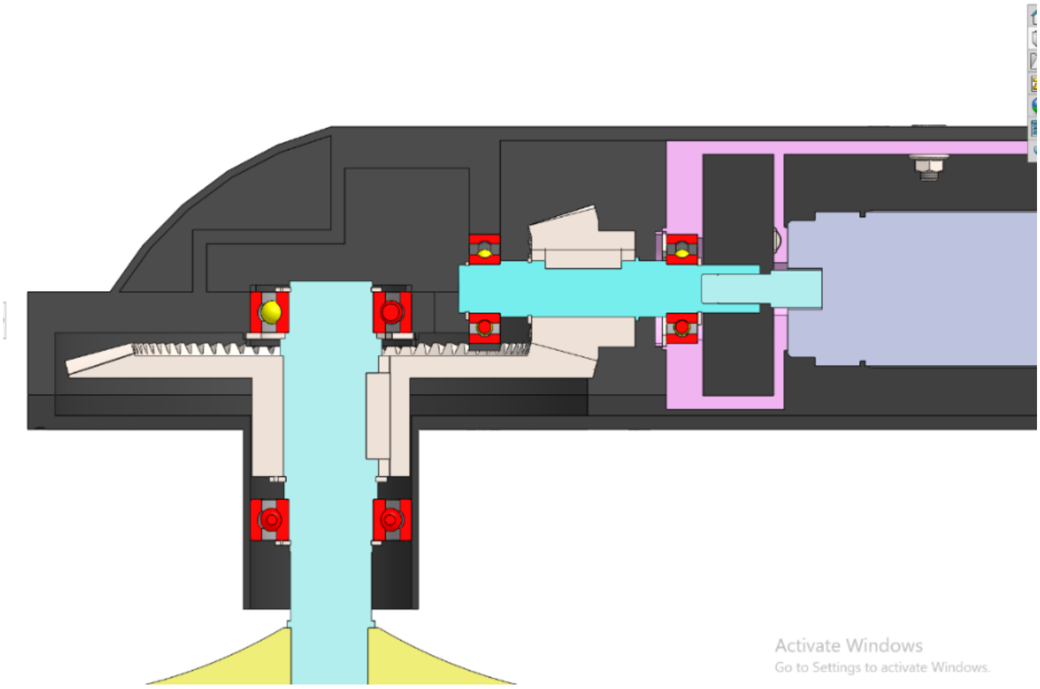

Figure 13 - Bearing Locations

The bearings provide the necessary shaft supports as well as providing location for the intermediate, output and motor shaft. Each gear is straddle mounted to ensure best transfer of force and to ensure equal forces are distributed to each bearing. Both the intermittent and output bearing pairs are located by means of a locating and non-locating pair.

The bearings which sit on the intermediate shaft are housed in the top casing and aluminium bracket (Figure 14 on the next page). The bearing on the left is non-locating and so the top races are not restricted. The inner races, however, are restricted by means of an external circlip and a shoulder on the intermediate shaft. The locating bearing has its inner races located by means of a shoulder on the shaft and an external circlip as well. The outer races of this bearing are located by means of a shoulder in the housing, and an internal circlip in the bracket. To assemble the configuration of the input to intermediate shaft, a key is placed on the input/motor shaft and the bracket is attached to the motor. The intermediate shaft has the locating bearing placed on it as well as its external circlip, before pushing the bearing into the housing and the cylindrical intermediate shaft's sleeve over the motor shaft, and onto the key. A key is then placed on the intermediate shaft and the gear is pushed over the key onto the shaft (which is then located by another external circlip). The non-locating bearing is pushed onto the other side and another external circlip is placed onto the intermediate shaft. Finally, the top casing slides onto the non-locating bearing and can be attached to the motor bracket to complete location of the motor and intermediate shaft. Figures 15 and 16 on the following page show a better view of each bearing in its housing with attached circlips.

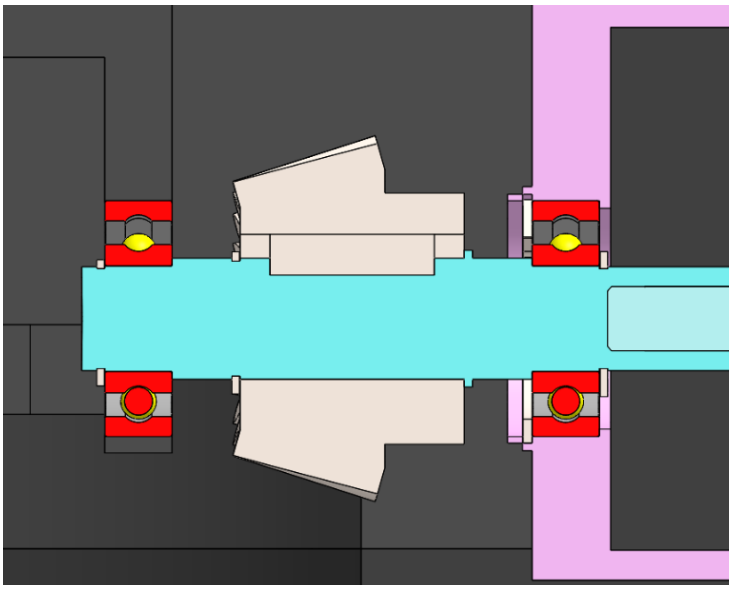

Figure 14 - Intermediate Shaft Cross Section

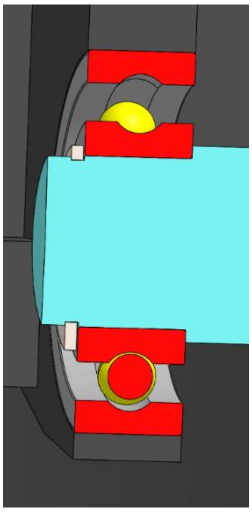

Figure 16 - Int. Shaft Non-Locating Bearing

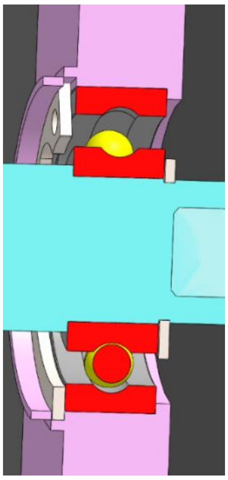

Figure 15 – Int. Shaft Locating Bearing

The bearings on the output shaft were designed and work in a similar fashion to those on the intermediate shaft. The top bearing in Figure 18 on the following page is the locating bearing, while the bottom bearing is the non-locating bearing. The two bearing configurations together, provide location of the output shaft relative to the housings.

The locating bearing has both its inner and outer races restricted. The inner races are restricted by means of an external circlip and shoulder on the shaft. The outer races are once again restricted by means of an internal circlip (with a groove made inside the top casing) and a shoulder provided by the top casing. The bottom bearing is the non-locating bearing and has its inner races restricted by means of a shoulder on the shaft and an external circlip placed on the shaft. A good view of both bearings can be seen in Figures 17 and 19 on the following page.

Figure 17 - Output Shaft Locating Bearing

Figure 19 -Output Shaft Non-Locating Bearing

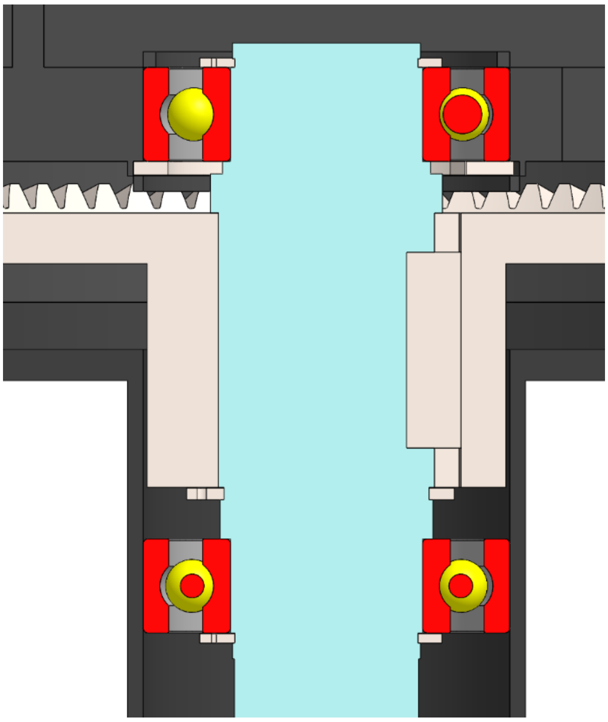

Figure 18 - Output Shaft Cross Section

The intermediate to output shaft configuration is set up as follows: the top bearing is pushed onto the shaft and the external circlip is placed on the shaft. The shaft-and-bearing configuration is then pushed into the top casing and the internal circlip is placed into the housing to fully locate the bearing. The key is placed in the output shaft and the gear is pushed upwards onto the shaft to mate with the pinion. The external circlip is then placed behind the gear and the bottom bearing is pushed onto the shaft, where another circlip is placed to locate the inner race of the bottom bearing. The bottom casing is then pushed upwards and over the shaft before being attached to the top casing. The rotating head can then be attached, and the battery can be inserted and covered too (and this completes assembly).

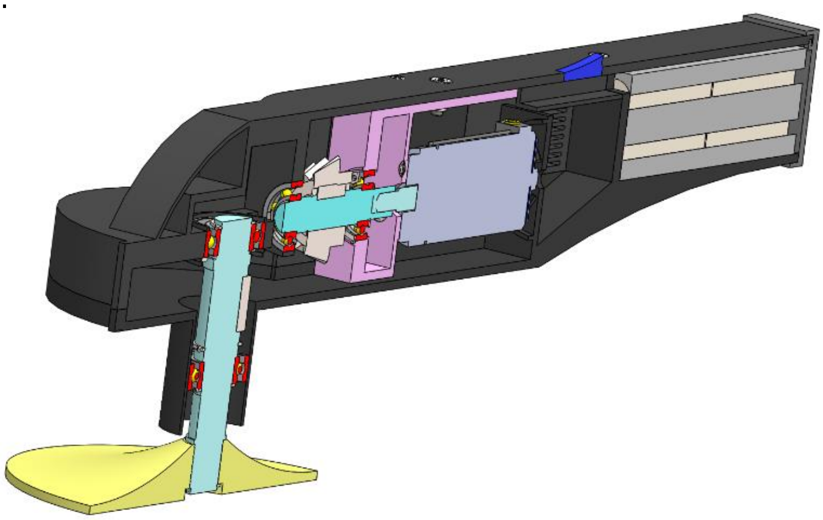

Figure 20 - Cross Section of Completed Assembly

3.2.1 Load Path Design

Figure 21 below shows the analysis of how load path design was incorporated into the detailed design of the product so that the shortest, most direct path with the smoothest force flow lines could be maintained. The lines in orange show predicted flow of force from the polisher head and gears, through the shaft, through the bearings, and into the housing. The forces will then be transferred to the hands of the operator. These forces have been made as low as possible to ensure maximum comfort of use.

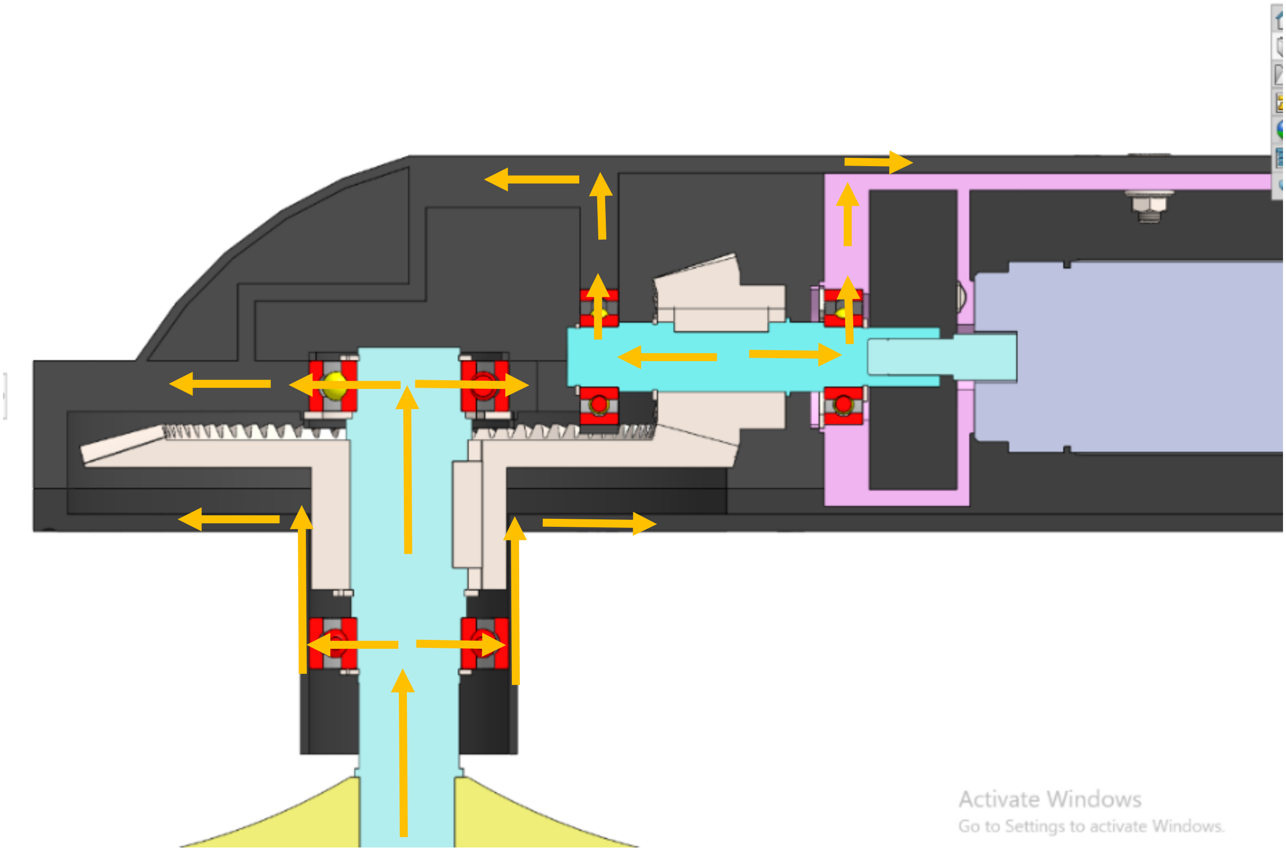

Figure 21 - Load Path Design

3.3 Gearbox/Linkage Design and Tool/End Fixture Interface Design

A gearset provides a means of transmitting power from one shaft to another (intermediate to output in this case). Gear reductions are also an important feature provided by gears and have been taken advantage of in this design to provide a higher torque output and lower output rotational speed.

This design makes use of two bevel gears to transmit power. The intermediate and output shaft axes are perpendicular in this design case and allow for rotation to be translated from the horizontal plane to the vertical plane.

As can be seen in Figure 22 below, the pinion (driving gear) is smaller than the driven gear and allows for a reduction in rotational speed while increasing torque at the output. The rotational speed of the intermediate shaft is 7986 rpm under operational conditions (and is 4108 rpm at maximum load conditions). The rotational speed of the motor shaft was calculated after the battery configuration was determined. This can be viewed in Appendix A: Section 11.1 (note that these calculations were done in more detail, but the final battery configuration chosen was the same as that chosen in the basic design phase). The maximum and operational conditions were calculated too and can be found in Appendix A: Section 11.2. This section also confirms that the motor limits are never exceeded in any operating condition.

An output rotational speed of 2200rpm was desired (as stated in the final specifications table in basic design). This meant that a reduction ratio of 3.63 would be required. This was rounded to 3.5 to ensure that a whole number of teeth could be provided and as a reduction ratio of 3.5 is a more standard reduction ratio (meaning sourcing the gears or producing the gears would be easier). This implies that the output speed would be 2281.71rpm (which meets the final specification).

