Automatic Cordless Fish Scaler

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EXECUTIVE SUMMARY

The market has a need for a new and innovative rechargeable cordless hand tool for people with impaired mobility, strength, or manual dexterity to aid in alleviating everyday tasks. Together with the client, it was decided that a product catering to the needs of the aging population would be designed. This report details the structured design process in designing a tool based on engineering considerations. With the client being a leading manufacturer of battery-operated products, the product was designed appropriately for mass manufacture of a projected 10,000 units per annum and to have many synergies with existing products. Multiple initial product ideas catering to several common everyday tasks were presented to the client with the Fish Scaler being selected as the product idea to be developed further. Based on market research carried out, this product was identified as the one with the highest potential due to its demand, functionality, practicality and versatility of the application in that it has the potential to be sold beyond the target market.

Presently, fish scaling is done manually using a knife or knife-like scaling tool placed at an angle and moved at quick speeds. This method is inadequate in three respects: firstly, it is a rigorous and time-consuming task. Second, it is a very dangerous application. Lastly, the outcome can be unsatisfactory and messy. The possibility of a fish not being scaled cleanly is very high, and the clean-up that comes along with the task requires makes this a dreaded one.

The objectives set out in the Product Design Specification were aimed at addressing these issues. The comprehensive engineering analysis done ensured carefully selected design features from detailed morphological analysis were safely and successfully implemented. This resulted in a final product in which all its functional systems have been streamlined to significantly reduce scaling time. The functional head is driven by a simple rotating shaft based on planetary gear transmission system. A dimensionally optimal tool size was also achieved (ϕ50 by 250 mm) making for a portable, compact and sleek build. The ergonomically designed slider and rocker switches for straightforward interface, lasting battery life of up to 1.7 hours, and scaling tool guard promises a user-friendly, technically functional, and reliable product whilst still prioritising user safety. Additionally, the implemented removable head and water-resistant feature further enhances the user interface making the usage and cleaning of this tool effortless.

Figure 1: Alternate view of the conceptually designed tool

The product is also priced attractively at £45.00 to ensure value for money. The profitability of the product was confirmed through a costing analysis which results in profits of £14.37 per unit. Uncertainties were analysed and actions to mitigate these risks were discussed in the event of unpredictable circumstances during production and launch of the product. It was concluded that the Cordless Fish Scaler, efficient and user-friendly design, has the capability to breakthrough in the market.

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1. INTRODUCTION

Scaling fish is one of the most essential methods in cooking; whether in restaurants or at home. However, a reliable automated tool for this task is still missing in the market to aid the traditional methods of scaling in preventing the unpleasant experience of eating scales. Hence, this would represent a feasible and marketable product, especially in the context of people suffering from hand impairments. Statistics further show that the kitchen is one of the main areas in need of adaptive tools.

Traditional methods of manual scaling are very labour intensive and time-consuming. It involves repetitive, quick, back and forth motions to de-scale the fish. The wrist dexterity, grip strength of the hand and fingers, and arm strength required for this application would make this an extremely difficult even for a healthy adult. Scaling also involves radial and ulnar deviation which are common planes of movement Rheumatoid and Osteo arthritis patients struggle with and have pain. Present methods of scaling could easily cause cuts which suggests safety issues. Additionally, the difficulty of removing the scales varies with the species of fish with some requiring a more gentle and refined stroke while others may require a more pressured stroke. The hassle of cleaning up the mess that comes along with this task is also something related with these methods. The product was designed to be reliable at solving these problems whilst providing a user-friendly and convenient experience. In short, this tool makes the essential yet overlooked task of scaling a fish a simple, quick and safe job.

2. PRODUCT DESIGN SPECIFICATION

Based on the client's requirements and data on target market preferences, a PDS was developed as shown in Table 1.

Table 1: Product Design Specification

ASPECT		OBJECTIVE	CRITERIA	TEST PLAN
Performance	Frictional Force	To effectively remove scales of the fish easily with minimal downward force	Minimum of 45 N	Literature and Estimation
	Motor Power	Must maintain constant angular frequency/speed	Minimum of 12.5 W	Calculation
	Duration of operation	Continuous use	Minimum of 1.5 hours of runtime for a single charge	Calculation
	Charging time (per charge)	Maximum of 30 minutes	To recharge to full battery capacity	Calculation
	Design Life	4 years frequent usage	Main output shaft must have a fatigue safety factor>2.0	Calculation
Dimensions and size (Aesthetics and Ergonomics)	Mass of device	Mass of whole product	Maximum of 2 kg	Measurement
	Length of device	Can fit in camping bag or kitchen drawer	250mm	Measurement
	Average diameter of handle grip	Ideal diameter for carrying and optimal grip strength	5.0 cm (based on literature)	Measurement and literature
	Shape of Handle	Comfortable hold with minimal grip force required. Be able to hold for extended periods without user fatigue.	Organic ergonomic shape which fits comfortably in the hand	Public feedback. Mock prototype and carry out market research.
Operation	User-Friendliness	Only two switches to operate	Press of a button to switch on and a slider to change speed	-
	Cleaning	Does not require additional tools to remove head	Removable head which can easily be wiped down or washed	Observation
	Versatility	To operate on varied sizes and types of fish	Multispeed (3 Speed) transmission	-
Manufacture	Quantity	Mass production	10 000 per annum	-
Safety	Protection from transmission	No moving and rotating transmission parts visible	Cover transmission completely Plastic Casing/Housing	-
	Protection from electrical components	Able to wipe down or wash lightly the body for cleaning purposes.	Withstand 1m immersion in water for 30s. Splash proof.	Submerge prototype in water. Open casing to check parts.
	Protection from Cutting Tool	Clear polymer cover snaps on and off for easy cleaning	Scaling tool shielded at the top	Observation
	Safety Factor of Transmission	Lasts the design life	All gear stresses have safety factor > 1.5	Calculation
Cost	Manufacturing Cost	10 000 units per annum	Less than £30 per unit	Calculation
	Retail Price	Value for Money	Less than £50	-

3. DESIGN PROCESS

Data Collection and Experimentation: Research was done on various ergonomic principles and functionalities of the proposed hand tool. Initial design choices (see Figure 2) were then made based on the data obtained. Table 2 also summarises a selection of data values obtained that were incorporated into the design and engineering analysis while Figure 2[8] displays an example of useful research data key to the ergonomics of the tool:

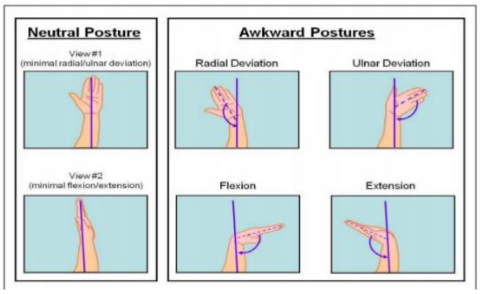


Figure 2: Neutral and awkward wrist positions

Table 2: A few values obtained from data collection

Max. force to pierce a fish scale[7]	Optimal grip diameter[6]
3.2 N	5 cm

The product was divided into multiple systems which were defined based on the functionality which reflects the PDS requirements. Based on the data collected and research, multiple solutions to each systems function were developed to select the most suitable combination of solutions. This was achieved through a morphological analysis (figure 3) represented using an IBIS chart, concept development, and in the final design analysis below.

