Design of an Action Camera Stabilizer for Hand-held Use

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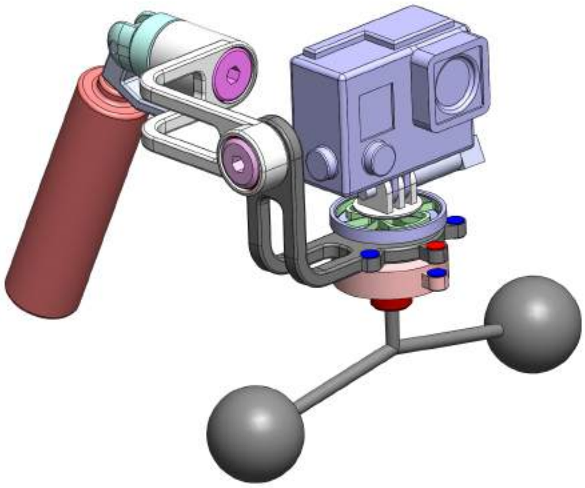

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May 2016

Abstract

Following the advent of portable action cameras in recent years, a great need for camera stabilization has arisen. Despite efforts of sophisticated optical stabilisation schemes, physical stabilisation will always result in better overall video quality. This report delves into the design of a solely mechanical stabilisation system for an action camera operated by a runner. Methods used range from magnetic damping, multi degree-of-freedom flexure damping and a 3-axis gimbal system. The result is a simulation validated design that will allow users smooth and stable video capture in an ergonomic fashion.

Acknowledgements

I would like to thank my supervisor Mr Francesco Pietra for his guidance regarding the design process followed for this design as well as his willingness to help throughout the semester. A special thanks also to Prof Cor-Jacques Kat for his contribution to my understanding of dynamics and the design of dynamic systems. I would also like to thank Mr Wim Murray for his great support with the technical drawing and CAE aspect of the project as well as his guidance regarding manufacturing procedures and graphical communication.

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List of Symbols

Greek Symbols

α	Rotational acceleration	$rad.s^{-2}$
η	Loss factor	
ω	Rotational velocity	$rad.s^{-1}$
ω_d	Driving frequency	$rad.s^{-1}$
ω_n	Natural frequency	$rad.s^{-1}$
ϕ	Angle	rad
σ	Normal stress	MPa
θ	Angle	rad
ζ	Damping ratio	

Roman Symbols

a	Acceleration	$\mathrm{m\,s^{-2}}$
B	Rotational viscous damping	$N.m.s/rad$
C	Linear viscous damping	$N.m.s$
C_r	Critical viscous damping	$N.m.s$
E	Modulus of Elasticity	MPa
F	Force	N
h	Height	m
I	Second moment of inertia	m^4
J	Rotational Inertia	$kg.m^2$
K	Linear spring constant	N/m
K_t	Torsional spring constant	$N.m/rad$
l	Length	m
M	Mass	kg
n_f	Factor of safety	

S_e	Fatigue endurance limit	MPa
S_{uts}	Ultimate tensile strength	MPa
T	Torque	$N.m$
w	Width	m
g	Graviational Acceleration	$m.s^{-2}$

Superscripts

$\ddot{}$	Second derivative
$\dot{}$	Derivative

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Chapter 1

Introduction

1.1 Background

With the drastic improvement of camera-sensors, image-processing capabilities and digital stabilization technologies over the recent years, action cameras can now record in ultra-high resolutions at very high frame rates. The GoPro Hero 4, for example, costs around R6500 and can shoot footage in 4K resolution at 30 FPS [GoPro, 2016].

The use of action cameras has extended to almost all extreme sports activities. The factor limiting the quality of the video footage has thus become the camera-operator and the way it is connected to the operator (e.g. hand-held or body-mounted). Due to sport-camera's increasing popularity, the successful stabilization of sports-cameras has recently become a large focus of research and product engineering.

This design report gives a detailed account of the design of such a hand-held camera stabilization system.

1.2 Problem Statement

A hand-held GoPro stabilization system needs to be developed which will sufficiently isolate the GoPro camera from the user's movement in order to improve video footage quality. The product needs to be compatible with GoPro accessories while at the same time being as small and as cost efficient as possible.

1.3 Objectives

The objective of this design is to supply the client with a product that conforms to the defined set of user requirements. This Design Project will be seen as a success if a design is developed that conforms to all of the user requirements, or alternatively, is deemed acceptable by the client when it is finally presented.

The objective of the design is also to explore alternative methods (other than electronically controlled gimbal systems) to stabilize a camera.

1.4 User Requirements

The client's requirements have been interpreted as follows:

General Requirements:

- The focus of the design should be such that it can be used by joggers (for running activities);
- The design should be focused on the GoPro Hero 4 Silver;
- The stabilization system must be handheld;
- The device should be of a mechanical nature (as opposed to electronically controlled);
- The design should stabilize and dampen the GoPro's movement relative to the operator;
- The operator should still be able to point the GoPro to what he/she wants - the camera should therefore not be entirely isolated from the operator's movement.

Compatibility and Physical Requirements:

- The design should allow full compatibility with the existing GoPro Hero 4 Skeleton case;
- The design should have an ergonomic interface with the operator;
- The design should be as small as possible.

Manufacturing Requirements:

- The design should be as light as possible;
- The device must be water resistant such that activities near water will not damage the device;
- The design must allow the client to offer customers a 1 year guarantee;
- Be as cost-effective as possible.

Chapter 2

Literature Study

This literature study comprises a variety of information needed for the design of a GoPro stabilization system. Items such as GoPro camera parameters, current market solutions, high frequency vibration elimination and handheld device ergonomics are researched and investigated.

2.1 GoPro Hero 4 Specifications

GoPro Camera Variations

The user requirements state that the design should focus on the GoPro Hero 4. This range consists of the silver and black versions [GoPro, 2016] which are shown in Figure 2.1 below.

Figure 2.1: GoPro Hero 4 Black and Silver [GoPro, 2016]

Physical Parameters

Mass and geometric parameters are critical for the design of dynamics. Table 2.1 shows the the mass and dimensions of the different GoPro Hero 4 cameras.

Camera Model	Weight (with Case)	Weight (Camera Only)	Nude Dimensions
Hero 4 Black	152 grams	89 grams	41x59x21/30mm
Hero 4 Silver	147 grams	83 grams	41x59x21/30mm

Table 2.1: Physical Parameters of Different GoPro Cameras [GoPro, 2016]

Casing Variations

GoPro sells a variety of different casing options for their Hero 4 Black and Silver edition cameras. The three main housing variations are called 'Standard', 'Skeleton' and 'Diving' casings. The differences of each housing type are highlighted in Table 2.2 below.

Casing Type	Waterproof	Waterproof Depth	Access to Ports	Housing Thickness
Standard	Yes	10m	No	3 to 5 mm
Skeleton	No	n/a	Yes	3 to 5 mm
Diving	Yes	40m	No	5 to 7 mm

Table 2.2: GoPro Hero 4 Case Variations [GoPro, 2016]

Depending on the particular use of the camera, a user can swop the casing for the camera. The Skeleton case is convenient if the camera is guaranteed not to come into contact with water as it allows the user access to all the ports such as micro-USB and screen buttons. However, if the camera will come into contact with water, the Standard casing will suffice. If the intended use of the camera is diving where the casing will be exposed to high water pressures, then the Diving case is the correct case [GoPro, 2016].

2.2 Current Solutions on the Market

There are a variety of products on the market that deal with GoPro camera stabilization. Additionally, there are also a number of film camera stabilizers which pose great interest for this design project.

2.2.1 GoPro Specific Products

Figure 2.2 shows current products developed for GoPro cameras specifically. Table 2.3 summarizes the critical parameters of each of these products.

	Curve	Smoothee	Cellfie Stick	Roxant Pro
Weight	230 g	350 g	500 g	820 g
Price	$99	$149	$79	$59
Rotation device	rotary ball	ball bearing	3-axis gimbal	rotary ball
Mounting Mechanism	clip	generic clip	screw clip	generic screw

Table 2.3: Important parameters of different products

Figure 2.2: Existing Products [GlideGear.net, 2015] [roxant.com, 2016] [BHPhoto, 2016] [tiffen.com, 2014]

2.2.2 Film Camera Specific Products

Filmography equipment has also been investigated. This is largely due to the fact that the film-industry is a $ 546 billion US dollar industry [statista.com, 2014] and a lot of research and development is done regarding video quality and camera stabilization. A product of special interest is the SNOW I.3 Black mechanical camera stabilizer shown in Figure 2.3.

Figure 2.3: Existing Products [snowcamerastabilzer.com, 2015]

The critical parameters for this stabilization device are shown in table 2.4.

	SNOW I.3 Black
Weight	900 g
Load Capacity	2 kg
Price	$599
Rotation concept	3-axis gimbal system
Rotation medium	Low frequency bearings
Mounting Mechanism	Camera thread

Table 2.4: Important parameters of SNOW I.3 Black

The most notable innovations of this product are as follows:

1. **Magnet damping system:** this product uses a magnet damping system to absorb movement in the yaw direction and aids in the smoothing of footage.
2. **Water pendulum:** the SNOW camera uses a water pendulum. This means the counterweight is partially filled with water. SNOW claims that this results in a return-to-position time that is 475% faster than a simple pendulum counterweight of the same mass. [snowcamerastabilzer.com, 2015].

2.3 Ergonomics

In 'A Check-List for Handle Design', Michael Patkin [Patkin, 2005] discusses the critical aspects that need to be considered when designing equipment that needs to interface with humans. Patkin's work is based on John Mackenzie's analysis of handles where hand grip types are divided into numerous different categories [Mackenzie and Iserhall, 1994].

Patkin goes on to describe the ultimate check list for handle design. The important aspects for handle design are:

1. **Size:**
   A length between 10-15cm is recommended to fit palm width. thickness must allow the thumb to just cover the end of the index finger and middle finger. Drury [Drury, 1980] recommends a diameter approximately 3-4cm.
2. **Shape:**
   A cylindrical shape is recommended. Thickened centrally if sliding is to be avoided. A bezel on the edge to avoid the hand from accidentally slipping off the device. Flattening for the thumb is useful as it gives guidance and allows for more precision. Other aspects that need to be considered are gentle finger grooving and central thickening to allow handle to be trapped in the palm. McCormick delves into greater detail regarding this topic [McCormick, 1983].
3. **Surface:**
   Insulation against vibration and ensuring a good grip with the device and the user.
4. **Surroundings:**
   Adequate clearance needs to be done concerning other objects surrounding the handle. An awkward posture should be avoided as this will cause discomfort and ineffective use of the tool.
5. **Storage:**
   Provision for storing the device must be considered. This is specifically relevant to this project as balancing will need to be done when the device is stationary and a device stand could help this a lot.
6. **Special other features:**
   Cleanability, replacement for left-handers (if asymmetric grip) and add-ons such as straps are all important considerations for the design.

2.4 High Frequency Vibration Elimination

Martin L Culpepper, Precision Engineering Professor at MIT, co-authored an article in 2011 regarding the design of flexural systems to allow for designed stiffness values in multiple degrees of freedom [Hopkins and Culpepper, 2011]. A flexure can be defined as a structure which has high compliance in certain degrees of freedom and high stiffness in other degrees of freedom such that it is able to fulfil certain stiffness requirements [Culpepper, 2015]. Figure 2.4 shows an example of a flexure with designated degrees of freedom.

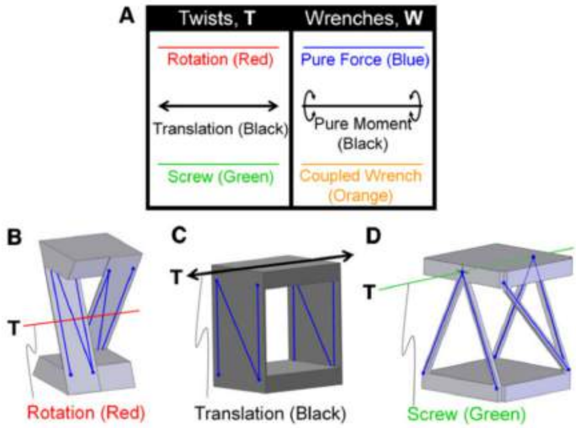


Figure 2.4: Example of flexures and their compliant degrees of freedom [Hopkins and Culpepper, 2011]

Culpepper [Hopkins and Culpepper, 2011] goes on to detail how serial synthesis principles for flexure design can be used to obtain multiple degree of freedom flexures. This article can be utilized to design high frequency vibration absorption systems for the GoPro stabilization system. Figure 2.5 shows a very simplified visual presentation of how a flexure would be designed to allow for multiple degree of freedom compliance.