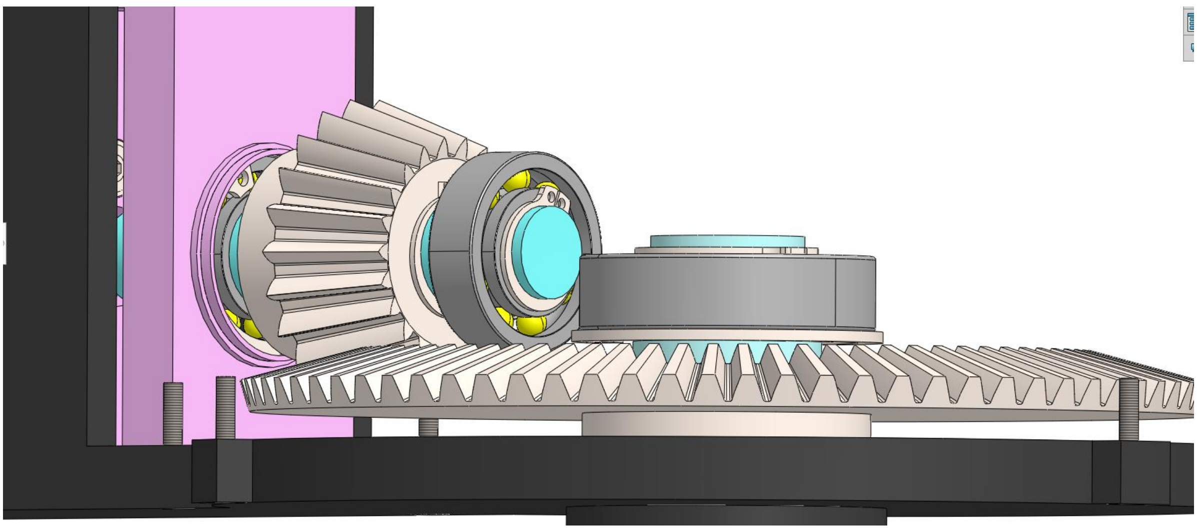

Figure 22 – Close-Up of Mating Gears

The motor torque at operating conditions was found to be 0.087Nm (see calculations in Appendix A: Section 11.1). With the reduction ratio of the gearbox, this translates to an output torque of 0.305Nm.

The maximum output torque was found to be 5.432Nm while the maximum motor torque was found to be 1.552Nm. These occur when the motor and output rotational speeds are 4109rpm and 1174rpm respectively. This output torque at maximum load is relatively high for a cordless polisher and means that enough torque can be provided by the machine in required conditions. An example of this would be when the polisher encounters a deep cut or heavily raised imperfection.

The module of the gearset was chosen to be 2mm. This was chosen as it ensured the that the forces on the gear and pinion are low (and thus the forces on the shaft are low as well). It was also chosen so that both the gear and pinion could be overhung (the bearing sitting on the end of the pinion needs enough space to be held in the housing without interfering with the face of the gear). Another detailed description of this can be found in Appendix A: Section 11.4.

A face width for the teeth was chosen to be 20mm as this is approximately half the diameter of the pinion and greater than nine times the module. Note that there is no standard for calculating face width and these are common reasonings used in industry based on experience and trying to minimize backlash [4][5].

In order to meet the gear ratio and required characteristics of the system, the pinion was chosen to have 22 teeth, and thus a diameter of 44mm. This followed from the reasoning behind choosing the module, particularly noting that the bore of the pinion needed to be large enough so that a key could fit in without intersecting the teeth and so that the overhung bearings could be housed properly. This means that the gear has 77 teeth and a diameter of 154mm (see Appendix A: Section 11.4).

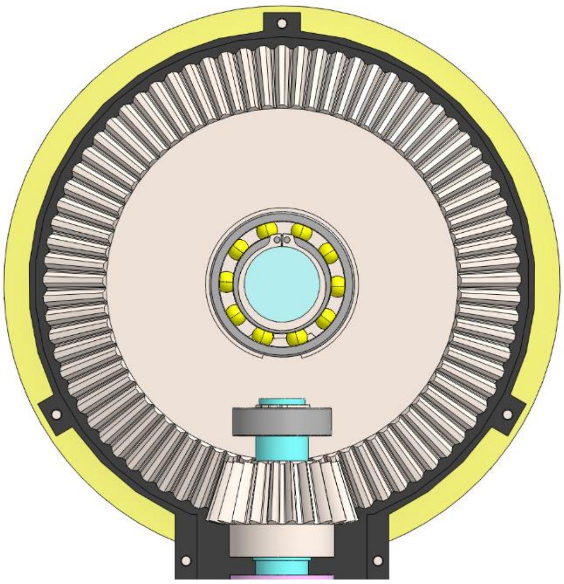

Figure 23 - Top View of Gears

A pressure angle of 20 degrees was assumed as this is the standard for bevel gears. It should be noted that a similar Chiaravalli gear [6] with a reduction ratio of 3 and module of 2 was used as the basis of sizing and was initially the gear that was going to be used until analysis deemed it not to be fit for this design.

Gear forces were found by first doing a force analysis on the gears (which can be seen in a sketch in Appendix A: Section 11.4) and then using engineering formulae to determine the forces based on a typical bevel gear analysis. In the maximum operating case, the tangential gear forces were found to be 80.62N and that the radial force on the pinion (which is the same as the axial force on the gear) was found to be 28.214N. The axial force on the pinion (radial force on the gear) was found to be 8.061N. The presence of an axial force and the fact that the radial forces are greater than 25N on the pinion (the largest radial load the motor can handle), mean that a bearing configuration or some other means of force transfer away from the shafts and gears, had to be used.

The gears were analysed and assumed to be steel (as steel provides a sufficiently high strength, a long life, is not expensive and is easily machined). A surface strength and life analysis of the gears found that the gears would run for approximately 9*10^10 cycles (Appendix A: Section 11.4), more than enough life required. Figure 23 on the previous page and Figure 24 below, show the placement and sizing of the gears relative to the shafts and bearings.

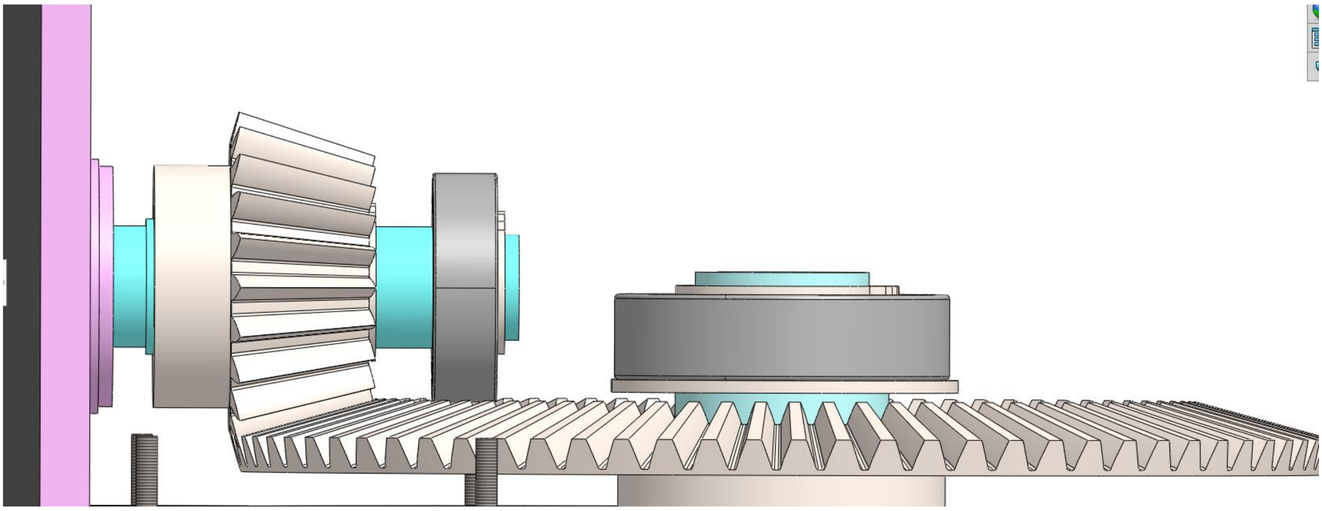

Figure 24 - Side View of Gears

The tool fixture interface is another component designed for this device. No calculations were done regarding this, as the shaft was already deemed to be strong enough and the focus of design was placed on the motor. However, a spline interface was designed for the output shaft to connect to the polishing head. The polishing head is thus radially located by the spline. A shoulder on the output shaft and an external circlip provide axial location of the polishing head. The spline design can be seen in the output shaft sketch in the Appendix A: Section 11.5). Figure 26 below shows a visual representation of the locating features and how the polishing head is attached to the shaft while Figure 25 shows the spline in more detail.

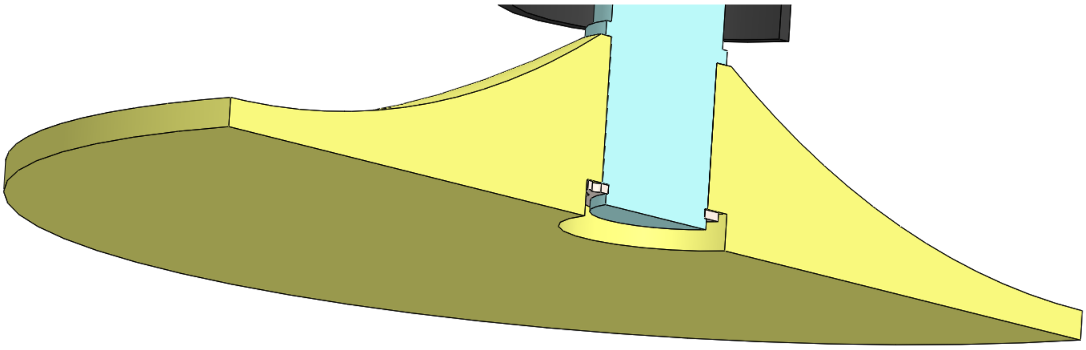

Figure 26 - Tool-End Locating Features

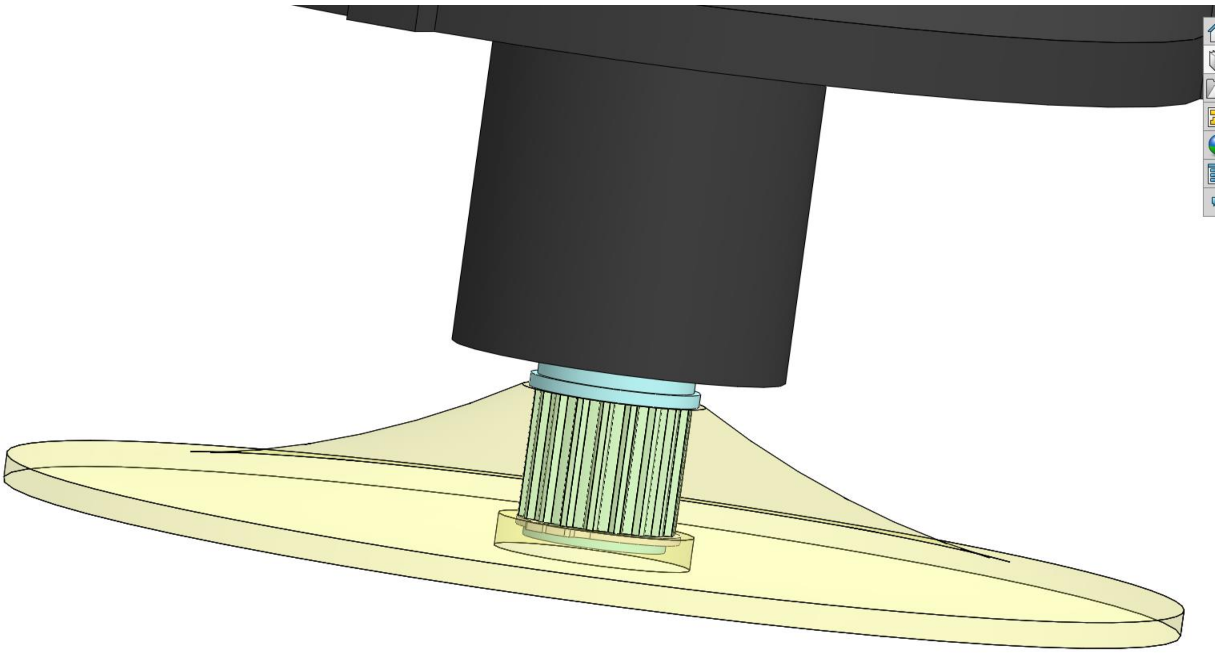

Figure 25 - Spline Design and Interface

3.4 Component packaging, including ergonomics considerations

The top and bottom casing of the product contain most of the packaging for machine components. This design method was taken in order to minimize the amount of parts and make assembly as simple and fast as possible. The design was based on the foundations of design for manufacturing and assembly, which aims to reduce the number of components, simplify the design, maintain symmetry and combine features. The overview of the assembled casings over the internal components can be seen in Figures 27 and 28 below. The outer casings aim to provide a sleek, ergonomic and simplistic design, that protect the inner mechanical components and ensure safety of operation. The top casing is symmetrical about one plane and so is the bottom casing. The casings fit onto the internal components easily and fit together seamlessly to create many integrated internal packaging and features (such as housing all bearings, housing the battery configuration, protecting the output shaft and providing space for the switch and wires).

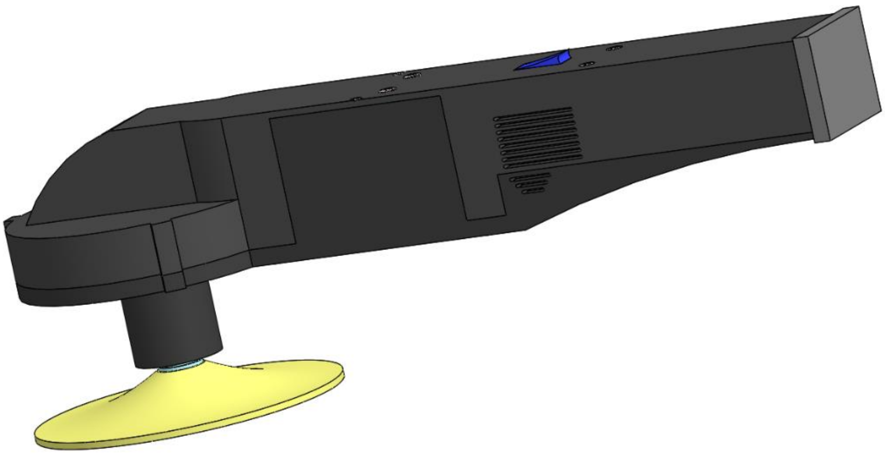

Figure 27 - Full Casing View 1

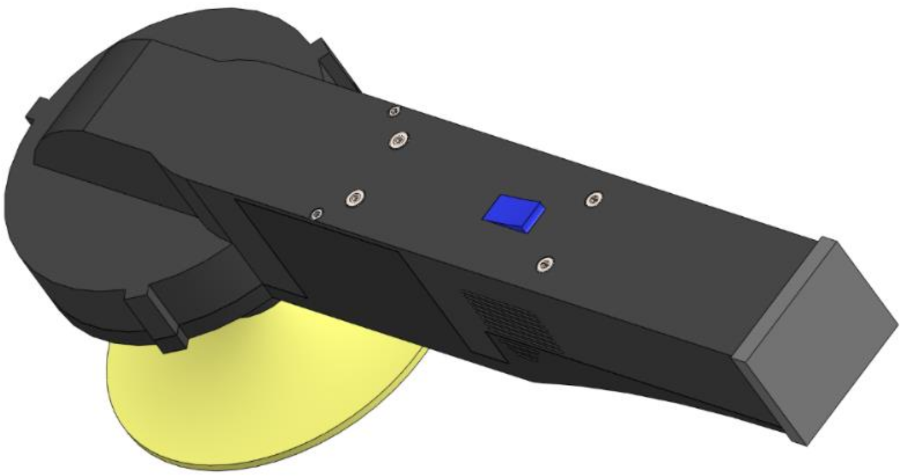

Figure 28 - Full Casing View 2

The top casing provides locating features for the output shaft's locating bearing as well as providing a non-locating housing for the intermediate shaft's non-locating bearing. This can be viewed well in Figure 30 below. A groove is made on this casing for the locating bearing to allow for an internal circlip to be inserted, and a shoulder is created to locate the other outer race of the bearing. Protection of the large gear is provided mainly by the top casing as well (as this covers the teeth of the gear). The pinion is protected by the outer wall of this casing as well. The large gap in between the walls of the top casing (seen in Figure 30 below) ensures that the motor bracket (as described in previous sections) can be properly attached to the top casing during assembly. The bracket is attached to the top casing by means of two countersunk M4 bolts and two M4 nuts.

The top casing contains most of the ventilation holes/slots that provide air to the motor however, all other cooling provisions are found on the bottom casing. Space for the wires to taken from the battery configuration to the motor is made and can be seen in Figure 29 and 31 below. A hole for the switch is made just above this.