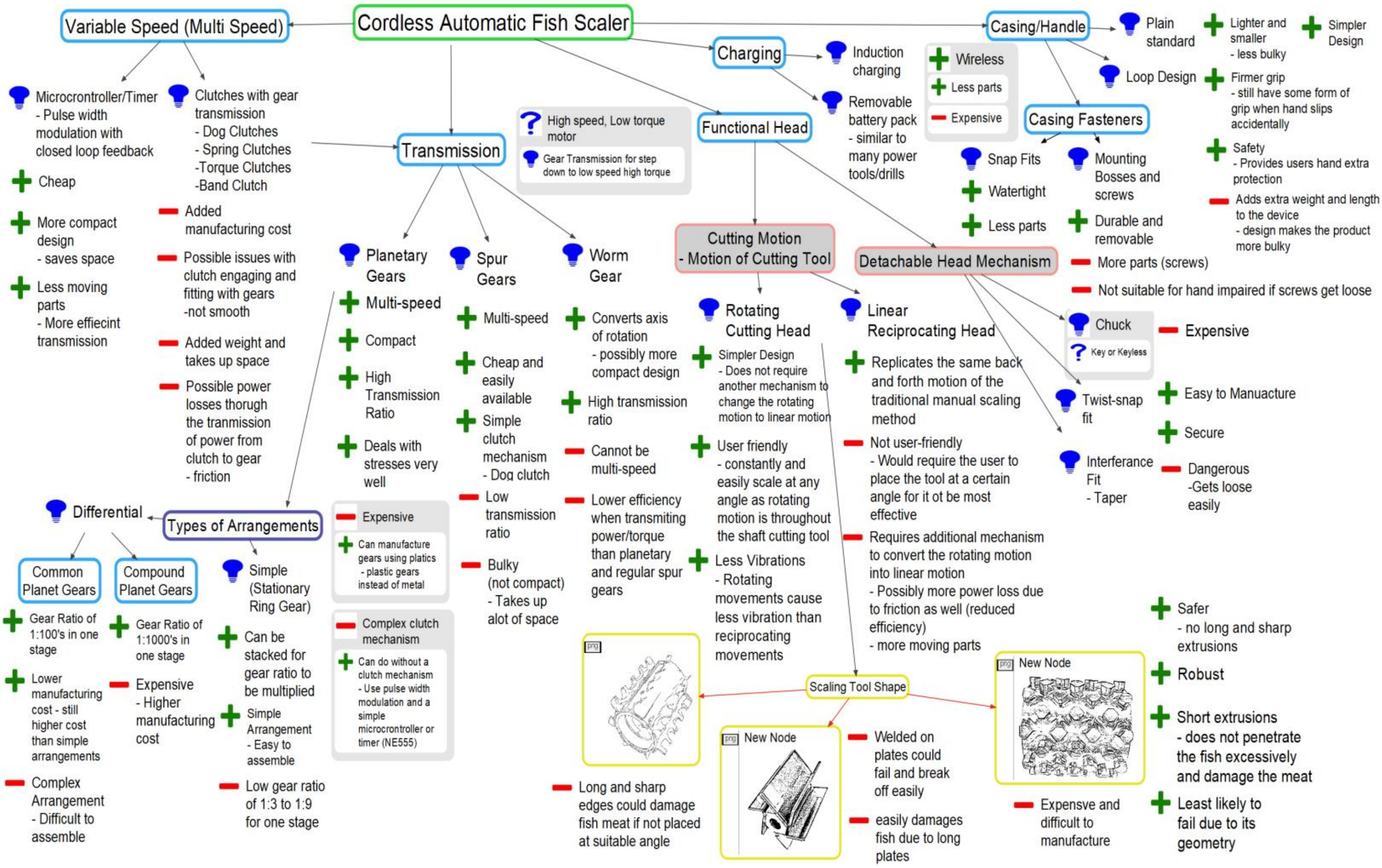
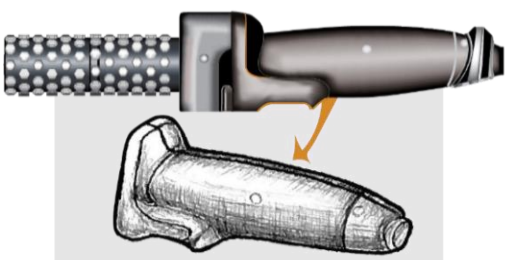


Figure 3: Morphological Analysis - IBIS Chart

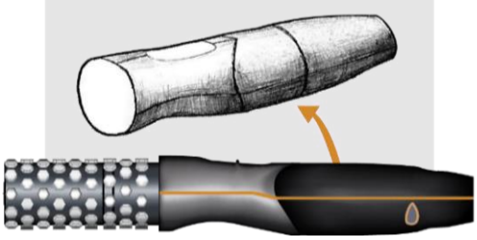
CONCEPT DEVELOPMENT

Multiple product concepts were then produced based on various combinations of the functional solutions.



- Pressure sensitive button to switch on and control speed
- Uneven weight distribution

- Manoeuvrable
- Compact
- Incorporates all requirements



- Bulky and heavy
- Provides more support
- Safer

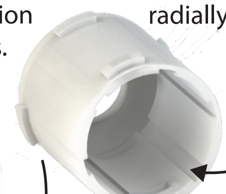
FINAL DESIGN ANALYSIS

TRANSMISSION ASSEMBLY

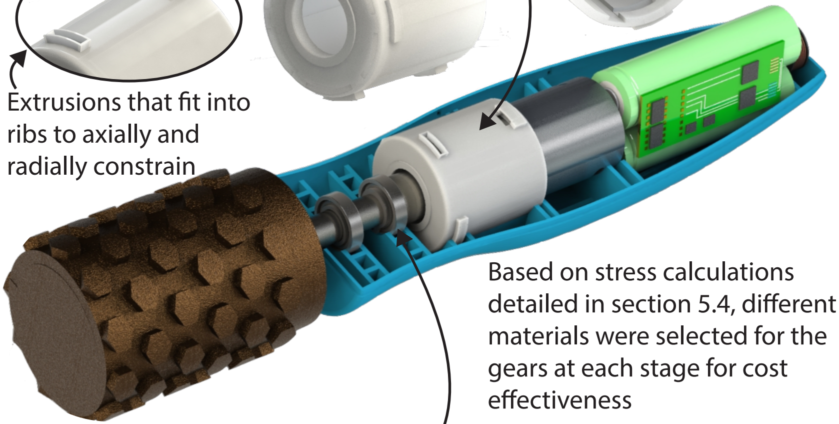
Planetary gear set selected due to compactness, high efficiency, and ability to handle high stresses.

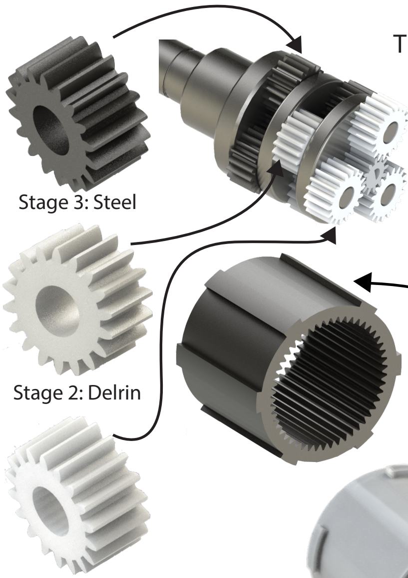
Gearbox casing holds all transmission components.

Internal grooves to radially constrain ring gear



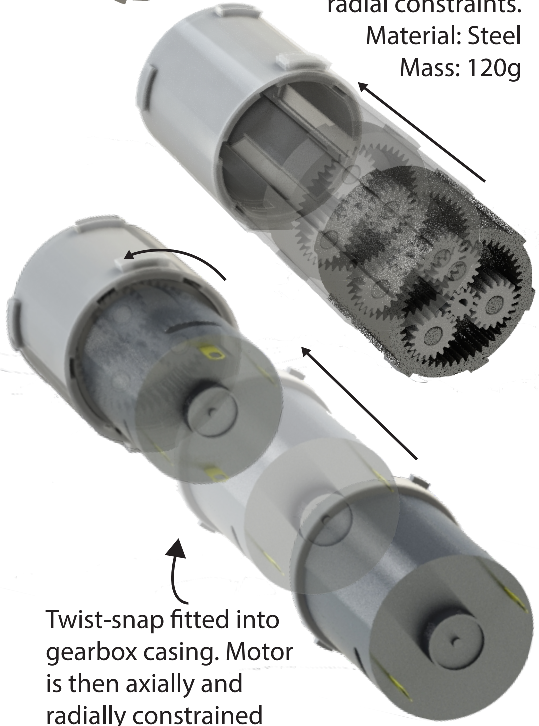
Extrusions that fit into ribs to axially and radially constrain





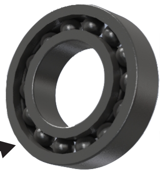
The gearbox was designed with identical reduction stages to ease manufacturing and lower costs. 3 stages are stacked in series in a multiplier arrangement to obtain desired reduction ratio.

