POWERED KEY TURNER

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EXECUTIVE SUMMARY

The goal of this project is to design an innovative, aesthetic, and useful hand-tool that aids consumers with impaired mobility, strength, and manual dexterity. The client, who is a leading manufacturer for battery operated products in the DIY market, is looking to enter the market for products made for the aging population. A key turner is a device that helps patients, with impaired strength and manual dexterity, to unlock common household and vehicle doors.

The product is expected to be sold in quantities of around 10000 per annum. To recover the overall costs of design, manufacture, distribution, financing, and warranty, it's essential to sell at a rate that will meet these costs in five years. The retail price would need to be competitive, therefore needs to be less than £100.

Three unique product ideas were explored and evaluated as hand tools. From this, the key turner concept was selected. After this, through a morphological analysis, critical features were specified and selected. Once the final design was chosen, engineering analysis was performed on the various dynamically loaded components to ensure that the design was within the limit of failure and safe to use.

Finally, after estimating the total cost of the product, labour and parts included in the costing analysis, a selling price of £18.05 was decided upon.

INTRODUCTION AND MARKET ANALYSIS

A key turner is a very useful device for people with musculoskeletal conditions like Arthritis, Rheumatoid Arthritis, Osteoarthritis, Gout, etc. According to arthritisresearchuk.org, the pain and functional limitation of arthritis makes it difficult for consumers to perform simple everyday tasks such as grasping small objects, twisting, and turning. Therefore, our product will aim to help solve some of the problems.

To assess the economic viability of a keyturner, it is important to consider the projected potential of the product and the competition in its current market. Whilst most current products, such as the Apex Medical 'Enablers Easy Key Turner' (figure 1) and the Queralto UK Key Turner (figure 2), do provide a more stable gripping surface or a large handle to reduce the required force application, none of these products are powered. Therefore there would be a great demand for a product that motorises the key turning process entirely.

Figure 1. Apex Medical 'Enablers Easy Key Turner' ($14 US Dollars)

Figure 2. Queralto 'KeyTurner' (£10 GBP)

Figure 3. 'Easy Grip Turner' (£6 GBP)

Improvements in design and price over years

Alternative designs, such as the 'Easy Grip Key Turner' (figure 3) looked at curved nylon handles at the back providing an easy, secure grip on the key for the user and good leverage for turning. The chunky handle is designed to aid those with a weak grip or those who suffer from Arthritis or tender joints. This product only costs 6 pounds today. These designs are much better than the Apex as they reduce the involvement of the consumer in key turning. However, a powered key turner incorporating current design trends, would be significantly more useful and popular in the modern market. Although the retail price will be higher, the design and utility benefits for consumers will outweigh this cost.

Product Design Specification

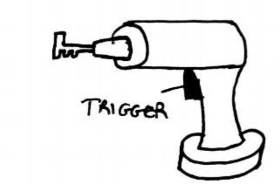
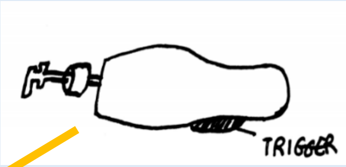
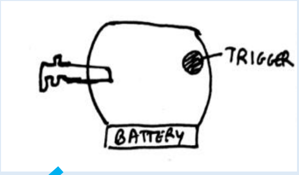
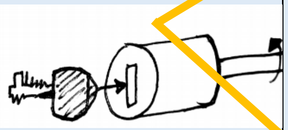
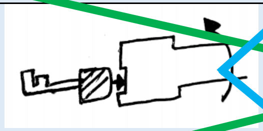
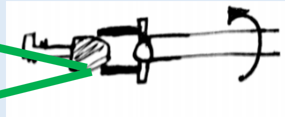
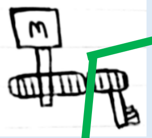
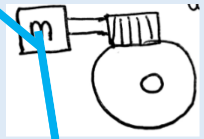
Once a Keyturner product idea was decided, a PDS was drafted up to specify the functional and aesthetic requirements of the product such that it addresses the needs of the client. The PDS has been broken down into 4 main categories: dimensions, performance, durability and marketability.

