Automatic Continuously Variable Transmission

For a commuter bicycle

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Executive Summary

Group 2 DMT has successfully designed, made and tested an Automatic Continuously Variable Transmission (ACVT) for a commuter bike. This has been achieved through 2 rounds of prototyping beginning with a static proof-of-concept prototype and culminating in a full-scale dynamic prototype. The promising final testing results indicate potential for further research and development of the concept. The anticipated market for this product is the growing hire bicycle sector where ease of use and freedom from maintenance are highly desired. This is particularly pertinent given the growing pressures on existing public infrastructure and higher levels of environmental health consciousness. Additionally, the number of novice cyclists with little experience in maintaining traditional bicycle groupsets has dramatically increased.

In the first round of prototype Design, Make and Test the group demonstrated at low torques, using a portable demonstration assembly, that the initial concept of a sprung ACVT could work. In the process, several possible product designs were eliminated.

In the second round of prototype Design, Make and Test the group created a functional model of the transmission mounted on a typical bicycle frame. After a period of testing and de-bugging, the prototype consisted of a tool that can be used to demonstrate the mechanical principle at full size.

Due to the greater than expected impact of a marginally misaligned belt, the prototype exhibits a jolting behaviour that makes the prototype uncomfortable to operate and extinguishes the possibility of collecting quantitative data.

The jolting can be eliminated by designing the prototype to utilise a straight belt line. This can only be achieved with the aid of custom bicycle geometry and so regrettably the possibility of smooth operation within the constraints of the PDS is beyond the resources of this project.

Overall, this project largely fulfils its stated aim; it has been demonstrated that a half-Reeves drive could be a viable alternative to traditional chain drive systems.

With gratitude to Graham Gosling for his feedback and guidance; Richard Van Arkel for his feedback at the design review; all of the workshop technicians, for their invaluable advice and help, particularly Eddie Carter & Neil Beadle for their CNC wizardry.

Contents

1. Introduction

1.1 Project Aims

This report presents the outcomes from the project titled Automatic Continuously Variable Transmission (ACVT) for a commuter bicycle and undertaken by DMT Group 2. The project set out with the following aim [1];

> *"To design make and test a drivetrain for commuter cyclists that is zero maintenance and automatically selects an appropriate gear ratio"*

This report can be broken into five main sections. The introduction details the project background and market need, which allowed the group to set objectives. The introduction also outlines the strategy undertaken by the group when approaching this project and details the PDS. The report then goes on to discuss the initial development, manufacture and testing of the first-generation proof-of-concept static prototype in section 2. This is followed by commentary on the development of the bicycle mounted dynamic prototype in section 3, which made up the bulk of this project. The final section of this report briefly discusses how this concept could be further developed in the context of mass market public hire and commuter bicycles.

1.2 Project Background

There are a range of factors that have led to a rapid rise in the use of the bicycle as a commuter vehicle in recent years. Higher levels of urbanisation, particularly in developing nations such as China and India [2, 3], has hugely increased the strain on existing public infrastructure. This has led to increased congestion on roads and chronic overcrowding on public transport at peak times [4]. Increasing capacity on these systems is expensive and projects can take years to come to fruition [5]. This has led city planners and government organisations around the world to invest heavily in cycling infrastructure [6, 7] in a bid to encourage commuters out of their cars and onto a bicycle. Such developments are often far less costly than rail or roads and can enable the transport of significantly more passengers [8]. However, infrastructure alone is insufficient to encourage commuters onto two wheels. Bicycle owners in cities face the perpetual threat of bike theft [9] which is only exacerbated by the inadequate supply of secure bicycle storage.

Government efforts to encourage more bicycle journeys are also partly driven by environmental reasons. Governments are legally obliged to support international efforts to keep global temperature rise below $2°C$ [10], and large scale change in transport provision goes some way towards achieving this [11]. Recent increased public awareness surrounding air quality in cities has further driven governments to encourage commuters to adopt more sustainable forms of transport [12]

Furthermore, in many developed nations there has been a trend in the general population towards people becoming more health conscious and it is arguable that this trend, only encouraged by government health campaigns, has further increased the numbers of cyclists on our roads [13].

These new cyclists, whether driven to cycle for health benefits, convenience or environmental reasons are often inexperienced in how to correctly maintain and use their bicycle. Most users rarely make use of all their gears on a flat city commute and regular derailleur and chain maintenance is beyond the comfort zones of many.

Such trends have played a huge part in driving one of the most fundamental changes in inner-city transport provision of the last few decades with millions [14] of public hire bicycles springing up in both government and private schemes across the globe. These bicycles offer users increased convenience, enabling them to find a bicycle wherever they are in a city, act to remove the threat of theft and replace continual cost of bicycle maintenance.

It is this market that this bicycle ACVT aims to satisfy by removing the need for the user to know how to select an appropriate gear and removing the need for regular chain and derailleur maintenance by bicycle hire operators. The transmission should be a ‘get on and pedal’ product, leaving the cyclist free to think solely about the road ahead.

1.3 Bicycle Terminology

A number of bicycle specific terms are used throughout this report and these are detailed below. Broadly, a bicycle can be described as consisting of four major assemblies; the frame, the groupset (or drivechain), the finishing kit and the wheel-set.

The frame is the structure from which all other components are hung. Traditionally manufactured from steel tubes brazed to one another at joins or lugs. More recently, moulded carbon fibre and welded aluminium have become more popular on racing and recreational bicycles. Traditionally frame shapes are of double diamond construction as illustrated in figure 1.1. This design has endured since the turn of the 20^{th} century and is still mandated for racing cycles by the UCI[1] today.

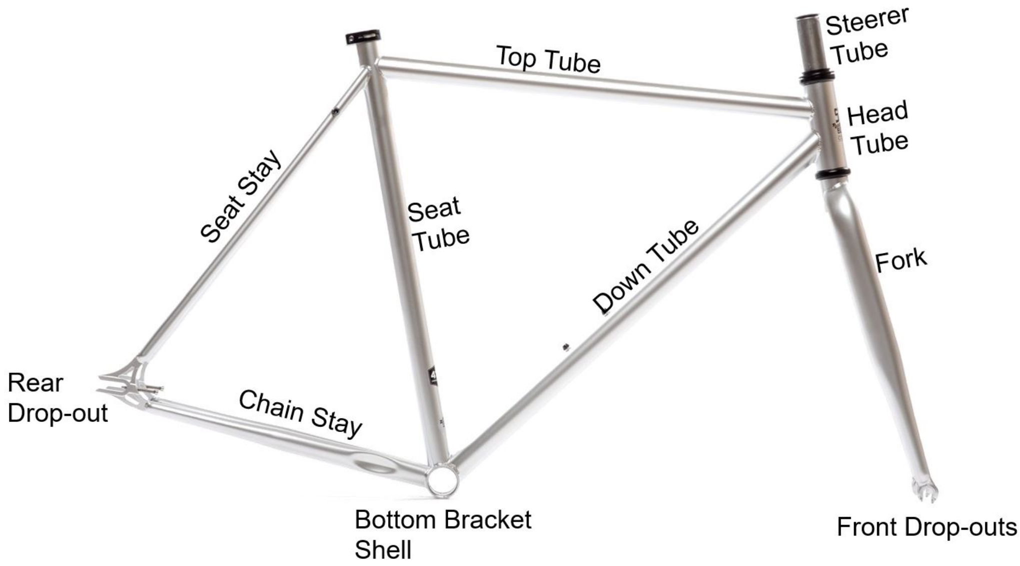

Figure 1.1: Traditional double diamond steel bicycle frame with key parts labelled (image sources given on page 63);

Also of interest is the more upright position used on English ‘roadster’ town bikes, shown in figure 1.2, this frame design still endures today in more modern bicycles such as the MoBike and Santander cycles which have recently begun appearing in cities across the globe. The dropped top-tube allows for mounting of the cycle to be far easier and many find the upright position to be far more comfortable.

The rear ‘drop-out’, the point where the rear wheel is attached to the frame. Of particular interest in this project, it is typically spaced at 130 mm in modern bikes, although this is not universally standardised.

A further definition of note is that of the ’Q-factor’ which is the horizontal distance between the pedal attachment points measured parallel with the bottom bracket spindle.

The finishing kit consists of the saddle, handlebars and associated supports. Its arrangement is not of particular importance to this project.

The groupset or drivetrain is the assembly which has the purpose of transferring the power

[1] Union Cycliste Internationale;

Figure 1.2: Traditional roadster steel bicycle with a dropped top tube frame shape;

generated by the cyclist to the wheelset, propelling them forward. This assembly is almost always based around the use of a chain drive. The forward set of sprockets, usually between 1 and 3 in number, are known as the chainrings. These are driven by the user pedalling on the cranks. Attached to the rear wheel the second, driven, set of sprockets (up to 12 on the most modern of groupsets) are known as the cassette. The chain which runs between these sets of sprockets is moved from one sprocket to the next by derailleurs which are controlled by the cyclist. A labelled image of these components is given in figure 1.3. The mech hanger is the piece of material which acts as the interface between the rear derailleur and the frame.

The wheel-set consists of two wheels, one at the front and one at the rear of the bike. These typically consist of a rim, on which the tyre is seated, laced to a hub by up to 36 spokes. The hub contains the bearing arrangement which allows the wheel to revolve on the axel and, at the rear, acts as the point onto which the cassette is attached. The cassette is mounted to a body known as the freehub by a series of splines, which makes up part of the rear hub. The precise arrangement and length of these splines is not standardised across all manufacturers, but the market leader, Shimano's hyper-glide design is by far the most common. The freehub contains a ratchet mechanism which allows the user to coast, i.e. the bike continue to move forward without the user turning the pedals. This is illustrated in figure 1.4.

For most standard road bicycles a rim diameter of 622 mm is used. This is known as 700C wheel size.

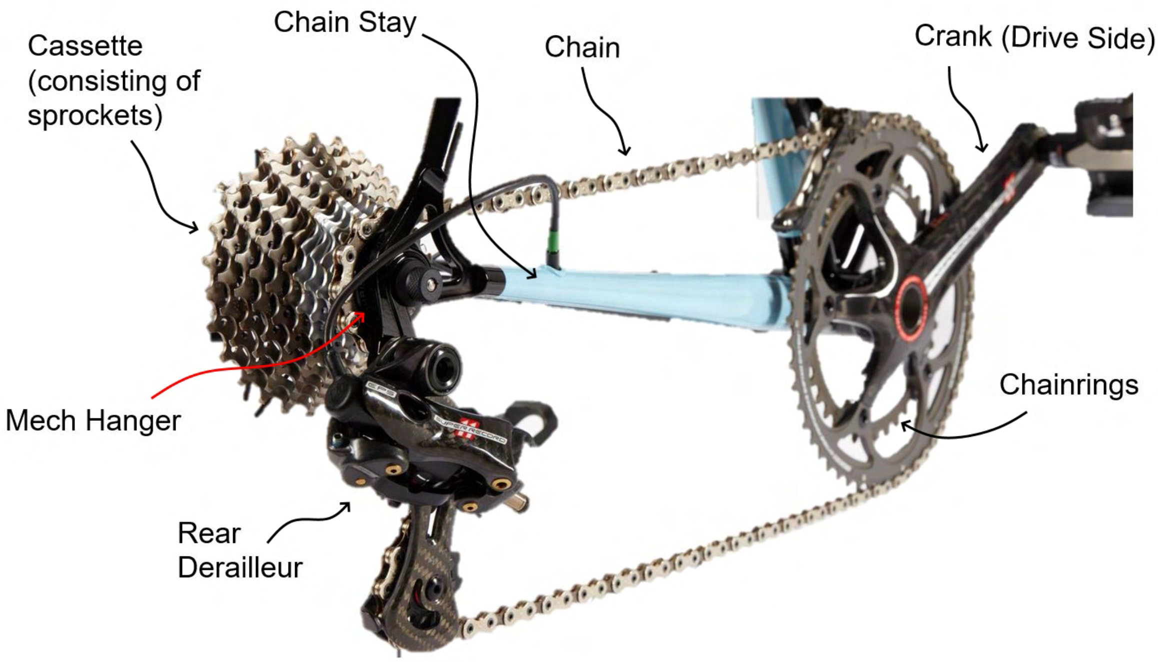

Figure 1.3: Labelled image of a bicycle groupset (image sources given on page 63);

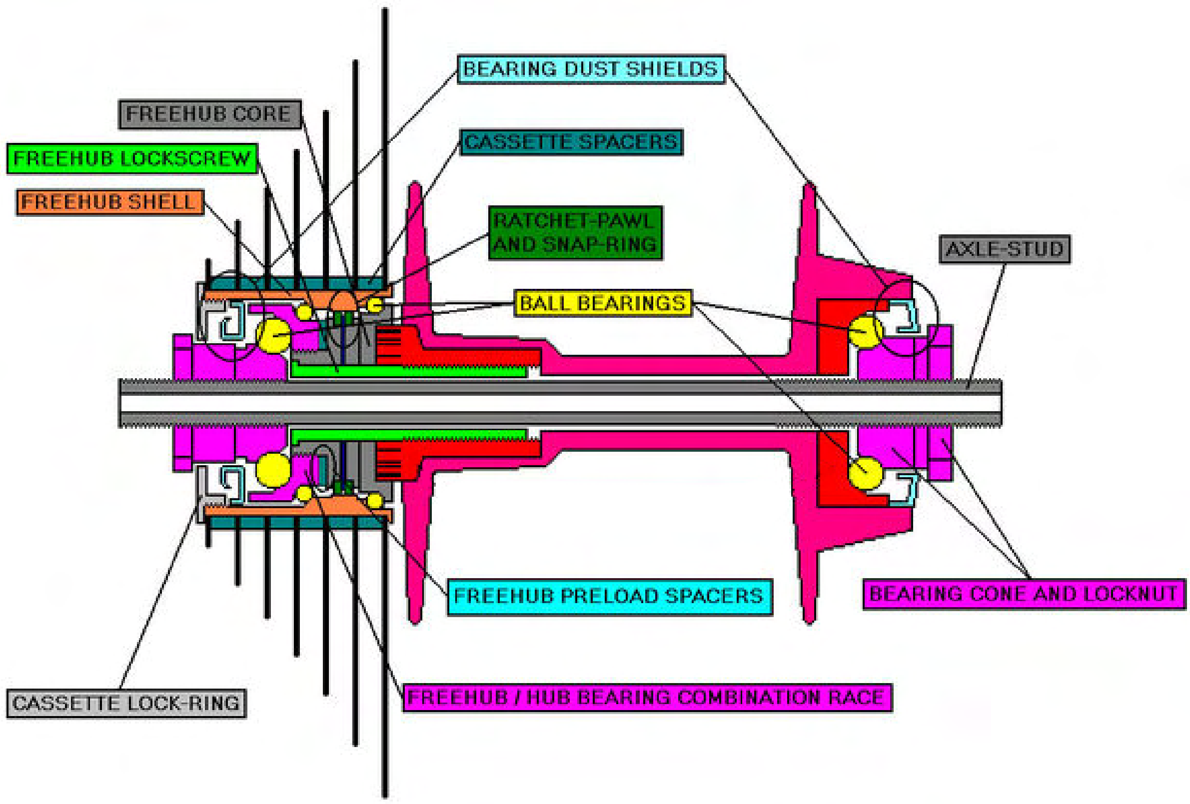

Figure 1.4: Section view of a standard free-hub;

1.4 Project Management

1.4.1 Team Roles

Broadly, responsibility for project tasks among the group members has been broken up as follows: Thomas Fisher has taken on a project management role; allocating tasks, maintaining the Gantt chart and ensuring that project deadlines are met. Charles Titmuss has taken on a secretarial and editorial role; responsible for making and distributing meeting minutes and acting as the editor for all written submissions. Henry Hart has acted as the technical lead on the project, with the ultimate responsibility for ensuring that the design remains coherent and undertaking the bulk of the calculations-based design work. Ieuan Swainston acted as the manufacturing lead; responsible for ensuring that all designed parts were manufacturable and performing the bulk of the more challenging manufacture and assembly. Responsibility for internal (i.e. through internal college systems) purchases fell to Charles Titmuss and external purchases to Thomas Fisher. Despite ultimate responsibility for various tasks being broken up among the group, all members were expected to participate in all tasks ensuring a relatively consistent workload for all members.

1.4.2 Project Supervision

The supervisor of the project was Graham Gosling.

1.4.3 Gantt Chart

The Gantt chart acted as a 'live document' throughout the project. It was regularly updated as tasks fell behind or got ahead of schedule. The Gantt chart in its final form is presented in Appendix B.

1.4.4 Budget

At the project outset the total budget of £1,000 was allocated as follows;

Test Rig	£250
Inital Prototypes	£300
Final Prototypes	£300
Contingency	£150

1.5 Existing Product & Literature Review

Whilst there are no known attempts to produce an ACVT for a bicycle, there have been a number of products aimed at solving similar problems.

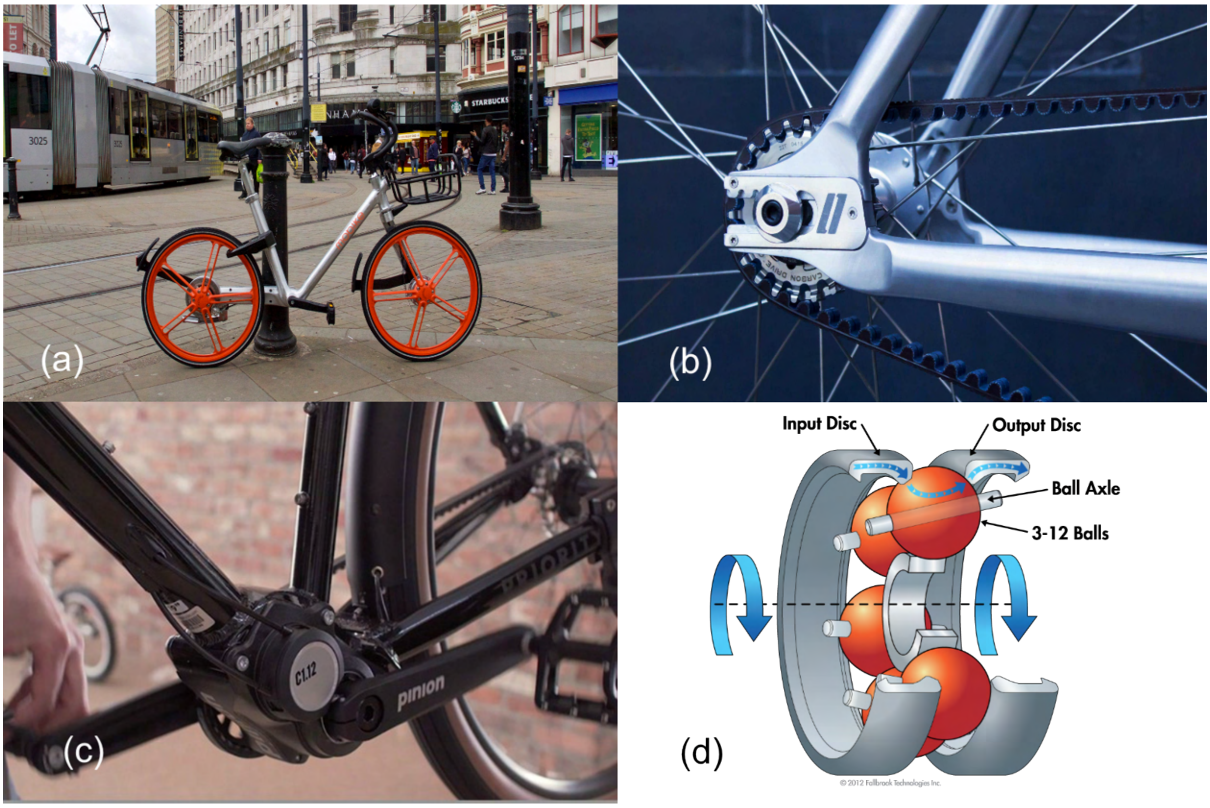

Figure 1.5: Similar products currently on the market; (a) MoBike; (b) Gates Carbon Drive; (c) Pinion gear box; (d) NuVinci CVT concept(image sources given on page 63);

1.5.1 Belt drives

The German manufacturer Schindelhauer produces all of its bikes using a 'Gates Carbon Drive' [15]. This is a belt drive system and is shown in figure 1.5 (b). It is claimed that it requires zero maintenance and has a much longer life span than a standard chain drive system. This drive system is popular in the mountain bike and bike trekking communities where it is less susceptible to performance degradation from mud and debris.