Ring gear - extrusions that fit into the grooves of the motor casing for radial constraints. Material: Steel Mass: 120g

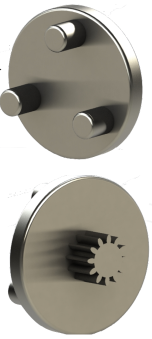


Based on stress calculations detailed in section 5.4, different materials were selected for the gears at each stage for cost effectiveness

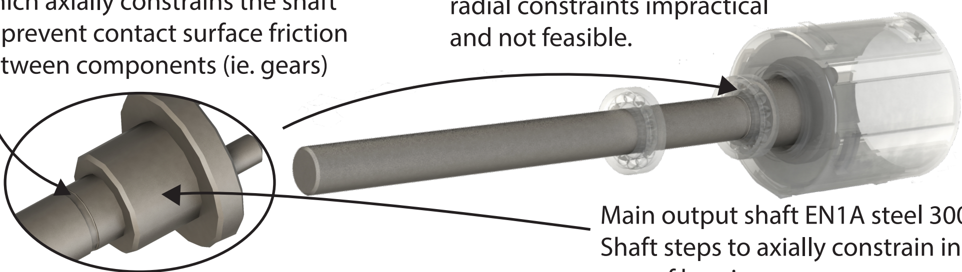
Ball bearings used to prevent axial and bending loads into the gearbox. Arranged in a 3-fixed-1-free configuration detailed in Appendix 1



Carrier: Material - steel. 3 pins press-fitted onto carrier. Sun gear is manufactured by high precision stamping. Common in many garden power tools. The small and compact nature of the gears make keys and other forms of radial constraints impractical and not feasible.



Circlip groove holds the circlip which axially constrains the shaft to prevent contact surface friction between components (ie. gears)

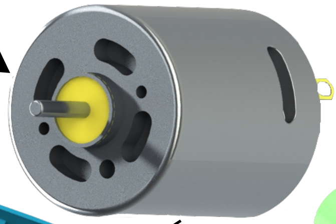


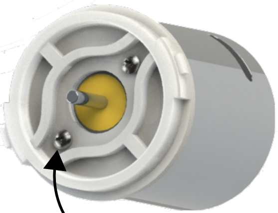
Main output shaft EN1A steel 300g Shaft steps to axially constrain inner race of bearing

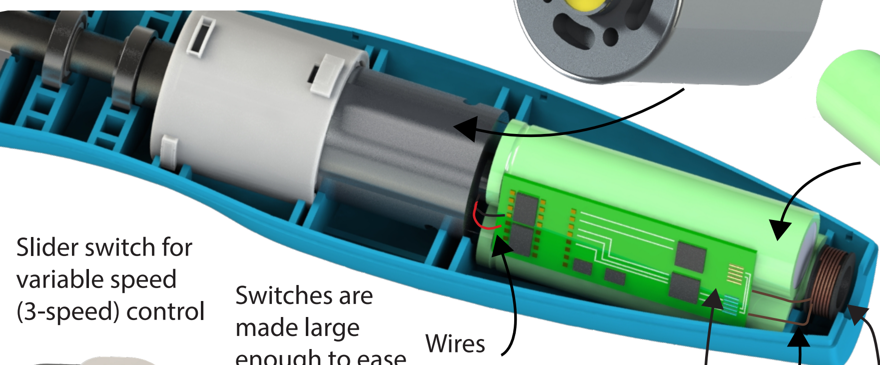
Twist-snap fitted into gearbox casing. Motor is then axially and radially constrained when assembled

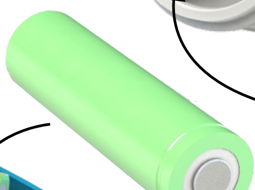
ELECTRICAL COMPONENTS

Mabushi RS-365SH DC Motor 18V nominal voltage. Mass: 54g Motor selection detailed in section 5.2









Motor cap screwed onto motor

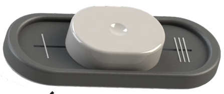
Slider switch for variable speed (3-speed) control

Switches are made large enough to ease usage

Wires connecting ECU to battery and motor

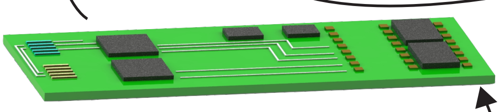
18650 Li-ion battery. 5 cells are arranged in series to obtain an 18V 2.0Ah battery pack. Frequently found in many power tools

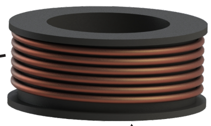
Strengthening ribs to improve structural integrity



Coil wires connected to ECU

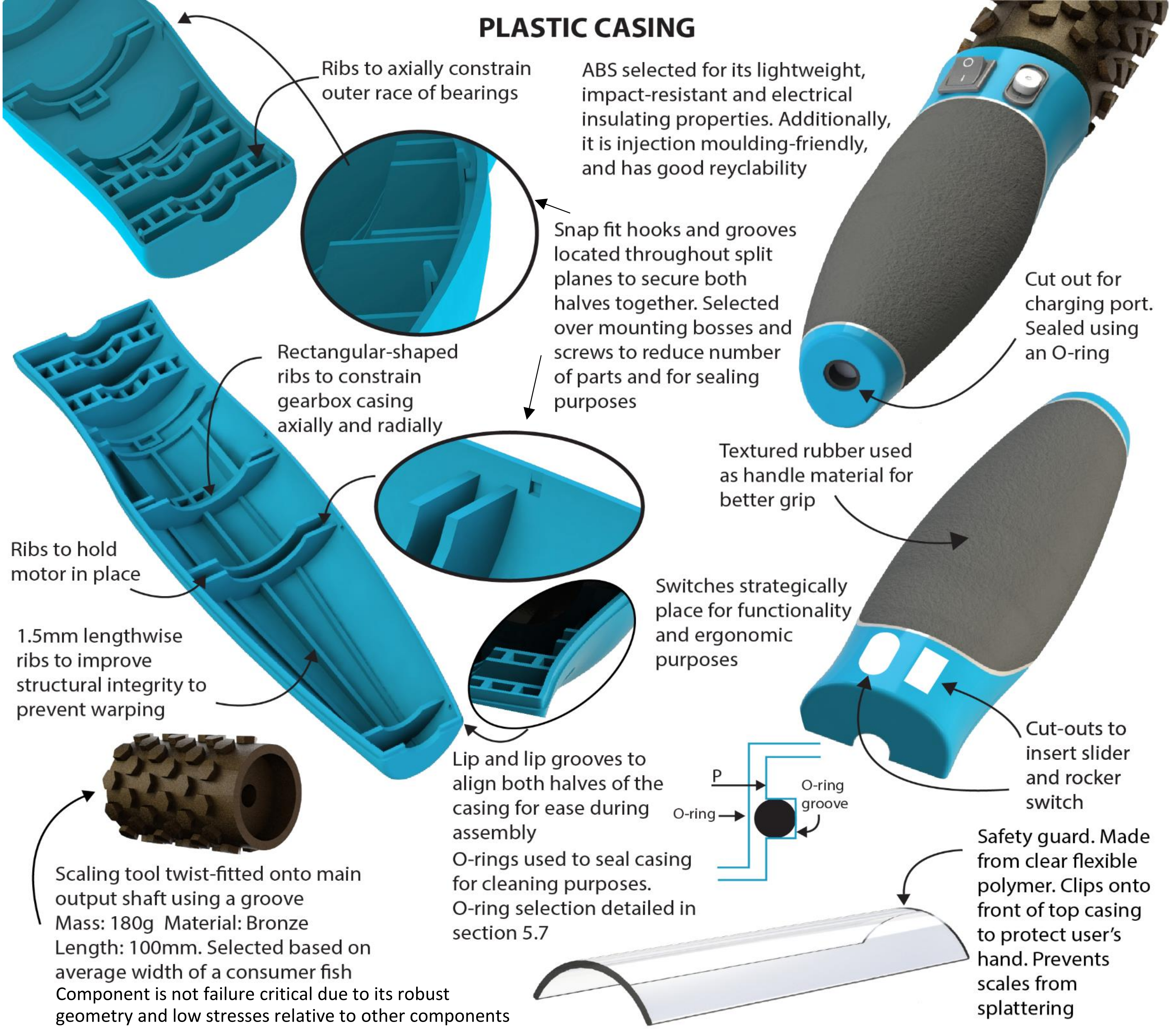
Rocker switch selected to prevent accidental switching on