Table 1 – KeyTurner's Product Design Specification

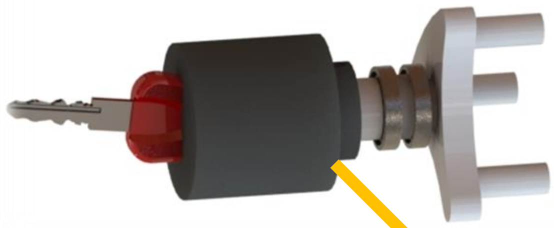
Aspects		Objective	Criteria	Testing
Dimensions	**Total Height**	Up to 250 mm	• Should be able to fit into a hand bag	Ruler Measurement
	Weight	0.5-1kg	• Lightweight, easy to carry	Weighing Scale Measurement
	Handle Diameter	80-100mm	• Broad, comfy grip for stability	Ruler Measurement
	Key Slot Size	20-30mm x 5mm	• Must be able to fit the range of most household keys	Ruler Measurement
Performance	**Angular Velocity**	60 revolutions per minute (3.14 rad/s)	• Should be able to open a lock (180 degrees) in half a second	Stopwatch/Torque measurement
	Output Torque	2-3 Nm	• Enough torque to unlock most common household door locks without snapping.	Calculation
	Transmission	Spatially Compact, as small as possible.	• Should be a compact transmission mechanism for ease of design process and to minimize the size of the tool.	Calculated Design based on gear ratio.
	Trigger Mechanism	Easy to squeeze, lock/unlock setting.	• Light squeeze of the padding to initiate motor • Two switches for free rotation in either direction (to lock and unlock)	Measure/approximate force required to squeeze.
	Battery Life	At least 20 lock rotations	• Should last for 3-4 days before charging.	Calculation
	Charging Time	Up to 5 hours	• Should charge overnight	Calculation
Durability/Usability	**Product Lifespan**	5 years	• Should last the length of manufacturers' warranty	Calculation
	Ease of changing keys	Replace old key with a new key in 10-15 seconds	• Replacing keys should be a simple process, using frictional rubber paddings to keep keys in place.	Qualitative Test
	Mass producibility	10000 products every year	• Features should be simple such that standard workshop tools can be used to mass produce the product.	Design Decision
Marketability	**Materials**	Strong, lightweight, shatter resistant material choices for different parts.	• Steel is a strong material, lowering the chances of failure due to fatigue. • Plastic is a lightweight material, that can be aesthetically pleasing. • Rubber for increasing friction (key placement) and comfortable gripping (handle)	Calculation and Design Choice
	Noise level	<40 dB from motor + transmission	• Noise levels should be minimised such that noise doesn't hinder usage of the device.	
	Market Competitiveness	Aesthetic appeal, price, and key features	• Specific design choices (trigger and material) for patients with arthritis • Stands out on market shelf due to aesthetic design	Design Decision
	Safety	Low risk	• Rigid casing and firm electrical wiring	Safety factor calculations Maximum Load Test

Morphological Analysis and Final Concepts

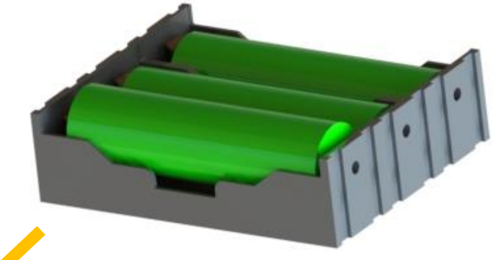
The product design was split into 5 functions: shape, functional head, transmission, switch and charging. For each function, three unique solutions were explored and evaluated. Different paths amongst these functions led to three final concepts as shown below:

Function	1	2	3	GREEN CONCEPT	BLUE CONCEPT	ORANGE CONCEPT
SHAPE	 GUN SHAPED	 STREAMLINED SMOOTH SHAPE	 BALL SHAPE, DETACHABLE BATTERY	+ Unique aesthetics + Easy trigger placement -Lacks sensible motor position - Difficult to carry around	+Compact shape +Easy way to attach func head - May require small transmission - May be tough to hold	+Compact shape +Unique aesthetics +Easy to control -May need to be long in order to contain all parts
FUNCTIONAL HEAD	 ROTATING CYLINDER	 SHAFT STEP	 EXTERNAL GRIPS	+Easy to axially constrain -Not very aesthetic -Not very flexible	+Easy manufacture +No axial constraint -Weakens shaft	+Comfortable shape +Easy to axially constrain -Not very aesthetic
TRANSMISSION	 SPUR GEARS	 PLANETARY GEARS	 WORM AND WHEEL	+Very simple mechanism +Low cost -Different shaft axes required -Takes up space	+Large ratios in a small number of steps +Silent, smooth -Expensive -Low efficiency	+Capable of high reductions +Compact, quiet -Can be expensive
SWITCH	 TRIGGER	 BUTTON	 FLIP SWITCH	+Capable of variable speed/direction -Hard to be precise with the speed	+Clear on and off settings +Easy to press -Not capable of variable settings	+Clear on and off settings -No variable settings -Difficult for users with low manual dexterity
CHARGING	 PLUG IN	 REMOVEABLE BATTERIES	 BATTERY PACK	+Batteries are easily replaced +Can be charged -Would need many batteries	+Easily detachable +High capacity +Batteries away from water	+No need to remove batteries +Charged anywhere -Unsafe during wet conditions