1.5.2 Gear 'Boxes'

The use of a belt drive means that traditional derailleur systems cannot be used to change gear. Most belt drive bicycles on the market today make use of one of two systems.

Firstly, the hub gear. These fit a gear box inside the rear hub of the bike and allow the user to change gear with a handlebar mounted control. By far the most popular of these systems is the ‘Sturmy Archer’ 3 speed hub which has been popular on city and commuter bicycles for decades.

The alternative and more recent innovation in this field is the use of bottom bracket mounted gear boxes. These were originally developed for the downhill mountain bike market to allow the user to switch into very low gears for steep technical climbs. The ease of maintenance and reliability are both major advantages in the commuter market and this has resulted in some manufacturers switching their product focus to the commuter market in recent years. A popular line of products is produced by the Pinion brand.

1.5.3 NuVinci

There have also been a number of innovations in the e-bike[2] market which offer advantages similar in nature to those of our product. These include the NuVinci hub which is a continuously variable system that fits inside the rear hub of the bike [16]. Figure 1.5 (d) shows a schematic of this concept. The input and output discs contact with differing positions on the ball bearing’s surface depending on the orientation of the ball’s axis. The transmission ratio between these two rings can be changed by varying the ball bearing’s orientation. In the bicycles using this transmission currently on the market, a microcontroller either responds to user input or automatically selects an appropriate gear based on the riders preferred cadence.

1.5.4 The Reeves drive

A Reeves drive, or Variable Diameter Pulley (VDP), is a type of continuously variable transmission whose gear ratio is controlled by the contracting and spreading of opposing cones to cause a V-shaped belt to move radially inwards or outwards. Integral to the operation of this system is the ability of the belt to slowly slip radially along the surfaces of the cones while changing gear. In most applications the movement of the cones is manually controlled with two pairs of cones acting in reverse of one another to maintain belt tension. However, ‘half’ Reeves drive designs have been developed which use of alternative methods for maintaining belt tension. So far as the group are aware, there have been no previous efforts to apply this transmission to a bicycle.

[2] electric bike

1.5.5 MoBike

MoBike is the largest public bicycle hire operator in the world. They operate in over 200 cities with bike similar to those depicted in figure 1.5 (a) [17]. These bicycles are of particular interest to this project due to many of their unconventional design features. Firstly, rather than using a regular chain drive the MoBike is shaft driven. This requires less maintenance for the operator, helping to keep costs low. Secondly, the frame is of a cantilevered design, allowing the bikes to be stacked more-closely together for ease of storage and transport [17]. A similar frame design may open up significantly more design freedom.

In London, the most popular public hire scheme is the Santander cycles scheme. These bicycles are publicly supported by Transport for London and operate on a docked basis.

1.6 Product Design Specification

At the initial stages of this project it was envisaged that the drivetrain developed as part of this project would be aimed at the private retrofit market i.e. a relatively low cost purchase to be made by individuals and fitted to their existing cycle, reducing the required maintenance and improving the ease of operation. The initial product design specification (PDS), as presented in the Quality Plan [1] reflected this fact.

However, as the project progressed it became increasingly clear to the group that the product may be better targeted at public hire cycles with its main improvement over current models being a reduction in complexity of operation for the user. As a result the product design specification had evolved in line with the changing project aims.

The most recent revision of the PDS is tabulated in table 1.1.

Table 1.1: Product Design Specification;

Element	**Statement or Criteria**	**Verification by**
Customer		
Needs	Fits a standard 68 mm ISO threaded bottom bracket, i.e. ISO 6696:1989 [18]	Design review
	Fits a standard 130 mm spaced rear drop out	Design review
Competition	Automatic continuously variable transmission to simplify bicycle operation. Market research shows there are no current products on the market that are automatic and continuous	Testing
Operation	Must operate without any user input other than pedalling	Testing
Performance	Must be capable of gear ratios from 1.2:1 to 2.8:1	Testing
	Capable of transmitting powers up to 95th percentile acceleration	Testing
	Equilibrium cadence: cadence at which the transmission will match the target power at steady state = 60 to 90 RPM	Testing
Environment	Must be suitable for outdoor operation	
	Temperature range -5 to 30$^{\circ}\mathrm{C}$ Testing (pre-ride freeze for lower bound, noted summer temperature for upper bound)	Testing
	Must operate adequately in wet conditions (rain and splash)	Testing
	Design to avoid crevices where debris can collect	Design review and testing
Weight	Transmission weight (not including bike) $<$ 5 kg	Design review and testing
Size	Crank Length of 170 mm	Design review
	Q-factor $<$ 0.2 m	Design review
Life		
Product life	Bicycle groupset life of 4 years as is typical of the industry	
Service life	5 year life at 70 miles per week (18,200 miles)	Calculation based on fatigue
Maintenance	No maintenance for service life (fit and forget)	Testing
Producer		

Table 1.1: Product Design Specification;

Element	Statement or Criteria	Verification by
Quantity	Design must be adaptable 10,000 units per year	Design review
Manufacturing constraints	Must be capable of manufacture using Imperial College Facilities in addition of approved vendor components	Design review
Prototype cost	<£1,000 including cost of test rig	Design review
Regulatory		
Safety Standards	Maximum score 1 on a 5 point risk matrix	Risk assessment
Product Regulations	As far as is practicable meet the requirements of the relevant sections of BS EN ISO 4210-2:201 Cycles. Safety requirements for bicycles. Requirements for city and trekking, young adult, mountain and racing bicycles	Design review
End of life disposal	Any plastic components should be recyclable	Design review

There were numerous pieces of research and calculation, detailed herein, that informed the product design specification. This ensured that the criteria given were specific, achievable and quantifiable.

Firstly the lowest gear ratio specified is 1.2:1 between the chainrings and sprockets for a 700c wheel size. This is equivilent to the lowest gear on a Santander public hire bike, and at 60 RPM gives a minimum speed of approximately 10 kph within the acceptable cadence range. The highest gear specified on the PDS is 2.8:1, again on a 700c wheel. This is the gear ratio given as standard on most ‘off the shelf’ single speed bikes and gives a maximum speed of approximately 30 kph at 90 RPM. The top gear is much lower than the gear given on most traditional groupsets. However, based on the target market of inexperienced, novice cyclists, making journeys in cities, the range is appropriate.

The cadence range given in the PDS are based on the experiences of group members and is supported by the literature [19].

170 mm is around standard for bicycle crank lengths. Naturally, this must be standard for a publicly available hire-bicycle.

The Q-factor affects the handling, power and safety of the bicycle (as further discussed in section 3.2.1). While the Santander public hire bike provides a good starting point for Q-factor, measured at 180 mm, the proposed integration of a continuously variable transmission will certainly increase required Q- factor and so an upper bound of 200 mm was chosen. Allowing a higher Q-factor would reduce comfort and potentially lead to injuries [20].

2. Static Prototype

2.1 Project Ideation

Following the development of the PDS it was then necessary to begin high level concept development. The problem was divided into three different systems:

1. The continuity mechanism, the mechanism that would allow the gear ratio to vary continuously over a defined range;
2. Automation mechanism, the mechanism that would cause the gear ratio to change in response to a change in the user input;
3. Power transmission, the mechanism that would allow for the transfer of power from the user to the rear wheel.

Each of these problems was then approached separately and the ideas generated then combined in order to produce a complete solution to the problem.

The different ideas considered are summarised in tables 2.1 through 2.3.

Table 2.1: Continuity mechanisms considered;

Name	Description	Advantages	Disadvantages
Reeves Drive	Two sets of two cones. The cones may splay apart and come together allowing a continuously variable gear ratio.	Many possible implementations. Widely used.	Tendency to slip.
Evans Drive	Two cones sat parallel and reversed against one another. By moving the belt along the axis of the drive cone, the gear ratio can be continuously altered.	Simple gear change mechanism.	Multiple off-axis shafts required. Hard to control. Tendency to slip.
NuVinci type ball bearings	The NuVinci uses two rings, ball bearings and an idler. The idler moves from left to right, changing the contact radius of the ball bearing on each ring, allowing for a continuously variable gear ratio.	Compact, tried and tested. Can be used with a chain, therefore no slip.	IP issues. Complex to manufacture. No obvious automatic mechanical control option.
Chainring diameter variation	A mechanism that allows the adjustment of the chainring diameter. This particular example consists of multiple sprockets that adjust radially.	Low possibility of slip.	Complex. Only compatible with electrical control.

Table 2.2: Product Design Specification;

Name	Description	Advantages	Disadvantages
Spring torque control.	Uses a spring to control the gearing in response to torque changes.	Robust. Requires little maintenance.	Unpredictable long-term behaviour. Hard to tune.
Electronic RPM sensor/actuator.	Detects RPM electronically and responds to changes using an electronic actuator	Behaviour can be precisely specified and tuned with ease.	Requires power supply.
Mechanical RPM sensor/actuator	Detects RPM mechanically and responds to changes using a mechanical actuator	No power supply.	Complicated. Heavy.
Electronic torque sensor/actuator	Detects torque electronically and responds to changes using an electronic actuator.	Behaviour can be precisely specified and tuned with ease.	Requires power supply.

Table 2.3: Product Design Specification;

Name	**Advantages**	**Disadvantages**
Toothed V-belt.	Can mesh with a pulley. Can achieve tighter wrap angles for a given surface area of contact.	Wedge angle restricts design freedom.
Untoothed V-belt.	Cheap and ubiquitous.	Slip. Wedge angle restricts design freedom.
Chain.	Strong. Unlikely to slip.	High maintenance.
Drive shaft.	High efficiency.	Complexity and heavy.
Electrical.	Space efficient.	Inefficient Low torque for acceleration.

Following consideration of the options discussed above against the PDS (table 1.1), it was deemed that a solution involving a Reeves drive using a spring torque controller held the most promise. This idea was selected as a basis for further investigation. Some initial sketches of this idea are presented in figure 2.1.

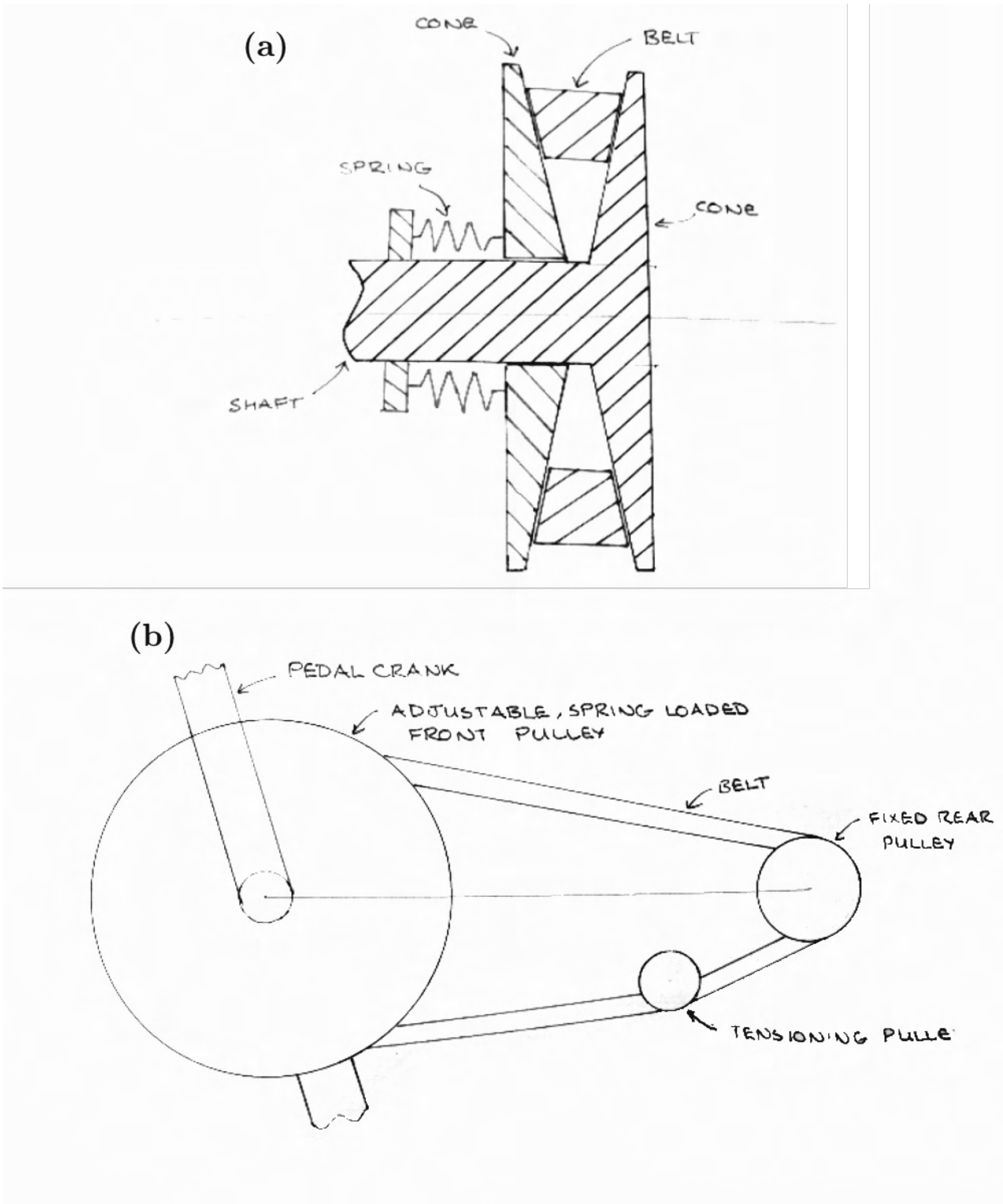

Figure 2.1: Initial concept sketches; (a) cross section (b) side view of a Reeves drive spring torque controlled concept;

The concept uses a balance of forces on the belt drive to dictate the movement of the cones and therefore the belt. When a higher torque is exerted by the rider, increased belt tension will force the cones apart and move the rider to a lower gear ratio to maintain optimal torque and therefore cadence. The reverse occurs when the rider inputs a lower torque. Given that a rider's power-cadence characteristic is roughly parabolic, the torque can be derived as always decreasing with increasing cadence.

At the rear end of the bicycle, a tensioner, similar in operation to a bicycle rear derailleur takes up the slack belt length and maintains a near constant belt tension for the portion beneath the chain stays. This element is required as the amount of slack belt varies as the drive radius of the front cones changes.

2.2 Proof of Concept Prototyping

Due to the lack of literature concerning half Reeves drives for bicycles (to be expected as so-far as we are aware, this is a novel design), the need to prove the concept was established. The prototype was to be used to demonstrate the principles of operation of a half Reeves drives, to test whether the assumptions made as a result of the morphological analysis were correct and to identify any immediate issues that had been overlooked. It was envisaged that this prototype would be a bench top rig. A PDS was written for the prototype and is tabulated in table 2.4.

The prototype was based around two plywood sideboards between which a drive shaft and an output shaft were mounted. The drive shaft represented the crankshaft on the bicycle, whilst the output shaft represented the rear wheel. The drive shaft was designed with customisation in mind, in order to fit multiple types of cone when testing the different flexible transmission methods. The cones would splay apart when a load was applied at the output shaft and were returned together by an axially mounted spring. A tensioner mechanism prevented the belt from going slack.