Induction charging coil. Common charging method in electrical appliances

Electrical control unit (ECU) used as a variable speed system to maintain optimal operating conditions. Based on PWM and closed loop feedback



5. ENGINEERING ANALYSIS

5.1 OUTPUT TORQUE AND POWER REQUIREMENTS

Analysis of output requirements were conducted based on traditional methods of fish scaling (Figure 4):

Optimal Length of one scaling stroke = 50 mm

Average time taken for efficient stroke = 0.2s

Minimum Linear Velocity = $\frac{0.05}{0.2} = 0.25\ ms^{-1}$

Angular Velocity = $\frac{Linear\ Velocity}{Radius\ of\ Cutting\ Tool} = \frac{0.25}{20 \times 10^{-3}} = 12.5\ rads^{-1}$

Conservative estimate of tangential force required= 50N.

Output Torque Required:

$Tangential\ Force \times Radius\ of\ Cutting\ Tool = 50 \times 0.02 = 1\ Nm$

$Power = Torque \times Angular\ Speed = 1 \times 12.5\ rads^{-1} = 12.5\ W$

Figure 4: Picture illustrating traditional scaling methods

5.2 MOTOR AND BATTERY SELECTION AND ANALYSIS

An important parameter as set out in the PDS was battery life. Therefore, motor selection was done with efficiency as priority. MATLAB was used to analyse available motors and output operating parameters at maximum efficiency. Motor efficiency plots were then used to select a suitable motor. The specifications of the selected motor are listed table 3:

Table 3: DC Motor Specifications

Model	Voltage		No Load		Maximum Efficiency				Stall		Dimensions	
	Operating Voltage (V)	Nominal (V)	Speed (rpm)	Current (A)	Speed (rpm)	Current (A)	Torque (Nm)	Power Output (W)	Torque (mNm)	Current (A)	Weight (g)	Dimensions (mm)
RS-365SH	6-20	18	19200	0.25	15910	1.14	0.0077	12.77	42.6	5.2	54	⌀27.7x32.6

Battery Life: The batteries were selected and arranged with specifications and configuration in Table 4 for operating voltage to be equal to nominal voltage. This ideal configuration ensures the motor was not oversized or undersized.

Table 4: Battery Specifications and Configuration

	Voltage (V)	Capacity (mAh)	Dimensions (mm)
One cell	3.6	2000	⌀18 x 65
Battery Pack –5 cells in series	18	2000	-

Assuming the tool runs at desired operating point at max. efficiency. This configuration gives the device a minimum run time of 1.76 hours per charge which satisfies the PDS.

Battery Life calculation:

$$I \times t = Capacity\ (Ah)$$

$$t = \frac{Capacity(Ah)}{I} = \frac{2}{1.139} = 1.76\ hours$$

5.3 GEAR SELECTION

Based on the motor output torque at the selected operating point and the output torque requirements, the reduction ratio of the planetary gears required was 129.87. A practical gear ratio was selected based on desired size considerations of the outer diameter of the ring gear and suitable and optimal number of teeth. The planetary gears were then designed with specifications in Table 5:

Table 5: Gear Ratio Calculations

Total Gear Ratio Required	Number of stages	Gear Ratio in a single stage	Practical gear ratio of one stage
$R = \frac{1}{0.0077} = 129.87$	3	$R_1 = \sqrt[3]{129.87} = 5.064$	5.081

Table 6: Number of teeth of each gear

GEAR	Sun	Planet	Ring
NUMBER OF TEETH	12	18	49

Table 7: Comparison of required and actual output parameters

Output	Output Torque	Output speed
Required	1.00	12.6
Actual	1.01	12.7

5.4 STRESS ANALYSIS OF GEARS (GEAR STRESSES)

Based on the Lewis formula[2] which considers the bending stresses, an acceptable face width was calculated:

$$V_{PLV} = \frac{d_p}{2} \times \omega \quad , \quad K_v = \frac{6.1}{6.1+V_{PLV}}$$

$$W_t = \frac{P}{V_{PLV}}$$

$$W_t = \frac{\frac{P}{3}}{V_{PLV}}$$

For planet gears: Power transmitted is split equally among the 3 planet gears

$$F = \frac{W_t}{K_v \cdot m \cdot Y \cdot \sigma_p}$$

d_p: Pitch Centre Diameter
ω: Angular Velocity
V_{PLV} : Pitch Line Velocity
W_t : Transmitted Load
K_v: Velocity Factor
P: Power Transmitted
F: Face Width
N: Number of teeth
R: Gear ratio of one stage
σ_p: Maximum Bending stress of material

Equations for speed of planetary gears:

$$\omega_{arm} = \frac{\omega_{sun}}{R}$$

$$\omega_{planet} = \left(\frac{N_{sun}}{N_{planet}} \times (\omega_{sun} - \omega_{arm})\right) - \omega_{arm}$$

$$\omega_{sun}(next\ stage) = \omega_{arm}$$

Note: $\omega_{carrier} = \omega_{arm}$

A script was written in MATLAB to iteratively compute the most suitable module and material for face width within the specified safety factor relative to available face width (HPC). Input parameters and results of this analysis is displayed in tables 8 and 9.

Table 8: Input Parameters of each stage of sun and planet gears

Gear	Number of teeth, N	Module (mm)	PCD, d_p (mm)	Lewis Form Factor, Y	Rotational Speed of Gear, ω $(rads^{-1})$		
					1st Stage	2nd Stage	3rd Stage
Sun Gear	12	0.5	6	0.22960	1666.667	333.019	64.550
Planet Gear	18	0.5	9	0.29327	564.412	111.083	21.862

Table 9: Intermediate output parameters and stress results of gears

STAGE	GEAR	MATERIAL	ALLOWABLE STRESS OF MATERIAL (MPa)	PITCH LINE VELOCITY, V (ms^{-1})	VELOCITY FACTOR, K_v	TRANSMITTED LOAD, W_t (N)	MINIMUM FACE WIDTH (mm)	FACE WIDTH AVAILABLE (mm)	SAFETY FACTOR, SF
1st Stage	Sun	Tufnol	31.0	5.00	0.5495	2.5540	1.3059	5	3.8288
	Planet	Tufnol	31.0	2.54	0.7060	1.6759	0.5281	5	9.5748
2nd Stage	Sun	Delrin	79.0	0.98	0.8611	12.9769	1.6389	5	3.0090
	Planet	Delrin	79.0	0.50	0.9243	8.5155	0.7951	5	6.2866
3rd Stage	Sun	Steel (81M40)	183	0.19	0.9692	65.9355	3.1389	5	1.5441
	Planet	Steel (045M10)	117	0.01	0.9841	43.2672	2.5218	5	1.9511

Table 9 shows how material selection of the gears were made based on the cost and weight considerations which satisfy the minimum face width values obtained through the Lewis Formula. Where stresses were low, plastic gears were chosen over metal gears. The stresses on the ring gear were calculated to be minimal relative to the other two gears.

Where life of the components is important, more rigorous standards are used. The relatively tight safety factors (highlighted in orange) in the 3rd stage obtained from the Lewis Equation analysis suggests further detailed analysis on these highly stressed parts were required to account for other modes of failure. Hence, AGMA calculations[2], derived from Hertzian contact theory, were done on this life critical component to analyse failure in the form of pitting which accounts for the other from of stress on gears – contact stresses. Pitting is surface fatigue failure due to repetitive high contact stresses. Two types of failure modes were considered, failure due to bending fatigue using quantities $\sigma_{b,all}$, $\sigma_{b,act}$ and failure due to pitting through quantities, $\sigma_{c,all}$, $\sigma_{c,all}$. AGMA is considered conservative as it evaluates stress with respect to 10^8 cycles. The input parameters were selected based on the following: uncrowned teeth, enclosed gearing, and the temperature of the lubrication is kept within its recommended range.