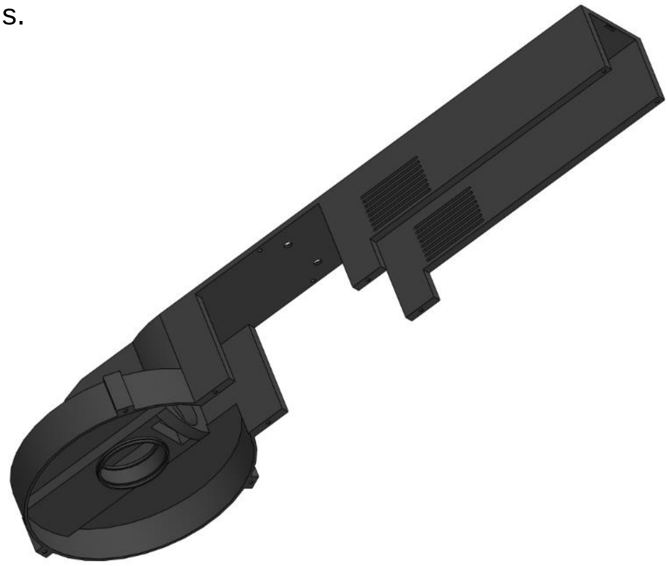

Figure 30 - Top Casing

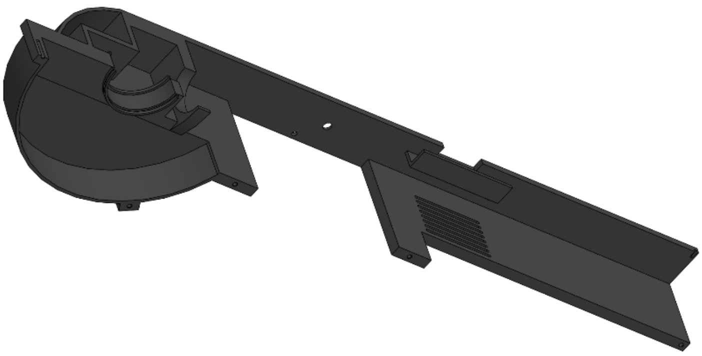

Figure 29 - Top Casing Section 1

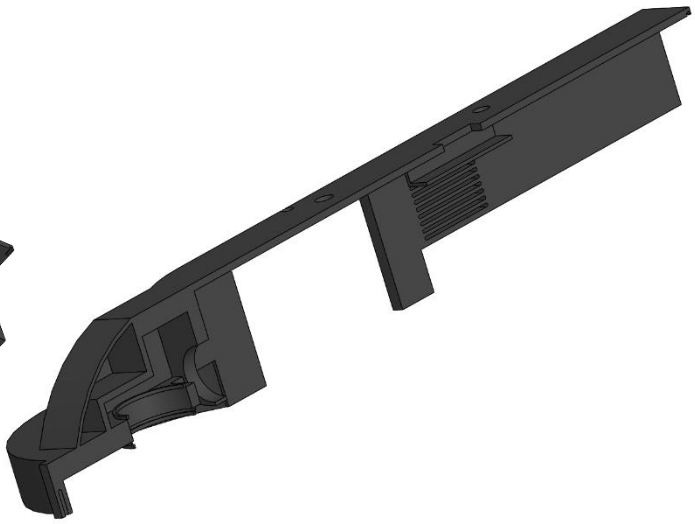

Figure 31 - Top Casing Section 2

The housing for the output shaft's non-locating bearing is part of the bottom casing, which also serves the dual purpose of protecting the output shaft. Most of the cooling provisions are found on the bottom casing too. Ventilation holes, the air flow extrude, and a wall to create an air chamber can be seen in Figure 33 below. This air chamber wall also locates the battery configuration and provides space for the top casing's wire provisions to fit in. The side walls in front of the air flow extrude provide protection of the motor.

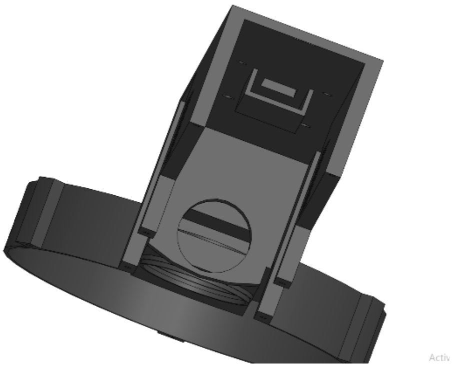

Figure 32 -Top Casing Back View

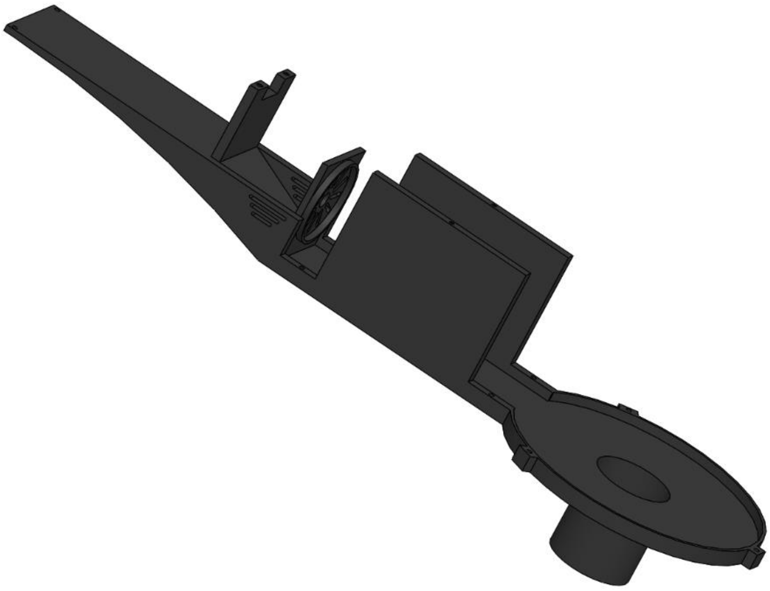

Figure 33 - Bottom Casing Full View

When the top and bottom casings are connected, a space for the battery configuration is formed. A removable battery cover seals the battery configuration in and allows for easy replacement and maintenance. The removable cover can be seen in Figure 34 below. The cover clips into the top casing to keep the battery configuration sealed in.

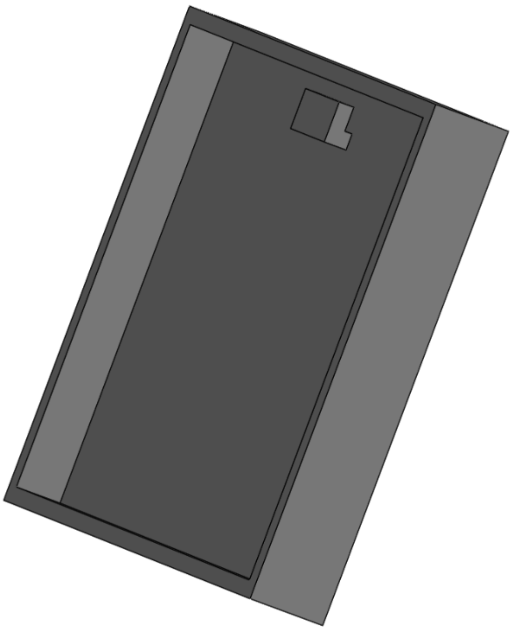

Figure 34 - Removable Battery Cover

The switch is located conveniently on the top of the device and allows for easy control of the switch with the thumb. A curved lower casing allows for the other four fingers to comfortably sit under the device. The rounding on the top casing which sits above the large gear allows for the other hand to grip the machine closer to the centre of gravity. The hand controlling the switch will control position of the machine while the hand closer to the centre of gravity will absorb more of the force as well as helping to balance the tool. This rounding is displayed in Figure 35 below while Figure 36 displays the switch and removable casing assembled onto the machine. Most ergonomics have been considered and explained and thus a 2D sketch of external ergonomic additions is not included in the model, however a hand grip on the bottom casing that allows for fingers to sit comfortably, could be included.

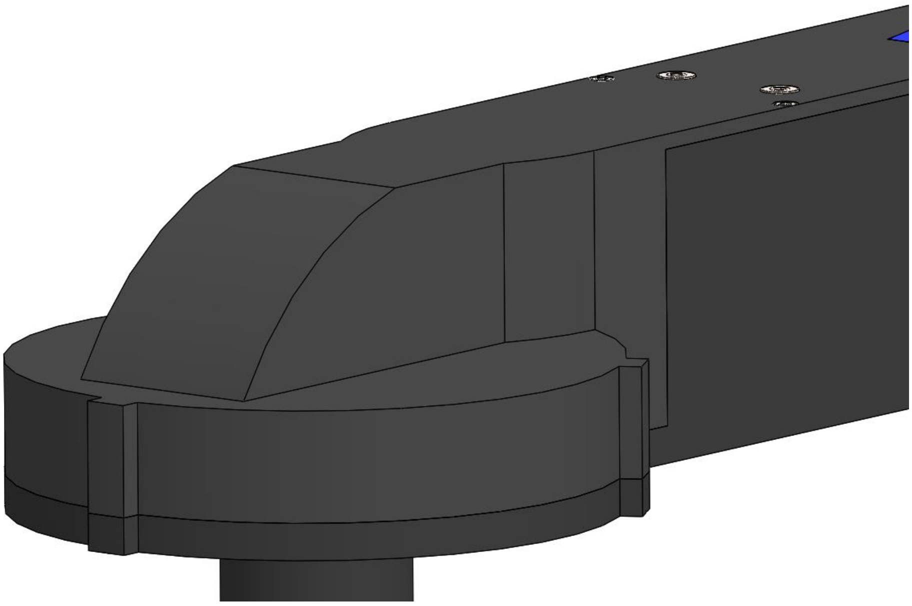

Figure 35 - Rounded Hand Grip of Top Casing

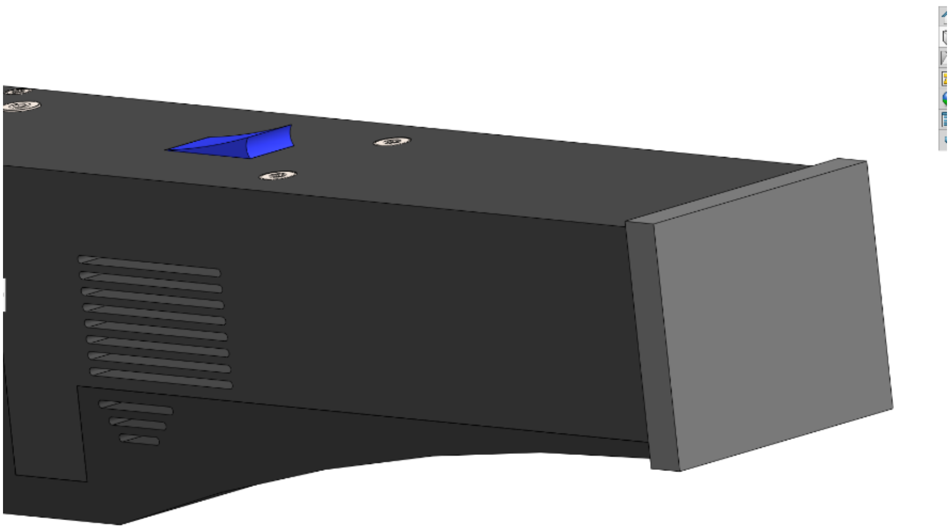

Figure 36 - Assembled Configuration

The two casings are connected by means of many M3 screws and matching machined threads as well as two M4 screws and paired threads machined into the casings. The placing of all screws can be seen without the casings in Figure 37 below, while Figure 38 shows the casings as transparent, where screws and threads can be seen too.

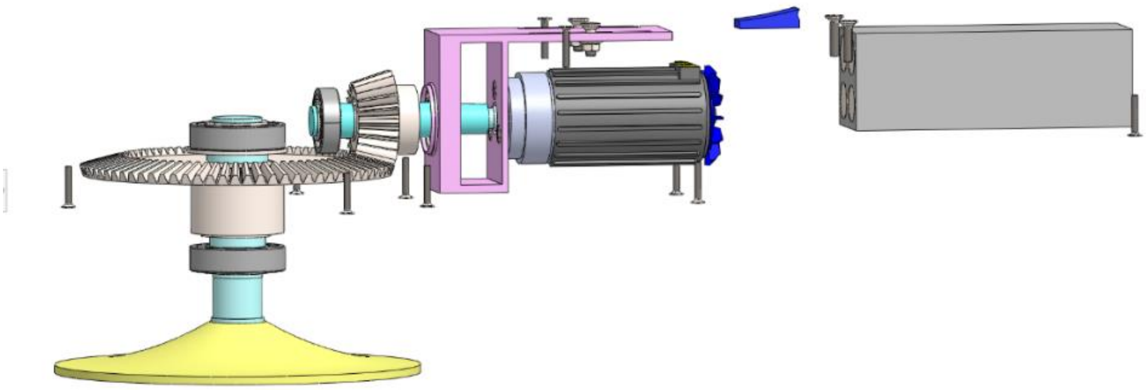

Figure 37 - View of All Screws

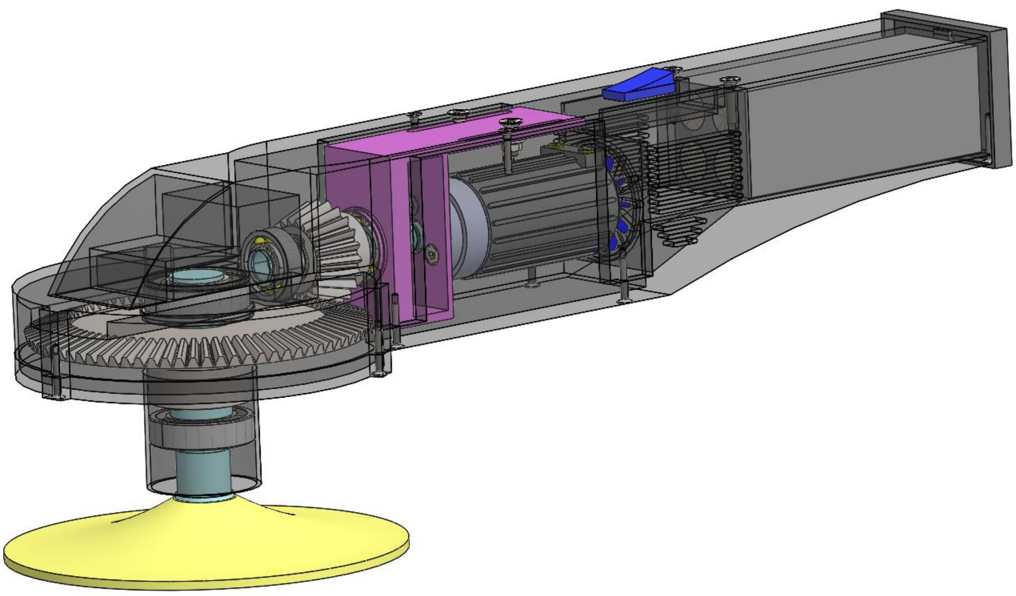

Figure 38 - Transparent Casing with View of Screws

3.5 Primary Material Selections and Manufacturing Considerations

Both the intermediate and output shafts are to be made from EN24 (T) alloy steel due to its excellent manufacturability, the ability to acquire it easily and its reasonable price [7]. This steel is also a typical material used in electric motor shafts. Calculations regarding strength of the shaft show that this material provides enough strength for use, even with stress concentration factors and induced key stresses (see Appendix A: Section 11.5). With regards to manufacturing of the shaft, a lathe will be used to produce the steps in the shaft as well as the bored hole. A broach is to be used to create the keyways on both the shafts as well as the external spline on the output shaft. The shafts can be seen in Figures 39 and 40 below. A surface grinder may be used to create the desired surface finish and final dimensions of the shafts.

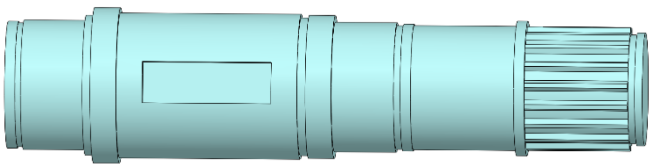

Figure 39 - Output Shaft

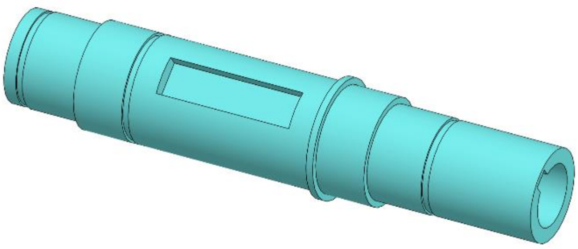

Figure 40 - Intermediate Shaft

The top and bottom casings, polishing head, battery configuration holder and removable battery cover are to be made from ABS plastic as it can be 3D printed easily for prototypes, is strong as a plastic material and can be injection moulded for mass production/value engineering in later phases. The price of the raw material is relatively good too. Most of the externals are made from plastic and this can be seen in Figure 41 below.