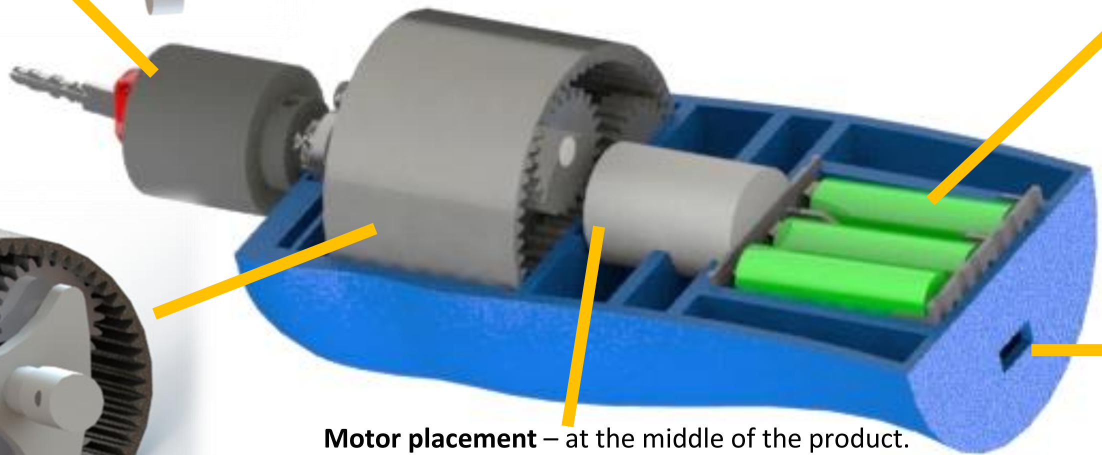
Final Concept and Motor Choice



A **rubber rotating cylinder** was selected as the **functional head.** This is axially constrained by a pin going through the hub of the cylinder and the main shaft. Rubber is chosen as the material as it provides a high-friction surface for the key to squeeze into a slot on the rotating cylinder and prevents it from falling out.



A plug-in **charging** battery was decided as the most appropriate. Calculations suggested the motor required a 24 nominal V and therefore 6 batteries were needed. It would be a pain to keep replacing these, charging would make the product much easier to use



A **charging port** for plug-in charging

A double reduction **planetary gear** train was selected for **transmission**. Not only would this achieve a high ratio in a small number of steps, but also would be much more compact than an equivalent spur gear train.

Motor placement – at the middle of the product. Directly connects into the planetary gearbox allowing for direct drive. Given the required output, and looking at the maximum power point, motor RS-545SH-18150 was selected.

Motor Specification

	Nominal Voltage	V	14.4
No Load	Speed	Rad/s	555.015
	Current	A	0.1
Max Power	Speed	Rad/s	277.507
	Torque	mNm	44.15
	Output	W	12.252
Stall	Torque	mNm	88.3
	Current	A	2.32

Motor Selection

Aspect	Formula	Units	Value
Desired Torque	Measured Force x Distance = Torque	Nm	2
Desired Output Angular velocity	Approximately 60 revolutions per minute	Rad/s	6.28
Power required	Desired Torque x Output Angular Velocity	W	12.56
Max Power Best Motor	Closest value (half) to the power required	W	12.252
MOTOR CHOICE = RS-545SH-18150			