Table 10: AGMA Equations and Input Parameters

AGMA Equations	Input Parameters					
$\sigma_{b,all} = \frac{S_t Y_N}{S_F Y_\theta Y_Z}$; $\sigma_{b,act} = W^t K_o K_v K_s \frac{1}{Fm_t} \frac{K_H K_B}{Y_J}$	K_s	K_o	Y_J	K_B	Q_v	C_e
$\sigma_{c,all} = \frac{S_c Z_N Z_W}{S_H Y_\theta Y_Z}$; $\sigma_{c,act} = Z_e(W^t K_o K_v K_s \frac{K_H}{Fd_p} \frac{Z_R}{Z_I})^{0.5}$	1	1.2	0.21	1	8	1

Table 11: AGMA rating results and safety factor for bending

Gears	Bending Stress (MPa)	Allowable bending stress (MPa)	Safety Factor
Sun	148.4	291.6	1.57
Planet	32.87	66.63	2.03

Table 12: AGMA rating results and safety factor for contact (pitting)

Gears	Contact Stress (MPa)	Allowable contact stress (MPa)	Safety Factor
Sun	161	758	4.71
Planet	85.9	681	7.93

Safety factors of bending stress and contact stress (Tables 11 and 12) meet the necessary minimum values required.

5.5 STRESS ANALYSIS OF SHAFT

The main form of failure was identified as fatigue failure of the shaft. Potential failure by fatigue is due to the possibility of frequent usage of the hand tool and the cyclic loading nature of the application. Hence, the hand tool was designed to be resistant against fatigue. The ASME procedure[1] used ensures the shaft was properly sized to provide adequate service life. Further analysis and calculations were then done to ensure the shaft was stiff enough to limit deflections of power transfer elements and minimise misalignments of bearings. A vibration analysis was then done to ensure the shaft's stiffness prevents unwanted vibrations in operating range. The forces acting on the were modelled as shown in the free body diagrams in figures 5,6,7 and 8.

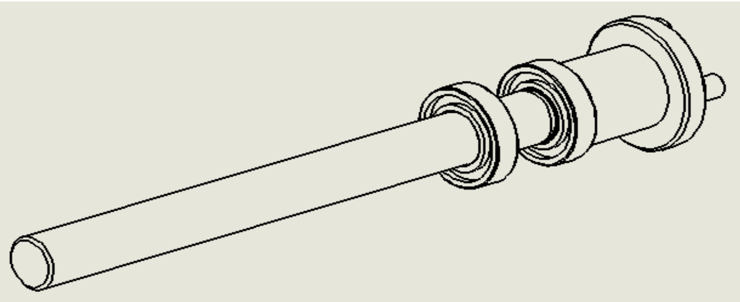


Figure 5: 3D Schematic View of Shaft

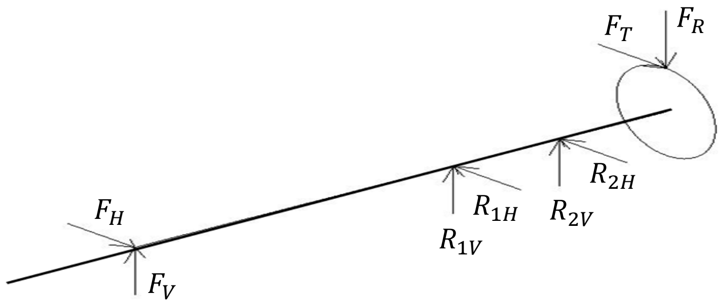


Figure 6: Shaft Loading Diagram

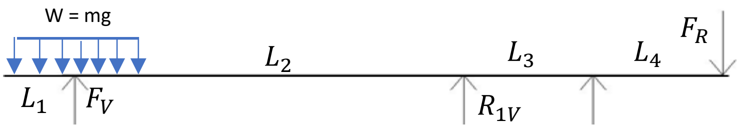


Figure 7: Vertical loading diagram

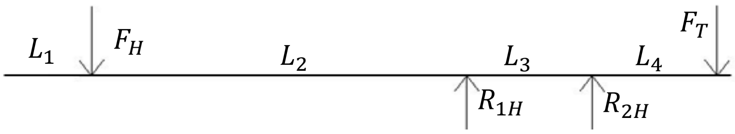


Figure 8: Horizontal loading diagram

F_r is the sum of the 3 radial loads of the final stage planet gears.