Notably, when injection moulded, the dimensional accuracy of an ABS component is within 0.0125mm (0.0005 inches) [8], better than any requirement for this product. Very complex parts can be produced in this process as well, labour costs are low and turnover times are fast [9]. The main negative is that the initial cost of the mould is quite high [9].

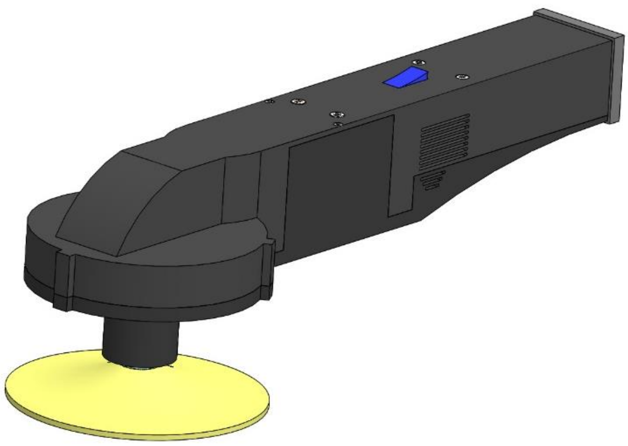

Figure 41 - ABS Components

The bracket housing is to be made from aluminium grade 6061 as it is easy to acquire, lightweight, machinable and could be cast in the future and is used in structural applications [10]. It also provides cooling for the motor. It will initially be machined by a milling machine out of a solid aluminium block and then will be cast and surface finished with a grinder for value-engineered production.

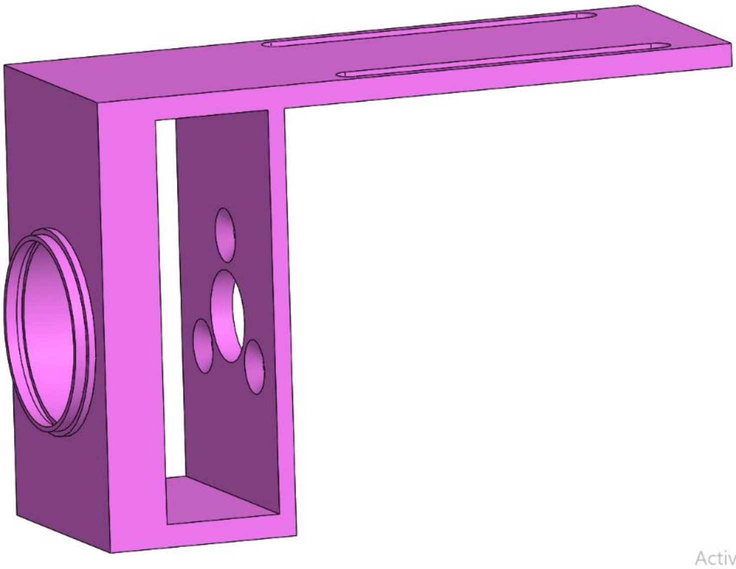

Figure 42 - Aluminium Bracket

The final consideration with regards to material and manufacturing is that of the gears. The gears are to be made from carbon steel as they have a high tensile strength, a high endurance and can be manufactured easily [11]. The prototype manufacturing will be done with use of a steel block, milled into a gear. The mass production/value engineered production will make use of steel die casting as it is the most practical and economical way of mass-producing gears.

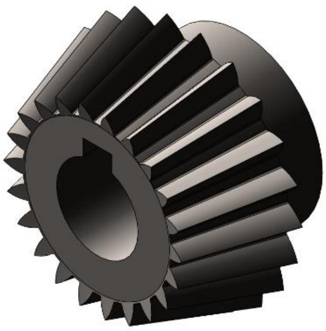

Figure 43 - Pinion

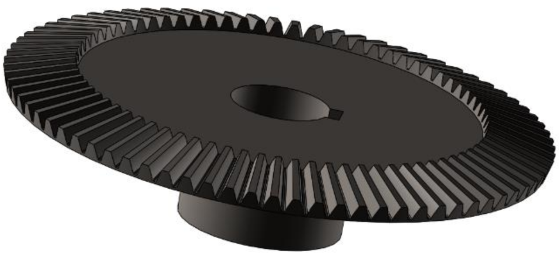

Figure 44 - Large Gear

All other components are bought, and their manufacturing considerations need not be considered. However, their material selections are still important. All circlips are made of spring steel as usual, keys are made from steel as this is the material of both the shafts and the gears. All screws, bolts, nuts and washers are made from carbon steel, as these are easy to acquire and provide a good strength compared to a plastic type fastener.

4. BOM Costing

The bill of materials has been created in the SolidWorks model and exported to Excel to use for prototype and value-engineered costing. Cost estimates are provided for each part, motivated either by means of sourcing component costs online or by estimating how much the component would cost to buy as a raw material and produce by machining. The former will generally refer to a site where the component is available while the latter will generally have a calculation reference and component and/or machining references. An assembly cost is included as well. No quotes have been solicited from companies and the SolidWorks costing tool has not been used.

4.1 Prototype Costing

The costing table can be seen below in Figure 45: Prototype costing (which continues onto the following page). A calculation reference is included which refers to the motivated calculations that appear below that. This is the estimated cost for the first prototype The cells highlighted in yellow refer to ABS plastic components, manufactured by 3D printing, the dark grey cells refer to prices acquired from the costing documents given, the light green cell refers to the aluminium bracket, the light red/orange cells refer to the gears, the pink cells refer to the shafts, the blue cells refer to the bought SKF bearings, the bright green cell refers to the switch and importantly, the light grey components that which would be bought from a supplier.

Note that some fasteners come in bags containing many components (i.e. a bag of 100 circlips). The cost for one fastener was estimated by dividing the cost of the bag by the number of fasteners in the bag. All masses were acquired from the mass calculation for the product found in Appendix A: Section 11.6 (masses were either taken from SolidWorks Mass Properties or from their masses available online).

4.1.1 Prototype Costing Table

NO.	PART NUMBER	DESCRIPTION/PART NAME	QTY	COST PER PART	COST	Note
1	MEC4103-019	Motor with Key	1	225	225	Cost from Motor Specification Document
2	DJNQUI001-001	Polishing Head	1	149.45	149.45	Calculation A1
3	DJNQUI001-013	Pre-Assembled Battery Configuration	1	N/A	N/A	N/A
3.1	DJNQUI001-002	18650 NMC Cell	8	90	720	Cost from Cell Options Document
3.2	DJNQUI001-003	Cell Holder	1	126.72	126.72	Calculation A1
4	DJNQUI001-004	Motor Bracket and Bearing Housing	1	277.2	277.2	Reference [BOM1] & Calculation A2
5	DJNQUI001-011	Small Pinion	1	20	20	See Calculation B1 & Reference [BOM2]

6	DJNQUI001-012	Large Gear	1	108.1	108.1	See Calculation B1 & Reference [BOM2]
7	DJNQUI001-005	Output Shaft	1	40.05	65.05	See Calculation B2 as well as the reference: [BOM3]
8	UCT-10308	Key 3×3	1	0.82	0.82	Bought Part, Available From [BOM4]
9	UCT-16019	Circlip, external 22 X 1.2	1	0.94	0.94	Bought Part, Available From [BOM5]
10	UCT-04010	Socket CSK HD Bolt M4	3	2.89	8.67	Bought Part, Available From [BOM6]
11	UCT-15005	Washer M4	5	0.94	4.7	Bought Part, Available From [BOM7]
12	DJNQUI001-006	Intermediate Shaft	1	31.5	56.5	See Calculation B2 and reference: [BOM3]
13	SKF-6002	15mm, 6002 Bearing	2	96.24	192.48	Bought Part, Available From [BOM8]
14	SKF-6005	25mm, SKF 6005 Bearing	2	129.09	258.18	Bought Part, Available From [BOM9]
15	UCT-16021	Circlip, external 25 X 1.2	2	1.3	2.6	Bought Part, Available From [BOM10]
16	UCT-16023	Circlip, external 28 X 1.5	1	1.5	1.5	Bought Part, Available From [BOM11]
17	UCT-16012	Circlip, external 15 X 1	2	0.62	1.24	Bought Part, Available From [BOM12]
18	UCT-16014	Circlip, external 17 X 1	1	0.69	0.69	Bought Part, Available From [BOM13]
19	UCT-10354	Key 6×6	1	1.13	1.13	Bought Part, Available From [BOM14]
20	UCT-10003	Key 8×7	1	1.3	1.3	Bought Part, Available From [BOM15]
21	UCT-17030	Circlip, internal 47 X 1.75	1	9.2	9.2	Bought Part, Available From [BOM16]
22	UCT-17021	Circlip, internal 32 X 1.2	1	1.77	1.77	Bought Part, Available From [BOM17]
23	DJNQUI001-007	Switch	1	10	10	Cost from Costing Input Document
24	DJNQUI001-008	Top Casing and Bearing Housing	1	381.92	381.92	Calculation A1
25	DJNQUI001-009	Bottom Casing and Housing	1	328.79	328.79	Calculation A1
26	UCT-07034	Cross CSK HD screw M3	11	1.72	18.92	Bought Part, Available From [BOM18]
27	UCT-04023	Socket CSK HD Bolt M5	2	3.83	7.66	Bought Part, Available From [BOM19]
28	UCT-07043	Cross CSK HD screw M4	2	3.58	7.16	Bought Part, Available From [BOM20]
29	DJNQUI001-10	Removable Battery Cover	1	15.23	15.23	Calculation A1
30	UCT-15003	Washer M3	11	0.86	9.46	Bought Part, Available From [BOM21]
31	UCT-15006	Washer M5	2	0.56	1.12	Bought Part, Available From [BOM22]
32	UCT-11000	Hex flange nut M5	2	1.5	3	Bought Part, Available From [BOM23]
				TOTAL COST	3016.5	Cost Without Assembly
					3242.74	Add 7.5% for Assembly

Figure 45 - Prototype Costing

Calculations referred to in the table can be found on the following page.

Calculation A1: From costing inputs document:

3D printing using ABS plastic: R700/kg

From the Mass Estimation Calculations:

Polishing Head: 213.25g

Cell Holder: 181.03g

Top Casing: 545.6g

Bottom Casing: 469.7g

Removable Battery Cover: 21.76g

Costs For These Parts Should Thus Be:

Polishing Head: R149.28

Cell Holder: R126.72

Top Casing: R381.92

Bottom Casing: R328.79

Removable Battery Cover: R15.232

Calculation A2: R118.01 for a plate of the correct diameters from Alustore [BOM1]. Multiplication factor of 3 for CNC milling (material removal) and for fine finishing. Not 4 as there is a lot of excess material that can be reused or recycled. Mass of motor bracket is 155.21g (Solidworks mass properties).

Calculation B1: Steel costs approximately R14.58/kg from stansteel. Round to R20/kg as acquisition is not in bulk.
Multiplication factor of 5 in order to blank/cast the gear shape and to acquire an adequate surface finish as well as for finishing machining
Pinion mass of 197.1g and Gear mass of: 1081.357g (Solidworks mass properties)

Calculation B2: R216.35 for a 20mm diameter steel rod, 6000mm in length.
Extrapolate to R389.43 for a 30mm diameter steel rod, 6000mm in length.
Divide the price for each rod by 6000mm to acquire a price per mm.
Multiply the length + 5mm, of the output and intermediate shafts respectively, to acquire a price for the length of the shafts.
Use a multiplication factor of 5 for the use of a lathe, broach to create keyways and to finish the product. Add R25 to each for labour as well.

Also add 7.5% to the total cost of the product for assembly cost.

4.2 Value-Engineered Costing

The process of value-engineered costing involves finding means in which production or acquisition of component parts can be reduced in cost. This follows a similar method and reasoning to the prototype costing; however, it is assumed that this will be the cost for producing parts to be distributed for sale and use, thereby implying components can be produced by more cost-effective means e.g. injection moulding instead of 3D printing.

4.2.1 Value-Engineered Costing Table

NO.	PART NUMBER	DESCRIPTION/PART NAME	QTY	COST PER PART	COST	Note
1	MEC4103-019	Motor with Key	1	225	225	Cost from Motor Specification Document
2	DJNQUI001-001	Polishing Head	1	8.54	8.54	Calculation VA2
3	DJNQUI001-013	Pre-Assembled Battery Configuration	1	N/A	N/A	N/A
3.1	DJNQUI001-002	18650 NMC Cell	8	90	720	Cost from Cell Options Document
3.2	DJNQUI001-003	Cell Holder	1	7.24	7.24	Calculation VA2
4	DJNQUI001-004	Motor Bracket and Bearing Housing	1	29.63	29.63	Calculation VA3
5	DJNQUI001-011	Small Pinion	1	10.206	10.206	See Costing VB1 [BOM2] R14.58/kg
6	DJNQUI001-012	Large Gear	1	55.16343	55.16343	See Costing VB1 [BOM2] R14.58/kg
7	DJNQUI001-005	Output Shaft	1	24.03	24.03	See Costing VB2 [BOM3] 6000mm x20mm is 216.35 thereforeR389.43 for 30mm x 6m
8	UCT-10308	Key 3×3	1	0.738	0.738	Bought Part, Available From [BOM4] 10% discount
9	UCT-16019	Circlip, external 22 X 1.2	1	0.846	0.846	Bought Part, Available From [BOM5] 10% discount
10	UCT-04010	Socket CSK HD Bolt M4	3	2.601	7.803	Bought Part, Available From [BOM6] 10% discount
11	UCT-15005	Washer M4	5	0.846	4.23	Bought Part, Available From [BOM7] 10% discount
12	DJNQUI001-006	Intermediate Shaft	1	18.9	18.9	See Costing VB2 [BOM3] 6000mm x 20mm is 216.35
13	SKF-6002	15mm, 6002 Bearing	2	90.7	181.4	Bought Part, Available From [BOM8] Discount on mass orders
14	SKF-6005	25mm, SKF 6005 Bearing	2	121.65	243.3	Bought Part, Available From [BOM9] Discount on mass orders
15	UCT-16021	Circlip, external 25 X 1.2	2	1.17	2.34	Bought Part, Available From [BOM10] 10% discount
16	UCT-16023	Circlip, external 28 X 1.5	1	1.35	1.35	Bought Part, Available From [BOM11] 10% discount
17	UCT-16012	Circlip, external 15 X 1	2	0.558	1.116	Bought Part, Available From [BOM12] 10% discount
18	UCT-16014	Circlip, external 17 X 1	1	0.621	0.621	Bought Part, Available From [BOM13] 10% discount

19	UCT-10354	Key 6×6	1	1.017	1.017	Bought Part, Available From [BOM14] 10% discount
20	UCT-10003	Key 8×7	1	1.17	1.17	Bought Part, Available From [BOM15] 10% discount
21	UCT-17030	Circlip, internal 47 X 1.75	1	8.28	8.28	Bought Part, Available From [BOM16] 10% discount
22	UCT-17021	Circlip, internal 32 X 1.2	1	1.593	1.593	Bought Part, Available From [BOM17] 10% discount
23	DJNQUI001-007	Switch	1	10	10	Cost From Costing Document
24	DJNQUI001-008	Top Casing and Bearing Housing	1	21.824	21.824	Calculation VA2
25	DJNQUI001-009	Bottom Casing and Housing	1	18.788	18.788	Calculation VA2
26	UCT-07034	Cross CSK HD screw M3	11	1.548	17.028	Bought Part, Available From [BOM18] 10% discount
27	UCT-04023	Socket CSK HD Bolt M5	2	3.447	6.894	Bought Part, Available From [BOM19] 10% discount
28	UCT-07043	Cross CSK HD screw M4	2	3.222	6.444	Bought Part, Available From [BOM20] 10% discount
29	DJNQUI001-10	Removable Battery Cover	1	0.87	0.87	Calculation VA2
30	UCT-15003	Washer M3	11	0.774	8.514	Bought Part, Available From [BOM21] 10% discount
31	UCT-15006	Washer M5	2	0.504	1.008	Bought Part, Available From [BOM22] 10% discount
32	UCT-11000	Hex flange nut M5	2	1.35	2.7	Bought Part, Available From [BOM23] 10% discount
				TOTAL COST	1648.58343	Cost Without Assembly
					1731.01	Add 5% for Assembly

Figure 46 - Value Engineered Table

From Costing inputs: Injection moulded ABS plastic: R20/kg x 2 for machine time

From the Mass Estimation Calculations:	Costs For These Parts Should Thus Be:
Polishing Head: 213.25g	Polishing Head: R8.54
Cell Holder: 181.03g	Cell Holder: R7.24
Top Casing: 545.6g	Top Casing: R21.824
Bottom Casing: 469.7g	Bottom Casing: R18.788
Removable Battery Cover: 21.76g	Removable Battery Cover: R0.87

Calculation VA3:

Use Aluminium Casting. $1.8 per kilogram of Aluminium. Multiply by 1.75 for buying from a supplier = $3.15 per kg. Translates to R54.62/kg. Multiplication of 3.5 for casting process and posting machining finishing process (grinding for fine finish).
Motor bracket is 155.21g in mass.
This results in a final cost of R29.63 for the motor bracket/housing

Calculation VB1:

Use steel casting to create gear and pinion.
Steel costs R14.58/kg from stansteel. (R14580/ton)
Pinion mass is: 197.1g
Gear mass is:1081.36g
Multiply by 3.5 for casting and for post surface finishing

Calculation VB2:
The same logic as B2 except multiplication factor is taken down to 3 for machining and the added labour cost is removed

Add 5% to the total cost for assembly now

4.2.2 Comparison to Target Cost

The target retail cost for this product was set as R2800 or lower (as stated in the basic design report). The final cost of the value-engineered product with assembly has been estimated to be R1731.01. A typical net profit margin in industry varies between 5% (low) and 20% (high). However, the addition of a profit margin onto this product (sale price to distributor) would not be the nett margin but rather the gross margin. The tooling costs calculated and described below will aid in developing an idea of overhead costs and thus a suitable profit margin to add to the value-engineered product.