2.2.1 Cone Geometry

The geometry of the main Reeves drive cones is integral to the automatic function of the CVT. The transmission is designed first by specifying cone geometry which then determines the specification of the springs. The important cone dimensions are defined completely by the maximum and minimum desired gear ratios, the belt width and the cone angle. The aforementioned gear ratios are specified in the PDS (table 1.1). Equations 2.1 and 2.2 relate cone dimensions based on these factors and allow belt-specific cones to be systematically

Table 2.4: Product Design Specification for the Proof of Concept Prototype;

Element	**Statement or Criteria**	**Verification by**
Customer		
Needs	Allows fitting of different cone iterations on a single test rig	Design review
	Allows adjustment of spring tensions	Design review
Operation	Requires force of less than 30N to operate	Testing
	Handle is comfortable to operate	Testing
	Can achieve cadences up to 80 RPM	Testing
Performance	Must be capable of gear ratios from 1.2:1 to 2.8:1	Testing
Weight	$<$ 8 kg	Design review and testing
Size	Must fit in DMT storage box 0.5 m x 0.5 m x 1 m	Design review
Life		
Service life	At least 10 hours of testing at 80 RPM	Calculation based on fatigue
Maintenance	No maintenance for service life	Testing
Producer		
Quantity	Single unit	Design review and compatibility with production methods
Manufacturing constraints	Must be capable of manufacture using Imperial College Facilities in addition of approved vendor components	Design review
Prototype cost	$<$£300	Design review
Safety		
Safety Standards	Maximum score 1 on a 5 point risk matrix	Risk assessment

designed and submitted for testing on a repeatable basis. This has helped the group's efficiently iterate cone designs during the first proof-of-concept prototype. The dimensions referenced in equations 2.1 and 2.2 are defined in figure 2.4.

$$R_i = \frac{w_t}{2.64 \cdot \tan\beta} - 1.258 \cdot D \tag{2.1}$$

$$R_o = \frac{w_t}{2 \cdot \tan\beta} + \frac{w_t}{2.64 \cdot \tan\beta} - 1.258 \cdot D \tag{2.2}$$

Some derived dimensions for different cone and belt combinations are given in table 2.5

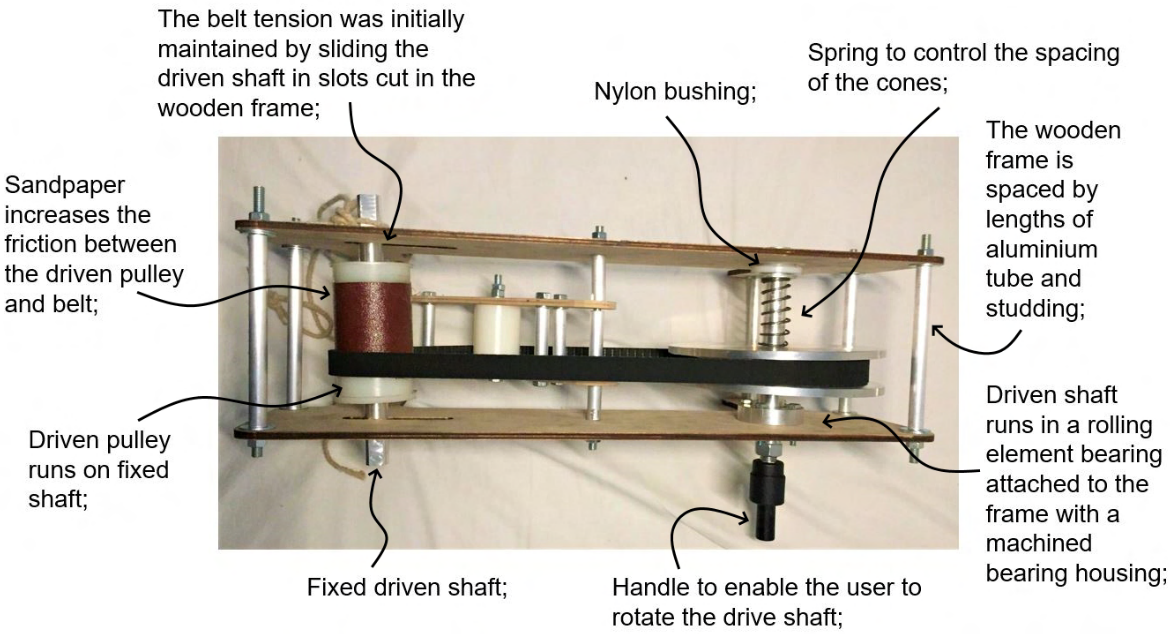

Figure 2.2: Top view of the static prototype;

Table 2.5: Various potential belts and their associated geometries (all units in mm unless otherwise stated);

Belt Model	W_t	D	$Beta$ [°]	R_i	R_o
Dayco HPX 5008 (Snowmobile Belt)	35.3	15.0	14.0	34.8	105.5
XPC Cogged	22.0	18.0	18.0	3.0	36.4
SPC	22.0	18.0	20.0	0.3	30.5
CX cogged	22.0	14.0	20.0	5.3	35.5
D Section	32.0	20.0	20.0	8.1	52.1
Fermer PU Belt (Round)	15.9	15.9	10.0	14.2	59.2
RS Pro PU Belt	8.0	8.0	10.0	7.1	29.8
Tesaflex Gas Pipe (Round)	24.0	24.0	10.0	21.4	89.4
RS Pro Timing Belt (Final Prototype)	20.0	6.0	10.0	35.0	91.5

2.3 Testing

Three different belts were tested; a snowmobile belt, a timing belt and a round belt. The snowmobile belt was a notched v-belt and the timing belt a standard toothed timing belt such as those used in an internal combustion engine. The round belt was created by drawing a section of rope through silicone tubing and splicing the ends together with a ‘long splice’. It is pictured in figure 2.5(b).

Due to the different section angles of the belts, the different belt profiles required different cones to use them. In this case, the timing belt and round belt utilised one set of cones whilst the snowmobile belt utilised another. The nature of the angled surfaces on the cones meant that they were most easily manufactured using a CNC lathe.

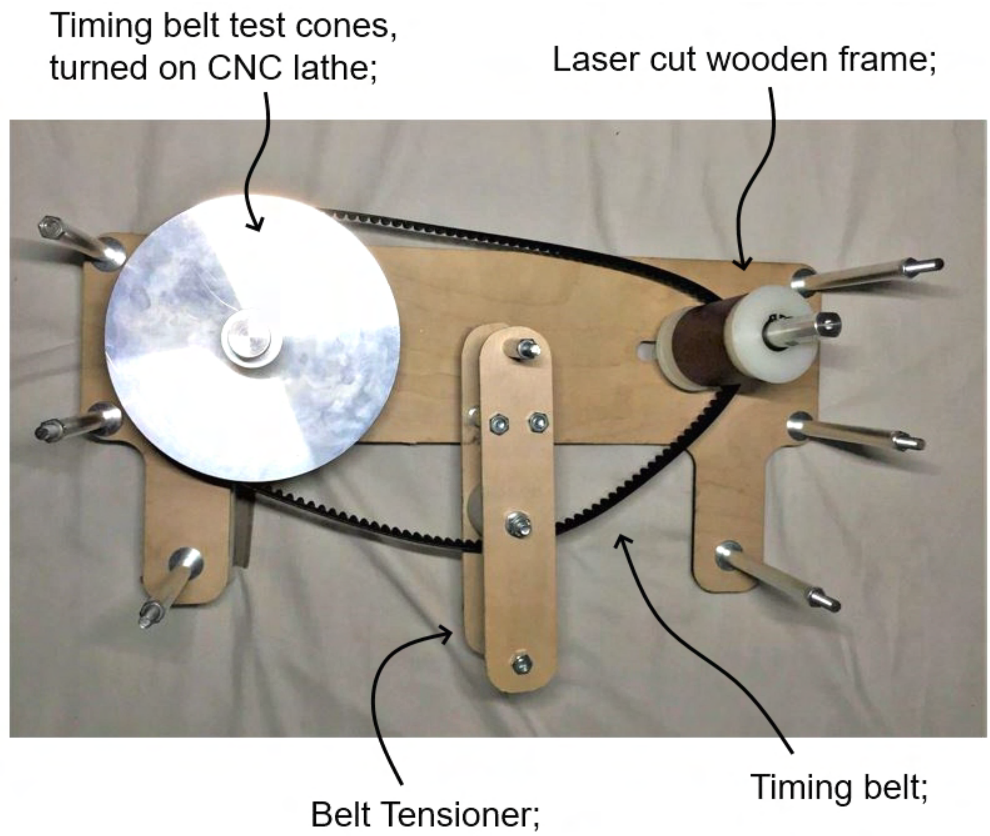

Figure 2.3: Photograph of the static prototype with one side-plate removed

Tests were carried out by installing the belts on the prototype along with their appropriate cones, the rig was then hand wound and observations as to the effectiveness of the cone-belt combination assessed. The best combinations were the ones for which the cones would most easily splay as the torque was increased.

The snowmobile belt was far too stiff to achieve the narrow wrap angles required as the gear ratio reduced. This resulted in any increase in torque causing flexing of the belt rather than any splaying of the cones. The round belt was not uniform due to the joins between the rope ends and silicone tubing. The lack of the constant diameter forced the cones apart as the lumpy section entered, causing the belt to slip and ultimately rendering it ineffective. The timing belt was the only belt that responded appropriately. It was capable of achieving the required wrap angle and was appropriately stiff in the axial direction[1] to cause the cones to splay.

[1] With respect to the shaft.

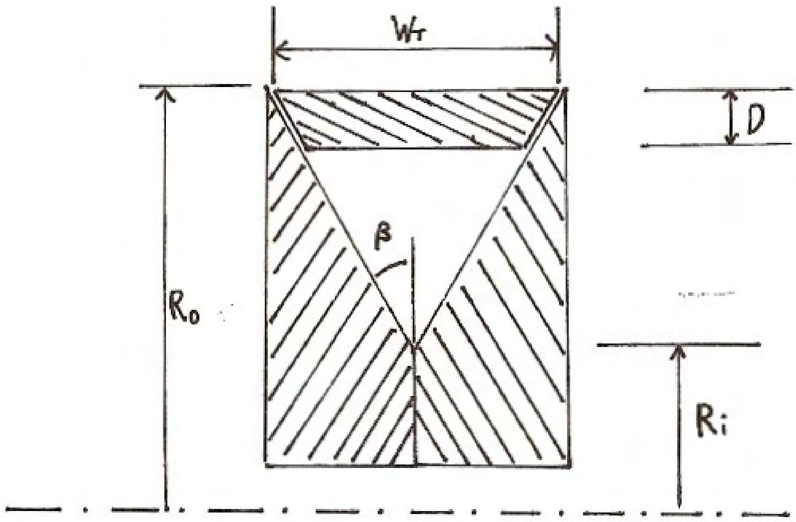

Figure 2.4: Schematic of cone cross section;

2.4 Evaluation

There were a number of lessons learned from the manufacture and testing of this initial prototype. When the cones were forced apart, the belt would become slack. In order to provide constant belt tension and prevent the belt from slipping it became necessary to introduce a tensioner mechanism of some form. Initially the tension was provided by moving the output shaft forwards and backwards by attaching it to a spring mechanism. However, this was found to be ineffective as it acted on both the top and bottom of the belt, whereas it was most important to take up slack at the bottom of the belt. A tensioner was introduced as shown in figure 2.3. It was designed so that its mass pulled the belt taut. From this it was understood that a tensioner mechanism would need to be incorporated in the final design.

During testing there where significant difficulties with the belt slipping on the output shaft, which was initially just a smooth roller. Sandpaper was added to increase the contact friction between the roller and the belt, which somewhat ameliorated the issue. A possible solution which could be considered going forward is the use of a timing belt pulley which would mesh with teeth on the belt. Slipping within the cones was also an issue, therefore the importance of correct spring selection was emphasised.

2.4.1 Expenditure

A summary of expenditure for the static prototype is given in table A.1.

The static prototype cost £340.10 in total. This was in excess of the £300.00 originally

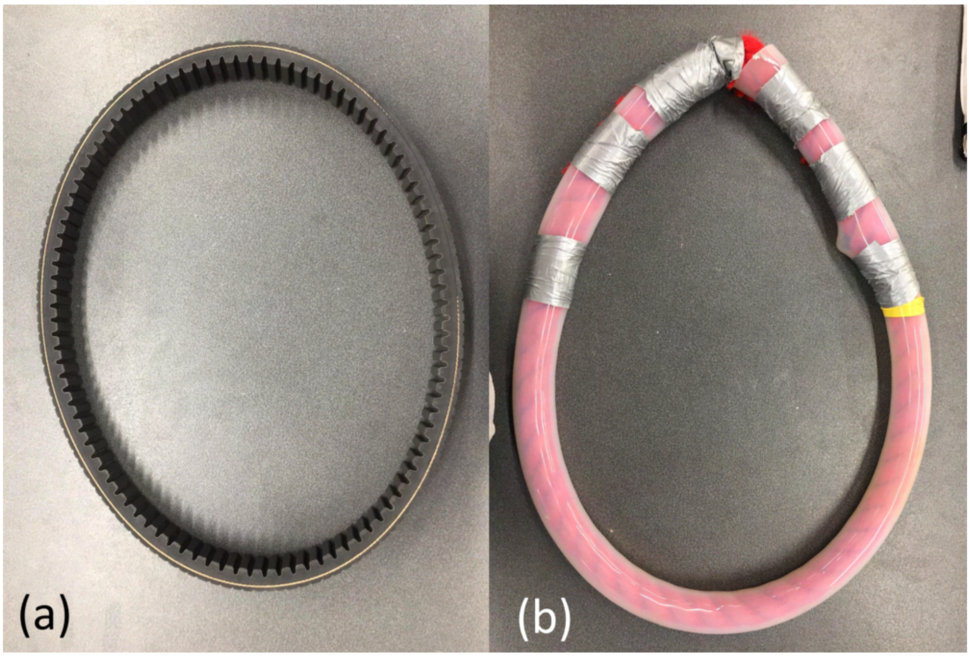

Figure 2.5: Photographs of the (a) snowmobile and (b) spliced rope belts tested;

budgeted for. This is likely to be mostly due to the fact that we used the rig to test a wider range of belts than initially anticipated. It was not deemed to be problematic at this stage in the project as the budget included sufficient contingency to cover this.

3. Dynamic Prototype

3.1 Revised Design

As the static prototype discussed previously had adequately demonstrated the principle of a mechanically controlled half Reeves drive, and shown that a timing belt performed satisfactorily, this concept was taken forward for more detailed design. The main challenges in the next design phase lay in determining how to retrofit this system to a bicycle and how to scale up the design from the low-torque prototype to one capable of dealing with the torque generated by a commuter.

It was identified that the solution would have to consist of three interdependent sub-assemblies; namely a bottom bracket assembly, a rear wheel assembly and a tensioner mechanism.

The morphological analysis and initial detailed design were carried out as a group over the course of two days. Feedback from the design review also fed into the design which was ultimately manufactured. This section aims to present the design as it was at the start of manufacture and to summarise any changes that have occurred since then and the drivers behind them.

Selected engineering drawings are appended to this report in Appendix D. A complete archive of engineering drawings and revisions can be found on the team Sharepoint at `DMT_Bicycle_ACVT/Documents/Drawings/`.

3.1.1 Test Bicycle

It was decided fairly early on in the project that the best way to test the design would be to mount it on a bicycle as this allowed for easy dynamic testing both on the road and on a turbo-trainer. The transmission is intended for use on purpose built commuter bikes however due to budget constraints the bicycle chosen was an old mountain bike and this necessitated a number of modifications. Firstly and most significantly, a standard bicycle frame is not designed to take a belt and so one of the seat stays had to be cut and then re-joined in order to locate the belt correctly. The re-joining method used was to use duct tape to align the two halves of the cut tube and then wrap it in 0.2 mm stainless steel shim and fix it with jubilee clips. This is not a good long-term solution however, it did allow adequate testing of the prototype. This is shown in figure 3.1.

Figure 3.1: Photograph of the drive side cut and re-joined seat stay on the test bicycle;

Secondly it was decided to change the wheels on the bike from standard 26” mountain bike wheels to 700C road bike wheels. This was in order to maintain the correct transmission ratios between the pedals and the road.

3.2 Crank-Set Sub-Assembly

The main design requirements of this sub-assembly were;

1. To incorporate a half Reeves drive sprung with the correct force;
2. To offer an interface with the pedals;
3. To interface with a bottom bracket;
4. To remain below the PDS limit on Q-factor;
5. To enable some adjustability to account for tuning. This was considered prudent given the uncertainties in the spring force calculations;

The spring forces required by this design were high, 700 N at maximum extension, and the compression was quite small. It was quickly identified that achieving the performance

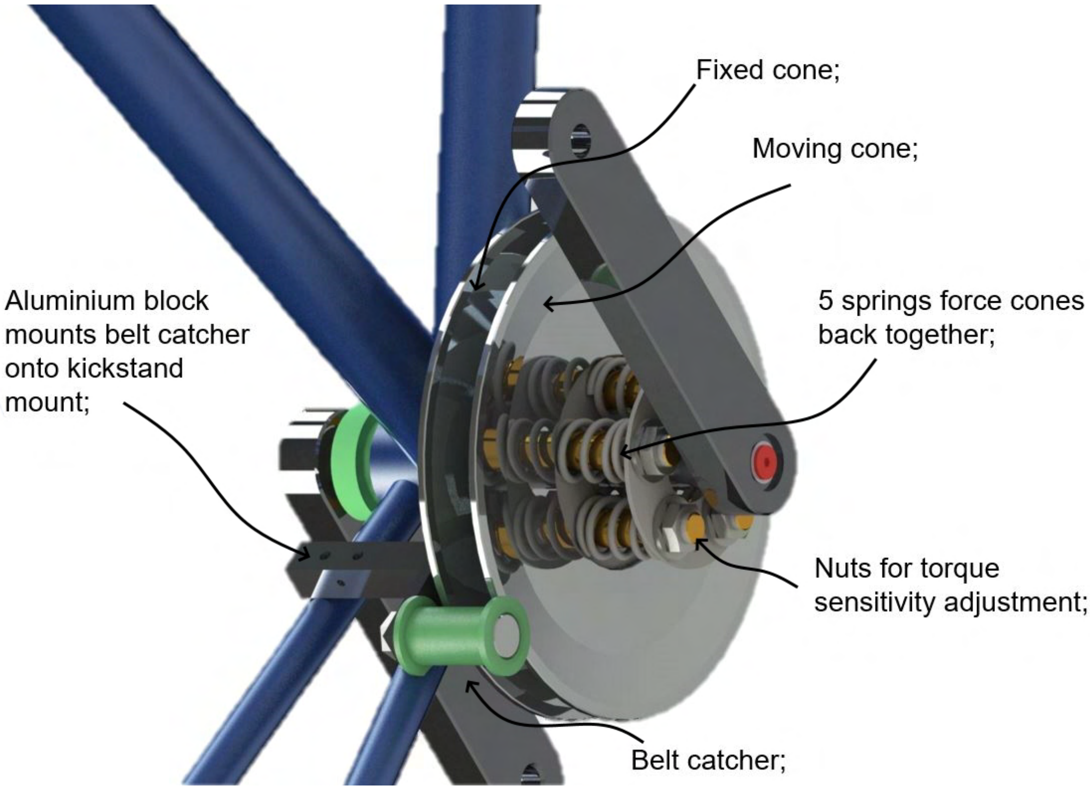

Figure 3.2: Render of crankest sub assembly

using a single large spring was not feasible. Instead a number of smaller springs equidistantly spaced around a circle concentric to the cones had to be used. This spring arrangement also had the advantage of precluding the moving cone from rotating independently of the fixed cone without having to incorporate a guide onto the main shaft.

It was decided that the sub-assembly should not incorporate cranks, rather that standard connections should be provided allowing any cranks to be used. Square taper connections were selected as they are widely used [21] and within the manufacturing capability of the STW.

Initial discussions about interfacing with the bottom bracket focused on different methods of appending the subassembly onto the end of a conventional bottom bracket shaft. Concerns were raised about the bulk of material that would be required to deal with the stresses generated by the connection during operation and the consequent increase in Q-factor. There were concerns about the practicalities of manufacturing the female square taper that would likely form the basis of the connection. It was decided that the the best solution would be to dispose of the connection and design the whole shaft as one part directly interfacing it with the bottom bracket bearings. Shimano Hollowtech series bottom brackets allow for this and hence the shaft was designed around this. Figures 3.2 and 3.3 show annotated views from the CAD model of this assembly.