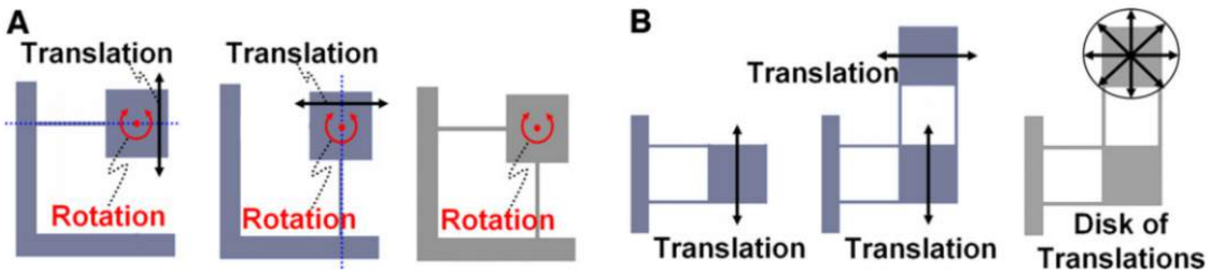


Figure 2.5: Example of series combinations of flexures to allow compliance in multiple degrees of freedom [Hopkins and Culpepper, 2011]

2.5 Isodamp Material

E-A-R Specialty Composites [Frankovich, 2010] manufactures a variety of damping materials. These materials are used for vibration absorption of electronic devices but could easily be adapted for mechanical stabilization [Frankovich, 2011]. Possible materials that could be used for high vibration damping from E-A-R Specialty Composites are [EARComposites, 2011]:

- ISODAMP thermoplastic material
- ISOLOSS®HD urethane
- VersaDamp™TPE
- ISOLOSS®HD urethane
- ISOLOSS VL Low Modulus Urethane Elastomer
- ISOLOSS LS High Density Urethane Foams

Flexures could possibly be made of Isodamp C-1002 thermoplastic. This is due to it's high loss factor which is critical for high damping applications [Frankovich, 2011]. Loss factors for different materials are shown in table 2.5.

Material	**Loss Factor**
Aluminum	0.005-0.007
Neoprene	0.1
Butyl Rubber	0.4
Isodamp®C-1002 thermoplastic	1.0

Table 2.5: Loss factors for common materials [Frankovich, 2011]

The loss factor η is related to the damping ratio ζ as $\eta = 2\frac{C}{C_r} = 2\zeta$ [Hertz, 1983].

2.5.1 Camera Stabilization and Precision

Denvel Garug dissects camera stabilization into two categories, namely precision and stability [Garug, 2014]. Garug defines precision as the focus of the camera on a certain point and describes it as a slower movement input response. On the other hand, stability is described as a devices's ability to eliminate the smaller amplitude, higher frequency vibration that is projected to the device.

Garug points out that most camera movement is hindered by rotation rather than translation and that eliminating vibrational movement in the rotational axes is far more effective than translational movement. This is because the camera's view is perturbed far more due to a unit rotation than a unit translation. Naturally, eliminating translational and rotational vibration is optimal.

2.6 Conclusion

This literature study investigated the GoPro cameras themselves as well as current stabilization products that are available. The study delves into literature regarding ergonomics, flexure design and elastomeric materials that could all potentially be used in this design project. In conclusion, this literature study revealed a variety of possible engineering methods and should considerably help synthesize the final design into the best possible solution.

Chapter 3

Experimental Data

3.1 Vibration Analysis

The aim of this analysis is to obtain the linear and rotational accelerations associated with a runner running with a GoPro Hero 4 camera. This analysis will yield the most significant accelerations and their associated frequencies which are of paramount importance when designing to minimize the accelerations experienced by the camera.

When using a GoPro, Denvel Garug [Garug, 2014] proposes that a runner's motion is split up into the following two categories:

- **Stabilization:** Motion due to the natural movement and vibrations associated with running in a straight trajectory without redirecting the camera. This will be defined as High Frequency Vibration.
- **Precision:** Motion that is realized when the runner is moving the camera in different directions to capture subjects in different directions. This is will be defined as Low Frequency Movement.

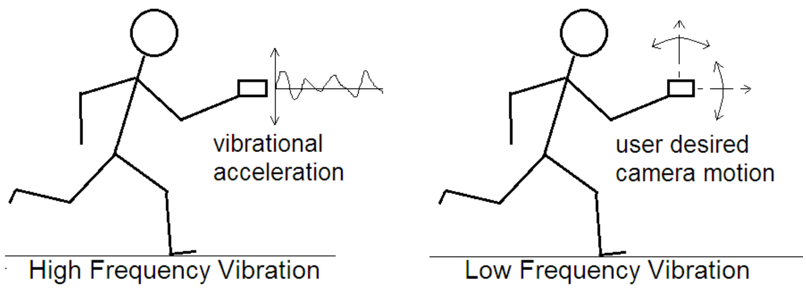


Figure 3.1: Normal high frequency vibration

In order to capture the accelerations, an Inertial Measurement Unit (IMU) will be used. The sensor data is extracted and filtered using SensorStream [SensorStream, 2016] after which a Fast Fourier Transform of the data is done in MATLAB. This reveals the critical frequencies and amplitudes that are necessary for the design.

To ensure brevity, the filtering process used to ensure accurate IMU data is cover in Appendix A.1.

3.1.1 Linear Accelerations

The linear acceleration is measured using the accelerometer sensors in the x,y,z directions. The filtered data is shown in Figure 3.2.

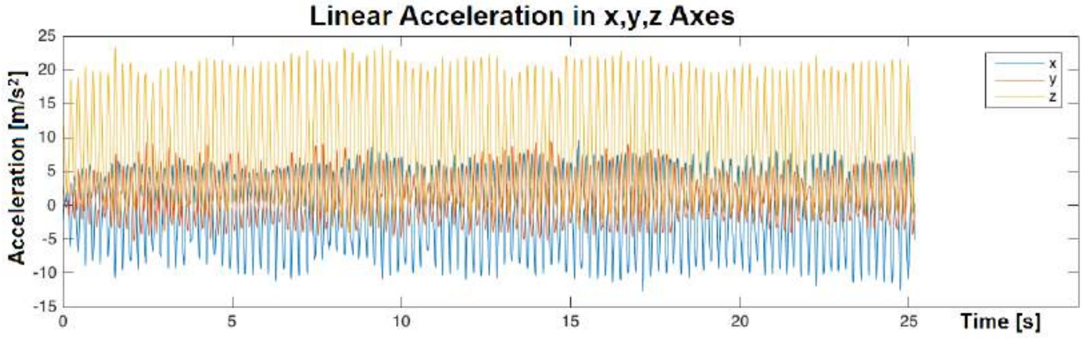


Figure 3.2: Acceleration of pure vibration case

The Fast Fourier Transform is performed on the data and is shown in Figure 3.3. The critical frequencies are quite obvious in the x,y and z cases and have been tabulated in table 3.1.

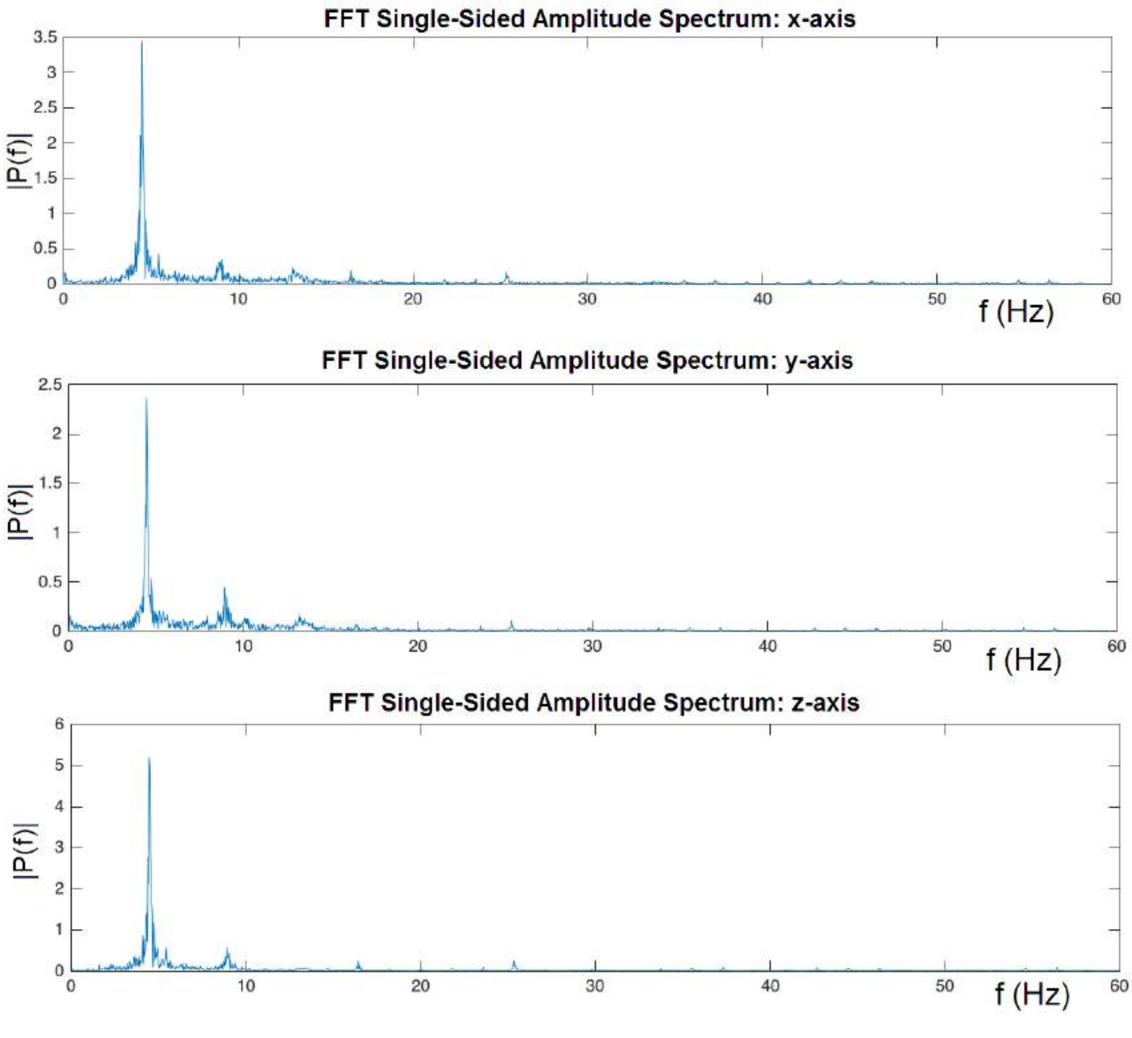


Figure 3.3: FFT of the x,y,z axis acceleration

	Frequency 1 (Hz)	Amplitude	Frequency 2 (Hz)	Amplitude
x	4.4	3.4	8.4	0.3
y	4.4	2.4	8.5	0.47
z	4.4	5.2	8.4	0.65

Table 3.1: Critical Frequencies and Amplitudes for Linear Accelerations

3.1.2 Rotational Accelerations

The rotational acceleration is measured using the gyro sensors in the roll, pitch, yaw directions. The filtered data is shown in Figure 3.4.

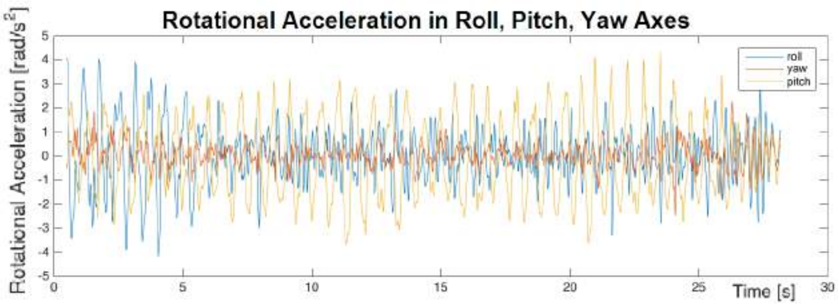


Figure 3.4: Gyrometer acceleration values

The Fast Fourier Transform is performed on the data and is shown in Figure 3.5. The critical frequencies for the roll, pitch and yaw cases have been tabulated in table 3.2.