ENGINEERING ANALYSIS

TRANSMISSION

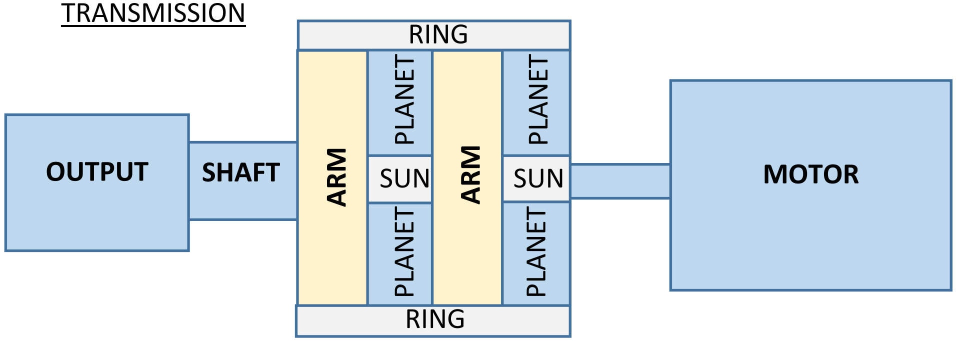


Figure 5 Model of the 2 stage planetary transmission gear set

Table 4 – Transmission Calculations and Specification

Aspect	Formula	Units	Value
Motor Output Speed	Motor Spec (1/2 of no load speed)	Rad/s	277.507
Desired Output Speed	PDS – Calculation of required speed	Rad/s	6.28
Reduction Ratio	Reduction = $\frac{Motor\ Output}{Desired\ Output}$	-	44.2
Motor Output Torque	Motor Spec (1/2 of stall torque)	mNm	44.1
Desired Output Torque	PDS - Force x Distance = Torque	mNm	2000
Ratio	Ratio = $\frac{Desired\ Output}{Motor\ Output}$		45.3
2 stage – equal reduction	$\sqrt{Reduction\ Ratio} = \sqrt{44}$	-	6.6
2 EQUAL STAGE REDUCTIONS – PLANETARY SETS			
Sun Gear 1 speed [ns1]	Motor Output	Rad/s	277.5
Arm 1 speed [nl1]	$nl1 = \left\lvert\frac{ns1}{stage\ reduction}\right\rvert$	Rad/s	42.0
Planet Set 1 speed [np1]	$np1 = \frac{NS}{NP}(nl1 - ns1) + nl1$	Rad/s	60.32
Sun Gear 2 speed [ns2]	$ns2 = nl1$	Rad/s	42.0
Arm 2 speed [nl2]	$nl2 = \left\lvert\frac{ns2}{stage\ reduction}\right\rvert$	Rad/s	**6.28**
Planet Set 2 speed [np2]	$np2 = \frac{NS}{NP}(nl2 - ns2) + nl2$	Rad/s	9.25

A two stage **planetary gearbox** was chosen in order to reduce the number of parts required, reducing the component cost. The motor output speed is reduced from 277.5 rad/s to 6.28 rad/s. The motor torque is increased from 0.0442 Nm to 2 Nm.

SHAFT YIELD FAILURE

The main shaft is one of the most important **dynamically loaded components**. It's essential to ensure that shaft failure due to yield is not a possibility. The rotating functional head provides a force at one end, the gear set provides a torque at the other and in between, 2 bearings support the shaft.

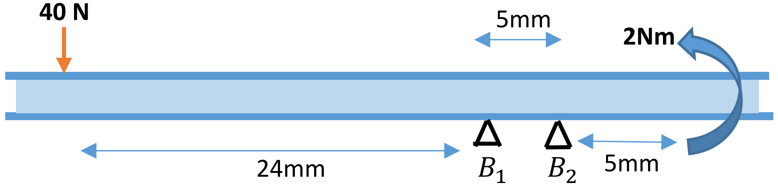
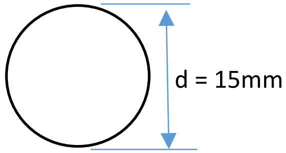


Figure 6 Free Body Diagram of the shaft and bearing support



$$I = \frac{\pi d^4}{64} = \frac{\pi}{64}\ x\ 0.015^4 = 2.48 \text{ x } 10^{-9}\ m^4 \quad (1)$$

$$J = 2I = 5 \text{ x } 10^{-9}\ m^4 \quad (2)$$

Max Torque = 2 Nm

The maximum torque output is calculated during the design process. The selected motor provides a torque of 0.0442 Nm which is increased to 1.92 Nm by the planetary gear ratio.

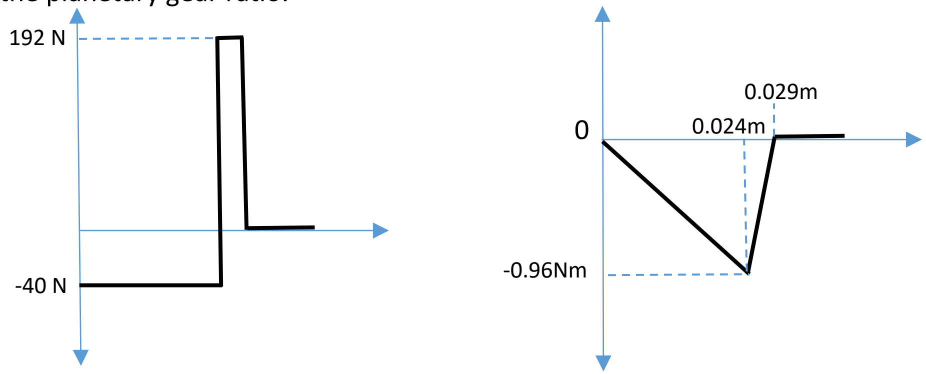


Figure 7 Shear Force and Bending Moment Diagrams of Shaft

We can use the value of maximum bending moment to calculate the axial stress (due to bending only as no horizontal axial forces) acting on the shaft:

$$\sigma_x = \frac{My}{I} = \frac{0.96*0.015}{2.5*10^{-9}} = 5.76\ MPa \qquad (3)$$

$$\tau_{xy} = \frac{Tr}{J} = \frac{2*0.0075}{5*10^{-9}} = 3.0\ MPa \qquad (4)$$

$$\sigma_y = 0\ MPa\ (no\ stress\ in\ direction)$$

Therefore we can find a Tresca value of principal stress (more conservative than Von Mises) by using equation 6 below:

$$\sigma_{1,2} = \frac{1}{2}(\sigma_x + \sigma_y) \pm \sqrt{(\sigma_x - \sigma_y)^2 + \tau_{xy}{}^2} \qquad (5)$$

$$\sigma_1 = 7.04\ MPa \qquad \sigma_2 = \ -1.27\ MPa$$

Tresca yield criterion Y = $|\sigma_1 - \sigma_2| = 8.31\ MPa$ (6)