4.2.3 Tooling Costs

NO.	DESCRIPTION/ PART NAME	QTY	Tooling Method	Tool Cost	Excess cost for size	Total Tool Cost	Reference
2	Polishing Head	1	Injection Mould (180mm diagonal)	35000	45000	80000	Costing inputs document & T1
3.2	Cell Holder	1	Injection Mould (146mm diagonal)	25000	36500	61500	Costing inputs document & T1
4	Motor Bracket and Bearing Housing	1	Casting (aluminium resin sand)	$3000	N/A	51000	[TO1] ($1 = R17)
5	Small Pinion	1	Casting (automatic moulding)	$2000	N/A	34000	[TO1] ($1 = R17)
6	Large Gear	1	Casting (automatic moulding)	$3000	N/A	51000	[TO1] ($1 = R17)
7	Output Shaft	1	Lathe	30000	N/A	30000	[TO2]
12	Intermediate Shaft	1	Lathe	Same Lathe	N/A	0	[TO2]
24	Top Casing and Bearing Housing	1	Injection Mould (520mm diagonal)	66000	130000	196000	Costing inputs document & T1
25	Bottom Casing and Housing	1	Can create cavity in top casing mould to create both casings in one and save money	0	0	0	Costing inputs document & T1
29	Removable Battery Cover	1	Injection Mould (95.74mm diagonal)	25000	23935	48935	Costing inputs document & T1
			Surface Grinder			5000	[TO3]
			Accessories and other tools			20000	
			TOTAL COST			577435	

Figure 47 - Tooling Costs

4.2.4 Calculation T1

R25 000 + R250/mm x diagonal size (max 150mm)

R35 000 + R250/mm x diagonal size (max 250mm)

R42 000 + R250/mm x diagonal size (max 350mm)

Using extrapolation to assume that a max 550mm mould would cost R66 000 + R250/mm diagonal size. One cavity shall be used for the top and bottom casing in order to save on tooling costs.

4.2.5 Overhead Costs

These are assumed by adding percentages of the value-engineered cost as the value chain as follows:

10% for receiving and 10% more for production (excluding tooling), and assuming shipping will be handled by the distributor/retailer.

Pre-distributor cost is R1731.01*1.2 = R2077.21

Adding 17.5% as the net-profit margin now, the sales price to the distributor will be R2077.21*1.175 = R2440.72, implying a net-profit of R363.51 per product (a high margin). This also adheres that the retail price should be close to R2800. The gross profit is then R1731.01/R2440.72 = 71%.

5. Development Plan

Quite a few assumptions are needed in order to create a product development plan and these can affect the cash-flow schedule in the manner that the break-even point can be manipulated to be earlier or later than a realistic date would be.

Development costs are the first inclusion of the development plan. This includes hiring engineers to develop the product and create solutions to the project. It is assumed that two engineers will be needed for two months to help develop the product and prototypes. A salary of R50000 a month is assumed for the engineers. It is assumed that 6 initial proof-of-principle prototypes will be required, split over the two months the engineers are working. These prototypes are assumed to be the same cost as that of the initial prototype, R3250. Two months of daily consultant marketers, analysts, surveyors and machinists are assumed to be required as well. This is assumed to cost R20000 a month. Tooling is assumed to be bought at month three and constant labour is assumed to be required after that, as well as rent and overhead costs for utilities, taxes and insurance. The purchasing of tools also implies the beginning of alpha and beta prototype testing. The production of product is assumed to start after the prototype productions and sales are assumed to begin two months after production of product begins. Labour, rent, utilities, insurance and taxes as well as cost of the producing the product are assumed to be constant expenses after sales begin, and constant income begins when sales begin. Figure 47 below shows the calendar in which the product development plan is planned to adhere to. Green cells represent the phase before constant expenses begin and a production team is set up. The orange cells represent the phase before sales begin and the yellow cells imply the beginning of the sales phase (to continue constantly past month 9).

Month ->	1	2	3	4	5	6	7	8	9
Development									
Consultant Engineers									
Labour (machinist consultants/marketing consultants etc)									
Prototypes	Proof-of-Principle	Proof-of-Principle 2 & 3							
Ramp Up									
Tooling									
Prototypes			Alpha	Alpha & Beta	Beta				
Production									
Rent									
Utilities, Insurance, taxes									
Labour									
Production of Product						Ramp-Up	Ramp-Up		
Sales									

Figure 48 - Product Development Plan

Cost	TOTAL			
Development Costs	159500	2 Engineers @ R50000/moth for 2 months	2 months other labour @ R20000/month for 2 months	6 prototypes @ R3250 over 2 months
Ramp-up Costs	648435	Once of payment of Tooling at R583435 (see tooling cost)	10 Alpha prototypes @2500 each over two months	20 Beta prototypes @ R2000 over 2 months
Rent of office/factory	8000	https://www.property24.com/to-rent/maitland/cape-town/western-cape/9153/108556242		
Utilities, Tax, Insurance	5000	62.5% of rent		
Volume Production cost per product	1731.01	See Previous Costing Section		
Labour costs per month	30000	Two machinists and one office staff		
Sales price to distributor per product	2440.72			
Steady State Production/sales per month	352	Only one tool per part, however the rest is bought. Assume 2 parts made per hour, 22 working days in a month, 8 hour days		

Figure 49 - Development Plan Costs

The table above shows all assumptions made to reach estimated development costs. These are used in the cash-flow schedule on the following page.

5.1 Cash-Flow Schedule

All values in the table are in kiloRands (kR) in order to reduce the length of numbers in cells so that the table is legible.

Indicates Break Even Point

Month ->	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Expenses	**59.75**	**79.75**	**658.9**	**75.5**	**63**	**164.17**	**406.51**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**	**652.32**
Development	59.75	79.75	20																					
Ramp-up			595.9	32.5	20																			
Rent, utilities, insurance and taxes			13	13	13	13	13	13	13	13	13	13	13	13	13	13	13	13	13	13	13	13	13	13
Labour			30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30
Production						121.17	363.51	609.32	609.32	609.32	609.32	609.32	609.32	609.32	609.32	609.32	609.32	609.32	609.32	609.32	609.32	609.32	609.32	609.32
Quantity						70	210	352	352	352	352	352	352	352	352	352	352	352	352	352	352	352	352	352
Income								**683.2**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**	**858.88**
Sales								683.2	858.88	858.88	858.88	858.88	858.88	858.88	858.88	858.88	858.88	858.88	858.88	858.88	858.88	858.88	858.88	858.88
Quantity								280	352	352	352	352	352	352	352	352	352	352	352	352	352	352	352	352
Nett Cost	**-59.75**	**-139.5**	**-798.4**	**-873.9**	**-936.9**	**-1101.1**	**-1507.6**	**-1476.7**	**-1270.1**	**-1063.6**	**-857.02**	**-650.46**	**-443.9**	**-237.34**	**-30.78**	**175.78**	**382.34**	**588.9**	**795.46**	**1002.02**	**1208.58**	**1415.14**	**1621.7**	**1828.26**
Stock inventory						70	280	352	352	352	352	352	352	352	352	352	352	352	352	352	352	352	352	352

Figure 50 - Cash Flow Schedule

Figure 49 on the previous page shows the cash-flow schedule with the same format followed as the calendar of the development plan. Values have been added appropriately per section and all expenses and incomes have been totalled together separately. The nett cost of the project is then acquired by subtracting the expenses from the incomes. This project plan shows that the largest total negative balance (debt/initial investment) is R1507.6 and that the project should break even on month 16. The profit should exceed initial investment on month 23.

6. Table of Final Specification and Comments on Success

Metric	unit	Value
Max no load rotational speed of the polisher	RPM	>=2200
Total mass of the tool	kg	<3.2kg
Length/Height of tool	mm	<450
Efficiency of Motor to Output	%	>=85%
Purchase Price	R	<2800
Continuous Power Output	W	>44.44
Torque Output	N.m	>0.3
Size of Polishing Head	mm	180
Duration of Full Power Operation Per Charge	mins	>=60
Current	A	3.055
Battery Chemistry		NMC
Battery Configuration		4S2P

Figure 51 - Edited Basic Design Final Specification Table

The table above was taken from the basic design report; however, duration of full power output and torque output have been changed after battery sizing calculations were re-done (Appendix A: Section 11.1). Mass of battery configurations had not been added to tools in the basic design and so a mass of 550g has been added to that to compensate for that error. All other specifications were kept the same.

The Table on the following page (Figure 51) shows the specifications of this product after the detail design, with green indicating meeting or exceeding the final specification, light orange indicating a small deviation and dark orange indicating a more critical deviation.

The cost, efficiency, power output, duration of charge and battery configuration all meet the final specification, with only the mass of the tool (1.49kg heavier than the limit) exceeding that of the final specification. This is a more critical problem and would need to be addressed after initial prototyping. The length of the tool is not too much greater than that of the final specification (83mm) and this would likely be decreased while mass minimization considerations are made.

For the most part, this design has produced a successful product, with only a few adjustments needing to be made. Costing targets have been reached and the ability for the product to produce a profit within two months has been shown.

Metric	unit	Value
Max no load rotational speed of the polisher	RPM	2282
Total mass of the tool	kg	4.69
Length/Height of tool	mm	536
Efficiency of Motor to Output	%	91.32%
Purchase Price	R	2800
Continuous Power Output	W	72.3
Torque Output (Operational)	N.m	0.305
Size of Polishing Head	mm	180
Duration of Full Power Operation Per Charge	mins	66
Current	A	3.055
Battery Chemistry		NMC
Battery Configuration		4S2P

Figure 52 - Final Specifications of Detail Designed Product

7. Risks, Drawbacks and Future Modifications needed for the Current Design

Although the detail design phase has been attributed as a success (for the most part) with regards to the final specifications set, there are many potential risks and drawbacks that have become apparent. Future modifications of the design would be required as well (after the initial prototype is built).

A heavy drawback and risk is that of the initial investment required to start full-production of the polisher. An initial investment of approximately R1.5 million rand would be required to get to the production stage (see the cost-flow table in the previous section). Profit would only begin around 16 months after the start of the project and it is likely that sales numbers will not be as high as expected in those months (low demand), pushing the break-even point further back. High cost of tooling is a drawback; however, these tools can be sold if the project is not successful.

This product is also a common power tool product and the market is likely saturated with vehicle polishers. The technology used in this design is not outstandingly creative or innovative and although it has a few unique features, it is not a breakthrough product that will crave a large demand from the market. A future modification could thus include more innovative features. A risk common to all businesses is that of failure to reach production due to the inability to produce a working product or the inability to fix problems in prototypes in the stipulated timeframe.

A modification that will be required in the future is that of total mass reduction. The total design mass exceeds the final specification by around 68%. Materials can be reconsidered and notably, the wall thicknesses of the outer casings and battery configuration holder can be reduced. The gear ratio is another modification that should be examined as the output gear is relatively large in comparison to the rest of the product, this would most likely result in a requirement for modifying and reconsidering the battery configuration too. Gear materials may then be considered to in order to reduce mass and size. The method of manufacturing products could be re-examined to minimize costs even further.

Another risk is not being able to steadily source components (as a lot of components are bought) and materials. It is also possible that assembly of the design does not work when manufacturing the real-life product and that a design flaw could go unnoticed before producing the design, resulting in a large step back in the design process. However, I do believe that this product has been designed adequately so that no surprises will become apparent during manufacturing and that the major risks are those of taking the product to market and the main drawbacks are that of reducing the mass and changing the gear sizes.

8. Conclusion

The detail design process has shown that this product can successfully be built and although some modifications will be required to meet the initial users' initial needs, these could be addressed without too much difficulty in future prototype development. The user needs are namely those of the polisher not scratching the surface of the vehicle, efficient and effective polishing, affordability, comfort of use and being able to use the polisher for extended periods of time. These all seem likely to be achieved given the detail design done on the product. The cost required to produce a profit or even break-even is a large risk and could deem the project too risky, unless starting a business centred around power tool creation.

9. Reference List

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10. Reference List for Costing & Tooling

Note that there is no particular manner for referencing products and their prices in IEEE, however, all available components referenced to in the costing and tooling sections can be found by following the links to their products below.

BOM1	https://alustore.co.za/project/aluminium-plates/
BOM2	https://www.stansteel.co.za/
BOM2	https://www.stansteel.co.za/
BOM3	https://www.metals4u.co.uk/materials/mild-steel/mild-steel-round/round-en24t/100818-p
BOM4	https://za.rs-online.com/web/p/keys/3023949/
BOM5	https://www.eis.co.za/index.php?route=product/product&path=6_56&product_id=4637
BOM6	https://za.rs-online.com/web/p/hex-bolts/2632110/
BOM7	https://za.rs-online.com/web/p/plain-washers/0189636/
BOM3	https://www.metals4u.co.uk/materials/mild-steel/mild-steel-round/round-en24t/100821-p
BOM8	https://za.rs-online.com/web/p/ball-bearings/3341534/
BOM9	https://za.rs-online.com/web/p/ball-bearings/2867669/
BOM10	https://www.eis.co.za/index.php?route=product/product&path=6_56&product_id=4640
BOM11	https://www.eis.co.za/index.php?route=product/product&path=6_56&product_id=4643
BOM12	https://www.eis.co.za/index.php?route=product/product&path=6_56&product_id=4630
BOM13	https://www.eis.co.za/index.php?route=product/product&path=6_56&product_id=4632
BOM14	https://za.rs-online.com/web/p/keys/3024172/
BOM15	https://za.rs-online.com/web/p/keys/3024223/
BOM16	https://www.eis.co.za/index.php?route=product/product&path=6_56&product_id=4705
BOM17	https://www.eis.co.za/index.php?route=product/product&path=6_56&product_id=4691
BOM18	https://za.rs-online.com/web/p/security-screws/4828149/
BOM19	https://za.rs-online.com/web/p/hex-bolts/2632211/
BOM20	https://za.rs-online.com/web/p/security-screws/6212552/
BOM21	https://za.rs-online.com/web/p/plain-washers/0189620/
BOM22	https://za.rs-online.com/web/p/plain-washers/0527397/
BOM23	https://za.rs-online.com/web/p/flanged-hex-nuts/0275535/

TO1	http://www.iron-foundry.com/mould-pattern-cost-cast-iron-sand-castings-steel-castings.html
TO2	https://www.adendorff.co.za/product/mac-afric-550-mm-professional-bench-lathe-excluding-stand/
TO3	https://www.adendorff.co.za/product/mac-afric-6-150-mm-universal-surface-grinder/

11. Appendix A – Detailed Calculations

11.1 Battery Configuration and Operating Load Scenario

Our output rotational speed is required to be: $w_{out} := 2200rpm$

Our output power is required to be: $P_{out} := 75W$

Thus our output torque will be: $T_{out} := \frac{P_{out}}{w_{out}} = 0.326\,N \cdot m$

Motor Constants: $R_a := 0.082ohm$ $K_T := 17.2N \cdot \frac{mm}{A}$

Motor Power: $P_{motor} := P_{out} = 75W$

If we take the output as the motor variables (ignoring the gearset for now)

$$I_a := \frac{T_{out}}{K_T} = 18.927A \qquad V_{ba} := \frac{P_{motor} + I_a^2 \cdot R_a}{I_a} = 5.515V$$

This would imply that for the NMC option (highest voltage and capacity option per cell):
6.769 (7 approx) cells in parallel to last for 1 hour
1.4894 cells (2 approx) in series to acheive required potential difference
Total of 14 cells
To last for 90 mins, more than 20 cells would be required, which would cost too much and take up too much space.