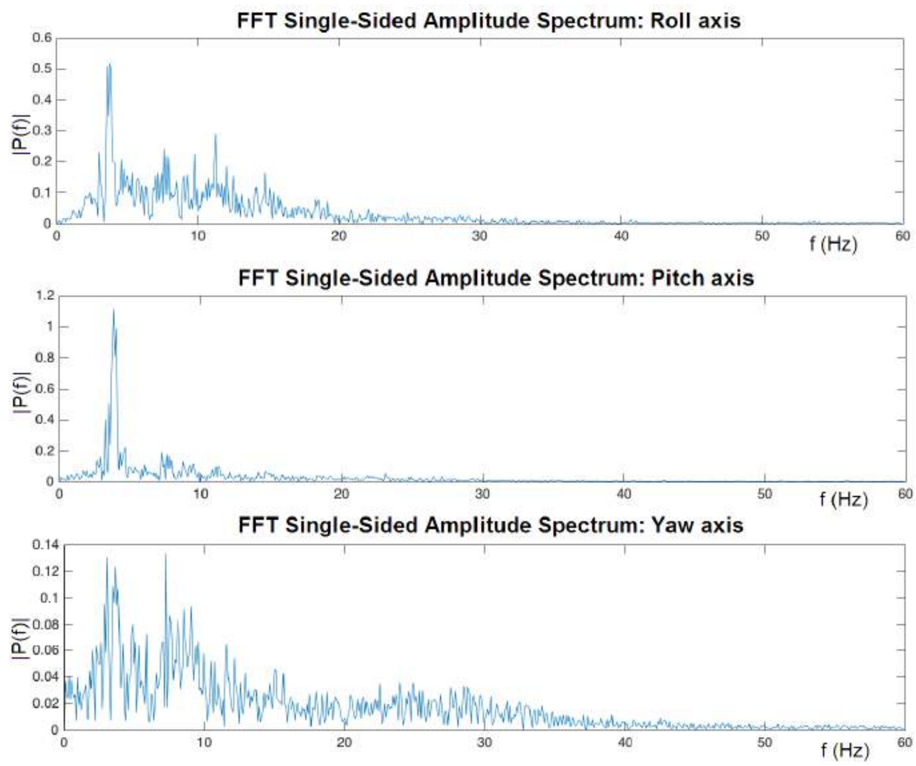


Figure 3.5: FFT of gyrometer accelerations in roll, pitch, yaw

	Frequency 1 (Hz)	Amplitude	Frequency 2 (Hz)	Amplitude
Roll	1.6	0.52	11.8	0.27
Pitch	1.6	1.11	0.76	0.19
Yaw	1.3	0.13	3.0	0.13

Table 3.2: Critical Frequencies and Amplitudes for Rotational Accelerations

3.1.3 Low Frequency Perturbations

Low frequency perturbations refer to low-speed rotational movement induced by the user. These perturbations are modelled as a sinusoidal function with amplitude of approximately 30 degrees ($\frac{\pi}{6} \approx 0.5$ radians) and a frequency of 1 rad/s.

$$U(t) = 0.5\sin(t) \tag{3.1}$$

3.1.4 Maximum Accelerations

The IMU was used to measure the maximum acceleration that could be imposed on it. This data is needed so that the design can account for extreme accelerations when considering the structural analysis of it's components. A simple experiment was conducted where the IMU was added to a weight of 0.5 kg (the approximate mass of the device) and was shaken by hand as fast as possible. The result was that the maximum acceleration in the x,y directions was $39m/s^2$ and in the z direction was $49m/s^2$.

3.2 GoPro Field of View (FOV)

In order to ensure that the device does not obstruct the view of the GoPro's camera, a view angle model was developed. The GoPro uses a special wide angle lense which is evidenced by the parabolic shape of the view profile shown in Figure 3.6: Isometric View.

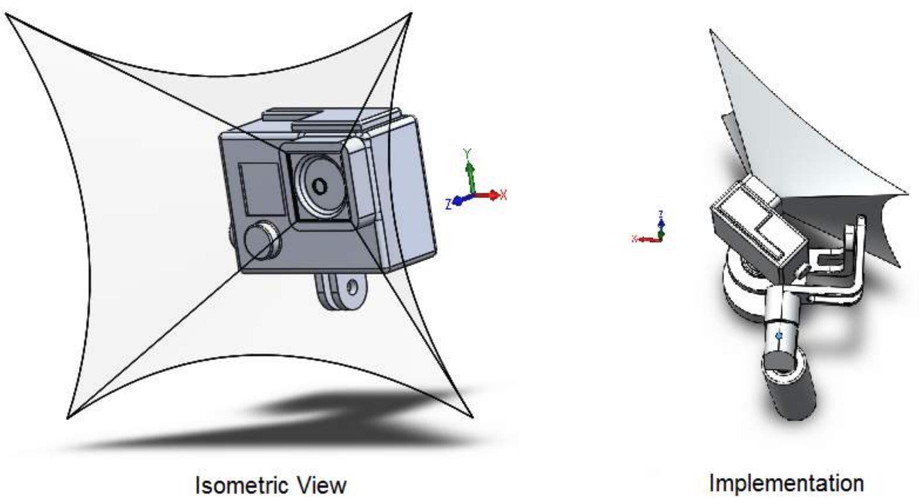


Figure 3.6: View Angle of the GoPro in as isometric view as well as an example of it's functional implementation

Chapter 4

Functional Analysis

The user requirements have been specified in section 1.4. In order to ensure a good design, a functional analysis of the functional requirements needs to be done. The main functional requirements are derived from the user requirements in section 4.1 and can be seen as a technical interpretation of the otherwise non-technical user requirements.

The objective of this analysis is to break down the relatively complex *main functional requirements* into sub-functions and even simpler base functions. This is because a better design is achieved when a solution is sought to fulfil each base function rather than finding a solution to the main functional requirements.

4.1 Main Functional Requirements

The user requirements specified in chapter 1.4 are decomposed into functional requirements and are as follows:

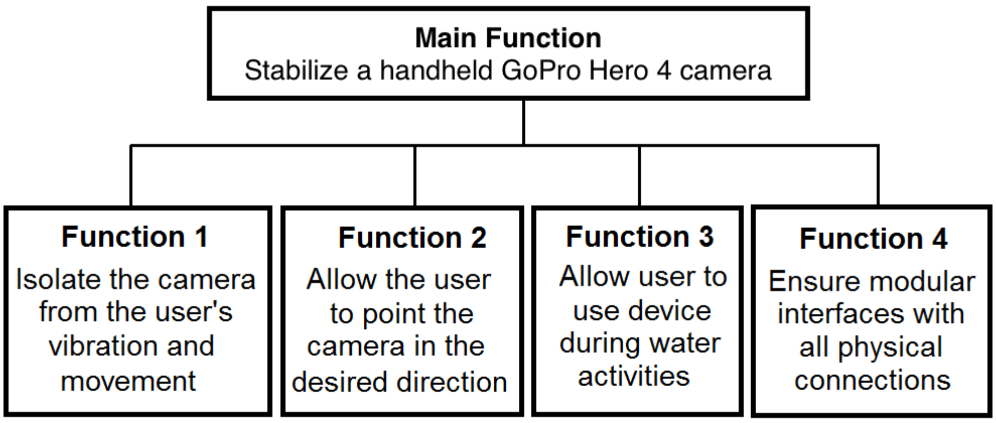


Figure 4.1: The main function and its direct sub functions

Figure 4.1 shows a function tree depicting the main function decomposed into its four main functional requirements. These four functions can each be further decomposed into their own sub-functions and base functions. This is shown in section 4.3.

4.2 Reference Axes for GoPro

In order to ensure that the orientation of the GoPro as referenced in the upcoming sections is clear, a referencing axis has been set and is shown in Figure 4.2.

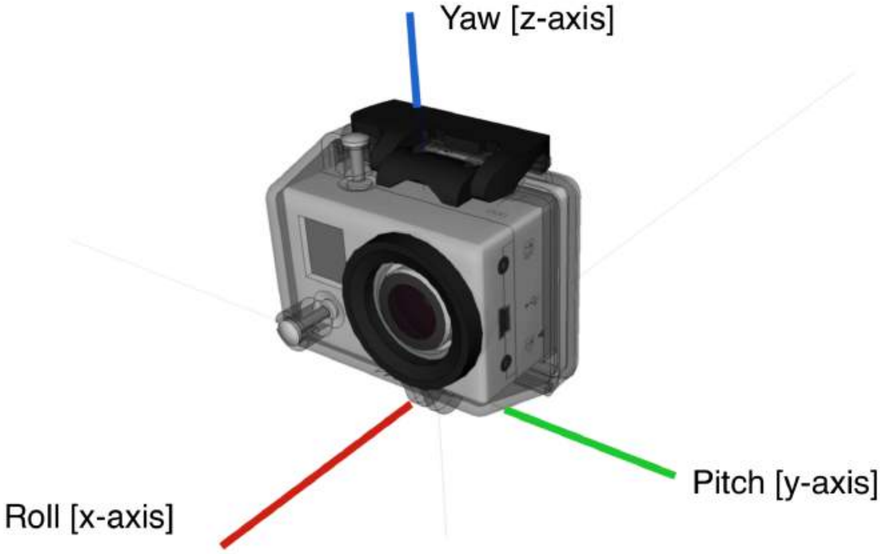


Figure 4.2: Reference axes for the GoPro Hero 4

4.3 Sub-Functions and Base Functions

Each of the functions listed in Section 4.1 can be further decomposed into simpler base functions. At this point it is important to recall that the base functions are made as simple as possible. This ensures that the conceptual design in chapter 6.1, which are derived from the functional analysis, are focused on solving specific functional requirements.

4.3.1 Function 1: Isolate the Camera

Isolating the camera from the user implies that all motions of the user should be experienced by the camera as little as possible. In order to simplify the functional analysis for Function 1, we subdivide the requirement into the yaw, roll and pitch axes and then focus on the movement seen by each of these axes. Figure 4.3 below shows a tree diagram for the decomposition of Function 1.

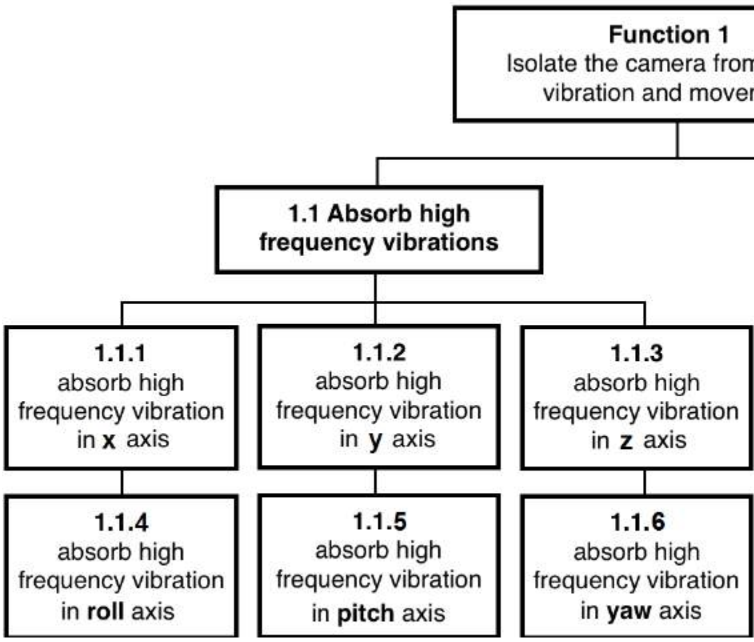


Figure 4.3: The functional tree for Function 1

4.3.2 Function 2: Allow User to Direct Camera

In order to allow the user to point the camera in the intended direction, certain axes need to follow input from the user. These axes have been identified as the yaw and pitch axes. The decomposition of Function 2 is shown in Figure 4.4.

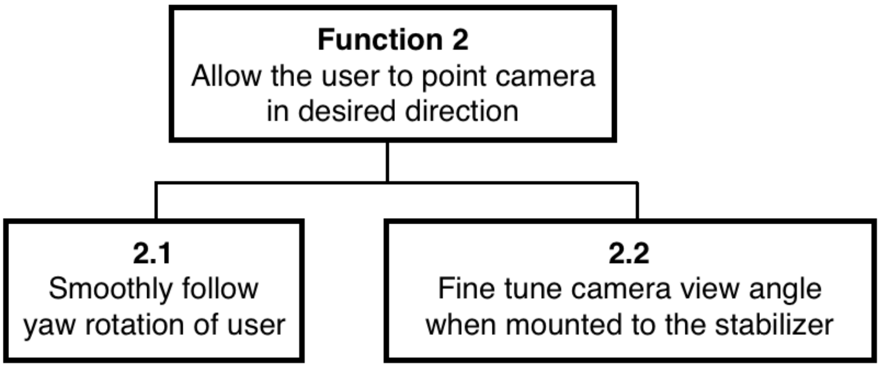


Figure 4.4: The functional tree for Function 2

4.3.3 Function 3: Water Resistant Requirement

The device has the requirement of being water resistant such that it can be used in wet areas. The function tree for Function 3 is shown in Figure 4.5.

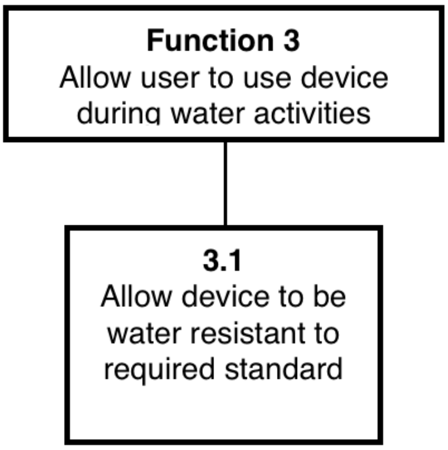


Figure 4.5: The functional tree for Function 3

4.3.4 Function 4: Interface with User, GoPro and other functions

Interfacing with the user and the GoPro Hero 4 is a very important functional requirement. Function 4 is decomposed into its base functions and is shown in Figure 4.6.