Setting a desired safety factor of 2 gives a Tresca yield stress of $\sigma_y = 16.62\ MPa$. Although this yield stress is far lower than the yield stress of a steel shaft being used in manufacture, using metal shafts in a plastic housing improves performanc and negates the use of bearings in our design. Moreover, by using lubricant on th shaft parts, friction can be minimised. Therefore, failure through yield is extremel unlikely and as such, cheap metal can be used to manufacture the shaft, reducing the price of the product.

Table 5 – Yield Stress of Aluminium and Steel

Yield Stress	Method	Units	Value
2014-T6 Aluminium	(ASM Matweb, 2017)	MPa	414
1045 Steel	(Azom, 2017)	MPa	300

As seen, the values of yield stress are significantly larger for both materials. In order to fully select the material, fatigue calculations need to be conducted.

FATIGUE FAILURE

A high cycle fatigue analysis was performed. The life of the shaft (30 hours at top speed) was converted into the number of cycles using the data for speed from table 4.

$$N_f = 30 * 60 * 6.28 = 1.16 \text{ x } 10^7 \text{ cycles} \qquad (7)$$

The stress amplitude was estimated from earlier calculations = 16.6 MPa

Therefore, fatigue analysis can be done by using these values in comparison to the typical S-N curve for both materials.

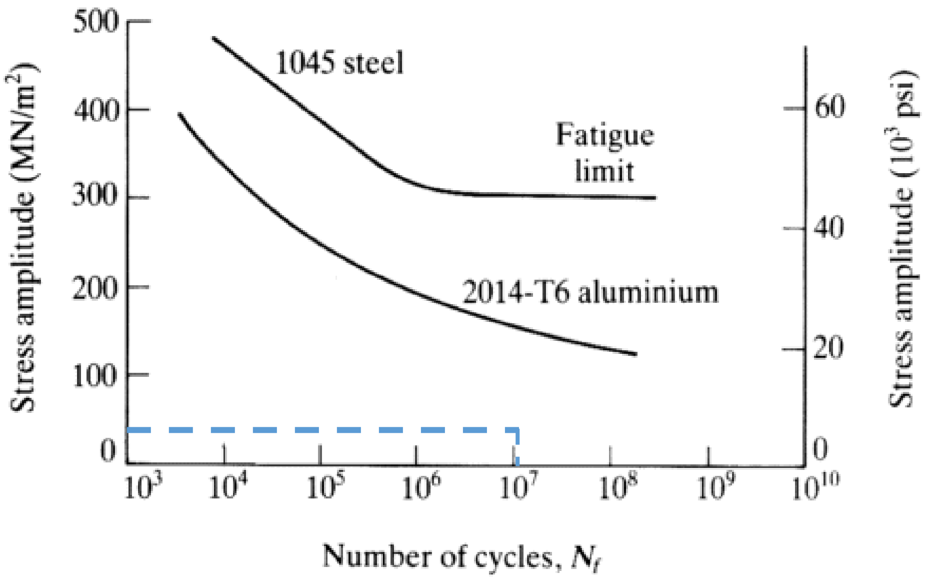


Figure 8 - S-N curve for high-cyle fatigue of Aluminium and Steel (Efunda, 2017)

Figure 8 displays that our shaft falls well within the failure limit of the curve, it is therefore unlikely to fail due to fatigue for both aluminium and steel. Thus, the final decision was based on mass and costing. Although steel has a greater density, overall, it's cost is significantly lower than aluminium. So with a slightly higher density but significantly lower cost than aluminium, steel was chosen.

GEAR STRESSES – DYNAMICALLY LOADED COMPONENT

The planetary gears in the transmission set are also important **dynamically loaded** components. Therefore, it is essential to ensure that the gears can withstand the major stress components.

Lewis Form Factor

The Lewis Form Factor is used to provide a minimum face width for the gears in a transmission. This was critical to deciding the module and material of the gears initially. In this 2-stage planetary transmission, each stage had an equal ratio and therefore approximately the same face widths. Below is the process to estimate the face width of 1 of the sets.