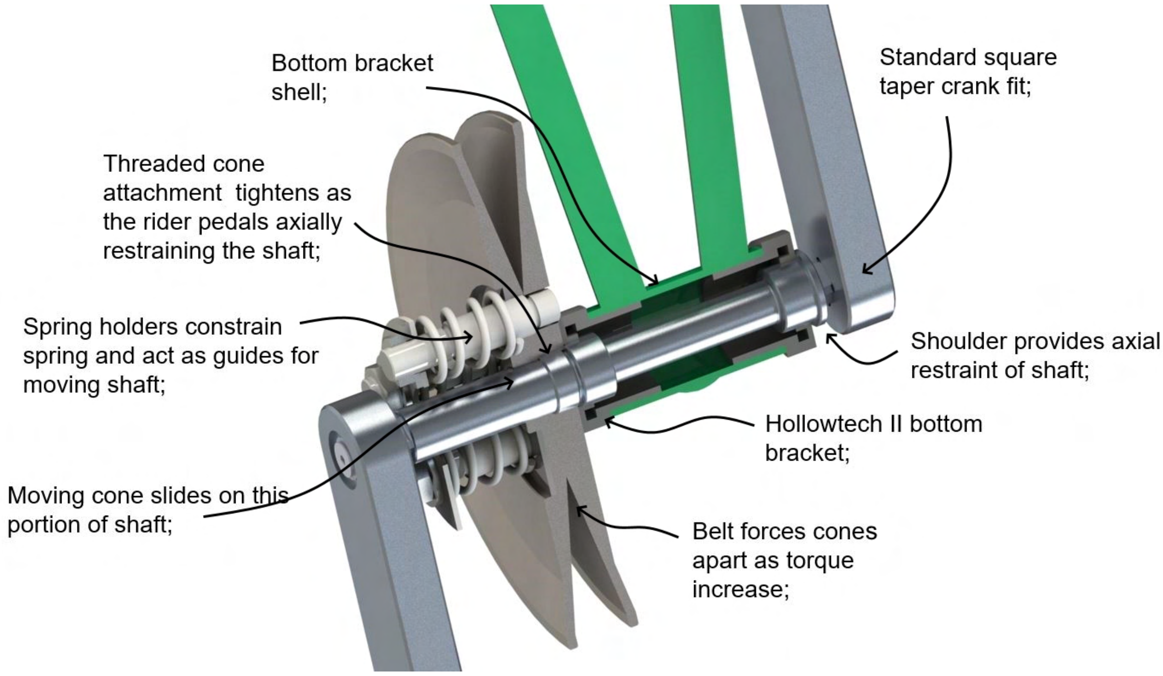

Figure 3.3: Render of crankest sub assembly

3.2.1 Bottom Bracket Spindle

For a fully functional mass market commuter bicycle the fatigue test for a bottom bracket spindle is detailed by ISO standard 4210 [22]. It specifies that a vertical pedalling force of 1300 N should be alternately applied on either side of the bicycle at a lateral distance of 65 mm from the crank outer face. This force should be applied with the cranks angled at 45 degrees below or above the horizontal as shown in figure 3.4. The standard specifies that the spindle should have a fatigue life in excess of 100,000 revolutions for this loading condition. While it is not possible for us to perform this test within the constraints of a DMT project, this test was used to inform the design of the spindle and the relevant calculations are detailed below.

This test introduces three modes of loading on the shaft. There is a bending stress as a result of the lateral cantilevering of the 1300N force to the right and left of the bicycle. There is a torsional shear stress due to the torque transmitted by the spindle to the belt. This analysis will also consider the comparatively small vertical shear stress caused by the force on the pedals. Before discussing bending, torsion and shear, the cross-sectional geometry of the shaft is discussed.

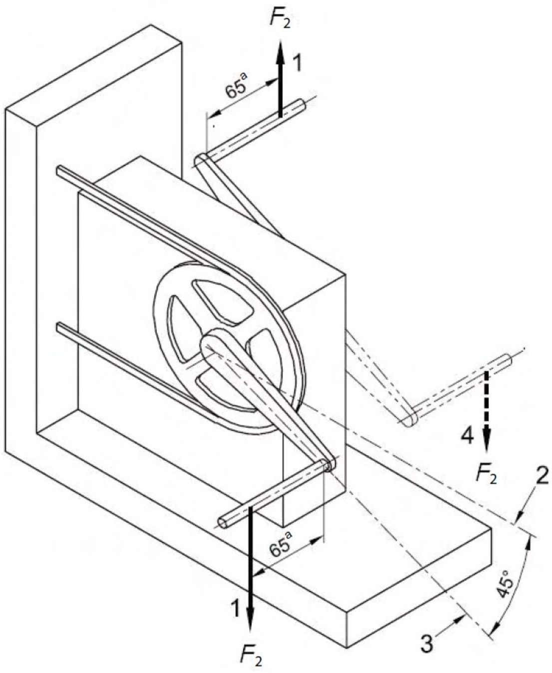

Figure 3.4: Schematic showing fatigue test arrangement as specified by ISO 4210 [22];

3.2.2 Geometry

Figure 3.5 shows a schematic side of the bottom bracket spindle. At each end of the shaft there is a square taper defined as per the relevant ISO standard [21] with an M8 straight tap concentric with the centreline. These exist so that the shaft can be used with ISO standard square taper cranks. The shaft has been manufactured in the final prototype to include fillets which reduce stress concentrations at the join between the square taper and the cylindrical section of the shaft. This is in line with the discussion given in section 2.4 of this group's DMT Progress Report [23].

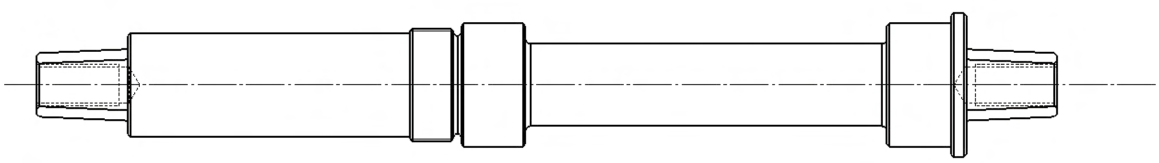

Figure 3.5: Schematic of the bottom bracket spindle geometry;

Beyond the square tapers, the shaft consists of several solid cylindrical cross sections and a threaded section. To determine the bending, shear and torsional stresses, the second moment of area, cross sectional area and torsional moment of areas were calculated. In the analysis it is important to include the effect of the M8 taps in the square tapers and this has been done by modelling them as 8 mm diameter holes.

3.2.3 Bending Stresses

The bending stresses are calculated by modelling the bottom bracket as short bearings which do not resist moments but can impose resisting forces as a couple. This is the conservative case as a bottom bracket with moment resistance would reduce the moment on the shaft between the bottom bracket bearings. Figure 3.6 shows the variation of shear force and resultant bending moment along the drive side (here shown on the left side) of the shaft. Only the drive side will be considered for this analysis since in the case of the square taper sections the analysis is identical and the cylindrical sections are critical only on the drive side.

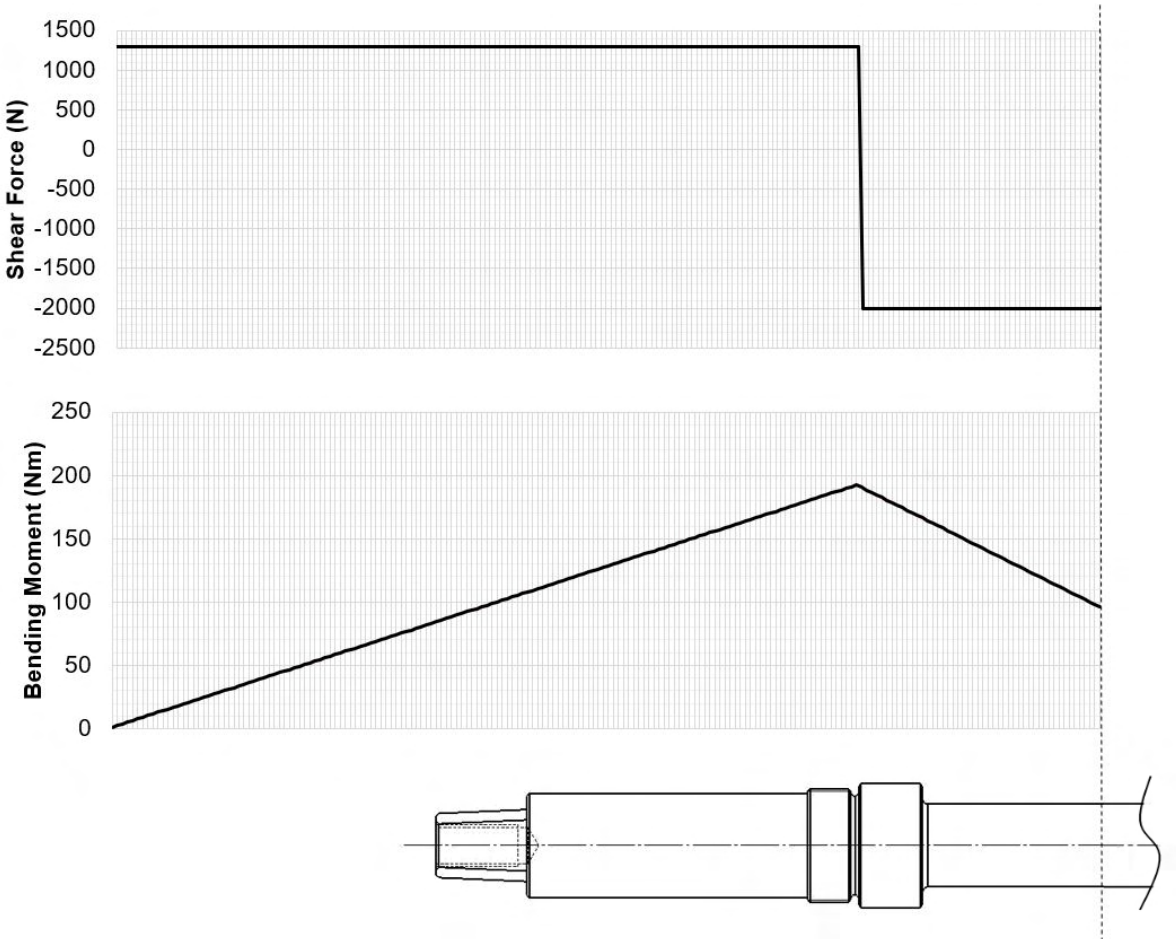

Figure 3.6: Shear force and bending moment along the length of the main shaft

This bending moment acts over the varying second moments of area and maximum radii[1] along the shaft to give a bending stress distribution as shown in figure 3.7.

[1] Taken as $\sqrt{2}R$ in the case of a square section [24];

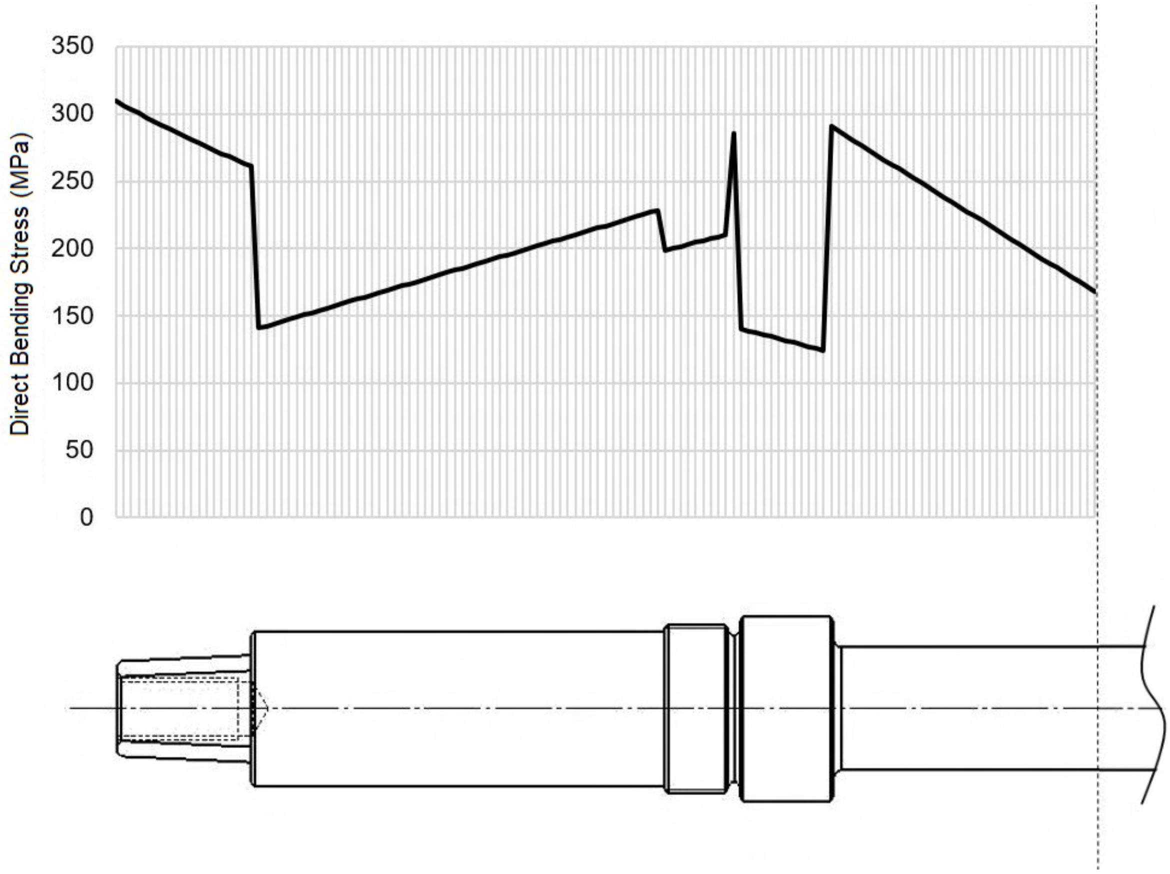

Figure 3.7: Variation in bending stress along the length of the main shaft

3.2.4 Torsional Shear Stress

The torque transmitted along the shaft is constant between the bases of the square tapers as it is transmitted from each pedal to the cones. The shear associated with the torque is therefore dependant only on the torsional moment of area and maximum radius. For the square tapers, it is assumed that the torsional shear is equal to the value calculated for the base since at failure the deformation would cause the limiting factor to be the torsion resisted at the base. The variation in torsional shear stress along the shaft is shown in figure 3.8.

3.2.5 Vertical Shear

The vertical shear force arises because of the force on the pedal. It does not depend on moment arm or angle of the shaft. The shear force transmitted in this mode is constant, therefore the vertical shear stress, $\tau_{vertical}$, as a result is inversely proportional to the cross-sectional area, and given by equation 3.1

$$\tau_{vertical} = \frac{shear\ force}{area} \tag{3.1}$$

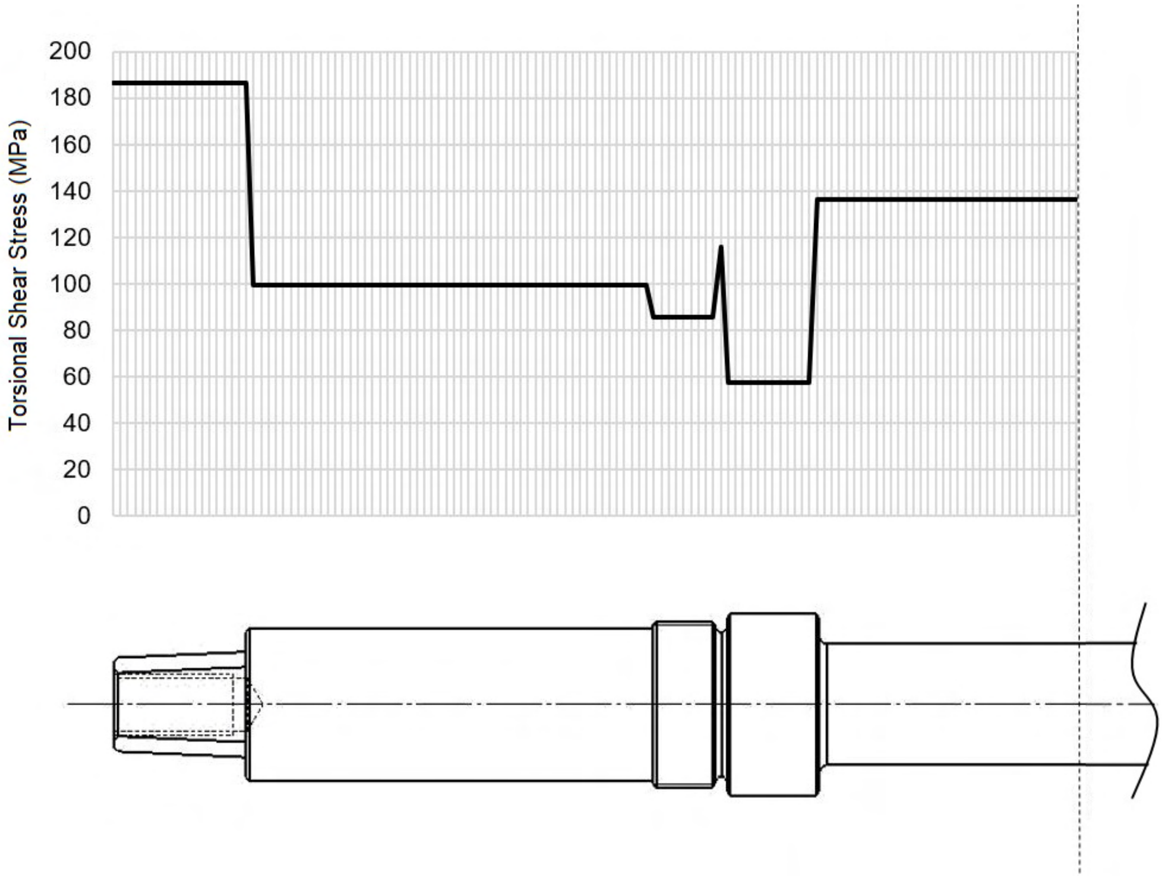

Figure 3.8: Variation of torsional shear stress along the length of the main shaft;

3.2.6 Total Resultant Stress

The effects of bending, torsion and vertical shear can be combined using a force balance. The resulting expression relevant to this situation is given in equation 3.2 [24]. The output from this calculation is an equivalent direct stress which can be compared to values given in direct stress tests.

$$\sigma_1 = \frac{1}{2}\sigma_{bending} + \sqrt{\left(\frac{1}{2}\sigma_{bending}\right)^2 + \tau_{total}} \tag{3.2}$$

Where σ_1 is the principal stress, $\sigma_{bending}$ is the bending stress. The total shear stress varies with shaft azimuth. The limiting total shear stress[2] is at the top and bottom of the shaft where the torsional and vertical shear are orthogonal. The total shear stress in this case is given in equation 3.3.

$$\tau_{total} = \sqrt{\tau_{torsion}^2 + \tau_{vertical}^2} \tag{3.3}$$

[2]Where the limiting total shear stress is taken as the shear stress at the same point as where total direct stress is at a maximum, not the maximum total shear stress at any point;

The variation in total resultant stress along the shaft is shown in figure 3.9.