$$F_r = 3 \times W_{T(planet)} tan\emptyset = 3 \times 43.27 \tan(20) = 47.24N$$

$$F_T = \frac{T}{r} = \frac{1.01}{11 \times 10^{-3}} = 91.81\ N$$

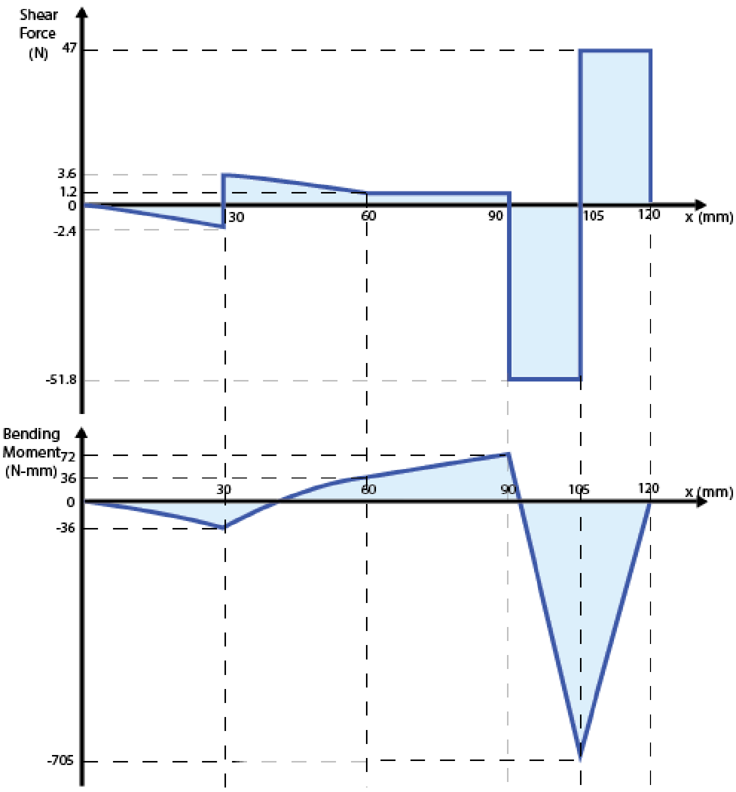


Figure 9: Vertical shear force and bending moment diagram

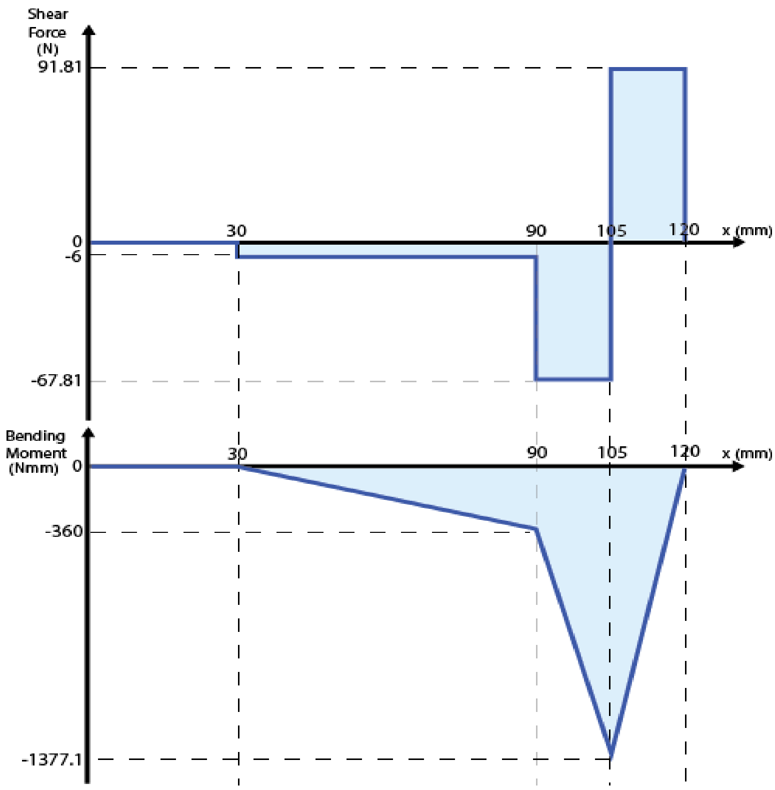


Figure 10: Horizontal shear force and bending moment diagram

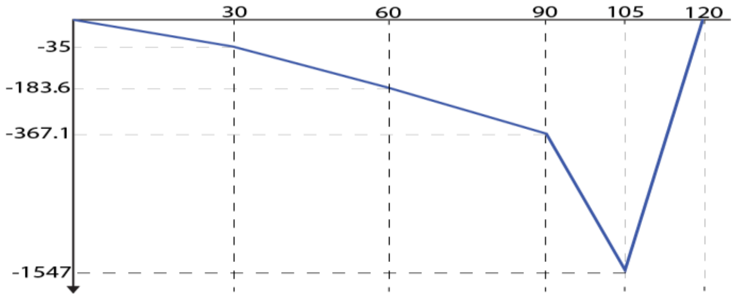


Figure 11: Resultant bending moment diagram

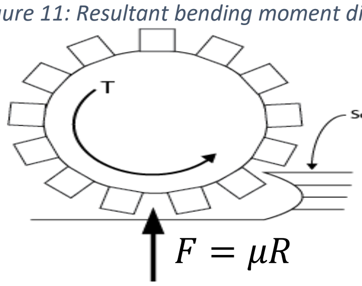


Figure 12: Schematic diagram illustrating the effect of normal force

The resultant bending moment across the beam was calculated by:

$$|M_R| = |M_H| + |M_V|$$

Figure 11 shows the resultant bending moment diagram across the shaft. The maximum bending moment is 1.58 Nm.

In reality, the torque would be a function of the normal force applied (see figure 12). A larger force applied by the user when applying downwards pressure would generate a larger frictional force. This requires a greater torque. This calculation represents an engineering model which simplifies this. Additionally, the torque and normal forces used in this analysis were conservative estimates to account for this.

5.5.1 FATIGUE ANALYSIS: ASME Design Code for Transmission Shaft[1]

Using the ASME Equation and the maximum moment calculated in the analysis above, a safe diameter designed against fatigue failure was obtained. This analysis also accounts for yield failure. The safety factor, SF against fatigue was set conservatively at 2. The correction factor was also σ_e taken as 0.2 of the UTS to be conservative. Table 13 details inputs and results of equation.

$$d = \left[\frac{32SF}{\pi}\sqrt{\left(\frac{M}{\sigma_e}\right)^2 + \frac{3}{4}\left(\frac{T}{\sigma_y}\right)^2}\right]^{\frac{1}{3}} = \left[\frac{32SF}{\pi}\sqrt{\left(\frac{1.58}{\sigma_e}\right)^2 + \frac{3}{4}\left(\frac{1.01}{\sigma_y}\right)^2}\right]^{\frac{1}{3}} = 7.57$$

Table 13: Input parameters and results of ASME Fatigue Analysis

	Maximum Moment (Nm)	Torque, T (Nm)	σ_{UTS} (MPa)	σ_y (MPa)	σ_e (MPa)	ASME Safe Diameter (mm)	Actual Diameter (mm)
Output Shaft	1.55	1.01	370	230	74	7.57	9

Analysis done confirmed design fatigue safety of the smallest diameter cross-section of the shaft (9mm). This represents the most critical section and hence, most likely to fail. The stresses in the shaft were then calculated for detailed future analysis.

Shear stresses $\tau = \frac{T}{J}r$; $J = \frac{\pi d^4}{32} = 6.44 \times 10^{-10}$; $\tau = \frac{1.01}{6.44\times10^{-10}}(4.5 \times 10^{-3}) = 7.057MPa$

Bending stresses $\sigma = \frac{M}{I}y$; $I = \frac{\pi d^4}{64} = 3.22 \times 10^{-10}$; $\sigma = \frac{1.58}{3.22\times10^{-10}}(4.5 \times 10^{-3}) = 22.08MPa$

5.5.2 DEFLECTION AND SLOPE

Macaulay's method was used to obtain the deflection and the slope at critical points along the shaft to ensure stiffness.

$EI\frac{d^2y}{dx^2} = M$; $M = F_H\{x - L_1\} - R_1\{x - (L_1 + L_2)\} - R_2\{x - (L_1 + L_2 + L_3)\}$

Slope: $EI\frac{dy}{dx} = \frac{F_H}{2}\{x - L_1\}^2 - \frac{R_1}{2}\{x - (L_1 + L_2)\}^2 - \frac{R_2}{2}\{x - (L_1 + L_2 + L_3)\}^2 + A$

Deflection: $EIy = \frac{F_H}{6}\{x - L_1\}^3 - \frac{R_1}{6}\{x - (L_1 + L_2)\}^3 - \frac{R_2}{3}\{x - (L_1 + L_2 + L_3)\}^3 + Ax + B$

The values of slope and deflection at the critical and loaded points were calculated by applying the boundary conditions in which assumes zero deflection at the bearings ($y = 0$ at $x = L_1 + L_2$ and $y = 0\ at\ x = L_1 + L_2 + L_3$). The results are shown in Table 14:

A = $\frac{F_H}{6L_3}(L_2^3 - (L_2 + L_3)^3) + \frac{R_1}{6}L_3^2$; B = $-\frac{F_H}{6}(L_2)^3 - A(L_1 + L_2)$; $I = \frac{\pi d^4}{64} = 3.22 \times 10^{-10}$; E = 200GPa

Table 14: Deflection and Slope values

Shaft	**OUTPUT SHAFT**			
Location	3rd Stage Carrier (x=120)	Bearing 1, R_1	Bearing 2, R_2	Functional Head
Deflection, y (mm)	2.0231×10^{-6}	0	0	1.907×10^{-5}
Slope, $\frac{dy}{dx}$	2.8128×10^{-4}	1.209×10^{-3}	-8.141×10^{-5}	2.491×10^{-4}

From literature, the limiting deflections allowed in components are as follows: Gears: deflection < 0.13 mm and slope < 0.03°; Rolling element bearings: slope < 2.5°. Hence, this ensures power transfer components (i.e. carriers, gears) mesh properly for power to be transferred efficiently and ensures rolling element bearings can rotate freely.

5.5.3 VIBRATION ANALYSIS AND CRITICAL SPEEDS OF THE SHAFT

This analysis aims to ensure critical speed is significantly higher than the operating (design) speed. The only possible vibrating mass on the shaft is the functional head. The bearings and the carrier were all constrained and will not be a source of vibration. Using the Rayleigh Ritz equation[1], the critical speed is calculated by:

$$\omega_c = \sqrt{\frac{g\sum_{i=1}^{n} W_i y_i}{\sum_{i=1}^{n} W_i {y_i}^2}} = \sqrt{\frac{9.81\ (6 \times 1.907 \times 10^{-5})}{6 \times (1.907 \times 10^{-5})^2}} = 5439.85\ rads^{-1} = 51947rpm$$

Therefore, the critical speed was calculated to be much sufficiently and significantly higher than the design speed which prevents critical vibrations which can cause failure of components (i.e. shaft). This analysis also ensures the ergonomic standards of the product and prevents any health and safety risks arising from vibration.

5.6 BEARING CALCULATIONS

All bearings have a finite life and will fail due to fatigue. This is due to load being exerted on a very small area which results in very high contact stresses. Bearings were selected from SKF catalogue based on required life, L of the bearing which fit available bearings with sufficient dynamic load rating and basic rated life, L_{10} detailed in table 15.