Given: Module m = 0.0015 metres

Maximum bending stress = 120 MPa

Motor Power output = 12.252 W

Table 6 – Lewis Factor Calculations

Symbol/Variable	Method	Units	Sun 2	Planet 2
Teeth, N	Solving simultaneous planetary gear equations	-	10	23
Pitch line diameter, PCD	PCD = mN	m	0.015	0.0345
Pitch line velocity, V	V = $\frac{PCD*\pi*n}{60}$	ms^{-1}	0.315	0.159
Transmitted Load, Wt	Wt = $\frac{P}{V}$	N	38.89	76.78
Velocity Factor, Kv	$K_v = \frac{6.1}{6.1+V}$	m^{-1}s	0.95	0.97
Lewis Factor, Y	Childs, 2014	-	0.201	0.333
Face Width, F	$F = \frac{Wt * SF}{K_v m Y \sigma_p}$	mm	**2.26**	**3.28**

AGMA Gear Stresses

The AGMA equations allow for bending and contact stresses to be calculated with greater accuracy. So having picked a value for face width using the Lewis factor, the failure can be evaluated through safety factors again. This is the AGMA process. The values were found as follows:

Table 7 – AGMA Calculations

Symbol/Variable	Method	Units	Sun 2	Planet 2
Bending Stress	$\sigma b = \frac{WtK_0K'_vK_sK_HK_B}{FmY}$	MPa	18.8	28.3
Contact Stress	$\sigma C = Z_e(\frac{WtK_0K'_vK_sK_mC_f}{FdZ})^{0.5}$	MPa	298.3	318.5
Permissible Bending Stress	$\sigma fp = 0.533Hb + 88.3$	MPa	216.2	194.9
Permissible Contact Stress	$\sigma hp = 2.22Hb + 200$	MPa	733	644
Allowable Bending Stress	$\sigma ba = \frac{\sigma fpY_N}{Y_\theta Y_z}$	MPa	241.7	223
Allowable Contact Stress	$\sigma ca = \frac{\sigma hpz_N z_w}{Y_\theta Y_z}$	MPa	785	727
Bending Safety Factor	$SFB = \frac{\sigma ba}{\sigma b}$	-	**12.84**	**7.88**
Contact Safety Factor	$SFC = \frac{\sigma ca}{\sigma c}$	-	**2.63**	**2.28**

Some of these constants were estimated from Childs 2014, the list of constants can be found in Appendix 1. The calculation process is referred to in Chapter 9, Childs, 2014.

BATTERY CHOICE AND PLACEMENT

The three major constraints to consider when selecting batteries are space, nominal voltage and lifetime. Given the required lifetime of 5 years, with an average of 10 door unlocks per day (3650 unlocks per year):

$$Activity\ time, Ta\ = \frac{time\ for\ one\ unlock\ x\ no\ of\ unlocks\ in\ 5\ years}{3600} = \frac{2\ x\ 3650\ x\ 5}{3600} = 10\ hours$$

The motor current (for maximum power) required is 2.32 A, which amounts to a total charge of:

$$Total\ charge, Q_T\ = total\ lifetime\ x\ operating\ current = 10\ x\ 2320 = 23200\ mAh$$

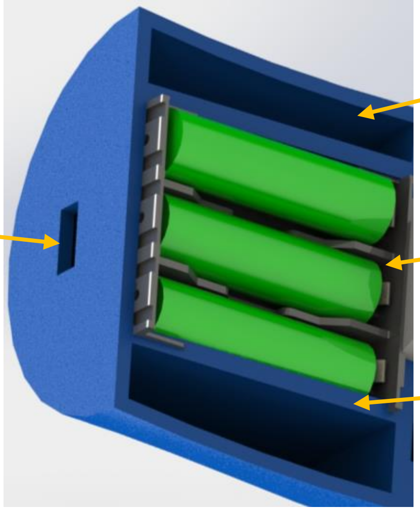
Given that the product should only be charged after 4 days (40 unlocks), this leads to a total number of 456 charging cycles across its lifespan.

$$Required\ lifetime = \frac{total\ charge}{total\ no\ of\ charges} = \frac{23200}{456} = 50.88\ mAh$$

Therefore given nominal voltage and capacity, we can decide the best battery choice. From the above calculations, I decided to choose 6 batteries of type 18650. Although these are slightly large in size, they meet the requirement and are the most powerful batteries. Since there was enough space at the back for 6 batteries, I designed a compartment to fit their size, 3 at the bottom half of the casing and 3 at the top. At the bottom there is also a charging port to charge these batteries regularly.

Table 8 – Battery Specifications

Battery Type	18650
Length (mm)	65.0
Diameter (mm)	18.0
Voltage (V)	3.6
Capacity (mAh)	2000



BUDGET AND COST ANALYSIS

Given there are 100000 units of every part order, the total costs of the project will have to be such that each item can be sold at a certain profit, with the price still being competitive with other turners in the market. As such, a break down of all parts and the individual cost for each part is required. The cost of each component will approximately be 10% of the single unit cost.