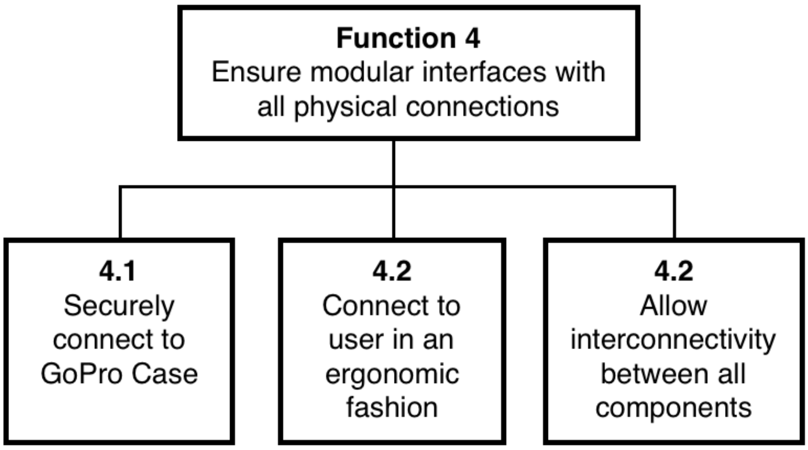


Figure 4.6: The functional tree for Function 4

4.4 Conclusion

The functional analysis has resulted in the acquisition of a list of base functions. These base functions can now be used to develop conceptual designs. The process of generating possible concepts is covered in chapter 6.1.

Chapter 5

Design Requirements

The objective of this chapter is to translate the functional requirements specified in chapter 4 into quantifiable design requirements. This will allow the final design to be evaluated in a technical manner and ultimately ensure that the user requirements are fulfilled.

Each base function is decomposed into it's relevant measurable parameters with the associated design requirement. The design requirements range from a maximum allowable value, a desired range that the design should fit into or a simple criteria that must be adhered to.

Functions 1.1.1 - 1.1.6: High Frequency Vibration

These basic functions have been lumped together as they all have the requirement of high frequency vibration damping, only in different directions namely x,y,z, roll, pitch and yaw directions.

ArduCopter proposes that high-rate sensors should have a minimum of 0.2G in the x and y directions and 0.4G in the z direction [Frantz, 2013]. Since the GoPro camera is fundamentally an optical sensor, we will use this criteria to define the design accelerations for high frequency vibration.

Similarly, for the rotational axes, for a shutter speed of 1/8192 seconds (which is the minimum shutter speed of the GoPro [GoPro.de, 2014]), the limiting rotational acceleration is recommended to not exceed 0.5 rad/s^2 [Tennis, 2011].

Table 5.1 summarizes the design accelerations.

Function	Parameter	Requirement Type	Design Specification
1.1.1	x acceleration	max allowable	2.0 m/s^2
1.1.2	y acceleration	max allowable	2.0 m/s^2
1.1.3	z acceleration	max allowable	3.9 m/s^2
1.1.4	roll acceleration	max allowable	0.5 rad/s^2
1.1.5	pitch acceleration	max allowable	0.5 rad/s^2
1.1.6	yaw acceleration	max allowable	0.5 rad/s^2

Table 5.1: Design Requirements for High Frequency Vibration

Functions 1.2.1 - 1.2.2: Low Frequency Vibration

These basic functions have also been lumped together as they all have the requirement of low frequency movement elimination for the roll and pitch directions. Table 5.2 tabulates these design requirements.

Function	Parameter	Requirement Type	Design Specification
1.2.1	roll low freq response	max allowable	0.1 rad amplitude
1.2.2	pitch low freq response	max allowable	0.1 rad amplitude

Table 5.2: Design Requirements for Low Frequency Vibration

Functions 2.1: Follow yaw rotation

The user must be allowed to point the camera in the desired direction. However, in order to ensure that the camera's footage is kept as smooth as possible, the camera should lag the user's input by approximately 2-4 seconds. Additionally, the camera should experience no overshoot as this means it will not point further away than what the user requires it.

Function	Parameter	Requirement Type	Design Specification
2.1	overshoot	criteria	0% overshoot
2.1	settling time	desired range	2-4 seconds

Table 5.3: Design Requirements for function 2.1

Function 2.2: Fine tune the camera

In order to account for manufacturing defects and the resulting imbalance, it is necessary that the counterweight's position be adjustable such that the user can ensure the GoPro is in pointing in the desired direction. In order to do this, the centre of gravity for the counterweight is set to be translatable from the centre axis of the GoPro. The criteria is summarized in Table 5.4.

Function	Parameter	Requirement Type	Design Specification
2.2	x translatability of CoG	desired range	15-25mm
2.2	y translatability of CoG	desired range	15-25mm

Table 5.4: Design Requirements for function 2.2

Function 3.1: Water Resistant

This base function simply requires that the materials that the final design will be made of will not be affected when brought into contact with water. The design requirement is therefore that the materials be non-rust metals or plastics or any other non-corrosive materials. Bearings used in the design should be made of stainless steel.

Function 4.1: Securely Connect to GoPro

Securely connecting to the GoPro is not a function that will be measured numerically. However, the final design should allow for the same security as current GoPro mounts do. This can be tested simply by shaking the designed mount and the official GoPro mount on the same platform and ensuring that the GoPro mount allows GoPro movement before the designed mount does.

Function 4.2: Connect to User Ergonomically

Following the ergonomics study done in the Literature study of section 2.3, the handle for the stabilizer should have the parameters as specified in Table 5.5.

Function	Parameter	Requirement Type	Design Specification
4.2	Handle Length	desired range	10-12cm
4.2	Handle Diameter	desired range	3-5cm

Table 5.5: Design Requirements for function 4.2

Function 4.3: Interconnectivity

This function is met following multiple CAD iterations that lead up to the optimal final design. There is no measurable quantity by which this function can be compared to. It is engineering judgement and the use of tools such as the Field of View angle from section 3.2 to which the successful synthesis of all the different components can be determined.

Conclusion

In conclusion, the design requirements have been determined for each base function. Once the final design is performed, the ability to perform the base functions can be measured by consulting the design requirements.

Chapter 6

Conceptual Design

This chapter gives details of possible solutions that could fulfil the functional requirements specified in chapter 4. In order to do this efficiently, a concept generation is performed in section 6.1. The concept generation is a compilation of a wide variety of possible solutions to ensure that each requirements posed by each base function is sufficiently met.

A concept selection is then performed in section 6.2 where concepts for each base function are chosen based on their ability to satisfy important criteria specified by the design specifications.

The final concept is shown in section 6.3 and shows the merged result of the concept selection.

6.1 Concept Generation

Function 1.1.1 - 1.1.6: Eliminate high frequency vibration

Base functions 1.1.1 - 1.1.6 require the elimination of high frequency vibration in the x,y,z,roll, pitch and yaw axes. Since the fundamental requirement for all six these base functions is identical with the only differentiator being the axis of rotation or translation, their concept generation has been merged into one. Four concepts have been developed and are shown in Figure 6.1. They are described as follows:

- **Concept 1** works on the principle of a flexure connection between the handle and the camera.
- **Concept 2** utilizes a design-specific torsional spring and low friction bearing and allows two parts to be segregated concerning the high frequency domain.
- **Concept 3** utilizes an elastomer based connection between two interfaces.
- **Concept 4** involves the design of a flexure that has certain damping characteristics in multiple degrees of freedom.

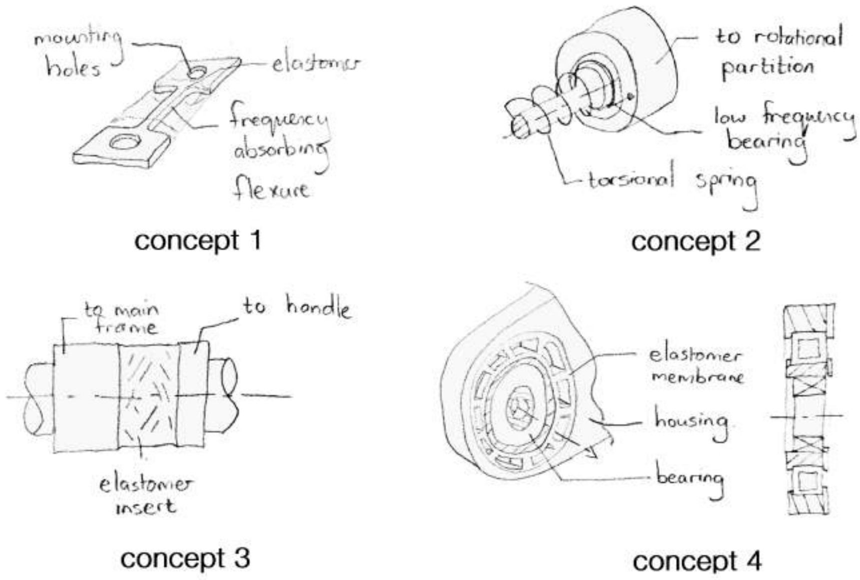


Figure 6.1: Concepts for high frequency vibration elimination

Function 1.2.1: Eliminate low frequency rotation about the roll axis

Base function 1.2.1 entails the elimination of low frequency rotation in the roll axis. The concepts are shown in Figure 6.2 and are described as follows:

- **Concept 1** lowered; single-sided arm. This configuration provides a lower centre of gravity but results in more complexity.
- **Concept 2** standard height; single-sided arm. This system allows for a simpler configuration but still utilizes a one-sided bearing design.
- **Concept 3** standard, double-sided arm. This system is the simplest solution but poses the biggest threat to performance.

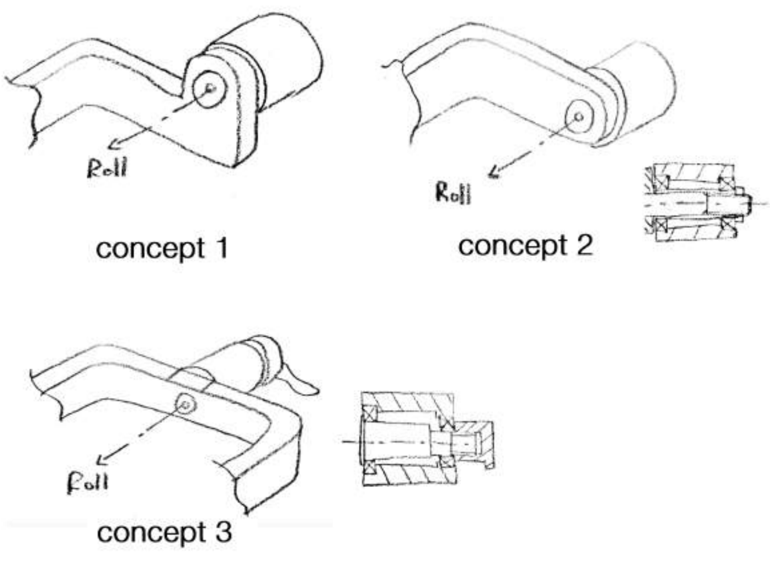


Figure 6.2: Concepts for low frequency rotation elimination in roll axis

Function 1.2.2: Eliminate low frequency rotation about the pitch axis

Base function 1.2.2 entails the elimination of low frequency rotation about the pitch axis. Since manufacturing precision and manufacturability is an important aspect of design, the concept generation has focused on the bearing and frame layout. The four concepts are shown in Figure 6.3. They are described as follows:

- **Concept 1** lowered; two-sided swing-arm. This configuration utilizes a single bearing per side.
- **Concept 2** standard height; single-sided swing-arm. This configuration utilizes a single bearing per side.
- **Concept 3** lowered; one-sided swing arm. This configuration makes use of dual bearing setup at the pivot.
- **Concept 4** lowered; one-sided swing arm. This configuration utilizes a single bearing design.

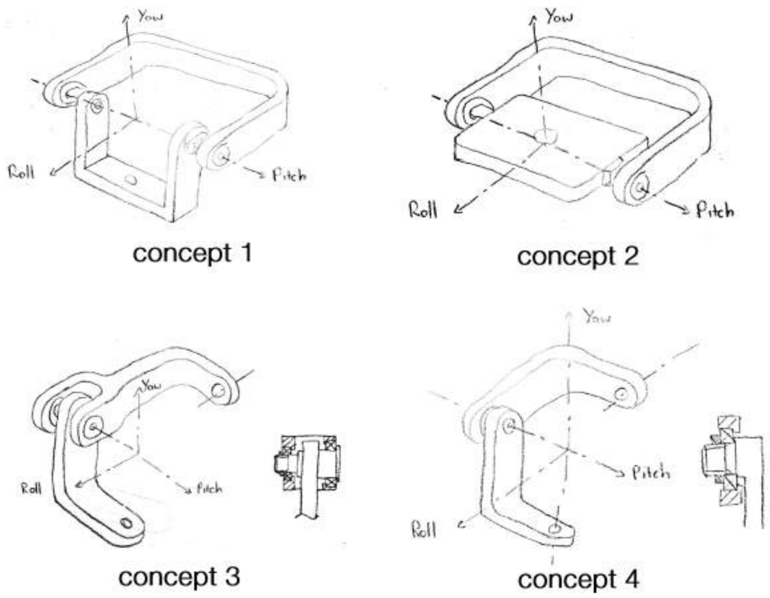


Figure 6.3: Concepts for low frequency rotation elimination in pitch axis

Function 2.1: Smoothly follow yaw rotation of user

Base function 2.1 requires a solution that smoothly follows the camera in the yaw rotation. The emphasis of this functional requirement is that low frequency movement such as the panning of the video camera by the user is transmitted to the GoPro, but that the high frequency vibration is not transmitted as well. The concepts are shown in Figure 6.4 and are described as follows:

- **Concept 1** eddy-current repelling system. This concept utilizes opposing magnets to allow the camera to slowly pan to its intended direction when the yaw rotation platform is rotated relative to its base.
- **Concept 2** torsional spring system. This concept employs a low stiffness torsional spring that will slowly react to panning inputs while eliminating unintentional yaw rotation.