$C = P\left(\frac{L}{10^6}\right)^{\left(\frac{1}{k}\right)}$; k = 3 for ball bearings

$$\boxed{L = L_{10} \times \omega = 10\ 000\ \times 121.33 \times 60 = 7.6 \times 10^6}$$

Resultant reaction force at bearings obtained by $|R_R| = |R_H| + |R_V|$ was used as the load, P in the equation above.

Table 15: Resultant reaction forces and Bearing Dynamic Load Rating Results

Bearing	Horizontal (N)	Vertical (N)	Resultant (N)	L (rev)	Dynamic Load Rating, C (kN)
1	53.0	61.81	81.42	7.6×10^6	0.16
2	98.8	159.62	187.7	7.6×10^6	0.37

Selected bearing: Designation - 618/9 based on inner diameter 9 and dynamic load rating, C = 0.64kN
Real service life can deviate from the calculated basic life equation. Hence, a modified life calculation[3] was done for a more conservative analysis to ensure that the gearbox is protected from axial and radial loads. The modification accounts for the lubrication, contamination, and misalignment factors among others. Table 16 displays input and results from SKF calculator[5]

Table 16: Modified life Equation Input Parameters and Results

Modified Equation	a_{SKF}	a_1	Load ratio, C/P	Reliability (%)	Condition	SKF rating life, L_{10m}
$L_{10m} = a_1 a_{SKF}\left(\frac{C}{P}\right)^k.$	2.38	1	4.6	90	Normal Cleanliness	$3 \times L_{10}$

5.7 O-RING SELECTION FOR WATER PROOFING

Internal perimeter of the casing was determined. For a static seal, the O-ring Internal Diameter (ID) was selected to be 4% smaller (range of 1-5%) than the groove ID to ensure a tight fit against the sealing surface and prevents the O ring from moving under external pressure conditions. Through an iterative process of changing the cross-section diameter, a suitable O ring was selected from the B.S. catalogue[4] – British Standard BS 4518 shown in Table 17.

Table 17: Reference numbers of O rings selected[4]

Shaft: 0081-16	Charging Port: 0071-16	Casing Perimeter: 0745-30

6. UNCERTAINTY ANALYSIS

Development of new and innovative products are associated with high risks and management challenges. This analysis aims to mitigate these risks to minimise any potential losses through strategic management while maximising any potential opportunities that arises with the uncertainties. Design related risks through potential technical failure have been addressed through the engineering analysis. Conservative values have been used throughout the calculations and design choices have been made practically. This has ensured that the product is not just theoretically feasible but workable in practice. An initial response to these uncertainties is to carry out a small-scale market test to obtain data. Additionally, assumptions made on the market are regularly and iteratively revisited and addressed to identify potential new uncertainties and validate the relevance of previous assumptions.

Table 18: Uncertainty Analysis

CATEGORY	UNCERTAINTY	ANALYSIS	RISK MITIGATION RESPONSE
Market	Market Demand	Entry in a non-peak season (no demand) could cause unnecessary losses.	Avoidance: delay entry until opportunistic time (i.e. fishing season)
	Product market	No product demand. New product not be well received by target market and public.	Cooperation: Work with Occupational Therapists and Physiotherapists to promote and advertise product
	Pricing	Priced cheaper than current market power tools and similarly or slightly more expensive than manual fish scalers.	Flexibility: Actual retail prices can be adjusted depending on first term revenue. Additionally, cost analysis was carried out conservatively.
	Market Perception	Only industrial tools which are not cordless and traditional fish scalers in current market	Control: Market the advantages and key features of the cordless product
	Competitive	Potential competitors' response would reduce duration of market exploitation	Avoid name calling and comparison in marketing advertising campaigns to prevent a response.
Production	Supplier	High dependence on supplier for purchased components	Quality checks to ensure components are up to the required standards
	Logistics	Outsourcing components would open the possibility of delayed arrival of components which will slow down the assembly line.	Cooperation: Constant close communication with supplier. Set up a more efficient system which ensures the quality of delivery and shipments on schedule.
	Manufacture	CNC equipment breaking down during production season.	Regular scheduled maintenance of the machines during to prevent break downs.

Firm	R&D	Potential for future iterations of the product	Obtain market feedback to improve current product. Introduce new features to maintain demand.
Environment	Macro-economics	Fluctuations and volatility in currency exchange rate particularly in Europe	Source for suppliers in other regions to prevent additional costs related with political uncertainty(Brexit)

COSTING ANALYSIS

Many design choices were made based on cost considerations. This is reflected in the analysis in Table 19. The analysis considers projected batch quantities of 10000. All labour costs have been accounted for in listed prices.

Table 19: Costing Analysis

Components	Quantity per tool	Price per unit	Price in bulk per unit	Price in bulk per tool	Source of Estimate	Description	
Motor	1	4.00	0.40	0.40	Mabuchi and eBay	Purchased components – Bought in bulk from supplier (Circlips and O-rings come in packs of 50)	
Lithium Ion Batteries	5	3.75	0.38	1.90	Samsung		
Rocker Switch	1	1.50	0.15	0.15	RS		
Slider Switch	1	1.00	0.10	0.10	RS and C&K		
Roller Bearings	2	7.12	0.71	1.42	SKF and bearingboys		
Circlips	1	0.16	0.02	0.02	RS		
O-rings	3	0.10	0.01	0.03	RS		
Plastic Casing	2 (L and R)	7500 (per half)	0.75	1.50	Plastic Cost Estimator for Injection moulding[9]. Price per unit refers to the total cost for bulk	Injection Moulded	Moderate complexity
Gearbox Casing	1	5000	0.50	0.50			Moderate complexity
Motor Cap	1	3000	0.30	0.30			Simple complexity
Plastic spur gears - planets	6	1.50	0.15	0.9	HPC and ACCU		Delrin and Tufnol - simple complexity
Sun gears and carriers	3	8.50	0.85	2.55	HPC		
Metal spur gears planets	3	9.51	0.95	2.85	HPC		
Ring Gears	1	10.0	1.00	1.00	HPC		
EN1A shaft (output)	1	8	0.8	0.8	In-house estimate	CNC machining	
Subtotal (£)	14.42					Per tool	
Overhead costs (£)	7.21					50% of cost price	
Retail Price (£)	45.00					Per tool	
V.A.T. (£)	9.00				GOV.UK	20% of retail price	
Profit per unit (£)	14.37						

A retail price was then set conservatively based on the analysis to breakeven and obtain a profit in not more than 3 years.

Profit per annum (One manufacturing run - 10 000 units) = £14.37 × 10 000 = £143 700

CONLCUSION

By conducting a detailed and comprehensive design and engineering analysis, the automatic cordless Fish Scaler has been successfully developed. Through an elegant and simple shaft-planetary system solution, the tool has been engineered to accomplish the required and desired Product Design Specifications (PDS) which include key features such as compactness and portability (total length of 250mm), lightweight (less than 2 kg), ergonomic user-friendly interface through well-designed switches and handles, variable speed, and safety factors above 1.5 of critical components. Other features include efficient charging capabilities and enhanced battery life. Beyond all these features, the primary purpose to design a simple, safe, and reliable device which would enable anybody including the aging and impaired to do this rigorously physical task was accomplished.

The detailed market analysis carried out showed a need for such a product which would attract customers from the target market (aging population and physically impaired) and beyond due to the product's versatility. Furthermore, it can be used at home, restaurants, and has the potential and capability to be used in industrial setting. Prototyping and market tests can be executed for a more comprehensive analysis before mass production. Predicted uncertainties are monitored and mitigation plans are put in place to ensure a successful product launch.