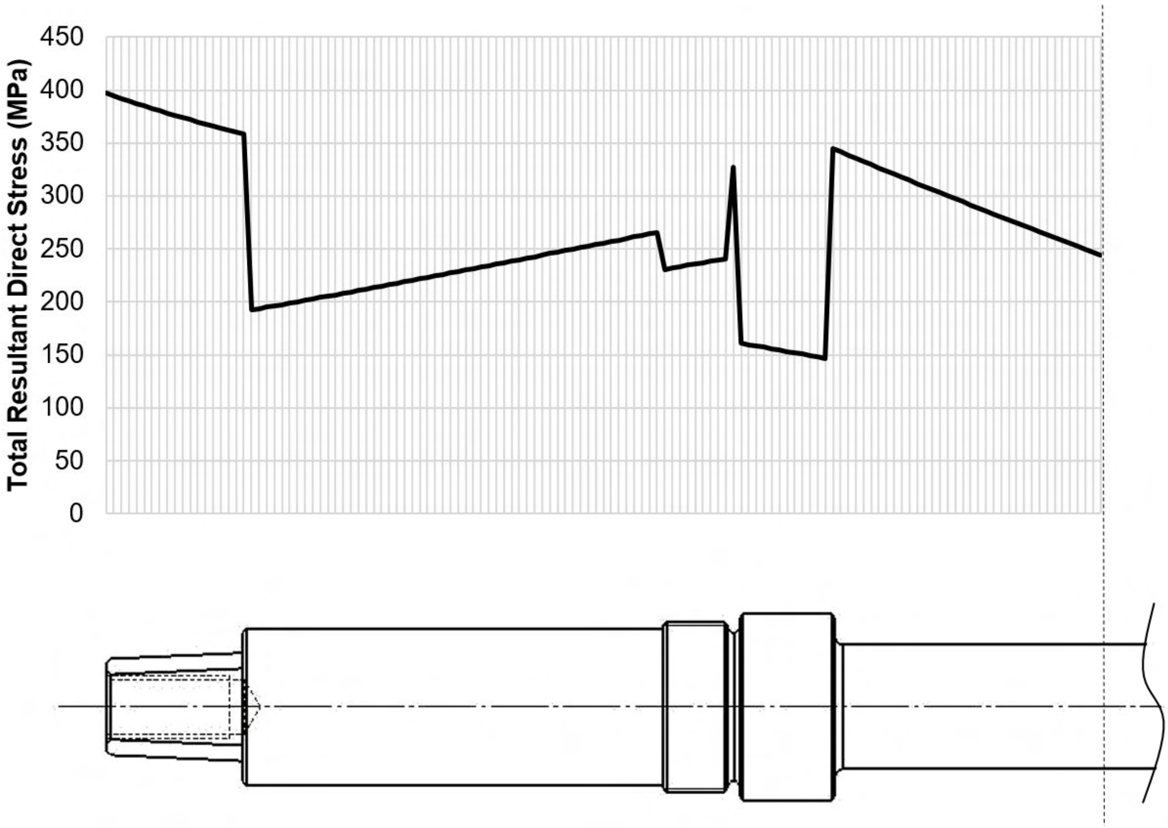

Figure 3.9: Variation in total resultant direct stress along the length of the main shaft;

The maximum total resultant stress is 400 MPa. In this loading condition this load is not fully reversed, however taking this to be fully reversed (and thus making a conservative assumption) figure 3.10 shows that the shaft would satisfy the ISO test requirement. This figure is based on use of AISI 1040 carbon steel which is typical for bicycle bottom bracket spindles. This is as expected because the most onerous loading is located on the square taper whose geometry is defined by standard ISO 6695:2015 [21].

In our prototype, which is not a bicycle suitable for long term use, we have not chosen a material with sufficient fatigue strength to satisfy the onerous ISO test requirement. We do not expect the bike to be subject to loading this extreme and so selected a lower grade of steel, EN1A. This material far easier to procure and machine yet is sufficient to demonstrate the viability of the drivetrain design.

3.2.7 Assembly

Compressing the springs to allow the backing plate and nuts to be assembled also proved challenging due to the high forces involved and the fact that the uncompressed spring length was longer than the guide length. Remaking the spring guides as longer parts was not an

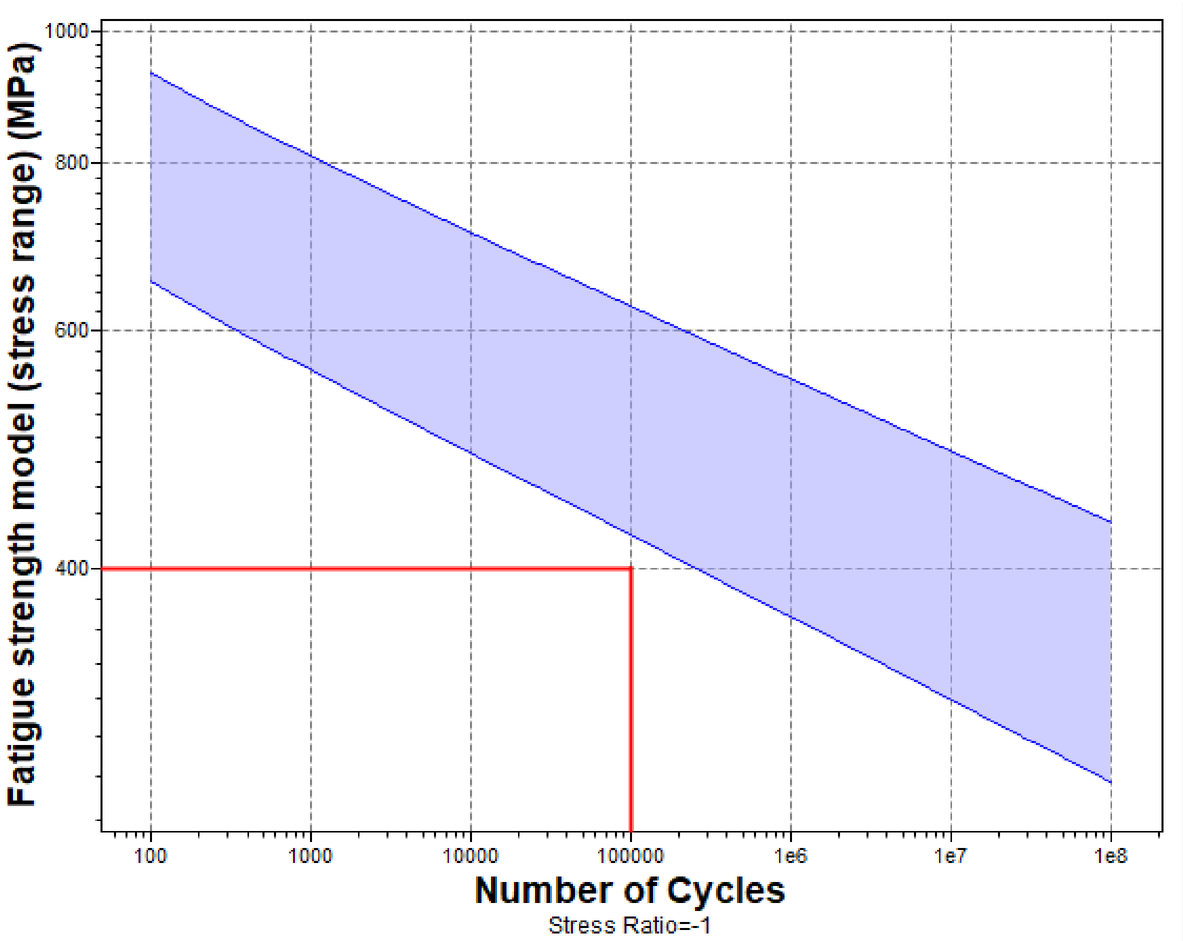

Figure 3.10: Fatigue curve for AISI 1040 carbon steel with the ISO test case loading condition plotted;

option due to the adverse impact on Q-factor. The springs were assembled by cable tying the spring compressed, then cutting the cable ties after assembly. Whilst this approach had some success it proved difficult to remove the cut cable ties and it was unclear whether these would impact operation, hence this approach was abandoned. The tactic then adopted was to use a hydraulic press to compress the springs and then put the nuts on. This method worked but it took a number of attempts for the parts to remain aligned during compression, hence it is clearly not a suitable strategy for mass manufacture. Dismantling these parts without forcibly ejecting components or damaging the threads was also challenging. Taken together these factors effectively preclude user maintenance as even if the assembly is safely dismantled it cannot be reassembled without access to a hydraulic press. This is not necessarily an issue as zero maintenance is a PDS requirement and hence user maintenance is not anticipated. Cycle hire companies, who would be responsible for maintaining a fleet of these bikes would be able to justify a specialist jig for performing this assembly.

3.2.8 Spring Tension

The design aim for the specification of the spring tension was such that the belt is in quasi-static equilibrium at the operating torque. Since the belt is continuously being run through the cones, it is assumed that the radial frictional forces on the belt are negligible. Clearly, the tangential friction cannot be ignored as it is required to transmit the torque to the rear wheel. In order to perform this equilibrium analysis, it will be assumed that the belt exerts an even pressure on the contacted area of the cone. This assumption is almost certainly inaccurate but using an average value acts as a good first pass analysis and avoids the need to consider complex belt deformation effects. Figure 3.11 defines the variables used to deduce the spring force, F_s required.

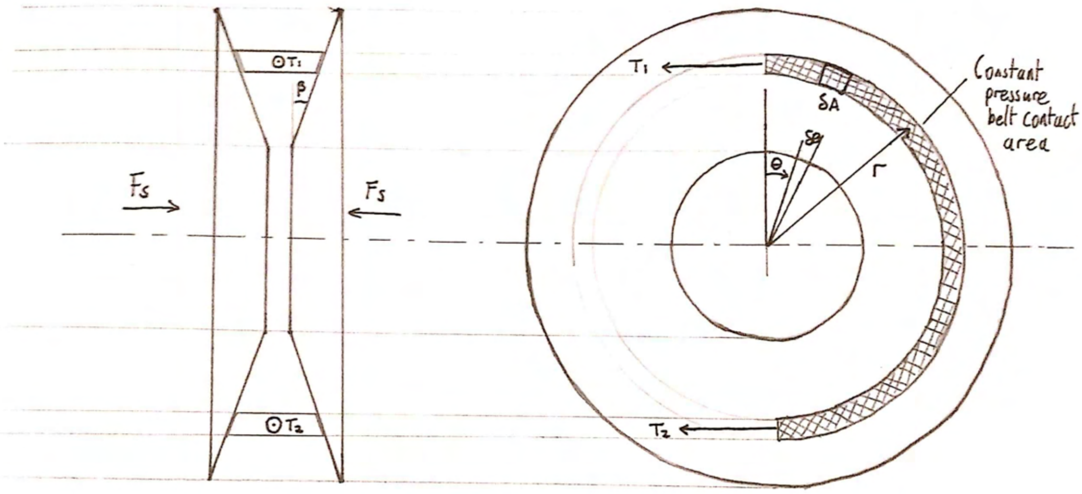

Figure 3.11: Schematic showing the variables used to calculate the required spring force;

By balancing the forces on the belt in contact with the cone, the component of the belt cone contact pressure in the horizontal direction can be expressed as the sum of the tension in the top and bottom of the belt. The contact force can be expressed as the integral of the pressure, p, over the contact area, A

$$\sum F = \lim_{\delta A \to 0} \left(\sum 2\, p\, \delta A\, \sin\theta\, \sin\beta \right)$$

$$= \oiint 2\, p\, \sin\theta\, \sin\beta\, r dr\, d\theta = \frac{r^2 \sin(\beta)\, F_s}{\oiint r dr\, d\theta} \left[1 - \cos\theta\right]_0^{\pi}$$

Where θ is the angle which is integrated over, r the radius from the cone centre and β the half wedge angle of the cone.

Where

$$p = \frac{F_s}{A_{total}} = \frac{F_s}{\oiint r dr\, d\theta}$$

Evaluating and equating this with the belt tensions, T_1 and T_2 for the taught and slack sides respectively, gives;

$$T_1 + T_2 = \frac{4\, F_s \sin\beta}{\pi} \tag{3.4}$$

It can be assumed that T_2 is negligible compared to T_1 as it is limited by the tensioner spring which is significantly weaker than a human cyclist. Therefore,

$$T_1 = \frac{torque}{belt\ radius} \tag{3.5}$$

Assuming a power output of 100 W and a cadence of 70 RPM as specified in the PDS the spring force is approximately equal to $\frac{60}{belt\ radius}$ N. Therefore, the spring must provide a higher force at smaller belt radius which is convenient as in this case the springs are compressed by the splaying of the cones. Ideally, given the cone dimensions and the variation in spring force previously derived, the spring mechanism would have a 'force rate' of 55 N/mm and a natural length of 68 mm. It would not be practical within the scope of this project to commission the manufacture of custom springs however a commercial implementation of this project would be sensible to do so.

The spring type eventually decided on, which was a compromise in terms of shaft length minimisation, simplicity and functional similarity to the figures mentioned above, was five M527312 springs from Flexo Springs Ltd. The combined force rate is 5 x 10.9 Nmm^{-1} = 54.5 Nmm^{-1} and the natural length is 60 mm. Ultimately, the assumptions made to get to this point in the design process unavoidably induce error and so a final implementation of the concept would use the final manufactured prototype as a test rig to define a custom spring.

3.3 Rear Wheel Sub-Assembly

This sub-assembly had two significant design requirements: to enable transfer of power from the belt to the rear wheel and to interface with a stand. Whilst other solutions were considered it seemed sensible, given the use of a timing belt, to use a timing belt pulley to achieve this design requirement. The majority of discussion was then on how to attach a timing belt pulley to a standard bicycle freehub. The complex geometry of a bicycle freehub meant that it was not feasible to bore this profile right through a pulley, hence the approach taken was to bore a clearance through the pulley and then sandwich it between two laser cut plates with the correct profile. The freehub used was a Shimano Ultraglide which has a complex splined profile.

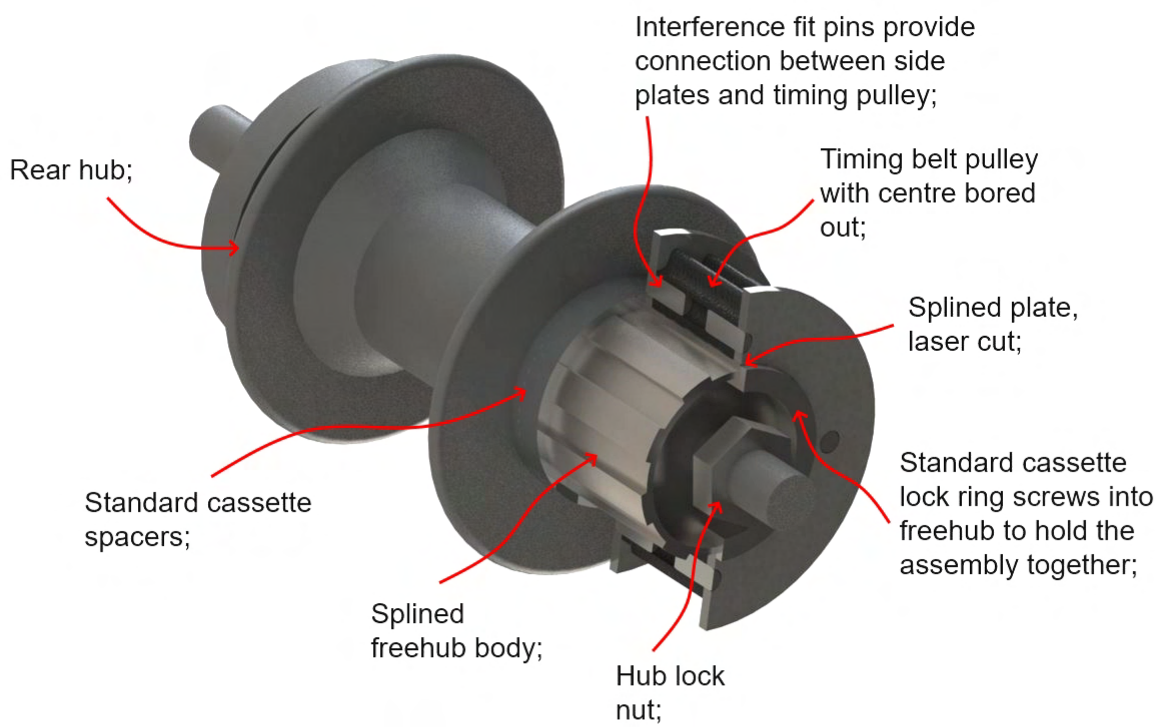

Figure 3.12: Annotated render of the free hub;

For reasons discussed in detail in the testing section it was decided to remake the rear pulley and this presented the opportunity to enact some minor design improvements. In the initial design the laser cut plates were each held onto the pulley by two countersunk M3 bolts and located by two 4 mm dowel pins. Tapping such small diameter holes into steel is challenging and after assembly onto the freehub they served no purpose as the part was properly constrained anyway, hence in the remake the two bolts were replaced by an additional two dowel pins.

3.4 Tensioner Sub-Assembly

The main design requirements of this sub-assembly were to adequately tension the belt slack and to attach onto a standard bicycle mech hanger.

The initial concept for this part was derived from the tensioner used in the static prototype. It consisted of a spring-loaded arm with a pulley on the end. However, it quickly became apparent that, due to the large difference in slack length between the highest and lowest

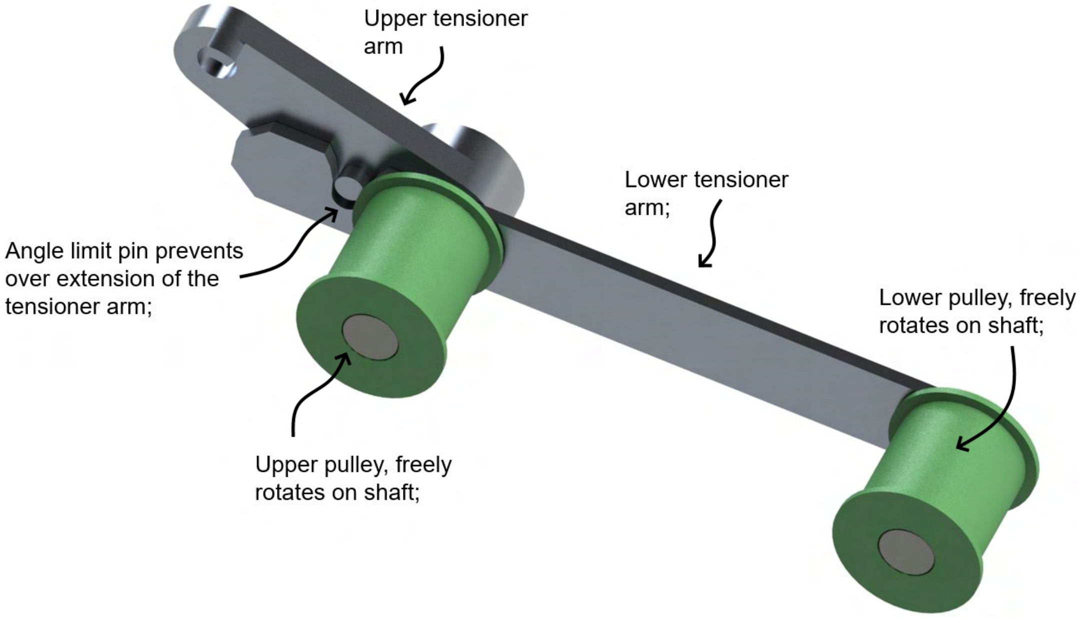

Figure 3.13: Annotated inside view of the tensioner sub assembly;

gears. This approach was insufficient to take up all of the belt slack. Instead two pulleys were used, in a similar manner to a conventional derailleur, to enable the slack to be taken up in a smaller working volume. This was developed into the design shown in the figures 3.13 through 3.15, in which the second arm is sprung via a torsion spring. The torsion spring around which the design was based was acquired by dismantling a rear derailleur. The tensioner allowed the pre-tension of the torsion spring to be adjusted. This was achieved by including a number of equidistantly spaced holes in the upper arm, all of which could be used to locate the end tab of the spring. This adjustment was needed to account for uncertainty in the calculations that determined the required tension. Mounting onto the standard bicycle mech hanger was achieved by simply ensuring that the tensioner top contained the standard mount.