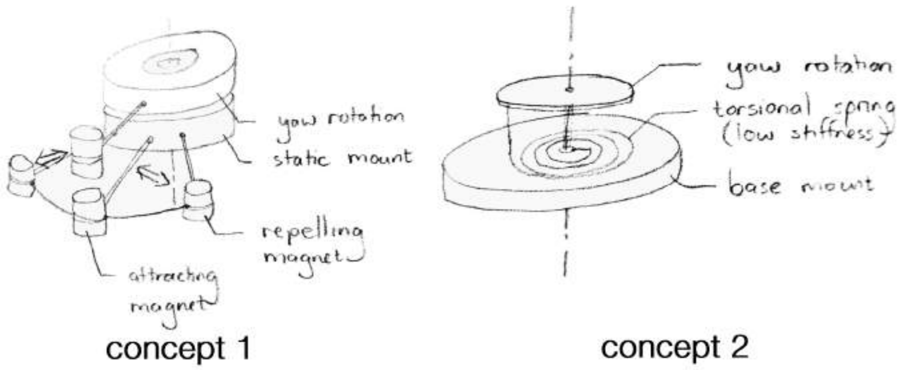


Figure 6.4: Concepts for low frequency rotation following about the yaw axis

Function 2.2: Fine tune camera view angle when mounted to the stabilizer

Base function 2.2 is of utmost importance. The concepts shown in Figure 6.5 involve the translation (in the x- and y-directions) of a stabilization weight which will allow the centre of gravity of the system to be such that the GoPro is pointed in the user's intended direction. This can also be seen as a calibration device. The three concepts are described as follows:

- **Concept 1** pinion-and-rack adjuster. This concept utilizes two rack-and-pinion adjusters that allows the translation of the weight.
- **Concept 2** flexure adjuster. This concept incorporates a pair of U-slots and bolts which allow the fine adjustment of the weight's position.
- **Concept 3** socket adjuster. This concept utilizes a sufficiently stiff socket-and-ball interface which is allows the weight to be moved such that it balances the camera above it.

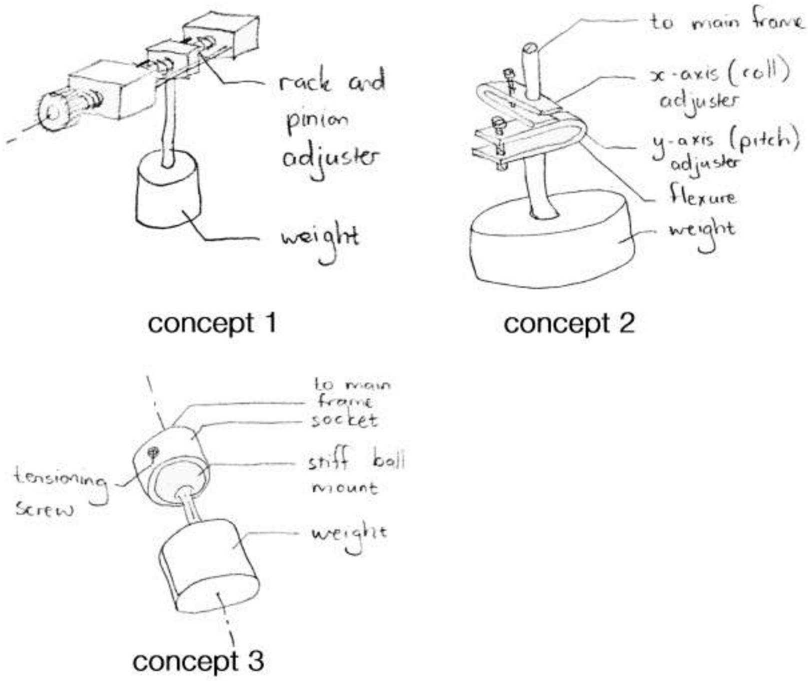


Figure 6.5: Concepts for the adjuster mechanisms to fine tune camera view angle

Function 3.1: Water Resistance Requirement

The device needs to be water resistant. Since this function does not comply with a specific component of the design, but rather the design as a whole, the water resistance will be ensured with the final concept.

Water resistance is also mostly a function of material choice. Materials that will be considered will be plastics, aluminium or stainless steels.

Function 4.1: Securely connect to GoPro Case

The GoPro case utilizes a simple two hole slot that merges with a three holed base mount and is fastened using a hand-tightened nut and bolt. The two concepts are shown in Figure 6.6 and are described as follows:

- **Concept 1** GoPro connector. This mount connects to the GoPro clip which connects to the GoPro case.
- **Concept 2** hybrid mount. This mount is a substitute to the current connector clip.

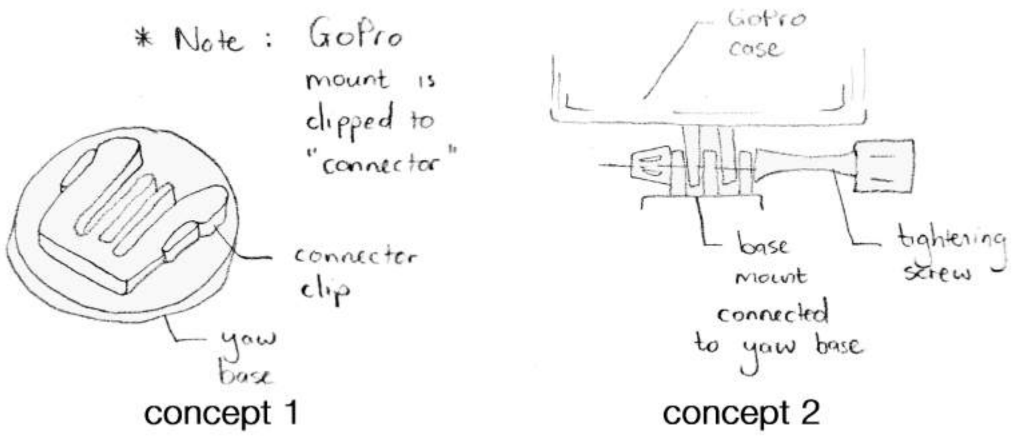


Figure 6.6: Concepts for the GoPro Hero case connection

Function 4.2: Connect to user in an ergonomic fashion

The user needs to connect to the design in an ergonomic fashion. Three concepts have been developed and are shown in Figure 6.7. They are described as follows:

- **Concept 1** hand-shaped grip. This concept utilizes an ergonomically shaped grip for extra user comfort.
- **Concept 2** rounded foam grip. This concept incorporates a simple, rounded grip.
- **Concept 3** rounded grip with strap. Similar to concept 2 but with the added security of a strap for extra support.

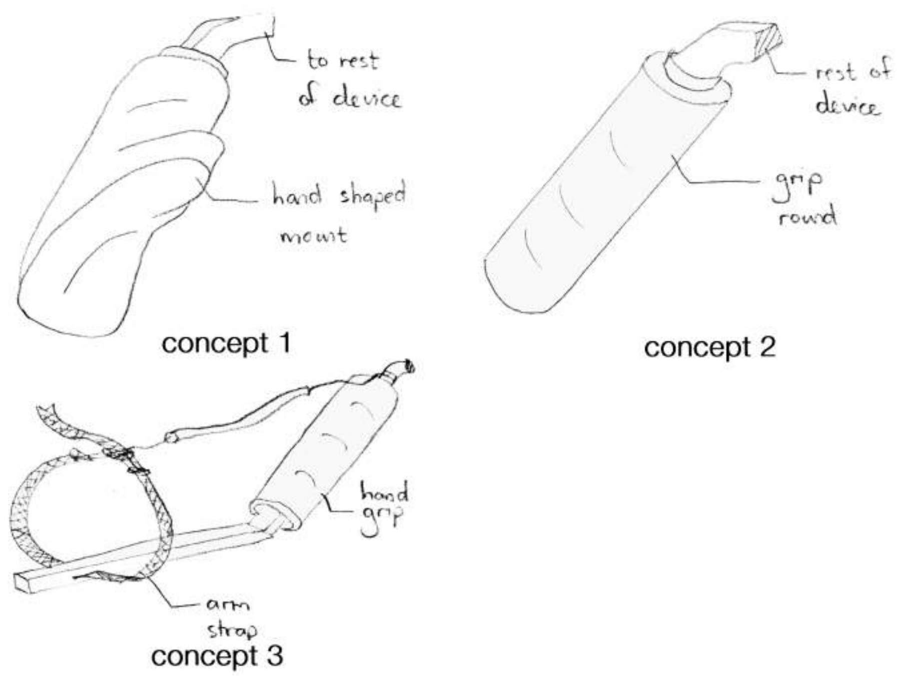


Figure 6.7: Concepts for the ergonomic connection with the user

Function 4.3: Allow interconnectivity between all components

Interconnectivity between all components is of major importance and is taken into consideration with each concept selection processes.

No concepts are specifically developed for this base function. Fulfilling this functional requirement can be seen as the successful merging of all the other concepts into one final concept.

6.2 Concept Selection

Method of Selection

Concept selection is done regarding the concepts generated in the section 6.1. The method of selection is by considering the important parameters required of each function. These parameters are each given a degree of importance. Each concept is given a score out of 5 for each parameter (5 is best, 1 is worst). This score is then multiplied by the importance factor of that parameter. The aggregate for each concept is then determined and the highest scoring concept gets selected.

Function 1.1.1 - 1.1.6: Eliminate high frequency vibration

It is important to note that since we have 6 degrees of freedom, we need to find the most compact method of merging all these vibration isolators into one. Additionally, performance is critical as it directly influences the camera's footage. This is all reflected in table 6.1.

Item	Importance	Score				Aggregate			
		1	2	3	4	1	2	3	4
Performance	4	3	2	2	5	12	8	8	20
Simplicity	3	5	3	4	2	15	9	12	6
Space	5	2	1	3	4	10	5	15	20
Total						37	22	35	**46**

Table 6.1: Concept Selection: 1.1.1-1.1.6

The concept selected is concept 4: flexure system. This is largely due to its compact nature and outstanding performance when executed correctly [Culpepper, 2015].

Function 1.2.1: Eliminate low frequency rotation about Roll Axis

Item	Importance	Score			Aggregate		
		1	2	3	1	2	3
Ease Implementation	3	3	4	4	9	12	12
Required Space	4	4	5	3	16	20	12
Simplicity	4	3	4	2	12	16	8
Potential Cost	2	4	4	5	8	8	10
Total					45	**56**	42

Table 6.2: Concept Selection: 1.2.1

The concept selected is concept 2: standard height, single-sided arm.

Function 1.2.2: Eliminate low frequency rotation about Pitch Axis

Item	Importance	Score				Aggregate			
		1	2	3	4	1	2	3	4
Ease of Implementation	3	4	3	3	3	12	9	9	9
Required Space	4	2	3	3	5	8	12	12	20
Simplicity	4	4	5	3	3	16	20	12	12
Potential Cost	2	3	2	3	3	6	4	6	6
Total						42	45	39	**47**

Table 6.3: Concept Selection: 1.2.2

The concept selected is concept 4: lowered, one-sided swing arm.

Function 2.1: Smoothly follow yaw rotation of user

The concept selected is concept 1: magnet damping system.

Item	Importance	Score 1	Score 2	Aggregate 1	Aggregate 2
Practical Implementation	4	4	2	16	8
Performance	5	4	3	20	15
Required Space	2	3	3	6	6
Total				**42**	29

Table 6.4: Concept Selection: 2.1

Function 2.2: Fine tune camera view angle when mounted to the stabilizer

Item	Importance	Score 1	Score 2	Score 3	Aggregate 1	Aggregate 2	Aggregate 3
Simplicity	5	1	3	4	5	15	25
Space required	4	2	3	4	8	12	16
Cost	3	2	4	4	6	12	12
Ease-of-use	4	5	2	4	20	8	16
Total					39	47	**69**

Table 6.5: Concept Selection: 2.2

The concept selected is concept 3: ball pivot adjuster. This system is simple and easy to implement, and should provide simple calibration of the device.

Function 4.1: Securely connect to GoPro Case

Item	Importance	Score 1	Score 2	Aggregate 1	Aggregate 2
Strength	4	3	4	12	16
Space required	5	3	5	15	25
Cost	3	5	2	15	6
Total				42	**47**

Table 6.6: Concept Selection: 4.1

The concept selected is concept 2: custom mount. This concept allows great space saving which is of paramount importance for this specific function.

Function 4.2: Connect to user in an ergonomic fashion

The concept selected is concept 2: round grip.