Noting that we can include a gear ratio.

We implore other options:

We also note that the typical cordless polisher has battery with a potential difference of 12-20V with a capacity of between 2Ah and 5Ah. This will be the basis of the following selection.

	NMC	LFP	LTO
Vnom	3.7V	3.2V	2.4V
Qnom	2.8Ah	1.9Ah	1.3Ah
Between 12V and 20V	Option 1: 5 cells in series (18.5V) Option 2: 4 cells in series (14.8V)	Option 1a: 4 cells in series (12.8V) Option 2a: 5 cells in series (16V)	Option 1a: 5 cells in series (12V) Option 2a: 6 cells in series (14.4V)
Between 2Ah and 5 Ah	Option 1b: 1 cell in parallel (2.8Ah) Option 2b: 2 cells in parallel (5.6Ah)	Option 1b: 1 cell in parallel (1.9Ah) Option 2b: 2 cells in parallel (3.8Ah)	Option 1b: 2 cells in series (2.6Ah) Option 2b: 3 cells in series (3.9Ah)
	Option 1 current: I = 75/18.5 = 4.054A Option 2 current:	Option 1a current: I = 75/12.8 = 5.859A Option 2a current:	Option 1 current: I = 75/12 = 6.25A Option 2 Current:

	I2= 75/14.8 = 5.0675A	I2= 75/16 = 4.6875A	I = 75/14.4 = 5.208A
Duration of power output	Least time: Time1 = 2.8/5.0675 = 0.55254hr Option 2b: T2 = 5.6/4.054 = 1.381hr	Least Time: Time1= 1.9/5.859 =0.324 hr Most time: 3.8/4.6875 = 0.7795hr	Least Time: Time1 = 2.6/6.25 = 0.416hr Most Time: Time2 = 3.9/5.208 = 0.74884hrs
	A good middle ground option is 4s2p, which means only 8 cells are needed and this will cost less.	A minimum of 15 cells would be required (5s3p) in order to achieve a runtime of at least 1 hour and thus this is not a viable option	To achieve the most time case, 18 cells (6s3p) would be required, and this still would not meet the hour mark for runtime. Therefore this battery configuration will not be viable.
Cost	5s2p: R900 4s2p: R720	5s3p: R675	6s3p: R990
Mass	5s2p: 450g 4s2p: 360g	5s3p: 630g	6s3p: 720g

Referring to the table: Battery Configuration Comparison above, it can be seen that the NMC cells are the only viable solution for a product needing a long duration of power operation between charges. The two most viable solutions are compared below and the best option is selected.

So we aim for between 3 and 5 cells in series and 1 and 2 cells in parallel for the battery configuration

Voltage for 3,4,5 cells in series respectively is: 11.1V, 14.8V, 18.5V

Capacity for 1 and 2 cells in parallel respectively is: 2.8Ah and 5.6Ah

If we use a 5s2p configuration: $I_{5s2p} := \frac{P_{out}}{18.5V} = 4.054A$ $\quad Time := \frac{5.6A\cdot hr}{I_{5s2p}} = 82.88min$

If we use a 4s2p configuration $I_{4s2p} := \frac{P_{out}}{14.8V} = 5.068A$ $\quad Time_2 := \frac{5.6A\cdot hr}{I_{4s2p}} = 66.304min$

The 5s2p has 10 cells and would cost: R900 Mass = 450 grams

The 4s2p has 8 cells and would cost: R720 Mass = 360 grams

The motor torque of the 5s2p configuration would be: $T_{5s2p} := I_{5s2p} \cdot K_T = 0.07 N \cdot m$

The motor torque of the 4s2p configuration would be: $T_{4s2p} := I_{4s2p} \cdot K_T = 0.087 N \cdot m$

The gear ratios of each are: $u_{5s2p} := \frac{T_{out}}{T_{5s2p}} = 4.669$

$$u_{4s2p} := \frac{T_{out}}{T_{4s2p}} = 3.735$$

A gear ratio above 4 would cause dimensions to become too large

Shaft diameter = 15mm (pinion diameter of minimum 30mm implies a gear diameter of minimum 150mm for the gear for a ratio of 5)

The output torque cannot be decreased further as there would be insufficient torque for the tool to function and thus the power requirements cannot be changed either.

Space and cost requirements infer that the only viable compromise is on functioning time of the tool.

Thus the 4s2p (total of 8 cells) configuration is chosen, with a gear ratio of 3.5

This would imply the following motor and functional specifications:

$V_b := 4 \cdot 3.7V = 14.8V$ $Q := 2 \cdot 2.8A \cdot hr = 5.6 A \cdot hr$ $u := 3.5$

$I := \frac{P_{out}}{V_b} = 5.068A$ $runtime := \frac{Q}{I} = 66.304 min$

where: $I_{stall} := \frac{V_b}{R_a} = 180.488A$ and I is less than Istall (limit not reached)

$T_{motorstall} := I_{stall} \cdot K_T = 3.104 N \cdot m$

$T_{motor} := I \cdot K_T = 0.087 N \cdot m$ $P_{motoractual} := I \cdot V_b - I^2 \cdot R_a = 72.894W$

$w_{limit} := 20000 rpm$

$w_{motor} := \frac{P_{motoractual}}{T_{motor}} = 7.986 \times 10^3 rpm$ and wmotor is less than wlimit (limit not reached)

$Mass := 8 \cdot 45g = 360g$ $Cost := 90 \cdot 8 = 720$

$E := I \cdot V_b = 75 W$ $Efficiency := \frac{P_{motoractual}}{E} = 97.192\%$

The output values should be:

$T_{out2} := T_{motor} \cdot u = 0.305 N \cdot m$ $w_{out2} := \frac{w_{motor}}{u} = 2.282 \times 10^3 \cdot rpm$

With the given gear ratio, the rotational speed is slightly higher, and the Torque is slightly lower than given

$$P_{out2} := T_{out2} \cdot w_{out2} = 72.894W$$

11.2 Maximum Load Case and Various Operating Scenario Cases

As calculated previously, a 4S2P configuration (8 NMC cells) will be used.
Operating conditions have been calculated previously, and are stated below.

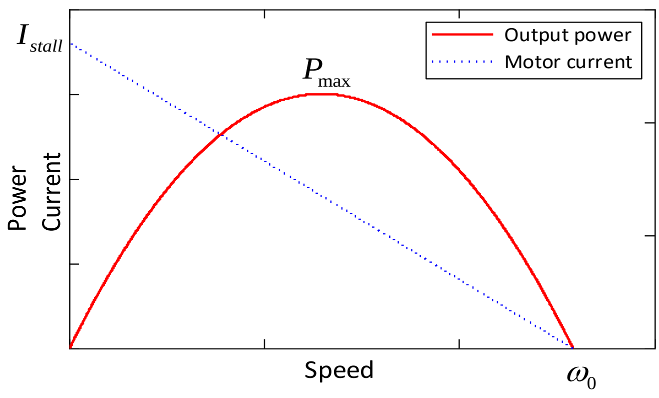
$T_{motoroperating} := 0.087N \cdot m$ $\qquad w_{motoroperating} := 7986rpm$

$P := 75W$ $\qquad T_{outoperating} := 0.305N \cdot m$ $\qquad w_{outop} := 2282rpm$

Calculations for Maximum Load Case Follow:

$V_b := 14.8V$ $\qquad Q := 5.6A \cdot hr$ $\qquad K_T := 17.2N \cdot \frac{mm}{A}$ $\qquad R_a := 0.082ohm$

No load speed: $w_o := \frac{V_b}{K_T} = 8.217 \times 10^3 \cdot rpm$ $\qquad I_{stall} := \frac{V_b}{R_a} = 180.488A$



$$P_{out} = V_b I - I^2 R_a$$

$$I_{stall} = \frac{V_b}{R_a}$$

$$\omega_0 = \frac{V_b}{K_T}$$

$$P_{max} \ at \ \omega = ½ \ \omega_0$$

P.max occurs when w=0.5wo $\qquad$ which is also where I=0.5Istall

$w_{pmax} := 0.5 \cdot w_o = 4.108 \times 10^3 \cdot rpm$ $\qquad I_{pmax} := 0.5 \cdot I_{stall} = 90.244A$

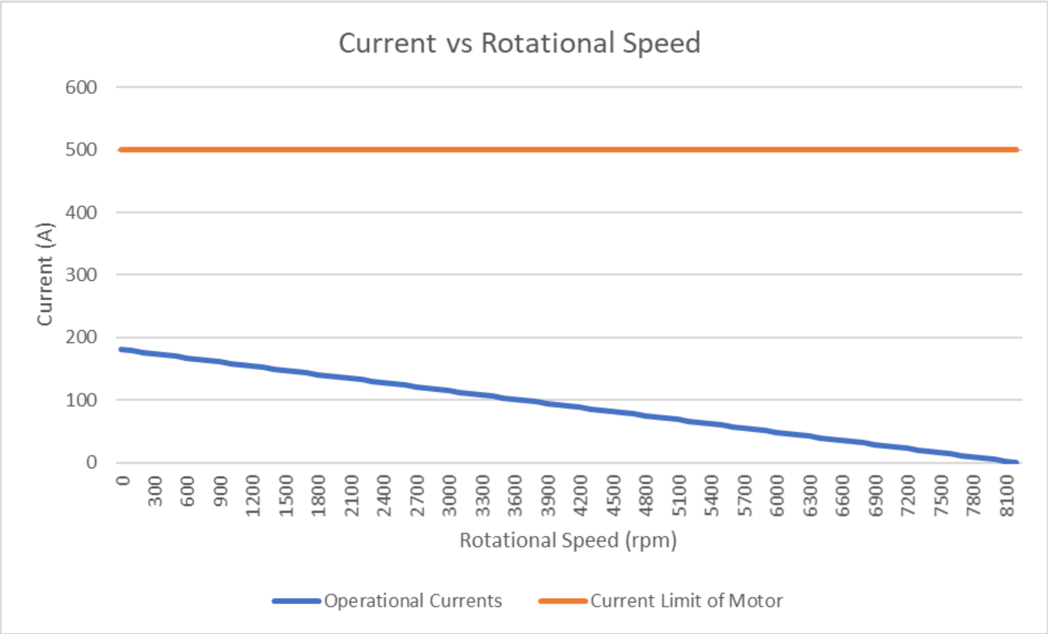
$T_{pmax} := I_{pmax} \cdot K_T = 1.552N \cdot m$ $\qquad$ maximum torque condition at the motor

$P_{outmax} := V_b \cdot I_{pmax} - I_{pmax}^{\ 2} \cdot R_a = 667.805W$

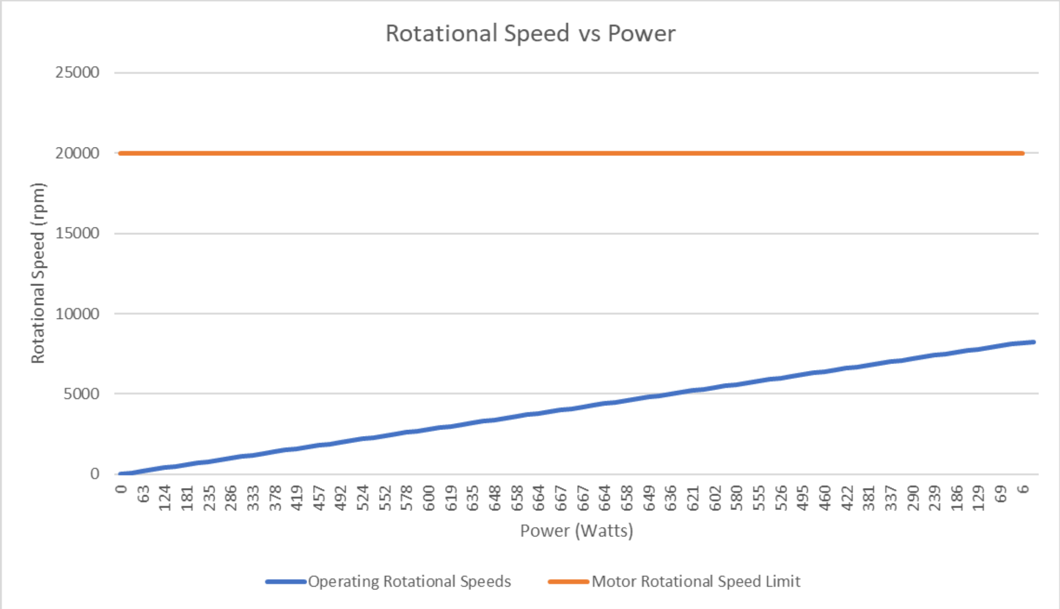
I.stall is less than I.limit and thus the maximum current limit will never be reached.

w.o is less than w.limit (20000rpm) and thus the maximum rotational speed limit will never be reached.

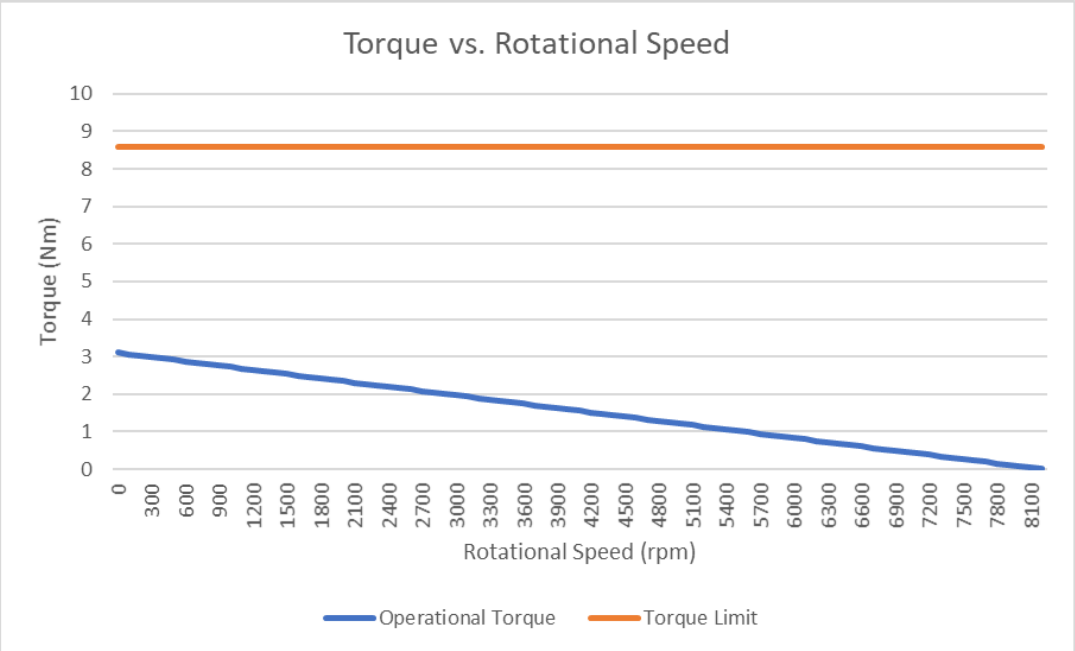
The following page shows three graphs which show the full range of operating conditions for Current, rotational speed, power and torque and confirm that none of the motor limits will be reached.