The manufacturing of this part was largely without issue however once it was assembled onto the bike it became clear that although the tensioner pulleys were aligned with the front cones, they were not aligned with the rear wheel pulley. To ensure that the belt alignment changed gradually over the tensioner the first shaft of the pulley was redesigned and made longer to move the pulley into a different alignment.

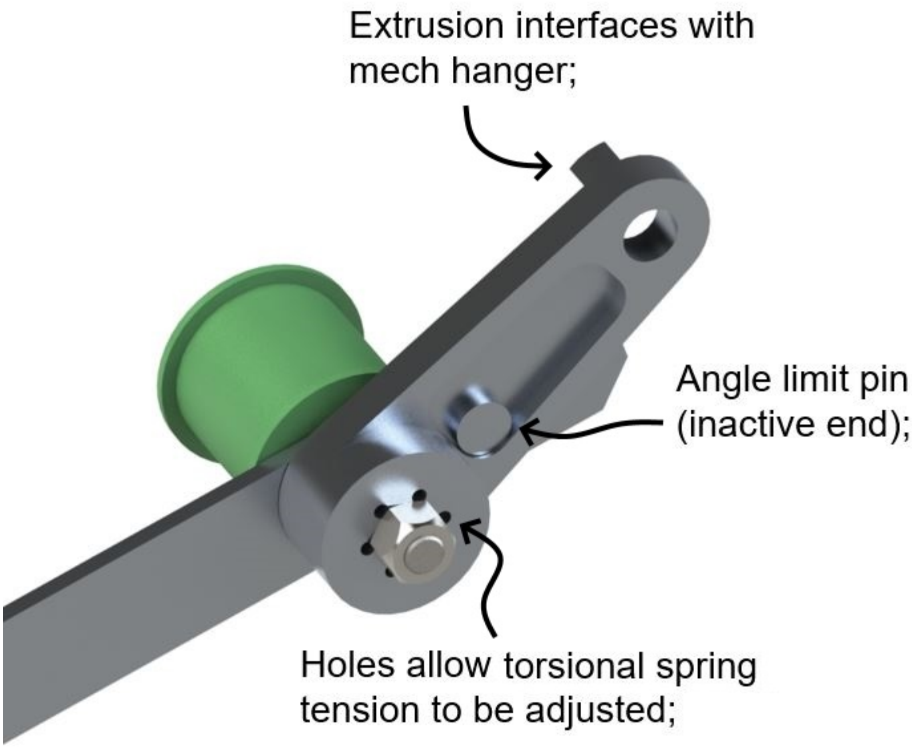

Figure 3.14: Annotated outside view of the tensioner sub assembly;

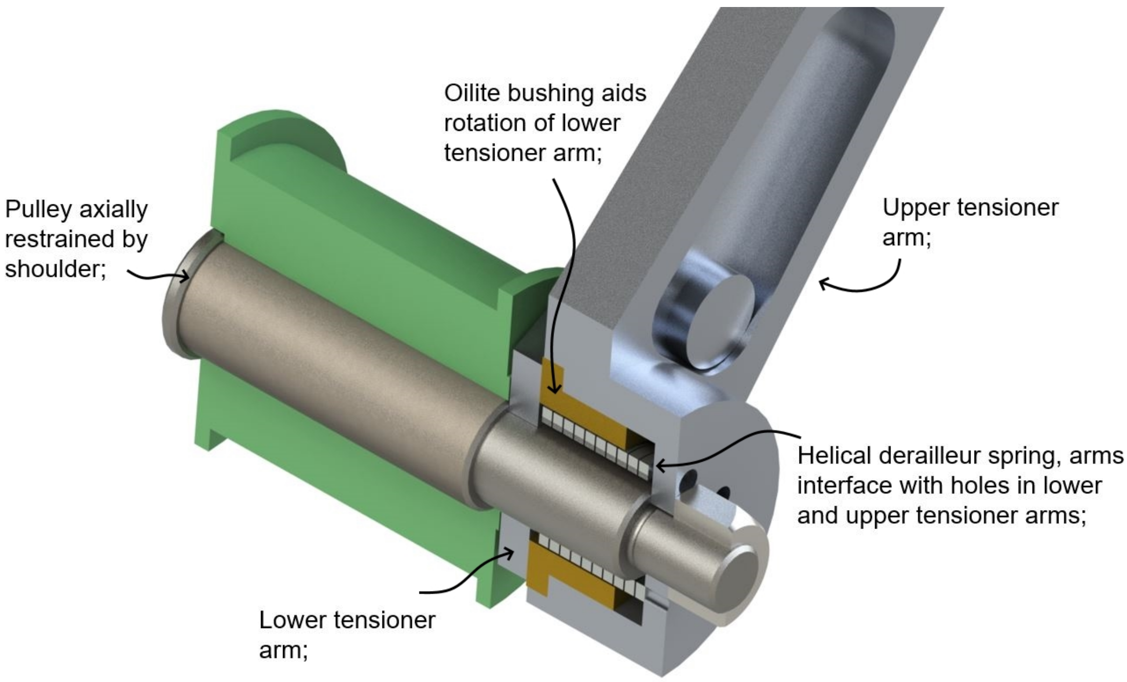

Figure 3.15: Annotated cross section view of the tensioner sub assembly;

3.5 Manufacturing

3.5.1 Material Selection

In order to ensure ease of material supply the project aimed, as far as possible, to use material grades held as stock in ME stores for the prototype. This meant that most components were manufactured from 6082T6 aluminium and a smaller number from EN1A mild steel. Emphasis was placed on weight minimisation to ensure that the weight limit given by the PDS was satisfied; several high load components were manufactured from high performance 7075T6 aluminium rather than EN1A steel. 7075T6 is an aircraft grade aluminium alloy with better strength characteristics and fatigue life performance than 6082T6, whilst having substantially lower density and better corrosion resistance than mild steel. The combination of low weight and good fatigue life performance ensured that this alloy was initially specified for all shafts within the design. Table 3.1 gives the material for all manufactured parts.

Table 3.1: List of manufactured parts;

Part no.	Part Name	Quantity	Material
Bottom Bracket Sub-Assembly			
101	Main Shaft[3]	1	EN1A Mild Steel
102	Fixed Cone	1	6082T6 Aluminium
103	Moving Cone	1	6082T6 Aluminium
118	Moving Cone Backing Plate	1	EN1A Mild Steel
119	Spring Holder	5	7075T6 Aluminium
129	Spring Backing Plate	1	7075T6 Aluminium
Rear Wheel Sub-Assembly			
104	Pulley Side Plates	2	EN1A Mild Steel
105	Pulley	1	EN1A Mild Steel
Tensioner Sub-Assembly			
107	Tensioner Top	1	6082T6 Aluminium
108	Tensioner Bottom	1	6082T6 Aluminium
111	Tensioner Intermediate Shaft	1	7075T6 Aluminium
112	Tensioner End Shaft	1	7075T6 Aluminium
115	Roller	2	Oilon
162	Tensioner Limit Pin	1	6082T6 Aluminium
Belt Guide Sub-Assembly[4]			
150	Main Mount	1	6082T6 Aluminium
151	Cantilever	1	6082T6 Aluminium
152	Guide Shaft	1	6082T6 Aluminium
153	Guide Roller	1	Nylon 6

3.5.2 Manufacturing Plan

Table C.1 given in Appendix C details the manufacturing plan as it was at the start of production of the dynamic prototype. The majority of the information contained within the plan proved to be correct. One significant exception to this was the material lead times for parts made of material not held as stock by ME stores, namely the oilon and 7075T6 aluminium. The estimates used in the creation of the plan were based upon the delivery times given on the suppliers website whereas the actual delivery times were longer. The material lead times for many of the CNC turned parts also proved to be overly optimistic. The length of these lead times was largely driven by the length of the CNC queue however a malfunction of the Haas ST10 CNC lathe in the STW in March resulted in longer than anticipated wait times.

Much of the impact of these longer than anticipated lead times could be absorbed by reprioritising the order of parts for manufacture and through use of contingency in the timeline. However, the volume of delays encountered did eventually impact on testing. Whereas the original plan gave the whole last week of the Easter term to testing the delays resulted in only one day being available prior to the start of the break.

Issues encountered during testing necessitated several parts be remade and resulted in a number of new parts and assemblies which were not included in the original manufacturing plan. A plan was drawn up for the manufacture of these parts and this is given in C.2 also found in Appendix C of this report.

3.5.3 Crank Manufacture

One challenge arising during manufacture of the final dynamic prototype was the question of how to manufacture the square taper holes on the cranks. Custom crank design was necessary for 2 reasons. Firstly, a compact profile with little lateral splay was required in order to keep the Q-factor below the PDS requirement of 200 mm (table 1.1). Secondly, a drive side crank with right-handed threading was required. A commercial non-drive side crank would be inappropriate because of the left-handed thread which would loosen the pedal during pedalling and a drive-side crank is designed to accommodate chainrings which are superfluous to the group's requirements.

The issue then confronting the group was the question of how to manufacture a square tapered hole using the CNC machine tools available. The smallest end mill available had a 4

[3]The main shaft was originally designed in 7075T6 aluminium but for reasons discussed subsequently was remade in EN1A mild steel.

[4]The belt guide sub-assembly was an additional assembly designed to resolve issues detected in initial testing and is discussed within the tensioner sub-assembly discussion.

mm diameter and so the taper had to be designed to overcut in the corners to allow clearance for the male part.

3.6 Testing

This section will detail the testing performed on the final prototype. It will cover the issues that arose with the prototype post-production and will describe the modifications implemented to resolve said issues. It will also include a brief summary of quantitative tests that would have been performed if the prototype was fully functional.

The testing performed was qualitative; the project did not progress to the point where quantitative testing would have provided significant useful information. Issues with the prototype could be easily identified through a de-bugging process- operating the prototype, identifying the area of failure and suggesting a solution.

The testing performed can be split into two parts: some first pass testing performed in March which identified some serious issues and more extensive detailed testing performed in late May through early June.

3.6.1 First Pass Testing

Initial testing was performed upon assembly of the prototype. An attempt was made to ride the bicycle on a level surface. The design initially appeared to be successful in transmitting power to the rear wheel. After around 15 metres it became increasingly difficult to pedal until eventually the bottom bracket assembly completely seized. Initial inspections indicated that this was a result of the fixed cone of the Reeves drive impinging on the bottom bracket housing. These issues are discussed in more detail within the design commentary, but the issues identified and solutions chosen are discussed below.

The main shaft was designed so that the cone assembly screwed onto a M22 left-handed thread, which was tightened by the operational torque against a shaft shoulder sitting just proud of the plastic bottom bracket race. Due to imprecise measurement of the bottom bracket width, the shaft was too short. This resulted in the thread tightening the cones against the race, causing it to fracture. This fracture then allowed the cone to interact directly with the bottom bracket housing, which does not rotate, hence the whole assembly was prevented from rotating. The damaged caused during this testing is shown in figure 3.16.

This was resolved by manufacturing a new shaft with correct lengths and tolerances to locate the cones on the shaft shoulder rather than the plastic bottom bracket bearing casing.

Figure 3.16: Damage caused to the bottom bracket by the first pass testing;

This resolved the issue. The original bottom bracket was significantly damaged, therefore a new one was purchased as a replacement.

A constant challenge in the design of this prototype was ensuring that the cones at the bottom bracket and the pulley on the rear hub were properly aligned. This would not be an issue on a custom designed frame geometry (as discussed further in section 4) but resulted in the 'belt-line'[5] being imperfect on this prototype. Correctly aligning the tensioner between these two points was difficult. This misalignment is illustrated in figure 3.17 (b).

It rapidly became apparent that for the prototype the group had got this wrong and that the mis-match in belt-line was sufficient to cause the belt to frequently de-rail. Two improvements were identified to mitigate this issue: adjusting the position of the first pulley on the tensioner and widening the rear pulley.

In order to move the intermediate tensioner roller it was necessary to re-manufacture the intermediate tensioner shaft. It was lengthened and a circlip groove was added to locate the now distanced pulley. This is shown in figure 3.17 (a). Whilst remaking the shaft did not entirely remove the belt-line issues, it did reduce belt misalignment on the slack side. The belt line changed more gradually over both tensioner rollers, rather than all of it between the rear

[5]the alignment between the rear pulley and the cones

pulley and the intermediate tensioner roller.

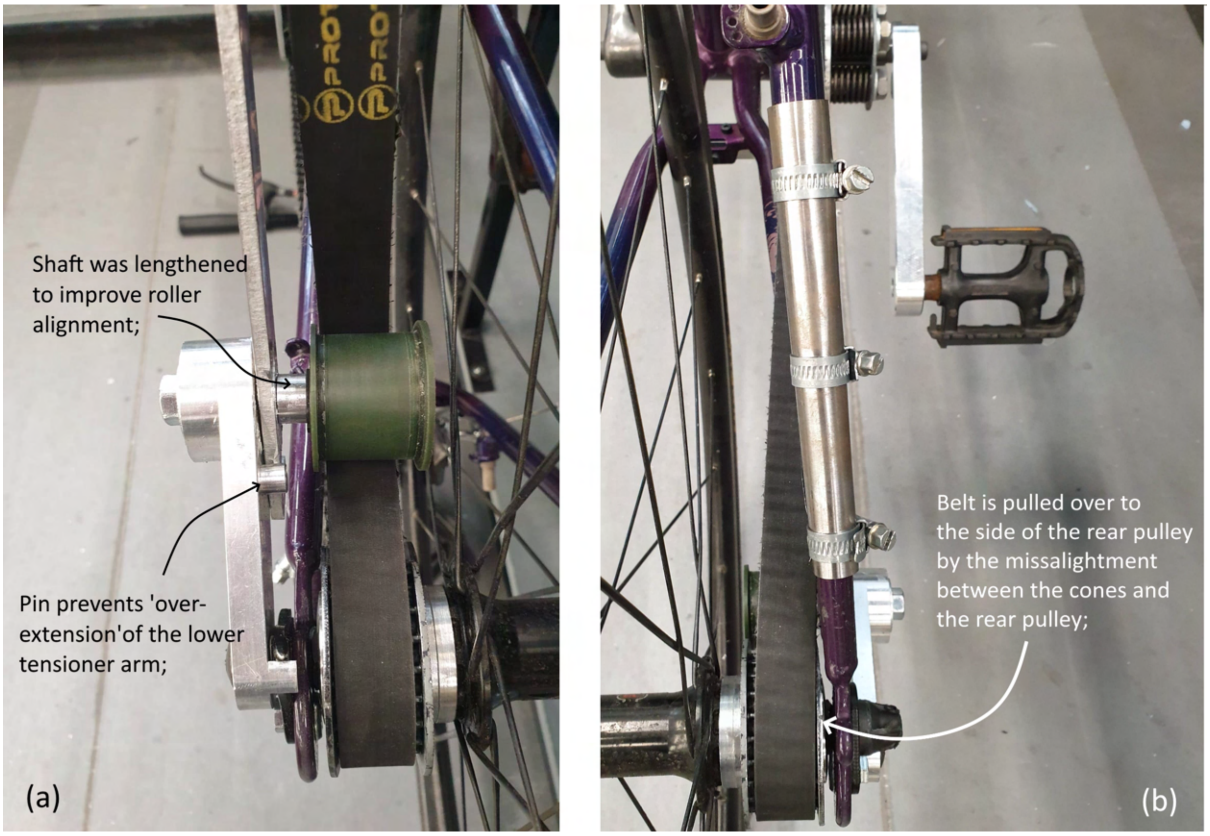

Figure 3.17: (a) Modification to the rear tensioner assembly; (b) Illustration of the misalignment between the cones and the rear pulley;

Initially, the rear pulley was designed to fit the belt exactly (20 mm). Widening the rear pulley meant that a greater degree of belt misalignment could be accommodated without the belt interacting with the pulley side plates and causing the belt to de-rail.

It became apparent even before the start of initial testing that the cones did not properly fit the belt. The belt would not sit easily in the top of the cones as they had been designed for the nominal belt width rather than a clearance. During the initial testing it was postulated that this poor tolerancing was causing the belt to tip over as the gear ratio reduced.

This issue was easily rectified by shimming the inner surfaces of the cones to move them apart enough to provide clearance for the belt at maximum radius.

3.6.2 Detailed Testing

Once the improvements made as a result of the initial testing had been implemented, a more rigorous testing programme was initiated. The testing carried out in this section was performed on a turbo trainer. This meant that the bike was static, enabling conditions to be better controlled and allowing easier observation of the transmission during operation. This testing was iterative and modifications were made to the bike as testing progressed.

Frequently during testing it was observed that the belt would get wedged deep between the cones and would subsequently not disengage and exit at a tangent towards the tensioner. This effect is shown in figure 3.18. The high cone force necessary for the gear change to occur over the correct range of torques exacerbated the issue, as did belt flexibility. It was postulated that increasing the force applied by the tensioner might rectify the issue. The pre-tension in the tensioner was therefore increased. This had some success in solving the issue, meaning that it occurred less frequently, however when it did occur it damaged the spring. Different solutions to the issue were therefore attempted.

Figure 3.18: A significant issue with the initial design was that the belt would become trapped in the cones;

The arms of the spring were bent back into their previous position. Redesigning the tensioner to accommodate a larger torsion spring was considered, but the group decided that it was unclear from available information whether the new construction would have a significantly higher failure load. Additionally, this redesign would have necessitated new CNC requests and there was concern that the lead time for manufacture would result in insufficient time remaining to pursue other potential solutions if the new spring did not rectify the issue. The original spring was therefore used in the tensioner resulting in sub-optimal belt tension; other potential solutions were then explored.

It was thought that the addition of an auxiliary roller close to the cone diameter would help to force the belt to leave the cones appropriately. The roller was mounted on two pieces of aluminium which attached to the bicycle's kickstand mount. This did not have the desired effect. Instead of helping to disengage the belt and cones, it simply caused it to happen at a steeper angle due to the flexibility of the timing belt. This is shown in figure 3.19.

Figure 3.19: Belt catcher introduced in attempt to rectify issue;

An adequately tensioned belt would be able to avoid the problem of the belt being sucked up into the cones as the bottom belt tension would be enough to pull the belt tangentially from them. The tensioner was not able to provide sufficient belt tension hence it was decided that the angle of the tensioner's bottom arm angle should be limited by a pin on the upper arm to prevent the belt from being able to travel too far around the cones. This is shown in figure 3.17 (a).

This solution did have the desired effect of limiting the belt's travel. Unfortunately it resulted in a high force being applied at the pivot between the tensioner upper and lower arms. This high force damaged the intermediate shaft and meant that the lower arm no longer rotated freely around the intermediate shaft.

Discussion amongst the group concluded that proper rectification of this issue would require a substantial redesign of the tensioner as it had now become clear that the tensioner would experience higher forces in operation than initially anticipated. This would need to be pursued to further develop the design.