The cost for each part is determined by firstly checking the cost range of the component on a manufacturer's website and taking the bottom value of that range (since a large number of parts are to be ordered). Next, we make reasonable estimates to those parts that aren't found on a manufacturers website. We can assume labour, for example, comes at £7 an hour, and on an assembly line the time taken between successive operations is 30 seconds. Therefore, approximating around 20 operations to manufacture the product from start to finish. So after an initial phase of (30 seconds x 20 operations) 10 minutes, one product is made every 30 seconds. Therefore, by multiplying the number of hours a day and the number of days a week, we can estimate the amount of labour altogether. My calculations suggest a cost of £0.12 per product is due to labour. In addition, an estimate cost for tooling is roughly £0.4 per product.

Part No.	Part Name	Manufacture	Price (£)	Quantity	Total Price
1	Body	Injection mould	0.15	2	0.3
2	Motor	Purchased	0.15	1	0.15
3	10T M1.5 F12mm Sun Gear	Purchased	0.4	2	0.8
4	23 T M1.5 F12mm Planet Gear x 6	Purchased	0.8	6	4.8
5	56 T M1.5 F 60mm Internal Spur Gear (Ring Gear)	Purchased	3.6	1	3.6
6	Button Switch	Purchased	0.13	1	0.13
7	Deep Groove Ball Bearing	Purchased/Lathed and milled	0.46	2	0.92
8	Pins	Purchased	0.01	2	0.02
9	Spacers	Workshop machining	0.12	7	0.94
10	Keys	Purchased	0.24	8	1.92
11	Gear Arms/Shafts	Lathed and milled	0.47	2	0.94
12	Labour	8 hrs/day. 5 days/week	0.12	1	0.12
13	Tooling	Purchased	0.4	1	0.4
					TOTAL COST FOR 1 PRODUCT: £**15.04**

As such, adding up all the part costs gives us a total of £15.04 per product. Given a 20% profit margin on each product, we can evaluate the retail price to be approximately £18.05. This is more expensive than current products, but is the only powered key turner in the market.

CONCLUSION

Overall, the powered key turner is a product with a huge potential in the current market for technologies that aid people with impaired mobility, manual dexterity and strength. Our product meets all the requirements of the client – dimensions, performance, durability, safety, marketability and maintenance. A thorough engineering analysis suggests that the product is well within the limits of fatigue or yield failure, and that the gears are appropriately greater than the minimum face width. Certain improvements I would focus on would be to reduce the transmission size through smaller stages of reduction. A 200 mm device is a bit too large to carry around comfortably, however is still manageable. Moreover, alternative electronics such as cheaper batteries could be explored to reduce costs.

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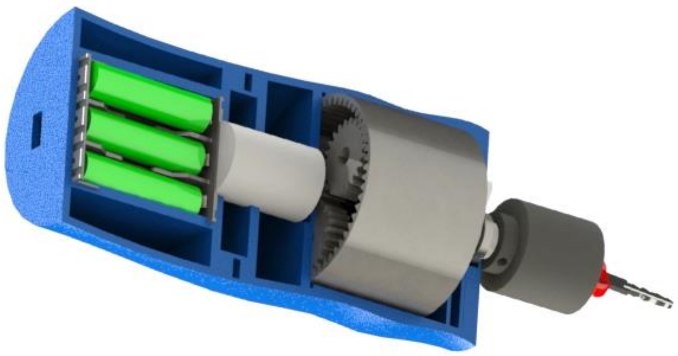
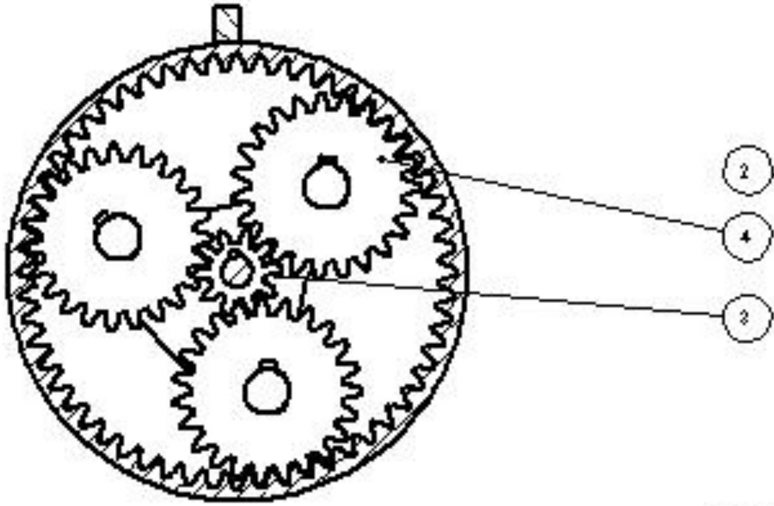
Appendix