Current vs Rotational Speed
Current (A)
Rotational Speed (rpm)
Operational Currents
Current Limit of Motor



Rotational Speed vs Power
Rotational Speed (rpm)
Power (Watts)
Operating Rotational Speeds
Motor Rotational Speed Limit



Torque vs. Rotational Speed
Torque (Nm)
Rotational Speed (rpm)
Operational Torque
Torque Limit

11.3 Gear Forces and Bearing Calculations

$P_{max} := 667.805W$ $\quad w_o := 8217rpm$ $\quad w_{atpmax} := 0.5 \cdot w_o = 4.109 \times 10^3 \cdot rpm$

$$T_{max} := \frac{P_{max}}{w_{atpmax}} = 1.552 \cdot N \cdot m$$

$modu := 2mm$ $\quad b := 20mm$ $\quad u := 3.5$ $\quad Pangle := 20deg$

This module was chosen as it ensures the gear diameter is not unproportionately large in comparison to the shaft and so it doesn't take up too much space but also so that the forces on the gear are not too large either

$$w_{ooutput} := \frac{w_{atpmax}}{u} = 1.174 \times 10^3 \cdot rpm \qquad Z_p := 22$$

$Z_g := Z_p \cdot u = 77$ $\quad d_p := modu \cdot Z_p = 44 \cdot mm$ $\quad d_g := modu \cdot Z_g = 154 mm$

$$deltap := atan\left(\frac{\sin(90deg)}{u + \cos(90deg)}\right) = 15.945deg$$

Note the gear and pinion sizes were compared to a similiar Chiravelli gear with a 3 gear ratio and 2 module with the same pressure angle, face width and pinion teeth amount

$$Rp := \frac{d_p}{2\sin(deltap)} = 80.081mm$$

$$r_m := 0.5\left(d_p - b \cdot \sin(deltap)\right) = 19.253mm$$

$$v_p := r_m \cdot w_{atpmax} = 8.283\frac{m}{s}$$

In maximum operating case:

$$F_{TP} := \frac{P_{max}}{v_p} = 80.62N \qquad F_{TG} := F_{TP} = 80.62N$$

$F_{RP} := F_{TP} \cdot \tan(Pangle) \cdot \cos(deltap) = 28.214N$ $\quad F_{XG} := F_{RP} = 28.214N$

$F_{XP} := F_{TP} \cdot \tan(Pangle) \cdot \sin(deltap) = 8.061N$ $\quad F_{RG} := F_{XP} = 8.061N$

Radial load not less than 25N and thus cannot be taken by the motor. Motor cannot sustain an axial load either.

In operating case

$w_{motor} := 7986rpm$ $\quad T_{motor} := 0.087N \cdot m$

$$w_{out} := \frac{w_{motor}}{u} = 2.282 \times 10^3 \cdot rpm \qquad T_{out} := T_{motor} \cdot u = 0.305N \cdot m$$

$P_{out} := w_{out} \cdot T_{out} = 72.757W$ $\quad P_{motor} := P_{out} = 72.757W$

$$F_{Tp2} := \frac{T_{out}}{r_m} = 15.816N \qquad F_{Tg2} := F_{Tp2} = 15.816N$$

$$F_{RP2} := F_{Tp2} \cdot \tan(Pangle) \cdot \cos(deltap) = 5.535N$$

$$F_{XP2} := F_{Tp2} \cdot \tan(Pangle) \cdot \sin(deltap) = 1.581N$$

See gears sketch

Design solution: use a rigid shaft coupling to allow more bearings to be mounted onto the shaft which can then support gear forces as well as cater for misalignment of the shafts, relative to the housing and each other.

However, this design was too costly (rigid coupling costs around R600) and thus a cylindrical sleeve with a key was used to connect the motor shaft to the intermediate shaft.

The shaft coupling used would have been a rigid shaft coupling from Ruland. This has preicsion honed bores to ensure the shafts are collinear. It has a keyway as well and a tightly controlled bored tolerance of +.05mm/.012mm

The resultant forces will be:

$$F_{resultantmax} := \left(F_{RP}^{2} + F_{XP}^{2} + F_{TP}^{2}\right)^{0.5} = 85.794N$$

at maximum load

and

$$F_{resultantnoload} := \left(F_{RP2}^{2} + F_{XP2}^{2} + F_{Tp2}^{2}\right)^{0.5} = 16.831N$$

at operating load

On both the gear and pinion

By using a cylindrical sleeve, bearings will be required to take all the load.

The shaft diameter where the bearings will sit for the shaft attached to the motor shaft by the sleeve is 15mm

see shaft sketch

We assume a 90% reliability and 140000 hour life. (Juvinall and Marshek) pg 609

$$cycles_{maxload} := 14000 \cdot 60 \cdot 4108 = 3.451 \times 10^{9}$$

where 4108rpm = Shaft speed at maximum load

$$cycles_{opload} := 14000 \cdot 60 \cdot 7986 = 6.708 \times 10^{9}$$

where 7986rpm = Shaft speed at normal load

$$C_{lifemaxlo} := (3415)^{\frac{1}{3}} \cdot \left(F_{resultantmax}\right) = 1.292 \times 10^{3} N$$

$$C_{lifeopload} := (6708)^{\frac{1}{3}} \cdot \left(F_{resultantnoload}\right) = 317.423N$$

From SKF Bearing Catalogue: 6002 Bearing has C = 5850N
Co = 2850N

The 6002 Bearing is a common and popular part and will be more than sufficient in any loading case and is thus chosen for the application.

The forces on the output shaft will be the same as that on the shaft above.

However, the rotational speed at the maximum and normal operational loads will be different, as follows:

$\text{cycles}_{\text{maxloadout}} := 14000 \cdot 60 \cdot 1174 = 9.862 \times 10^{8}$ where 1174rpm = shaft speed at max load

$\text{cycles}_{\text{oploadout}} := 14000 \cdot 60 \cdot 2282 = 1.917 \times 10^{9}$ where 2282rpm = shaft speed at normal laod

$$C_{\text{lifemaxout}} := (986.2)^{\frac{1}{3}} \cdot \left(F_{\text{resultantmax}}\right) = 853.978\text{N}$$

$$C_{\text{lifenormalout}} := (1917)^{\frac{1}{3}} \cdot \left(F_{\text{resultantnoload}}\right) = 209.081\text{N}$$

I choose an output shaft of 25mm diameter where the bearings will sit to ensure more than sufficient strength and to try maintain the lowest ratio of diameter sizes between the shaft and the gear but also because smaller bearings are generally cheaper and less volume shafts mean less material cost and less mass.

From the SKF bearing catalogue: we choose a bearing with a diameter of 25mm

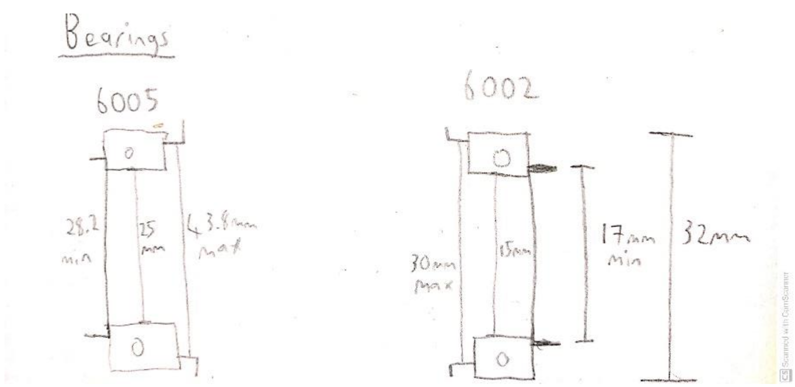
61805 Bearing has load ratings: C = 4360N Co = 2600 N

However, the 61805 bearings is a less common part and costs over R200 in South Africa [rscomponents]. The much more common 6005 bearings, which has a better static and dynamic load rating is about half the price.

6005 bearing loads ratings: C = 11900N Co = 6550N

These are both more than sufficient for the applied case and the 6005 bearing is a popular item so is thus even more suitable for this application

Note that the limiting speed of the bearing is 20000rpm (higher than this system will ever go).



11.4 Separate Gear Calculations and Reasonings

$w_{motor} := 7986 rpm$

To achieve output rotational speed of 2200rpm $u_{acheive} := \frac{7986}{2200} = 3.63$

However, 3.63 is a very hard gear ratio to obtain and so a gear ratio $u := 3.5$ was chosen

$w_{out} := \frac{w_{motor}}{u} = 2.282\times 10^{3}\,rpm$ $T_{out} := 0.305 N\cdot m$ $P_{out} := w_{out}\cdot T_{out} = 72.877 W$

$Pangle := 20deg$ $P_{motor} := P_{out} = 72.877 W$

$modu := 2mm$ $b := 20mm$ $pangle := 20deg$

This module was chosen as it ensures the forces on the gear and pinion are low.

It was also chosen so that there would be sufficient space for the bearings to sit on the intermediate shaft without interfering with the gear.

It was also chosen so that the bore of the pinion would not be too large (and so that a key could fit in without creating a bore so large that the teeth of the pinion would form part of the bore).

The shaft diameter where the pinion sits is 17mm, which implies that the bore of the pinion must be 17mm too. The key sits 4mm above the face of the keyway (8mm key in height). This means that the internal diameter of the pinion cannot be less than 21mm. The 6002 bearing has an outside diameter of 32mm and the housing surrounding it will need about 4mm each side to ensure proper location and clearance from the large gear. This means that the minimum diamter of the pinion should be 40mm.

The face width of gear teeth has no standard calculations but should be around half the diametrical pitch of the pinion. I based mine on a similiar Chiarevelli gear.

$Z_p := 22$ $Z_g := Z_p\cdot u = 77$ $d_p := modu\cdot Z_p = 44\,mm$ $d_g := modu\cdot Z_g = 154\,mm$

The diameter of the pinion is thus sufficiently large to ensure the bearing has proper clearance and these values for pinion teeth and gear teeth are selected.

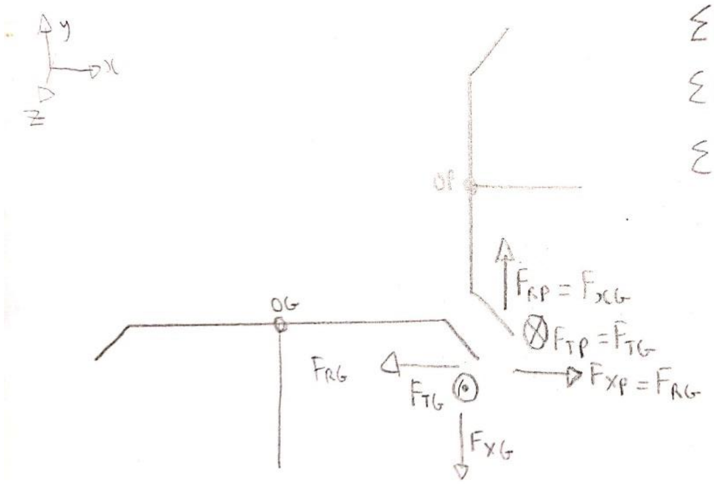
$$deltap := atan\left(\frac{\sin(90deg)}{u + \cos(90deg)}\right) = 15.945deg$$

$$Rp := \frac{d_p}{2\sin(deltap)} = 80.081mm \qquad r_m := 0.5\left(d_p - b\cdot\sin(deltap)\right) = 19.253mm$$

Under operational conditions:

$$v_p := r_m\cdot w_{motor} = 16.101\frac{m}{s} \qquad F_{TP} := \frac{T_{out}}{r_m} = 15.842N \qquad F_{TG} := F_{TP} = 15.842N$$

Bevel Gear Force Analysis



$$\sum F_{px} = F_{xp} = \sum F_{yG}$$

$$\sum F_{py} = F_{RP} = \sum F_{xG}$$

$$\sum F_{pz} = F_{TP} = \sum F_{zG}$$

$$\sum M_{op}^{+\circlearrowleft} = F_{xp}\frac{d_p}{2} + F_{TP}\frac{d_p}{2}$$

$$\sum M_{OG}^{+\circlearrowleft} = -F_{xG}\frac{d_G}{2} - F_{TG}\frac{d_G}{2}$$

See bevel gear force analysis above to describe acquisitions below:

$F_{RP} := F_{TP}\cdot\tan(Pangle)\cdot\cos(deltap) = 5.544N$ $\qquad$ $F_{XG} := F_{RP} = 5.544N$

$F_{XP} := F_{TP}\cdot\tan(Pangle)\cdot\sin(deltap) = 1.584N$ $\qquad$ $F_{RG} := F_{XP} = 1.584N$

$$F_{res} := \left(F_{TP}^{\;2} + F_{XP}^{\;2}\right)^{0.5} = 15.921N$$

As seen in the calculations before:

$F_{Tmax} := 80.62N$ $\qquad$ $F_{RPmax} := 28.214N$ $\qquad$ $F_{XPmax} := 8.061N$

Thus maximum moments will be:

$$M_{op} := F_{XPmax}\cdot\frac{d_p}{2} + F_{Tmax}\cdot\frac{d_p}{2} = 1.951N\cdot m$$

$$M_{og} := -F_{XPmax}\cdot\frac{d_g}{2} + -F_{Tmax}\cdot\frac{d_g}{2} = -6.828N\cdot m$$

For pinion:

$K_{vp} := \frac{6.1 + 16.101}{6.1} = 3.64$ $\qquad$ $K_o := 1.25$ $\qquad$ $K_m := 1.1$ $\qquad$ both gears starddle mounted, not maximum rigidity

$$I := \frac{\sin(pangle)\cdot\cos(pangle)}{2}\cdot\frac{u}{u+1} = 0.125$$

Assume the use of steel with 850MPA $\qquad$ $C_p := 191\cdot MPa^{0.5}$ $\qquad$ $C_R := 1$

99% Reliability as required from gears

$$S_{fe} := 2.8\cdot\left(\frac{850}{3.45}\right) - 69 = 620.855$$

$$C_{li} := \frac{1.23\cdot 191}{620.855\cdot 1}\cdot\left(\frac{80.62\cdot K_{vp}\cdot K_o\cdot K_m}{20\cdot 44\cdot I}\right)^{0.5} = 0.725$$

Approximately 9*10^10 cycles $\qquad$ More than sufficient life

11.5 Shaft Calculations

I choose to use EN 24 (T) as it is used often in industry for electric motors and shafts and has high tensile strength, excellent machinability and is of high quality (resource)

$S_u := 850MPa$ $\quad S_y := 680MPa$ $\quad E := 203MPa$ $\quad BHN := \frac{850}{3.45} = 246.377$

Using a safety factor of: $RF := 3.5$ see reference

Using the maximum shear stress theory

$$tau_{max} := \frac{S_y}{2 \cdot RF} = 97.143MPa$$

$M_{tmax} := 1.552N \cdot m$ $\quad M_{tnoload} := 0.305N \cdot m$ See diagram

use maximum torque to determine minimum diameter, given as follows:

$$d_{min} := \left(\frac{16 \cdot M_{tmax}}{\pi \cdot tau_{max}}\right)^{\frac{1}{3}} = 4.333mm$$

The minimum diameter of the output shaft must thus be 4.333mm

The smallest diameter to be used in the input shaft is 14.8mm. Much greater than the minimum allowable diameter. Attached diagram explains why FB is half of Fg and causes the maximum bending moment

$F_{gmax} := 95.74N$ $\quad F_{gnormal} := 18.782N$

$F_{Bmax} := 0.5 \cdot F_{gmax} = 47.87N$ $\quad F_{Bnormal} := F_{gnormal} \cdot 0.5 = 9.391N$

Shaft life calculations:

$Si_n := 0.5 \cdot S_u = 425MPa$

$C_L := 1$

$C_T := 1$

$C_s := 0.73$

$C_D := 0.97$

$C_R := 0.897$

$S_n := Si_n \cdot C_L \cdot C_T \cdot C_s \cdot C_D \cdot C_R = 269.945MPa$

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Using Soderberg method and assuming worst case loadings as stipulated below

$K_T := 2.2$ bending

$k_f := 1.35$ Square key for worst case $K_{ts} := 1.65$ Torsional

For input shaft

$La := 29.6mm$ where La is the distance between the centre of the pinion and bearing

$M := F_{Bmax} \cdot La = 1.417 N \cdot m$

$$d := \left[32 \cdot \frac{RF}{\pi} \cdot \left[\left(M \cdot \frac{K_T}{S_n} \right)^2 + \left(M_{tmax} \cdot \frac{K_{ts}}{S_y} \right)^2 \right]^{0.5} \right]^{\frac{1}{3}} \cdot k_f = 10.213mm$$

For Output shaft

$Lb := 22.6mm$

$M_b := F_{Bmax} \cdot Lb = 1.082J$

$$db := \left[32 \cdot \frac{RF}{\pi} \cdot \left[\left(M_b \cdot \frac{K_T}{S_n} \right)^2 + \left(M_{tmax} \cdot \frac{K_{ts}}{S_y} \right)^2 \right]^{0.5} \right]^{\frac{1}{3}} \cdot k_f = 9.439mm$$

Both the input and output shaft diameters are greater than these diameters

Goodman Equation

$sigma_b := \frac{S_n}{2} = 134.973MPa$ using a safety factor of 2 as there is already a 3.5 safety factor on the diameter

Input shaft for the minimum diameter of d = 14.8mm

$$sigma_{bmax} := \frac{32 \cdot \frac{F_{Bmax}}{2}}{\pi \cdot (14.8mm)^3} = 7.521 \times 10^7 \frac{kg}{s^2 \cdot m^2}$$

$$sigma_{bnormal} := \frac{32 \cdot \frac{F_{Bnormal}}{2}}{\pi \, (14.8mm)^3} = 1.475 \times 10^7 \frac{kg}{s^2 \cdot m^2}$$

$$L_{max} := \frac{sigma_b}{sigma_{bmax}} = 1.795m$$

$$L_{maxnormal} := \frac{sigma_b}{sigma_{bnormal}} = 9.148m$$

Maximum shaft length will certainly be under this maximum length and thus shaft length will not be an issue.