A further issue with the design was a persistent light jolting of the drivechain whilst in operation. The issue originates from the belt initially contacting the timing pulley at an angle from the tensioner pulley meaning that the pulley and belt do not always mesh properly. This caused a jolt for the rider. The angular entry of the belt is inevitable because of the necessity for the prototype to retrofit onto an existing bicycle. This forces the design to utilise an out-of-line transmission assembly arrangement.

Whilst the prototype had to work within the constraints of typical bicycle geometry given lack of access to frame-building facilities or expertise, it would be possible to eliminate the issue of belt jolting if the belt-line could be designed as straight. In order to achieve this, the outer face of the bottom bracket should be moved towards the centreline of the bicycle to reduce the Q-factor and the rear timing belt pulley should be moved away from the centreline to be in line with the contact faces of the cones when closed. This would require a larger rear drop out spacing so that the timing pulley assembly does not interfere with the seat stays on the bicycle frame.

3.7 Evaluation

This section aims to evaluate, for each PDS criterion, whether that criterion was met. The PDS is tabulated in table 1.1 of this report; this section discusses criteria in the order they are given in that table. Where a criterion has not been met, some discussion is provided as to why it was not met and what modifications could be implemented on future product iterations to achieve compliance.

Customer

The first identified requirement under customer needs was that the product must be compatible with a standard 68mm ISO threaded bottom bracket, as detailed in ISO 6696:1989 [18]. This requirement was met as the design utilised a Shimano Hollowtech II series bottom bracket, which conforms to the standard. A second requirement identified under customer needs was to fit a standard 130mm spaced rear dropout. This requirement was met, with the caveat that although the product was mountable there were some belt-line issues, as discussed in section 3.6 of this report. Both requirements were included simply to ensure that the product was compatible with a standard bicycle for testing.

The only identified competition requirement was that the product must be an automatic continuously variable transmission thereby simplifying bicycle operation. This requirement formed the basis of the product and arose, as detailed in the introduction, from an identified gap in the market. The requirement was met, with the dynamic prototype adequately demonstrating that transmission operated both automatically and continuously, as detailed in section 3.6.

The prototype as been shown to operate without any user input other than pedalling as detailed in section 3.6 of this report.

With regard to performance, the first design requirement identified in this section is that the transmission must be capable of achieving gear ratios from 1.2:1 through to 2.8:1. The geometry of the cones was chosen to ensure that this requirement could be met however, due to the issues encountered during testing, it was not possible to confirm whether the constructed prototype could achieve the full theoretical range of gear ratios.

The PDS also detailed as a requirement that the transmission must be able to transmit powers up to the 95th percentile acceleration. Whilst the transmission was designed to meet these requirements, design issues encountered during testing, which precluded quantitative testing, prevented it from being confirmed whether this was the case for the manufactured product. The final performance requirement was that the equilibrium cadence of the transmission be between 60 and 90 RPM. During testing a stopwatch was used to check the cadence and confirmed that the equilibrium cadence lay around the 70 RPM value with some variation.

Whilst two of the performance requirements were not met this was not the result of any fundamental flaw in the design, but rather was the result of a number of minor issues preventing full successful operation of the transmission. It is judged likely that, were sufficient time available within the project to produce another iteration of the product, then significant improvements would be observed.

All of the requirements in the environmental section fall under the remit of ensuring that the transmission is suitable for outdoor operation. To that end they require it to: operate in a -5$^\circ\mathrm{C}$ to 30$^\circ\mathrm{C}$ temperature range, operate when wet and avoid crevices in which debris can collect. The product had no issues with dry operation at either normal temperature or the upper-bound ambient temperature of 30$^\circ\mathrm{C}$. In wet weather the system did operate, but with a notable degradation of performance due to an increased propensity of the belt to slip. It was not practicable to test the low ambient temperature operation as removing the cones from the bicycle after use proved challenging and hence prevented their temperature being lowered. However, given that at the low ambient temperature the cones would quickly become covered in condensation the performance at this temperature is anticipated to be similar to that in wet weather.

The design did not avoid crevices in which debris can collect, however this was a conscious decision made due to a change in the envisaged use of the product. Initially it had been anticipated that this product would be a final prototype in the development of after-market retrofit product, hence it needed adequate ingress protection to ensure its safe operation. Once it was realised the product might be better suited to purpose-built city hire bicycles it was no longer deemed an essential requirement.

The PDS required that the transmission did not weigh more than 5kg. This requirement was met by the initial design; the design did not then exceed the requirements due to the various modifications and additions made during testing. The final mass of the entire system was approximately 4.2 kg.

Two requirements were given under this heading; that the crank length be 170mm and that the Q-factor be less that 200 mm. The cranks were designed specifically for this transmission and hence ensuring that they were 170 mm long was a trivial task and this requirement was met. The Q-factor was confirmed from both the CAD model and the manufactured product to be exactly 200 mm; hence this requirement was also met.

Life

The PDS required that the product life should be four years. This requirement was again made prior to the change in product use. Whilst a product life is necessary for production products the end use of this product meant that this requirement was no longer relevant. The final requirement in this section of the PDS was that the product required no maintenance over its service life. This requirement was not met as issues occurring during testing required frequent maintenance.

Producer

The requirements detailed within this section of the PDS are as follows: the product must be adaptable for mass manufacture at 10,000 units per year, the product must be manufacturable using Imperial College facilities and vendor approved components and that the project cost must be under £1,000. Whilst a detailed redesign of the product for mass manufacture has not been carried out, discussion in section 4 of this report indicates that this is feasible; hence this criterion is met. The product was successfully manufactured using the facilities available in the STW and components were acquired from approved vendors; hence this requirement is met.

Regulatory

As required by the PDS, the transmission was designed in accordance with and assessed against relevant ISO standards. The bottom bracket spindle did not conform to the relevant ISO standard, this has been identified and justified elsewhere in this report (3.2.1). As detailed in those sections of this report whilst the product as manufactured does not meet the standard the design does. Material choices to reduce lead times resulted in poorly performing components which nonetheless perform adequately for the testing duties of the prototype. The PDS requires that any plastic components used in the design be recyclable at end of life. The only non-recyclable plastic component used in the design is the belt, due to the reinforcing fibres within it. This is unavoidable and it means that this PDS requirement is not met.

Financial

The expenditure by the group on the dynamic prototype came to £532.58 and a detailed breakdown of this is presented in appendix A of this report. The expenditure on this prototype was in excess of the value budgeted for in the quality plan[1]. The main reason for this was the fact that many of the components had to be manufactured or purchased twice due to damage in the initial testing. Fortunately, due to sufficient contingency the project was still delivered within budget at £872.68; hence this PDS requirement was also met.

4. Future development

4.1 Design Options

This Design, Make and Test project has taken the design development as far as a full scale, dynamic, bicycle mounted prototype. Further development would be necessary to develop a product which constitutes a market-ready design. Although the team has demonstrated the merits of the ACVT, there are several elements that require further development to develop a viable city hire bicycle:

1. The bicycle frame should be re-designed so that the belt can be installed without cutting the frame. Options include cantilevering the rear wheel (as explored in section 4.2) or having a removable section of frame;
2. The key mechanisms of the transmission should be guarded to prevent malfunction due to fouling of the mechanism by dirt or water. This guarding should be robust enough to protect the transmission from both accidental and malicious damage;
3. The prototype included some adjustability to enable fine tuning. These elements should be permanently fixed to achieve a one-size-fits-all product. It would also be prudent to utilise locking nuts to ensure that the bicycle could not be dismantled without specialist tools;
4. The frame size should be approximately the population average- around a 53 cm frame. The handlebars should be at a relaxed height and the saddle should be adjustable but not removable. All these elements are not particularly novel and are features of many public hire bikes available now;
5. The tensioner needs redesigning to increase applied force to prevent slack side belt issues. The drivers behind this change are discussed in detail within the testing section.

4.1.1 Modification for Mass Manufacture

The manufacturing and assembly methods for many of the transmission components may need to be rethought to reach a design suitable for mass manufacture. The high reliance on both CNC and manual machine tools during the manufacturing of the prototype would result in an uncompetitively expensive production product. Manufacture of the fixed and moving cones was a lengthy process. However it would be possible to rethink the design of these components to allow them to be manufactured by stamping of sheet metal. This approach would significantly reduce the manufacturing lead time and hence cost of the part. This would have the added benefit of reducing the component weight. Another long lead time component is the main shaft. In its present form this would inevitably require some machining due to the tolerances required for it to interface satisfactorily with the hollowtech bottom bracket. It may be beneficial to design a custom bottom bracket with a suitable integral shaft. This is unlikely to remove the need to machine some surfaces of the shaft though as it will still need to interface with some bearings.

The design of the rear pulley cannot be simplified much further. However at production volumes a correctly dimensioned pulley could be specified from the supplier, removing the need to machine a standard pulley to the correct dimensions. There is not deemed to be much benefit in discussing possible modifications to tensioner components because, as discussed previously, they will require a substantial redesign to create a commercial product.

The prototype contained a lot of components which were connected by screw threads to allow dismantling for modification and inspection. In the production design many of these could instead be welded, which would reduce the complexity of these components.

4.2 Cantilever Calculations

One possible frame option which would allow the remove the need for cutting the seatstays of a frame is cantilevering the rear wheel with a solid chain stay. This is not a new concept, the Boardman Lotus was the most prominent example of a legal track bike utilising a rear wheel cantilever. More recently, the MoBike [17] also successfully integrates a rear cantilever on one side of the rear wheel.

The ISO safety standard [25] for the required strength of the rear wheel support specifies that a force of 1200 N must be applied over 50,000 cycles 70 mm aft of the intersection of the seat tube and saddle as shown in figure 4.1.

For the purposes of this calculation, it was assumed that the applied force causes an

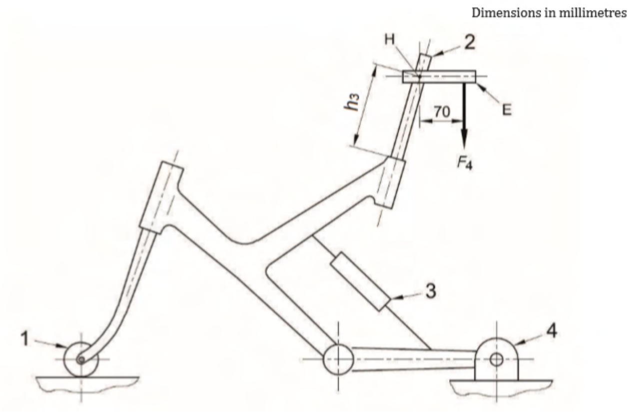

Figure 4.1: Schematic showing the test geometry of the cantilever specified by ISO 6696:1989 [18];

equal force at the rear wheel axle. This is a maximum for any given bicycle geometry as the front wheel condition is simply supported. This causes a linear moment distribution in the chain stay which is maximum at the join with the bottom bracket. It is important to emphasise that the suspension element or seat stay labelled ‘3’ in figure 4.1 is necessarily absent. The force at the rear wheel axle causes a maximum moment of 504 Nm when a chain stay length[1] of 420 mm is used. Defining the chain-stay geometry to interface with a standard 68 mm bottom bracket shell and allowing sufficient tyre clearance for a 32 mm tyre, the maximum stress the chain stay has been calculated to be 130 MPa. This figure is well below the fatigue strength of many engineering alloys. For example 7075T6 aluminium alloy has a fatigue limit of approximately 150 MPa and recent research by engineers at University of California, Los Angeles have developed new welding processes for this material in the context of bike frame manufacture [26].

With the chain stay element being the only novel aspect of this frame, the other elements of the concept can be designed with relative freedom to conform to standard geometry or a more futuristic sweep design.

[1] from the centre of the axle to the centre of the bottom bracket

5. Conclusions

5.1 Project Conclusions

The aim of this project was to develop an automatic continuously variable transmission for use on a commuter bicycle. A range of ideas were explored before a concept based upon a mechanically controlled half Reeves drive was selected for further development. As there are no products currently on the market using a half Reeves drive in a similar application it was decided to develop a simple static prototype to prove the concept. A hand-wound, static prototype was therefore designed and produced.

The prototype demonstrated that the concept worked as theorised . However, it operated at a far lower torque range than would be experienced by a bicycle transmission and hence, whilst it provided confidence in the concept, it did not adequately prove the feasibility of the design. The static prototype was not intended to be interfaced with a bicycle frame and hence did not address the design challenges presented by this task.

The next design challenges were; to address how to up scale the transmission from the low torque prototype to one capable of transmitting the torque of a commuter, and to determine how to retrofit the system to a bicycle. Once a feasible design had been developed that operated within the correct torque range, it was manufactured and mounted on a bicycle for testing. A number of minor modifications were required to rectify design oversights and overcome unforeseen challenges encountered at the very start of testing. Following completion of these modifications an extensive testing campaign was embarked upon which by and large proved that the design worked. Whilst the concept was proven viable, a number of issues were identified over the course of both the development and testing which would need to be overcome prior to the development of a commercial product. There are significant belt misalignment issues.

Several of the issues identified relate to the frame geometry and would all be addressed by designing a frame specifically for this transmission; as would be the case if it were deployed on city hire cycles. A first issue that could be addressed by adjusting the frame geometry is that of belt line misalignment. In the dynamic prototype, the crank and rear pulley did not align. This was unavoidable due to the constraints of the frame we selected, resulting in meshing issues at the rear pulley and a tendency for the belt to derail from the tensioner. Aligning the crank and rear pulley would remove this issue. The second frame geometry issue was that a conventional double diamond bicycle frame cannot accommodate a continuous belt; mounting a chain onto it relies on splitting the chain which is not possible with most reinforced belts. For the prototype this issue was overcome by splitting and then re-joining one of the seat stays however this would not be suitable for commercial use. The frame geometry employed would need to have either a cantilevered rear wheel, as many MoBikes do [17], or have a removable joint between the seat stay and chain stay, such as those manufactured by Schindelhauer [15].

Testing identified that the transmission operated better with higher belt tension on the slack side. This belt tension was provided by the tensioner. However, the tensioner design was limited in the amount of resistive force it was capable of applying due to the weak torsion spring. Torsion springs are difficult to acquire so it was judged prudent to concentrate improvement efforts elsewhere. A production product would need a redesigned tensioner capable of operating at higher tensions but would also have the advantage of manufacturing in large enough volumes to be able to specify custom springs at a reasonable unit price.

The design adequately proved the concept of an automatic continuously variable transmission for a commuter bicycle. The prototype produced also allowed the identification of issues to be rectified to develop a commercial design, enabling viable solutions to be proposed.

5.2 Individual Evaluation

5.2.1 Thomas Fisher

Whilst our prototype is not fully working, as this project draws to a close I feel it went rather well. The bicycle drivetrain is a system that has been iteratively improved by multinational companies for decades, which meant that there is relatively little room for innovation. That said I think our novel approach to the problem has been an interesting and enjoyable project to work on. I am a bit disappointed that our prototype is not completely functional and whilst it is easy to say in retrospect, I think that a bit more attention to detail in February would have gone a long way to solving the belt line issues.

As a group I feel we worked well together. I think this can be attributed to a range of factors, firstly the fact that all group members have previously undertaken a year in industry, which meant that we all had similar expectations as to the quality of work and how to behave in a professional environment. Secondly, as a self proposed project we were all working on a project that we were genuinely interested in and this resulted in a desire to make the project a success. As a team we were already familiar with one another's work prior to the project commencement. This meant we could skip over the forming and storming phases of the team development stages and play to one another's strengths from the outset.

As would be expected there were some disagreements within the group throughout the project, but I think these were resolved well with nobody walking away from meetings feeling personally attacked. It was definitely to our benefit that we knew each other well and could be frank with one another without the risk of causing offence.

As project manager, I worked hard to ensure that everyone knew what their responsibilities were at any given time and I think this was broadly successful. Project management is something I have experience in, from both my time working in industry and managing Imperial College Symphony Orchestra. I think I performed the role well achieving an even division of work amongst the group and avoiding a rush for project deadlines.

Another important function I performed in the group dynamic was encouraging the group to stop when faced with a problem. To take a step back and consider the problem more fully. There were a number of examples where in my opinion this resulted in us implementing far more effective fixes to problems that the first one that sprung to mind.

I learned to not let team members take on too many tasks at once in this project, something I think we were all susceptible to doing at different points in the project. For me the biggest example of this was volunteering to typeset project reports in LaTeXwhich improved the presentation of them considerably, but took a significant amount of time. I have a good attention to detail and think that made me an effective reviewer of both reports and engineering drawings.

In hindsight, I would have taken a bit longer to think about the possible effects of the imperfect belt line, which we did identify quite early in the project, but failed to properly fathom the consequences of.

5.2.2 Henry Hart

In evaluating this project I will discuss where things went well and badly with regard to both the teamwork and final engineering design.

Good Points

The primary benefit of self-proposing a project was that the team could be chosen by the respective members and this was a significant motivation to come up with a good project. The members of our team have worked together before on various matters, related or otherwise to our university work.

While in many ways our team shares some characteristics, our project-related skills in fact vary quite widely. This was known in advance and is reflected in the team roles that each of us took on cross-reference team roles. Generally, it has been to our advantage that each member can bring his own speciality in which he may be trusted to do the best job whether that be report formatting, engineering calculation, manufacturing or report editing.

The one significant feature that unites all members is that we all worked in industry for a year before university and every summer since. Additionally, we all score in the top half of the year group academically; an advantage that few groups can claim. I genuinely think that this has put us all on the same page with respect to quality expected and attained.

The design was originally intended to be a Reeves drive, and ended up being so. This is not because other ideas were dismissed flippantly. The Reeves drive was not decided upon until a meeting on the 28th of October based on a morphological analysis and hours of constructive criticism of various designs.

Bad Points

There are have been few drawbacks of this group in my opinion. If forced to pick one out, there have been some arguments particularly between the three members who live together although generally we have been able to brush it off quickly.

It is indeed the case that our final prototype does not function smoothly. It does serve to demonstrate the concept but is nowhere near being suitable for real commuting in its current state. It is possible that our project was initially a little too ambitious, particularly the aim to exclude electronic operation. This is an inevitable danger of self-proposed projects.

This project has been far more of a success than a failure. While the headline result is a non-fulfilment of the project aim in strict terms cross-reference aims, the project has been conducted in a professional manner similar to what I have experienced in two engineering firms. The output is suitable for further work and the core concept is certainly capable of success given more resources and time.

5.2.3 Ieuan Swainston

Designing a novel bicycle transmission system, when the current method has been refined and improved over many decades, was always going to be challenging. I believe that the group largely achieved its goal, of producing an automatic continuously variable transmission for a bicycle, although some minor improvements would be required to produce a fully functioning product ready for the end user.

The project was self-proposed giving the group the opportunity to self-select team members; rather than being randomly assigned. This had the advantage of meaning that we were all already reasonably aware of each other's strengths and weaknesses right from the start of the project. This made both assigning tasks appropriately and identifying any skills shortfalls easier. This is one aspect of the project in which I believe our group performed well, we recognised some team members greater knowledge in some areas and trusted their opinions accordingly. For example; I recognised that other team members had greater knowledge of stress analysis than myself whilst other team members recognised that I knew more about manufacturing.