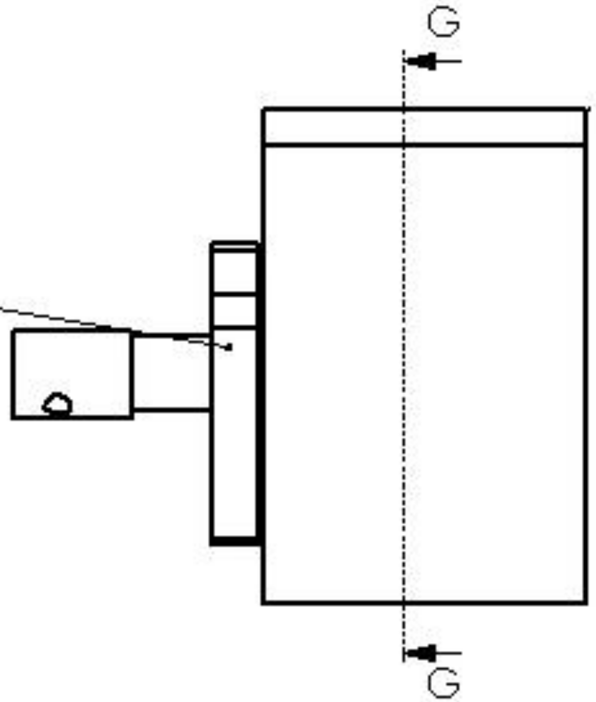
Table 10 – AGMA Factors and Values

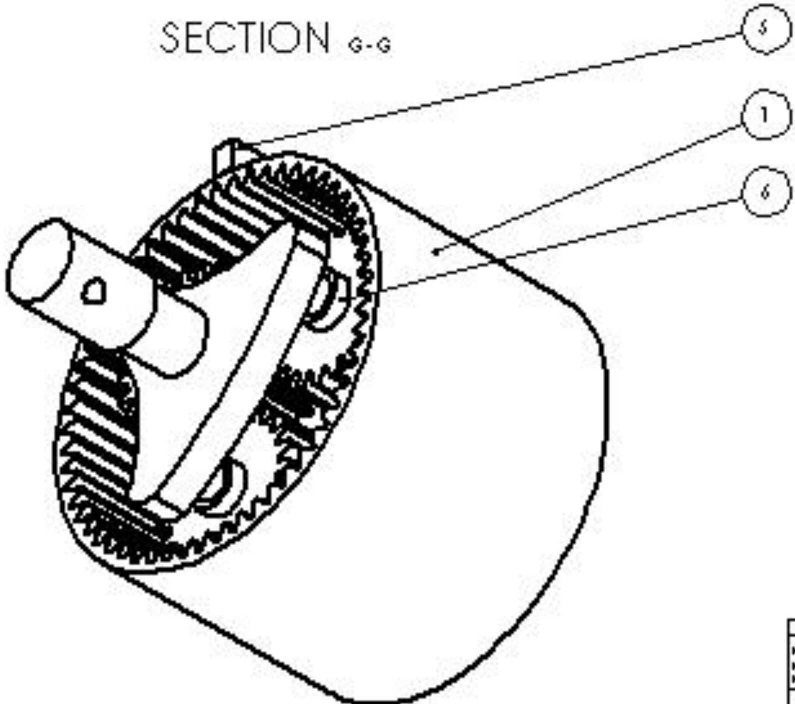
Aspect	Units	Sun 2	Planet 2
Overload Factor K0	-	1.50	1.50
Dynamic Factor K'v	-	1.112	1.141
Size Factor Ks	-	1.00	1.00
Load Distribution Factor Kh	-	1.12	1.10
Surface Condition Factor Zr	-	1.00	1.00
Geometry Factor Yj	-	0.25	0.29
Elastic Coefficient Ze	$(MPa)^{0.5}$	191	191
Rim Thickness Factor Kb	-	1.00	1.00
Geometry Factor Zi	-	0.12	0.12
Pressure Angle φ	Rads	0.35	0.35
Brinell Hardness Hb	HB	240	200
Bending Life Factor Yn	-	0.95	0.99
Temperature Factor Yθ	-	1.00	1.00
Pitting Life Factor Zn	-	0.90	0.96
Hardness Ratio Factor Zw	-	1.00	1.00





SECTION G-G





ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	G001	RING GEAR [1.5 M 56T]	1
2	P001	plate2	2
3	G002	SUN GEARS [1.5M 10T]	2
4	G003	PLANET GEARS [1.5M 23T]	6
5	R001	RADIAL CONSTRAINT RING GEAR	1
6	S001	SPACER	6

DRAWN
CHECKED
APPROVED

DO NOT SCALE DRAWING

A3 SCALE 1:1

TRANSMISSION

DWG NO. T001

Imperial College London
Department of Mechanical Engineering