Output shaft

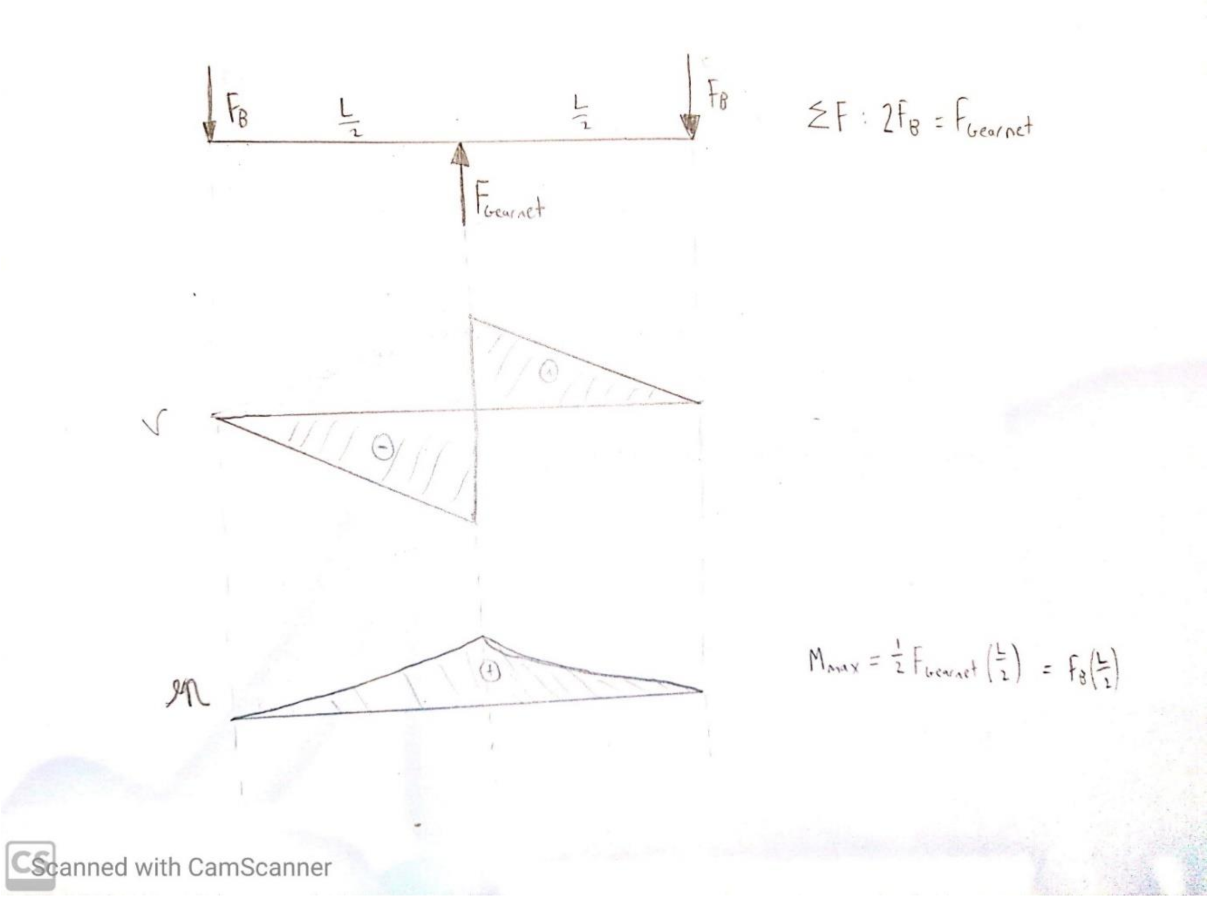
Smallest diameter is 22mm

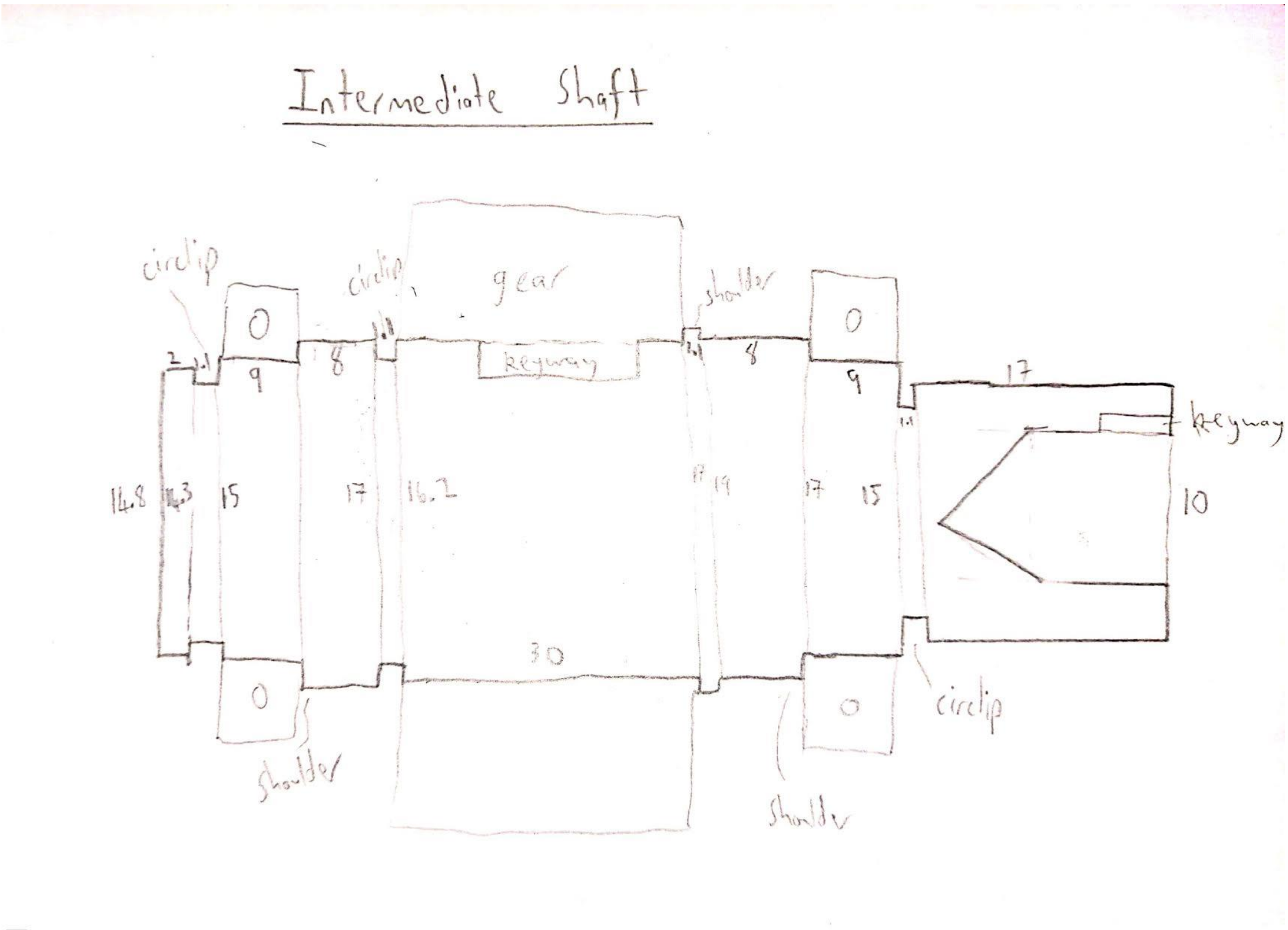
$$sigma_{bmaxo} := \frac{32 \cdot \frac{F_{Bmax}}{2}}{\pi \cdot (22mm)^3} = 2.29 \times 10^7 \frac{kg}{s^2 \cdot m^2} \qquad sigma_{bnormalo} := \frac{32 \cdot \frac{F_{Bnormal}}{2}}{\pi \, (22mm)^3} = 4.492 \times 10^6 \frac{kg}{s^2 \cdot m^2}$$

$$L_{maxo} := \frac{sigma_b}{sigma_{bmaxo}} = 5.895m \qquad L_{maxnormalo} := \frac{sigma_b}{sigma_{bnormalo}} = 30.049m$$

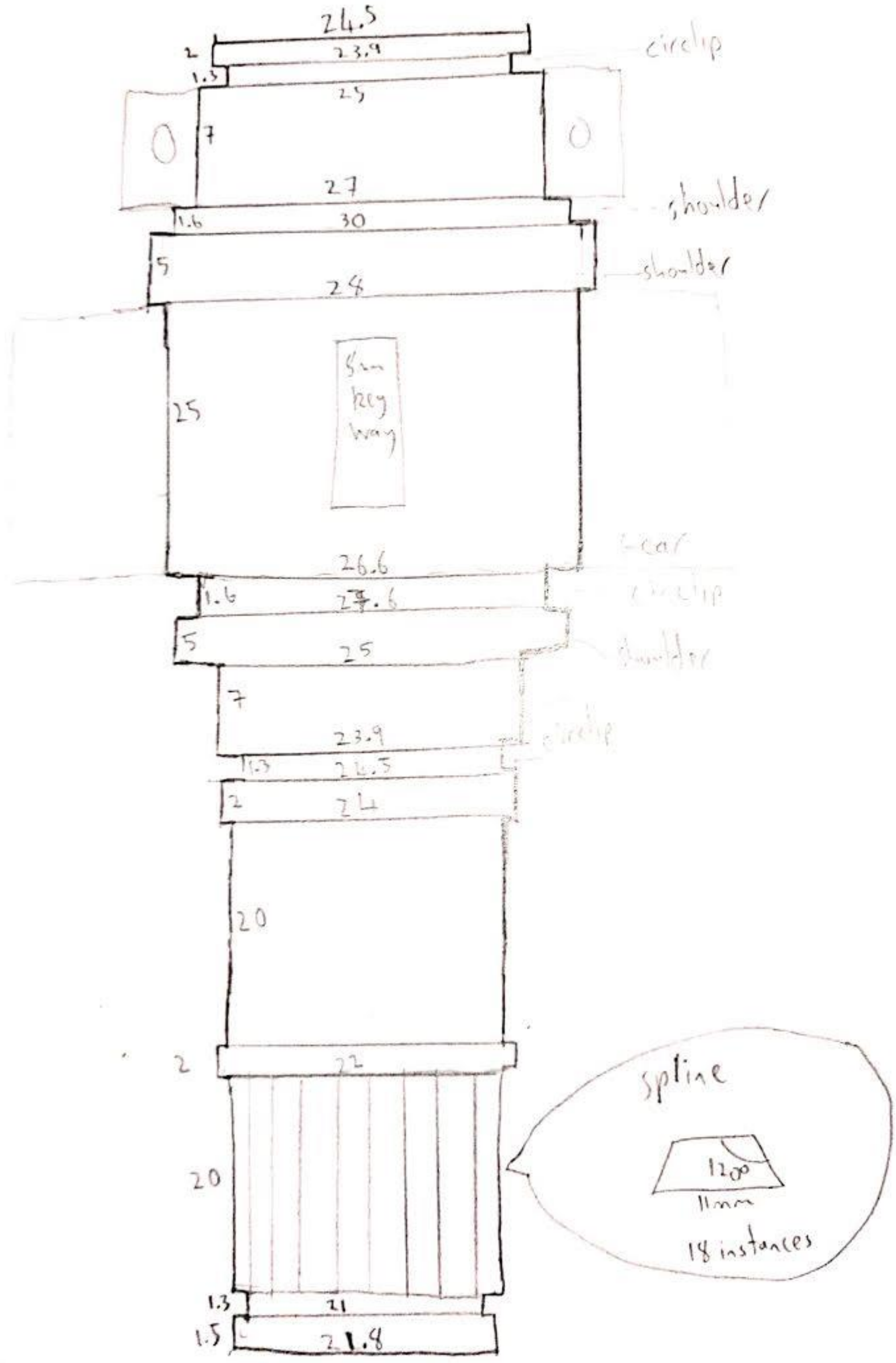
Maximum output shaft length will definitely be under 5.895m.

The following two pages contain the general moment diagram for the intermediate and output shaft as well as the intermediate and output shaft dimensions.


F_B
$\frac{L}{2}$
$\frac{L}{2}$
F_B
$\Sigma F : 2F_B = F_{Gearnet}$
$F_{Gearnet}$
V
M
$M_{max} = \frac{1}{2} F_{Gearnet} \left(\frac{L}{2}\right) = F_B\left(\frac{L}{2}\right)$
Scanned with CamScanner


Intermediate Shaft
circlip
circlip
gear
shoulder
keyway
keyway
17
10
30
circlip
shoulder
shoulder

Output Shaft



11.6 Mass Calculation

NO.	PART NUMBER	DESCRIPTION	QTY	MASS (grams)	MASS OF ALL	REFERENCE	MATERIAL
1	MEC4103-019	Motor with Key	1	630	630	See motor specification doc	N/A
2	DJNQUI001-001	Polishing Head	1	213.25	213.25	Solidworks	ABS
3.1	DJNQUI001-002	18650 NMC Cell	8	45	360	See cell options doc	N/A
3.2	DJNQUI001-003	Cell Holder	1	181.03	181.03	Solidworks	ABS
4	DJNQUI001-004	Motor Bracket and Bearing Housing	1	155.21	155.21	Solidworks	6061 Aluminium
5	DJNQUI001-011	Small Pinion	1	197.095	197.095	Solidworks	Carbon Steel
6	DJNQUI001-012	Large Gear	1	1081.357	1081.357	Solidworks	Carbon Steel
7	DJNQUI001-005	Output Shaft	1	484.44	484.44	Solidworks	En24T Alloy Steel
8	UCT-10308	Key 3×3	1	0.05	0.05	Solidworks	Carbon Steel
9	UCT-16019	Circlip, external 22 X 1.2	1	0.2	0.2	Solidworks	Spring Steel
10	UCT-04010	Socket CSK HD Bolt M4	3	0.189	0.567	Solidworks	Carbon Steel
11	UCT-15005	Washer M4	5	0.039	0.195	Solidworks	Carbon Steel
12	DJNQUI001-006	Intermediate Shaft	1	125.73	125.73	Solidworks	En24T Alloy Steel
13	SKF-6002	15mm, 6002 Bearing	2	30	60	SKF site	N/A
14	SKF-6005	25mm, SKF 6005 Bearing	2	78	156	SKF site	N/A
15	UCT-16021	Circlip, external 25 X 1.2	2	0.238	0.476	Solidworks	Spring Steel
16	UCT-16023	Circlip, external 28 X 1.5	1	0.352	0.352	Solidworks	Spring Steel
17	UCT-16012	Circlip, external 15 X 1	2	0.093	0.186	Solidworks	Spring Steel
18	UCT-16014	Circlip, external 17 X 1	1	0.107	0.107	Solidworks	Spring Steel
19	UCT-10354	Key 6×6	1	0.78	0.78	Solidworks	Carbon Steel
20	UCT-10003	Key 8×7	1	1.39	1.39	Solidworks	Carbon Steel
21	UCT-17030	Circlip, internal 47 X 1.75	1	0.854	0.854	Solidworks	Spring Steel
22	UCT-17021	Circlip, internal 32 X 1.2	1	0.299	0.299	Solidworks	Spring Steel
23	DJNQUI001-007	Switch	1	2.8	2.8	Solidworks	ABS
24	DJNQUI001-008	Top Casing and Bearing Housing	1	545.6	545.6	Solidworks	ABS

25	DJNQUI001-009	Bottom Casing and Housing	1	469.7	469.7	Solidworks	ABS
26	UCT-07034	Cross CSK HD screw M3	11	0.153	1.683	Solidworks	Carbon Steel
27	UCT-04023	Socket CSK HD Bolt M5	2	0.39	0.78	Solidworks	Carbon Steel
28	UCT-07043	Cross CSK HD screw M4	2	0.3	0.6	Solidworks	Carbon Steel
29	DJNQUI001-10	Removable Battery Cover	1	21.76	21.76	Solidworks	ABS
30	UCT-15003	Washer M3	11	0.015	0.165	Solidworks	Carbon Steel
31	UCT-15006	Washer M5	2	0.056	0.112	Solidworks	Carbon Steel
32	UCT-11000	Hex flange nut M5	2	0.266	0.532	Solidworks	Carbon Steel
				TOTAL MASS	4693.3g		