I learnt a lot from the project, most notably about effective time management. Initially the group proposed overly ambitious timescales, largely as a result of underestimating lead time, as the project progressed the accuracy of proposed timescales improved. As head of manufacturing I gained an appreciation of project management, prioritising manufacturing tasks as appropriate, and assigning certain people to make certain parts to ensure the best possible outcomes. I was grateful of the opportunity to improve my manufacturing skills and to gain a greater understanding of the limitations of both my own manufacturing ability and the workshop machine tools.

The very wide project brief meant that many different potential solutions were available to us. Over the course of the project there were frequent differences of opinion about the best way forward or the best solution to particular problems. The subsequent discussions amongst the group allowed me to develop my communication skills, as I had to succinctly explain why I believed a certain course of action was appropriate. In most discussions everyone in the group had different initial opinions, the process of discussing to bring multiple people over to one point of view increased my confidence that we were making the best possible decisions. Whilst it was at times frustrating to encounter so many differing strong opinions, it allowed us to produce a better and more well thought out product.

5.2.4 Charles Titmuss

Overall the project went well. The group designed, made and tested a static prototype and a dynamic prototype, allowing us to prove the concept of a half-Reeves drive as a viable alternative to other types of transmission.

The project was completed in a timely and organised fashion, with a fairly consistent workload over the whole project. The only points where this could be said to be untrue were where deadlines approached, and it was necessary to raise the tempo of work for a brief period of time. There were multiple reasons for this. Firstly, the fact that all group members had previously completed a year working in industry meant that we had experience in open-ended, self-driven projects where time management is key. Furthermore, the good use of a Gantt chart, taking into account the other commitments that the group had ensured that the workload was never overwhelming. Finally, the regular meetings and accurate and concise minutes provided with our supervisor ensured that the project remained focused and on track, and that everyone knew what they were meant to be doing at any one point.

An improvement for the group would be to have more group meetings when writing the final report, where each team member could review the report and suggest sensible additions

My main role within the group was as group secretary. This involved ensuring that the group was appropriately coordinated by releasing weekly minutes, as well as taking on a significant amount of responsibility for every single written assignment in an editorial role. An improvement I could have made while performing this role was that the minutes should have been more consistently released, as I would sometimes fail to compile and release them on the same day each week. However, the work I undertook as the editor allowed for all submissions to be delivered with clarity and accuracy.

I also took responsibility for the assembly drawings. As this was a largely mechanical project, the assembly drawings were a crucial part of explaining the project to a wider audience. I learnt a lot whilst completing these drawings and I shall never use an exploded view ever again.

The main lesson I have learnt from this DMT project it is imperative to have good time management and to play to each of the group members strengths. The group worked very well together throughout the year, with a roughly even split of work completed on the project between each member overall.

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List of Image Sources

Figure 1.1	`https://www.statebicycle.com/products/silver-frame-set`
Figure 1.3	`https://cycling-passion.com/breadwinner-lolo-campagnolo-super-record-eps /breadwinner-lolo-campagnolo-super-record-eps-drivetrain/`
Figure 1.4	`https://www.instructables.com/id/Rebuild-a-bicycle-rear-hub/`
Figure 2.1(a)	`https://cornexchangemanchester.co.uk/app/uploads/2017/07/mobike-2.jpg`
Figure 2.1(b)	`https://www.schindelhauerbikes.com/en/Bikes/Previous-Year/Siegfried.html`
Figure 2.1(c)	`https://cdn-images-1.medium.com/max/1600/1*ABNGWEzpLm8Rnpa0cTN4Ww.jpeg`
Figure 2.1(d)	`https://www.fallbrooktech.com/nuvinci-technology`

A. Expenditure

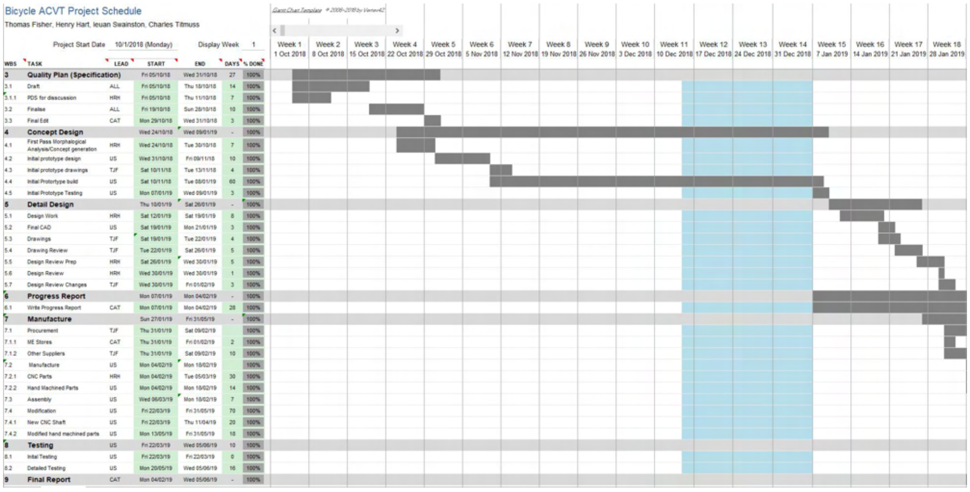
Table A.1: Summary of expenditure for the static prototype;

Order Supplier	Date	Quantity	Contents of Order	Cost
ME Stores	13/11/2018	1	Ø200 mm aluminium round x 80 mm	£50.18
ME Stores	13/11/2018	7	M8 studding x 200 mm	£3.36
ME Stores	13/11/2018	1	Ø25 mm aluminium round x 270 mm	£0.70
ME Stores	13/11/2018	1	Ø30 mm aluminium round x 50 mm	£0.21
ME Stores	13/11/2018	1	5 mm steel key	£0.32
ME Stores	13/11/2018	1	2 mm mild steel sheet 50 mm x 50 mm	£0.10
ME Stores	13/11/2018	1	Ø20 mm aluminium round x 270 mm	£0.55
Dore Metals	19/11/2018	2	Aluminium tube 4 m	£80.00
Dore Metals	19/11/2018	1	Carriage	£22.00
Direct Plastics Ltd.	19/11/2018	1	45 mm nylon 250 mm	£7.20
Direct Plastics Ltd.	19/11/2018	1	65 mm nylon 250 mm	£9.90
Direct Plastics Ltd.	19/11/2018	1	Carriage	£5.95
ME Stores	19/11/2018	1	60 mm aluminium round bar x 40 mm	£0.61
Direct Plastics Ltd.	05/12/2018	1	45 mm nylon Bar 0.25m	£4.46
Direct Plastics Ltd.	05/12/2018	1	65 mm nylon Rod 0.25 m	£9.32
Direct Plastics Ltd.	05/12/2018	1	Carriage	£14.50
ME Stores	14/12/2018	1	Ø200 mm aluminium round x 70mm	£43.91
ME Stores	14/12/2018	2	25 mm aluminium round x 270mm	£1.40
Chem Eng Stores	17/12/2018	1	1 m Ø25 mm silicone tubing thin	£1.00
Chem Eng Stores	17/12/2018	1	1 m Ø25 mm silicone tubing thick	£1.00
Amazon	17/12/2018	1	Snowmobile belt	£37.00
Arthur Beale	07/01/2019	1	Rope	£16.74
RS	14/01/2019	1	Timing belt	£16.51
Arthur Beale	15/01/2019	1	Rope	£13.20
			Total	**£340.10**

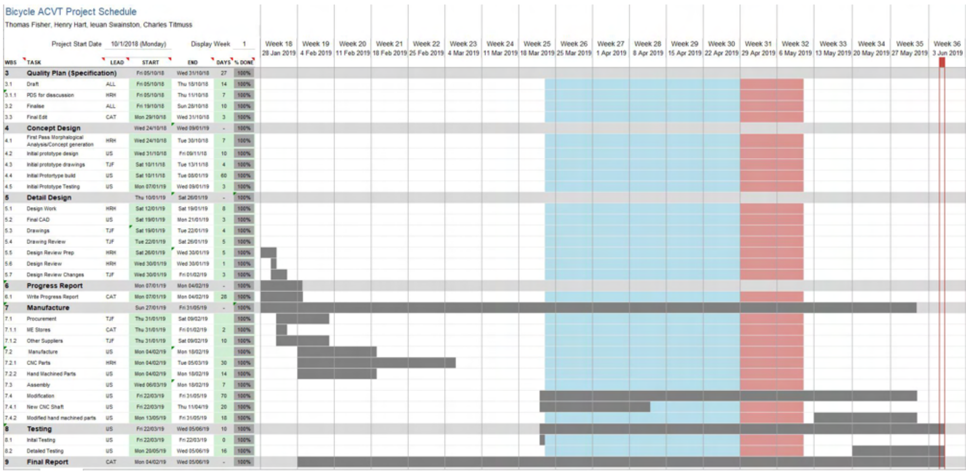
Table A.2: Summary of expenditure for the dynamic prototype;

Order Supplier	Date	Quantity	Contents of Order	Cost
Ebay	16/01/2019	1	Shimano Hollowtech Deore ii bottom bracket	£13.00
Ebay	22/01/2019	1	Shimano Ultegra Hollowtech ii bottom Bracket	£15.80
Ebay	24/01/2019	1	Bottom bracket spanner	£3.98
ME Stores	30/01/2019	1	Ø200 mm aluminium round x 100 mm	£62.73
ME Stores	30/01/2019	1	Ø40 mm aluminium square bar x 100 mm	£0.67
ME Stores	30/01/2019	1	Ø30 mm aluminium square bar x 100 mm	£0.41
ME Stores	30/01/2019	1	2 mm steel sheet 85 mm x 85 mm	£0.29
ME Stores	30/01/2019	1	2mm steel sheet 70 mm x 70 mm	£0.20
ME Stores	30/01/2019	2	40mm aluminium square bar x 250 mm	£4.08
LEZ cycles	31/01/2019	1	Bicylcle spacers	£14.00
LEZ cycles	11/02/2019	1	Test bicycle	£50.00
RS	11/02/2018	1	Timing belt 1.28 m length x 20 mm width	£15.60
RS	11/02/2018	1	Steel timing belt pulley	£13.04
MetalEx	13/02/2019	1	Ø25.4 mm aluminium round x 600 mm 7075T6	£34.00
MetalEx	13/02/2019	1	Ø31.75 mm aluminium round x 250 mm 7075T6	£23.85
RS	13/02/2019	1	Galvanised spray paint	£11.64
ME Stores	14/02/2019	1	4 mm aluminium sheet 190 mm x 30 mm	£0.89
ME Stores	14/02/2019	1	Ø25 mm aluminium round x 500mm	£5.19
RS	14/02/2019	1	Shim	£31.09
MetalEx	15/02/2019	1	7075T6 aluminium round bar	£81.42
Direct Plastics Ltd.	25/02/2019	1	250 mm oilon round	£17.98
RS	25/02/2019	1	Oilite bearing	£10.68
Flexospring	26/02/2019	1	Springs	£43.06
Wiggle	01/03/2019	1	Taps	£18.48
ME Stores	14/03/2019	4	Jubilee clips	£3.32
RS	19/03/2019	1	Timing belt 1.44 m length x 20mm width	£18.05
RS	22/03/2019	1	Steel timing belt pulley	£13.04
ME Stores	22/03/2019	1	Ø30 mm x 250 mm steel bar	£1.03
Wiggle	21/04/2019	1	Shimano Ultegra Hollowtech ii bottom bracket	£16.99
Decathlon	20/05/2019	2	Brake cable	£3.98
Decathlon	20/05/2019	1	Brake housing	£2.99
ME Stores	24/05/2019	1	Ø20 mm x 70 mm aluminium bar	£0.14
ME Stores	30/05/2019	1	25 mm x 250 mm x 4 mm aluminium plate	£0.97
			Total	**£532.58**

B. Gantt Chart



WBS	TASK	LEAD	START	END	DAYS	% DONE
3	**Quality Plan (Specification)**		Fri 05/10/18	Wed 31/10/18	27	100%
3.1	Draft	ALL	Fri 05/10/18	Thu 18/10/18	14	100%
3.1.1	PDS for disscussion	HRH	Fri 05/10/18	Thu 11/10/18	7	100%
3.2	Finalise	ALL	Fri 19/10/18	Sun 28/10/18	10	100%
3.3	Final Edit	CAT	Mon 29/10/18	Wed 31/10/18	3	100%
4	**Concept Design**		Wed 24/10/18	Wed 09/01/19	-	100%
4.1	First Pass Morphalogical Analysis/Concept generation	HRH	Wed 24/10/18	Tue 30/10/18	7	100%
4.2	Initial prototype design	US	Wed 31/10/18	Fri 09/11/18	10	100%
4.3	Initial prototype drawings	TJF	Sat 10/11/18	Tue 13/11/18	4	100%
4.4	Initial Protortype build	US	Sat 10/11/18	Tue 08/01/19	60	100%
4.5	Initial Prototype Testing	US	Mon 07/01/19	Wed 09/01/19	3	100%
5	**Detail Design**		Thu 10/01/19	Sat 26/01/19	-	100%
5.1	Design Work	HRH	Sat 12/01/19	Sat 19/01/19	8	100%
5.2	Final CAD	US	Sat 19/01/19	Mon 21/01/19	3	100%
5.3	Drawings	TJF	Sat 19/01/19	Tue 22/01/19	4	100%
5.4	Drawing Review	TJF	Tue 22/01/19	Sat 26/01/19	5	100%
5.5	Design Review Prep	HRH	Sat 26/01/19	Wed 30/01/19	5	100%
5.6	Design Review	HRH	Wed 30/01/19	Wed 30/01/19	1	100%
5.7	Design Review Changes	TJF	Wed 30/01/19	Fri 01/02/19	3	100%
6	**Progress Report**		Mon 07/01/19	Mon 04/02/19	-	100%
6.1	Write Progress Report	CAT	Mon 07/01/19	Mon 04/02/19	28	100%
7	**Manufacture**		Sun 27/01/19	Fri 31/05/19	-	100%
7.1	Procurement	TJF	Thu 31/01/19	Sat 09/02/19		100%
7.1.1	ME Stores	CAT	Thu 31/01/19	Fri 01/02/19	2	100%
7.1.2	Other Suppliers	TJF	Thu 31/01/19	Sat 09/02/19	10	100%
7.2	Manufacture	US	Mon 04/02/19	Mon 18/02/19		100%
7.2.1	CNC Parts	HRH	Mon 04/02/19	Tue 05/03/19	30	100%
7.2.2	Hand Machined Parts	US	Mon 04/02/19	Mon 18/02/19	14	100%
7.3	Assembly	US	Wed 06/03/19	Mon 18/02/19	7	100%
7.4	Modification	US	Fri 22/03/19	Fri 31/05/19	70	100%
7.4.1	New CNC Shaft	US	Fri 22/03/19	Thu 11/04/19	20	100%
7.4.2	Modified hand machined parts	US	Mon 13/05/19	Fri 31/05/19	18	100%
8	**Testing**	US	Fri 22/03/19	Wed 05/06/19	10	100%
8.1	Inital Testing	US	Fri 22/03/19	Fri 22/03/19	0	100%
8.2	Detailed Testing	US	Mon 20/05/19	Wed 05/06/19	16	100%
9	**Final Report**	CAT	Mon 04/02/19	Wed 05/06/19	-	100%



Bicycle ACVT Project Schedule

Thomas Fisher, Henry Hart, Ieuan Swainston, Charles Titmuss

Project Start Date 10/1/2018 (Monday) Display Week 1

WBS	TASK	LEAD	START	END	DAYS	% DONE
3	**Quality Plan (Specification)**		Fri 05/10/18	Wed 31/10/18	27	100%
3.1	Draft	ALL	Fri 05/10/18	Thu 18/10/18	14	100%
3.1.1	PDS for disscussion	HRH	Fri 05/10/18	Thu 11/10/18	7	100%
3.2	Finalise	ALL	Fri 19/10/18	Sun 28/10/18	10	100%
3.3	Final Edit	CAT	Mon 29/10/18	Wed 31/10/18	3	100%
4	**Concept Design**		Wed 24/10/18	Wed 09/01/19	-	100%
4.1	First Pass Morphalogical Analysis/Concept generation	HRH	Wed 24/10/18	Tue 30/10/18	7	100%
4.2	Initial prototype design	US	Wed 31/10/18	Fri 09/11/18	10	100%
4.3	Initial prototype drawings	TJF	Sat 10/11/18	Tue 13/11/18	4	100%
4.4	Initial Protortype build	US	Sat 10/11/18	Tue 08/01/19	60	100%
4.5	Initial Prototype Testing	US	Mon 07/01/19	Wed 09/01/19	3	100%
5	**Detail Design**		Thu 10/01/19	Sat 26/01/19	-	100%
5.1	Design Work	HRH	Sat 12/01/19	Sat 19/01/19	8	100%
5.2	Final CAD	US	Sat 19/01/19	Mon 21/01/19	3	100%
5.3	Drawings	TJF	Sat 19/01/19	Tue 22/01/19	4	100%
5.4	Drawing Review	TJF	Tue 22/01/19	Sat 26/01/19	5	100%
5.5	Design Review Prep	HRH	Sat 26/01/19	Wed 30/01/19	5	100%
5.6	Design Review	HRH	Wed 30/01/19	Wed 30/01/19	1	100%
5.7	Design Review Changes	TJF	Wed 30/01/19	Fri 01/02/19	3	100%
6	**Progress Report**		Mon 07/01/19	Mon 04/02/19	-	100%
6.1	Write Progress Report	CAT	Mon 07/01/19	Mon 04/02/19	28	100%
7	**Manufacture**		Sun 27/01/19	Fri 31/05/19	-	100%
7.1	Procurement	TJF	Thu 31/01/19	Sat 09/02/19		100%
7.1.1	ME Stores	CAT	Thu 31/01/19	Fri 01/02/19	2	100%
7.1.2	Other Suppliers	TJF	Thu 31/01/19	Sat 09/02/19	10	100%
7.2	Manufacture	US	Mon 04/02/19	Mon 18/02/19		100%
7.2.1	CNC Parts	HRH	Mon 04/02/19	Tue 05/03/19	30	100%
7.2.2	Hand Machined Parts	US	Mon 04/02/19	Mon 18/02/19	14	100%
7.3	Assembly	US	Wed 06/03/19	Mon 18/02/19	7	100%
7.4	Modification	US	Fri 22/03/19	Fri 31/05/19	70	100%
7.4.1	New CNC Shaft	US	Fri 22/03/19	Thu 11/04/19	20	100%
7.4.2	Modified hand machined parts	US	Mon 13/05/19	Fri 31/05/19	18	100%
8	**Testing**	US	Fri 22/03/19	Wed 05/06/19	10	100%
8.1	Inital Testing	US	Fri 22/03/19	Fri 22/03/19	0	100%
8.2	Detailed Testing	US	Mon 20/05/19	Wed 05/06/19	16	100%
9	**Final Report**	CAT	Mon 04/02/19	Wed 05/06/19	-	100%

C. Manufacturing Plan

Table C.1: lorem

ipsum

filler

Table C.2: lorem

ipsum

D. Engineering Drawings