Item	Importance	Score 1	Score 2	Score 3	Aggregate 1	Aggregate 2	Aggregate 3
Manufacturability	3	3	5	3	9	15	9
Weight	4	4	4	3	16	16	12
Ergonomics	5	4	3	5	20	15	25
Cost	2	4	5	3	8	10	6
Total					53	**56**	52

Table 6.7: Concept Selection: 4.2

6.3 Final Concept

The final concept is the synergistic product of each of the concepts chosen to best fulfil each base function. The final concept is shown in Figures 6.8 and 6.9.

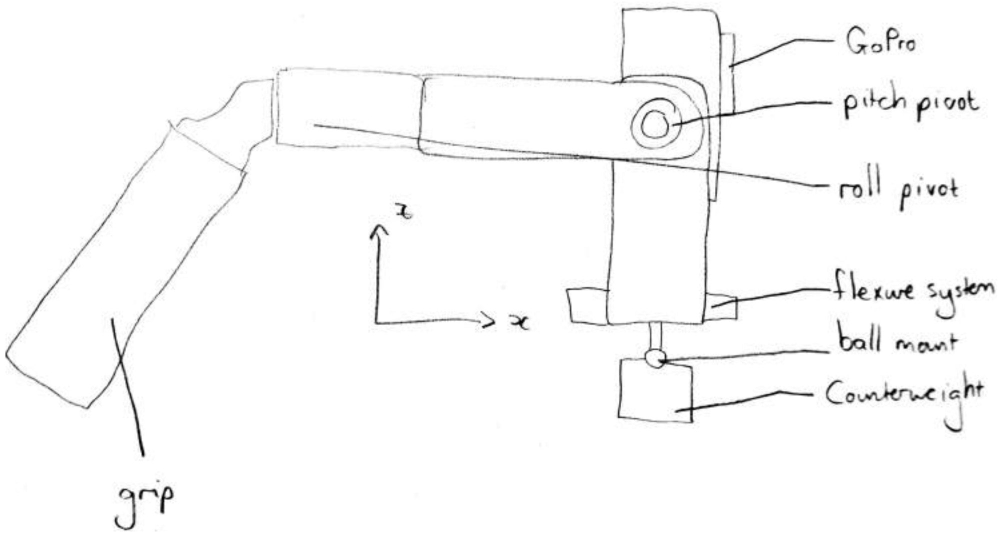


Figure 6.8: Concepts for the ergonomic connection with the user

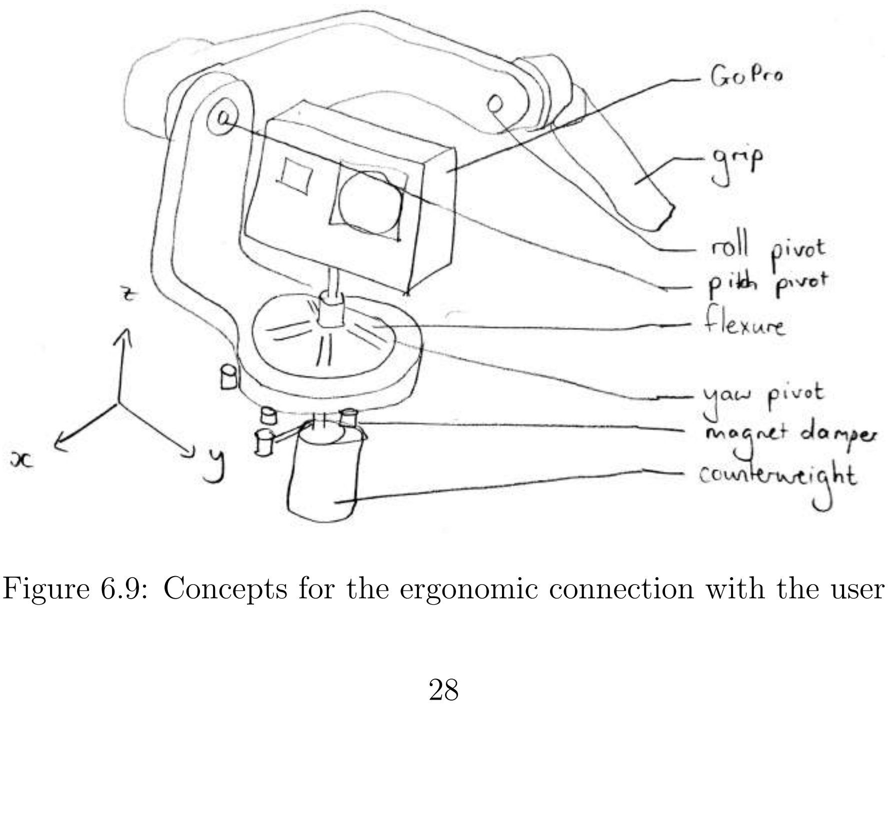


Figure 6.9: Concepts for the ergonomic connection with the user

Chapter 7

Final Design

This section entails the detailed design of various components or sub-systems of the final concept shown in section 6.3. Section 7.1 shows a breakdown of the various parts contained in the final design and sections 7.2 through 7.7 document the design procedure and results for the various sub-systems ranging from bearing selection to the low frequency analysis.

7.1 Final System Naming

Figure 7.1 depicts a CAD model of the final design. Each of the parts are named and serve as a reference for the rest of this chapter.

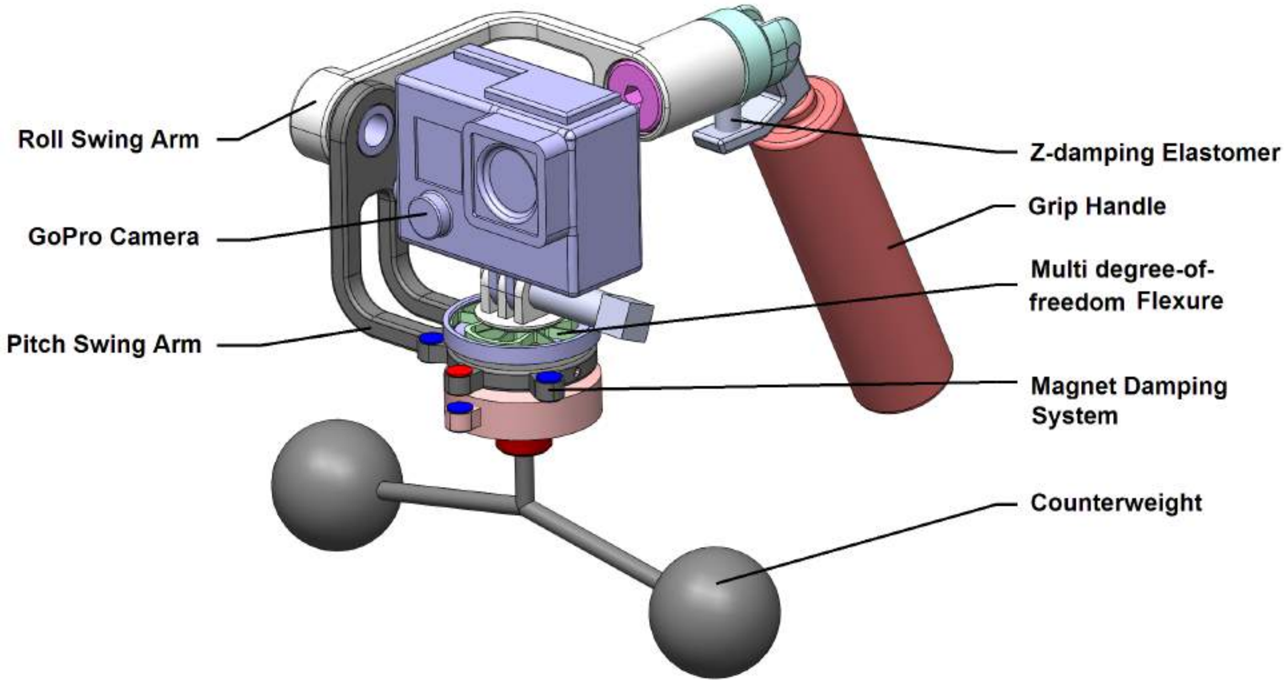


Figure 7.1: Magnet System Free Body Diagram

7.2 Magnet Damping System

A simple diagram of the magnet damping system is shown in Figure 7.2: 3D View. The design idea is based on magnetic repulsion and attraction and using this to provide restoring torques to the yaw platform.

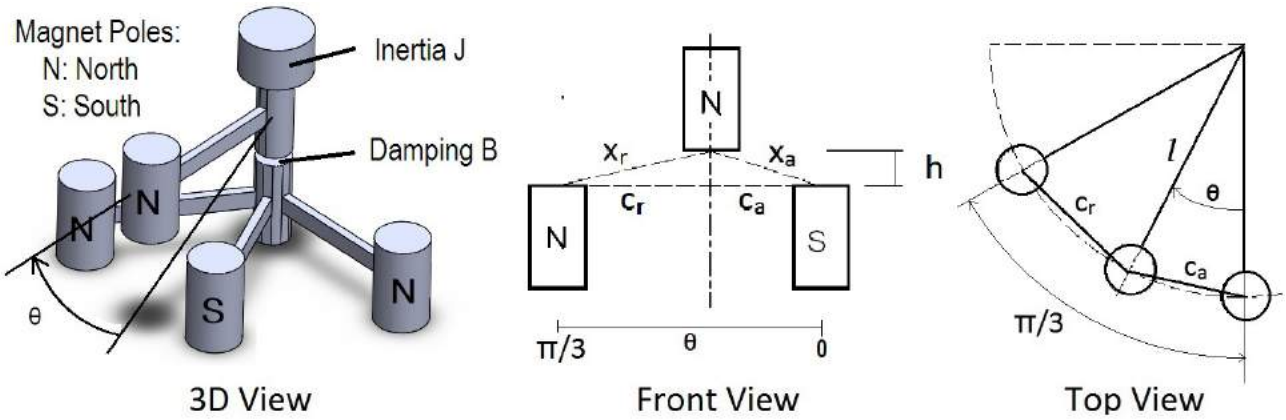


Figure 7.2: Magnet System Free Body Diagram

For the free body diagram shown in Figure 7.2: 3D View, J is the rotational inertia of the yaw platform about the yaw axis. B is the rotational viscous damping coefficient. Additionally, let $T_{\text{magnets}}(\theta)$ represent the restoring torque due to the magnets The governing differential equation that describes the motion of the yaw-platform is therefore:

$$J\ddot{\theta} + B\dot{\theta} + T_{\text{magnets}}(\theta) = 0 \tag{7.1}$$

Calculating $T_{\mathbf{magnets}}(\theta)$

Assuming that magnet repelling or attraction force is proportional to the distance between the two magnets, we can define $T_{\text{magnets}}(\theta)$ using basic trigonometry. Using the geometric diagrams shown in Figure 7.2: Front and Top View, the torque due to the magnetic forces can be expressed as:

$$T_{\text{magnets}}(\theta) = F_a l \cos\left(\frac{\theta}{2}\right) + F_r l \cos\left(\frac{\pi}{6} - \frac{\theta}{2}\right) \tag{7.2}$$

Note that a stands for attract and r for repel. The actual force between two magnets can be approximated [KJMagnetics, 2016] using equation 7.3.

$$F_a = F_{\text{a,max}} \exp(-C_a x_a) \text{ and } F_r = F_{\text{r,max}} \exp(-C_r x_r) \tag{7.3}$$

The coefficients $F_{\text{a,max}}$, C_a, $F_{\text{r,max}}$ and C_r are obtained using the Magnet Calculator from K&J Magnetics [KJMagnetics, 2016] and are discussed in Appendix A section A.4. For equation 7.3, the distances x_a and x_r are calculated using the relations specified in equation 7.4 and 7.5.

$$x_a = \sqrt{h^2 + c_a^2} \text{ and } x_r = \sqrt{h^2 + c_r^2} \tag{7.4}$$

$$c_a = 2l \sin\left(\frac{\theta}{2}\right) \text{ and } c_r = 2l \sin\left(\frac{\pi}{6} - \frac{\theta}{2}\right) \tag{7.5}$$

Solving the Non-Linear ODE

As evidenced in the previous section, $T_{\text{magnets}}(\theta)$ is a highly non-linear function. In order to solve this function, the ODE113 solver from MATLAB Simulink is used to plot the response to an initial displacement $\theta(t) = \theta_i$. Using the magnet catalogue from K&J Magnetics [KJMagnetics, 2016], different magnets were iterated until the design requirements were met. An example of this iteration can be seen in Appendix A section A.4.

Magnet Specification and System Response

Following numerous iterations, the magnet grade and dimensions as shown in table 7.1 were chosen. The response is shown in Figure 7.3. Note that the response has no return overshoot and a lag of at least 2 seconds to it's peak value.