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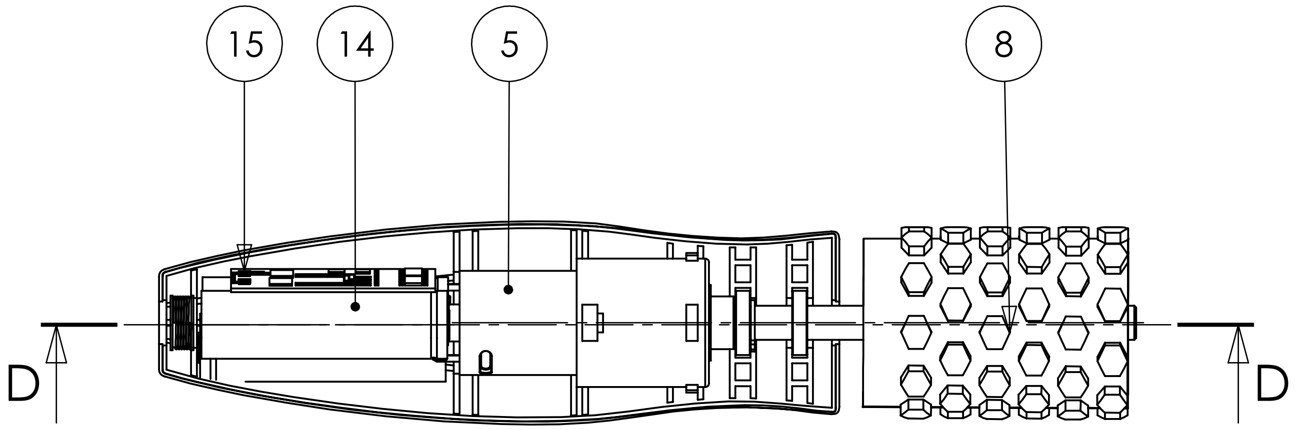
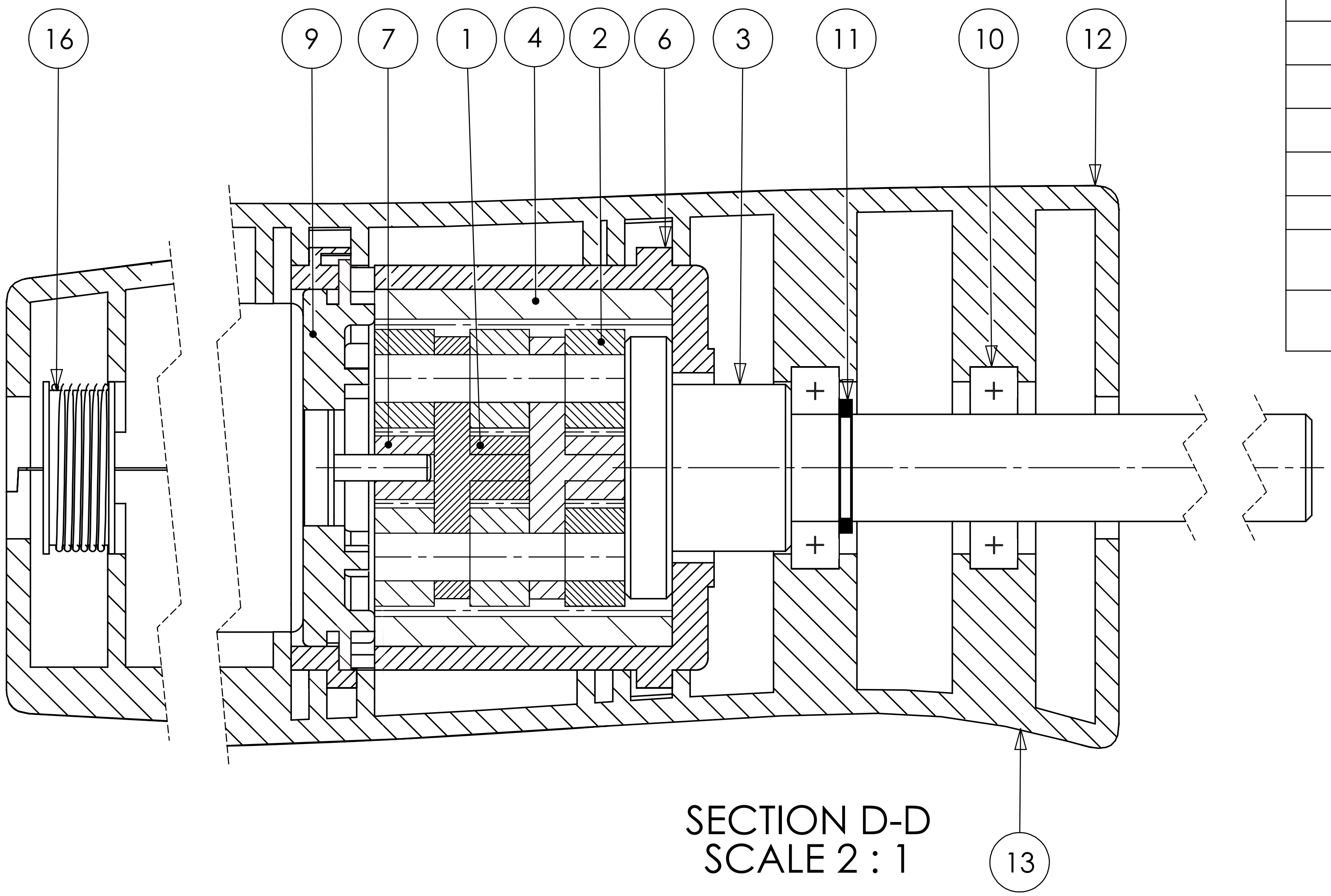
ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	AFS001	SUN GEAR AND CARRIER	2
2	AFS002	PLANET GEAR	9
3	AFS003	OUTPUT SHAFT (3RD STAGE CARRIER)	1
4	AFS004	RING GEAR	1
5	AFS005	MOTOR	1
6	AFS006	GEARBOX CASING	1
7	AFS007	1ST STAGE SUN GEAR	1
8	AFS008	SCALING TOOL	1
9	AFS009	MOTOR CAP	1
10	AFS010	SKF BALL BEARING SKF - 618-9 - 14,SI,NC,14_68	2
11	AFS011	9mm EXTERNAL CIRCLIP BS 3673-4 - S009M	1
12	AFS012	TOP HALF OF CASING	1
13	AFS013	BOTTOM HALF OF CASING	1
14	AFS014	18650 3.6V BATTERY	5
15	AFS015	ECU BREADBOARD	1
16	AFS016	INDUCTION CHARGING COIL	1
17	AFS017	ROCKER SWITCH	1
18	AFS018	SLIDER SWITCH	1





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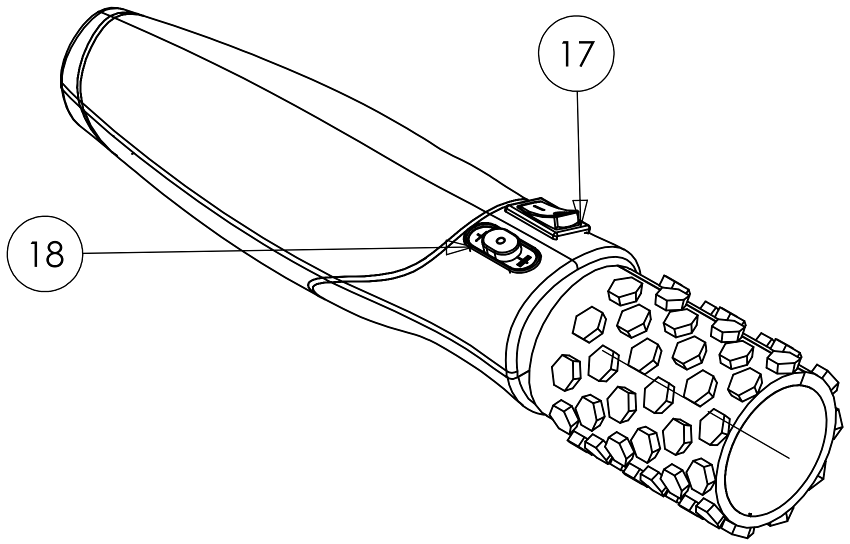
SCALE 2 : 1



TOLERANCES
X =± 0.5
X.X =± 0.1
X.XX =± 0.02
ANGULAR ±1°
SURFACE FINISH MACHINED FACES Ra 6.3
THIRD ANGLE PROJECTION

	NAME	DATE
DRAWN	BRYAN LIM	12.03.18
CHECKED		
APPROVED		

MATERIAL: N/A
ALL DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
A3 SCALE 1:2

TITLE: FISH SCALER ASSEMBLY
DWG No. ASM-001

Imperial College London
Department of Mechanical Engineering

SHEET 1 OF 1 REVISION 1