Parameter	Specification
Material	Neobydnym
Grade	N52
Magnet Diameter	7mm
Magnet Height	6mm
Separation Distance h	6mm

Table 7.1: Magnet Specifications

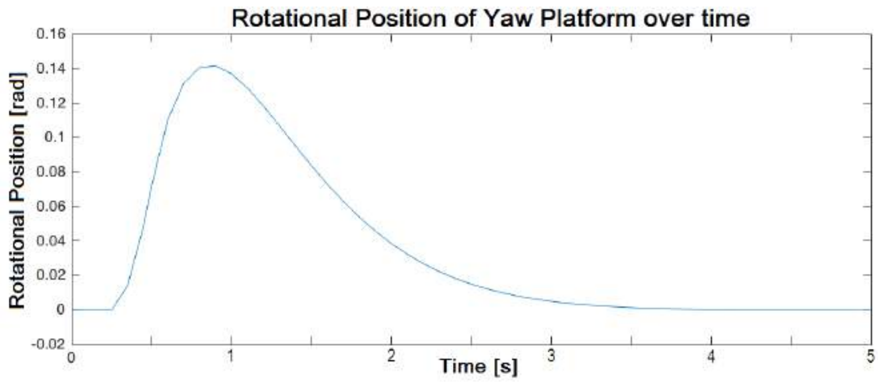


Figure 7.3: Magnet Response

Conclusion

In conclusion, the dynamics of the magnet damping system was represented mathematically and used to iterate multiple magnet specifications until a response was obtained that satisfied the design requirements.

7.3 Multi Degree-of-Freedom Flexure Design

This section entails the design of the elastomer flexure system which will absorb the high frequency vibration in multiple degrees-of-freedom. A flexure can be defined as a structure which has high compliance in certain degrees-of-freedom and high stiffness in other degrees of freedom such that it is able to fulfil certain requirements [Culpepper, 2015].

7.3.1 Basic Flexure Structure and Naming

This design requires relatively low stiffness in the x, y, roll, yaw and pitch directions and high stiffness in the z direction. The proposed solution is shown in Figure 7.4.

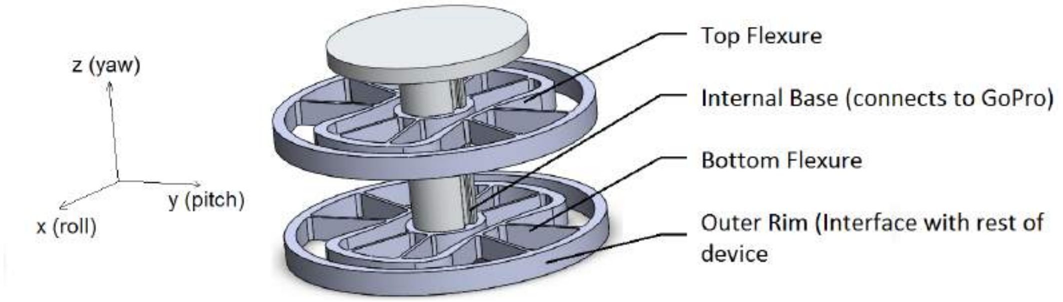


Figure 7.4: Flexure Design

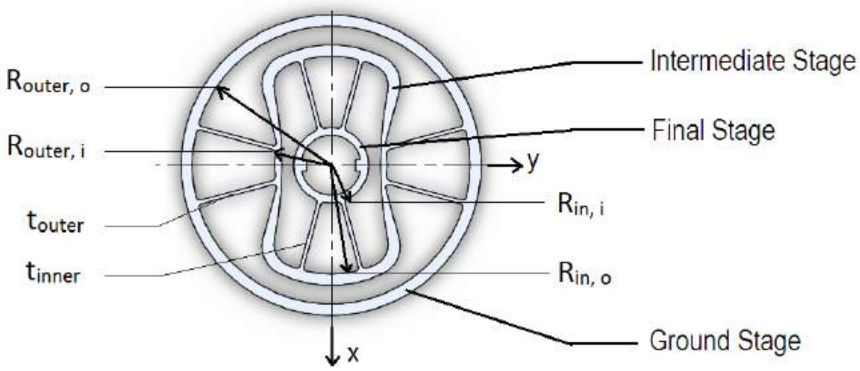


Figure 7.5: Critical Dimensions of the Flexure

7.3.2 Structural Parameters

The flexure will be manufactured from ISODAMP C-1002 material which is covered in section 2.5. The ground, intermediate and final stage will have a stainless steel ring inserted into it. This is so that the low-stiffness of the flexure is only caused by the inner and outer arms. The assumption is that the steel ring has infinite stiffness in comparison with the thermoplastic material.

ISODAMP C-1002 is a material with a very high loss factor of $\eta = 1.0$. As specified in the Literature Study, loss fact $\eta = 2\zeta$ where ζ is the damping ratio. Since the damping ratio

$\zeta = \frac{c}{\sqrt{km}}$ and the mass m is dependent on the rest of the device, the only variable is the stiffness k. Once the stiffness k is determined, it will be known that the system response will be that of a system with damping ratio of 0.5.

The Elastic Modulus E is also an important material parameter as it directly influences stiffness. The Elastic Modulus is obtained from the nomogram shown in Figure 7.6. At a temperature of 20^{o}C and with an input frequency of around 1 Hz, the Elastic Modulus is found to be $E = 10^{7.5} = 32$ MPa.

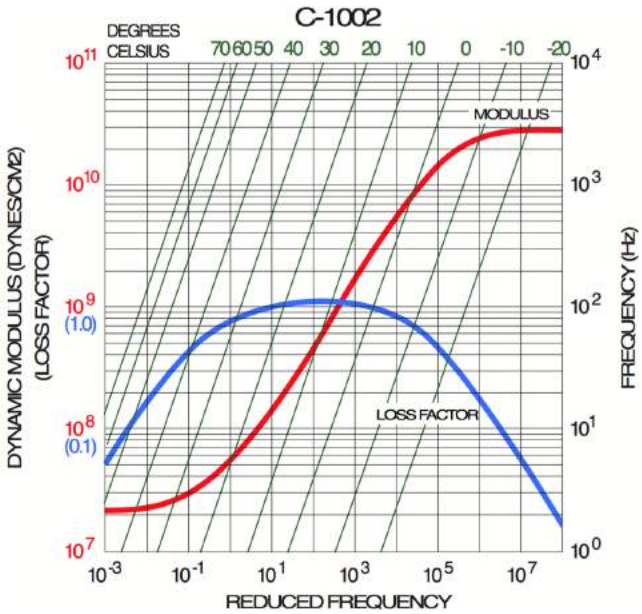


Figure 7.6: Nomogram of C-1002 ISODAMP [EARComposites, 2011]

7.3.3 General Calculation Approach

This section explains the design methodology. The example calculations are done for the y-translation degree-of-freedom.

1. **Calculate the transmissability:**
   For x, the amplitude was 3.4 m/s^2. The design amplitude is 2.5 m/s^2. The transmissability T is therefore:

$$\begin{aligned} T &= \frac{\text{design output Amplitude}}{\text{input Amplitude}} \\ &= \frac{2.5}{3.4} \\ &= 0.74 \end{aligned} \tag{7.6}$$

2. **Determine the required natural frequency:**
   Using the Transmissability graph in Appendix A.6, the transmissability is located on the vertical axis. For the graph associated with damping ratio $\zeta = 0.5$, find the required $\frac{f}{f_n}$ ratio.

   In this case, the ratio $\frac{f}{f_n} = 1.7$. From this, the required natural frequency can be determined. Keeping in mind that $\omega = 2\pi f$, this is calculated as follows:

$$\omega_n = \frac{\omega_{\text{driving}}}{(f/f_n \text{ ratio})} = \frac{2\pi(4.4)}{1.7} = 15.91 rad/s \tag{7.7}$$

3. **Determine the required stiffness:**
Knowing that $\omega_n = \sqrt{\frac{K}{M}}$ The stiffness required is then calculated as follows:

$$K_{\text{required}} = M\omega_n^2 = (0.476kg)(15.92rad/s)^2 = 120.6N/m \tag{7.8}$$

4. **Determine stiffness for given dimensions:**
For the y-direction, it is assumed that only the inner arms contribute a low stiffness. The arms are modelled as simple cantilever beams as shown in Figure 7.7.

Figure 7.7: Relevant deflection of a cantilever beam (Shigley, 2012)

The force-displacement of a cantilever is then determined. Since $F = ky$, the stiffness can also be calculated:

$$F = \frac{3EI}{l^3}y \text{ hence } K = \frac{3EI}{l^3} \tag{7.9}$$

For a rectangular section, the second moment of area is calculated as:

$$I_{\text{upper}} = \frac{1}{12}bh^3 = \frac{1}{12}(0.003)(0.0006)^3 = 5.4\text{E-}14m^4 \tag{7.10}$$

From the Isodamp®C-1000 technical data sheet [EARComposites, 2011], using the reduced frequency nomogram graph for Isodamp®C-1002, the elastic modulus E around 1Hz is 32 MPa. Since springs in series can simply be added together and noting that the upper and lower flexures both contribute to this degree of freedom, the stiffness is calculated as follows:

$$K_{\text{actual}} = \left(4 \text{ arms x } \frac{3EI}{l^3}\right)_{upper} + \left(4 \text{ arms x } \frac{3EI}{l^3}\right)_{lower} = \left(4\frac{3(32\text{E}6)(5.4\text{E-}14)}{0.01^3}\right)_{upper} + \left(4\frac{3(32\text{E}6)(1.62\text{E-}13)}{0.01^3}\right)_{lower} = 84N/m \tag{7.11}$$

5. **Compare results, iterate and optimize:**
   We can now compare the stiffness obtained in equation 7.11 with the required stiffness determined using equation 7.8. If the obtained stiffness K_{actual} is less than the required stiffness K_{required}, then the design objective has been met. This is because a lower stiffness results in a lower natural frequency which, in turn, results in a higher reduction factor.

7.3.4 Alterations to General Calculations

Not all the degrees-of-freedom stiffness values can be calculated using the equations mentioned in the previous section. For example, rotational stiffness relates torque and rotation and not force and displacement as was done in the calculations of the previous section. This section shows the alterations that need to be made for these cases.

X and Y translation

The x and y translation stiffness calculations are done in the same manner as mentioned in the previous section. The only difference being that the x direction uses the outer arms of the flexure which has different dimensions.

Yaw Calculation

Since the yaw direction is a rotational degree-of-freedom, a rotational rather than translational stiffness is required. This is determined as follows:

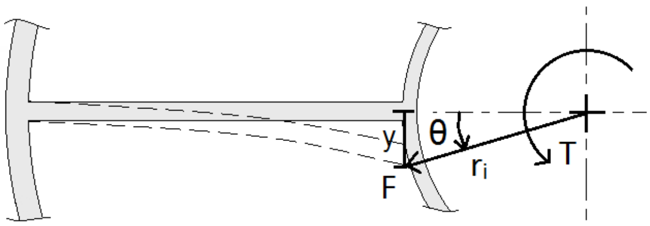


Figure 7.8: Torque to force conversion

By considering Figure 7.8 and noting that $T = Fr_i$ and (assuming small angles), $y = r_i\theta$, the cantilever equation mentioned in equation 7.9 converts to equation 7.12. Since the torsional stiffness K_t is defined as $T = K_t\theta$, the stiffness calculation of equation 7.9 is substituted with the rotational stiffness which is calculated as:

$$T = \frac{3EI}{l^3} r_i^2 \theta \text{ hence } K_t = \frac{3EI}{l^3} r_i^2 \tag{7.12}$$

Additionally, the required stiffness is determined using $K_{t\ \text{required}} = J\omega_n^2$ instead of equation 7.8.

It is also important to note that both the inner and outer arms contribute to the stiffness in the yaw direction. Since both these springs are modelled in series, the total yaw stiffness is calculated using equation 7.13/

$$K_{yaw} = \left(\frac{1}{K_{yaw,outer}} + \frac{1}{K_{yaw,inner}} \right)^{-1} \tag{7.13}$$

Roll and Pitch Calculation

The stiffness model for the roll and pitch rotational directions is shown in the figure below:

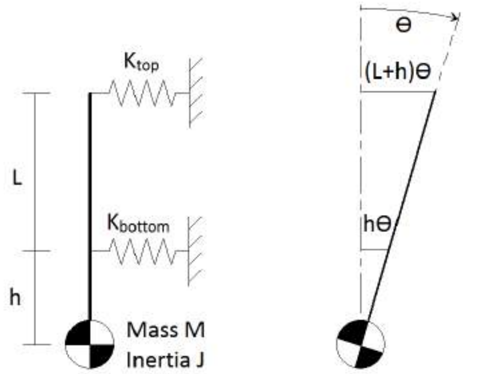


Figure 7.9: Stiffness FBD for roll and pitch

The inner arms contributes to the stiffness shown in the image for the roll case. The outer arms The centre of mass of the inner section is below both of the y-translational springs. Assuming small angles of θ, the equation of motion is given as:

$$J\ddot{\omega} + B\dot{\omega} + (k_{bottom}h^2 + k_{top}(h+L)^2)\omega = 0 \tag{7.14}$$

The roll stiffness is governed by the y-direction translational stiffness. Therefore, k_{bottom} is the stiffness of the lower arm in the y-direction and k_{top} is the stiffness of the upper arm in the y-direction.

Since there are 4 arms per flexure, the stiffness k_{bottom} = 4 x 15.6 N/m = 62.60 N/m and k_{top} = 4 x 5.22 N/m = 20.87 N/m since the stiffness of each individual arm for the lower and upper arms in the y-direction are 15.6 N/m and 5.22 N/m respectively (these values were obtained from the application of equation 7.9). Finally, the rotational stiffness around the roll axis is determined using equation 7.15.

$$\begin{aligned} K_{t,roll} &= (k_{bottom}h^2 + k_{top}(h+L)^2) \\ &= ((62.60 \text{ N/m})(0.010 \text{ m})^2 + (20.87 \text{ N/m})(0.01 \text{ m} + 0.018 \text{ m})^2) \\ &= 0.0226 Nm/rad \end{aligned} \tag{7.15}$$

Z translation

The z-translation direction is the only degree-of-freedom that has a minimum stiffness requirement. This is because the flexure needs to carry the whole yaw-platform (which include the counterweight and the GoPro) without deflecting more than 4mm.

Both the upper and lower flexure's inner and outer arms contribute to the stiffness in the z-direction. Due to the long, tall slender shapes of the arms, the arms are modelled as dual-cantilever beams as shown in 7.10.

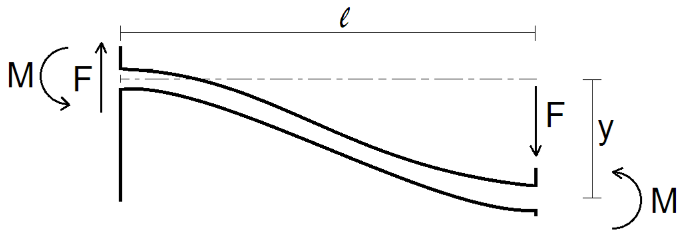


Figure 7.10: Z axis cantilever

The weight of the yaw platform is 0.43 kg (obtained from the SolidWorks CAD model). The static force of the yaw platform is therefore $F = mg = (0.43\text{ kg})(9.81m/s^2) = 4.22$ N. The stiffness of a dual-cantilever beam is derived from Shigley's Mechanical Design [Shigley, 2011]. The stiffness is given as $K = \frac{6EI}{l^3}$. The stiffness of the z-translation can therefore be calculated as:

$$K_z = \left(\frac{1}{K_{z,outer}} + \frac{1}{K_{z,inner}}\right)^{-1} \tag{7.16}$$

The value of $K_{z,outer}$ is determined using the equation below:

$$\begin{aligned} K_{z,outer} &= 4 \text{ x } \left(\frac{6EI}{l^3}\right)_{upper} + 4 \text{ x } \left(\frac{6EI}{l^3}\right)_{lower} \\ &= 4 \text{ x } \left(\frac{6(32\text{E}6)(1.69\text{E-}12)}{(0.0125)^3}\right)_{upper} + 4 \text{ x } \left(\frac{6(32\text{E}6)(4.56\text{E-}11)}{(0.0125)^3}\right)_{lower} \\ &= 18595 \text{ N/m} \end{aligned} \tag{7.17}$$

$K_{z,inner}$ is calculated in a similar fashion. Finally, the stiffness is calculated by applying equation 7.16. This yields the following result:

$$\begin{aligned} K_z &= \left(\frac{1}{K_{z,outer}} + \frac{1}{K_{z,inner}}\right)^{-1} \\ &= \left(\frac{1}{18595 \text{ N/m}} + \frac{1}{29212 \text{ N/m}}\right)^{-1} \\ &= 11400 \text{ N/m} \end{aligned} \tag{7.18}$$

The deflection is calculated as:

$$\begin{aligned} \delta_z &= \frac{F}{K} \\ &= \frac{4.22 \text{ N}}{11400 \text{ N/m}} \\ &= 0.37 \text{ mm} \end{aligned} \tag{7.19}$$

This deflection is less than the maximum allowable deflection of 4mm, hence, the design satisfies the requirements.

Note: Finally, it must be noted that this equation purposefully only takes gravitational acceleration into account when determining the deflection and not the maximum acceleration that could be imposed on the yaw platform. Close inspection of Figure 7.13 in section 7.5 reveals that the cups will stop the flexure from displacing more than 4 mm. Once the flexure displaces by this amount, the load is carried by the cups and the flexure will not fracture.

7.3.5 Final Flexure Design Specifications and Conclusion

All the stiffness equations for the 6 degrees of freedom mentioned previously were compiled into a large Excel spreadsheet. The different flexure parameters were then iterated until all the design requirements were met. The final specification for the flexure is shown in Table 7.2. See Figure 7.5 for reference to each parameter.

Parameter	Value (mm)
Top flexure h	3
Bottom flexure h	9
$R_{outer,o}$	20
$R_{outer,i}$	7.5
$R_{in,o}$	15
$R_{in,i}$	5
t_{outer}	0.75
t_{inner}	0.6

Table 7.2: Design Requirements

The final design specifications compared to the design requirements are shown in table 7.3. Note that the stiffness values are a maximum allowable value and that the final stiffness values are all below the requirement which means it is acceptable.

	Maximum Allowable Stiffness	Designed Stiffness
x	87 N/m	84 N/m
y	87 N/m	84 N/m
Yaw	0.00158 Nm/rad	0.00144 Nm/rad
Roll	0.0316 Nm/rad	0.0226 Nm/rad
Pitch	0.0316 Nm/rad	0.0226 Nm/rad

Table 7.3: Allowable stiffness and final stiffness of the design

In conclusion, a flexural system was designed that satisfies the design requirements for high frequency vibration in the x,y,roll, pitch and yaw directions as well as ensuring a sufficient z-translation stiffness to hold the camera steady.

7.4 Z-translation Damping

The flexure took into account the x,y, roll, pitch and yaw damping but, due to structural strength requirements, the z-translation could not be taken into account. The z-translation damping is implemented by an elastomer inserted at the handle mount. Figure 7.11 shows the actual CAD model as well as the mathematical representation of this model.

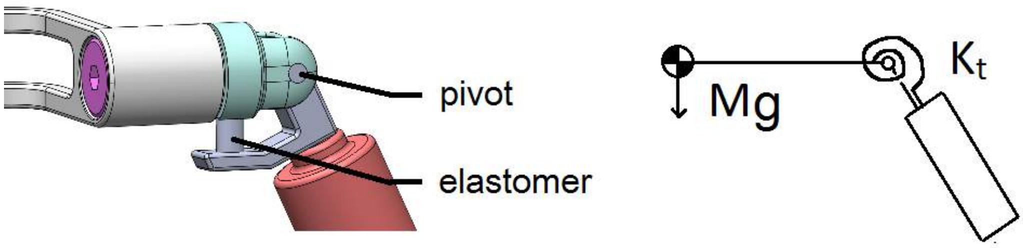


Figure 7.11: CAD Model and Physical Interpretation of Z-damping

To determine the required stiffness, we first require the transmissability using equation 7.6. In this case $T = \frac{3.9\text{m/s}^2}{5.2\text{m/s}^2} = 0.75$. Using the Transmissability graph in Appendix A.6, the frequency ratio $\frac{f}{f_n} = 1.6$. From the section 3.1.2, the input frequency for z-translation $f_{input} = 4.4$ Hz. From this, the natural frequency can be determined as:

$$\begin{aligned} \omega_n &= \frac{2\pi f_{input}}{\text{ration} f/f_n} \\ &= \frac{2\pi(4.4 \text{ Hz})}{1.6} \\ &= 17.3 \text{ rad/s} \end{aligned} \tag{7.20}$$

The required stiffness is calculated as follows:

$$\begin{aligned} K_t &= J\omega_n^2 \\ &= (0.0055 \text{ kg m}^2)(17.3 \text{ rad/s})^2 \\ &= 1.65 \text{ Nm/rad} \end{aligned} \tag{7.21}$$

Where J is the rotational inertia obtained from the SolidWorks CAD model.

The elastomer will be modelled as two springs on the opposite sides of a pivot as shown in Figure 7.12.

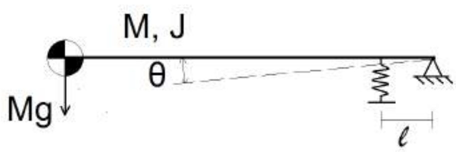


Figure 7.12: Equivalent to Torsional Spring

From K_t, the required stiffness of each spring can be determined. Let x be the spring displacement. Hence, assuming small angles of θ ($l\theta \approx x$), $K_t\theta = Kxl = Kl^2\theta$. The required translational stiffness of the elastomer is calculated as:

$$\begin{aligned} K &= \frac{K_t}{l^2} \\ &= \frac{1.65 \text{ Nm/rad}}{(0.006 \text{ m})^2} \\ &= 45833 \text{ N/m} \end{aligned} \tag{7.22}$$

Shape Factor S and the corrected Elastic Modulus are required to calculate the stiffness.

$$\begin{aligned} S &= \frac{\text{Area Under Load}}{\text{Area Free to Bulge}} \\ &= \frac{\pi(0.007^2 - 0.006^2) \text{ m}^2}{0.01(2\pi(0.007) + 2\pi(0.006) \text{ m}^2} \\ &= 0.05 \end{aligned} \tag{7.23}$$

As determined in section 7.3.2, the Elastic Modulus of C-1002 ISODAMP thermoplastic is 32 MPa. Using the Shape Factor $S = 0.05$, the corrected Elastic Modulus is calculated as:

$$\begin{aligned} E_{corrected} &= E(1 + 2S^2) \\ &= (32\text{E}6 \text{ MPa})(1 + 2(0.05)^2) \\ &= 32.16\text{E}6 \text{ MPa} \end{aligned} \tag{7.24}$$

Following the E-A-R Elastomer Design Manual [Frankovich, 2011], the stiffness of an elastomer can be calculated as:

$$\begin{aligned} K &= \frac{E_{corrected}(a_o^2 - a_i^2)}{t} \\ &= \frac{32.16\text{E}6 \text{ MPa}(0.007^2 - 0.006^2)}{0.01} \\ &= 41808 \text{ N/m} \end{aligned} \tag{7.25}$$

This calculated stiffness is lower than the required stiffness of 45833 N/m. Hence, the design requirement is satisfied. Iterative optimization yields the dimensions as specified in Table 7.4.

Parameter	Value
Outer radius a_o	0.007 m
Inner radius a_i	0.006 m
Thickness t	0.01 m

Table 7.4: Final Dimensions of Z-damping Elastomer

7.5 Fatigue and Stress analysis on Flexure Design

The flexure dead stop feature is shown in Figure 7.13. The principle is that the upper and lower cups ensure that the flexure does not displace more than 4 millimetres even under excessive loads.

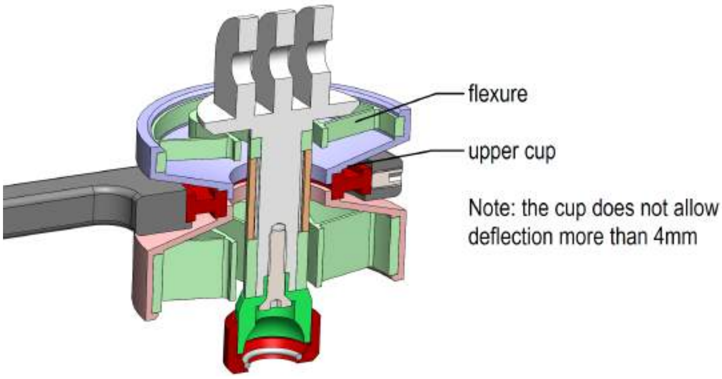


Figure 7.13: Section view of elastomer design

From the E-A-R Composites Design Manuals [EARComposites, 2011], C-1002 Elastomer Material has the following properties listed in table 7.5.

Property	Value
Ultimate Tensile Stress S_{uts}	45 MPa
Endurance Limit S_e	28 MPa

Table 7.5: Structural Properties of C-1002 Thermoplastic

Modelling the arms as long cantilever beams, the stress in the material is determined using equation 7.26. To simplify the calculation, we conservatively only consider the taller bottom flexure to carry the load. For a displacement of 4 mm, The moment $M = 3.2\text{E-}4Nm$ and $I = 1.62\text{E-}13m^4$. The maximum stress amplitude σ_a in the arm is calculated using equation 7.26.

$$\begin{aligned}\sigma_a &= \frac{M_a y}{I} = \frac{(3.2\text{E-}4Nm)(0.0045m)}{1.62\text{E-}13m^4} \\ &= 8.9\text{MPa}\end{aligned} \tag{7.26}$$

The mean stress calculated similarly and yields $\sigma_m = 1.2$MPa. Using the ASME Elliptic Fatigue Failure Criterion, we determine the safety factor against fatigue using equation 7.27.

$$\begin{aligned}n_f &= \sqrt{\frac{1}{(\frac{\sigma_a}{S_e})^2 + (\frac{\sigma_m}{S_{uts}})^2}} \\ &= \sqrt{\frac{1}{(\frac{8.9}{16.2})^2 + (\frac{1.2}{45})^2}} \\ &= 3.13 > \text{ design Safety Factor } = 3.0\end{aligned} \tag{7.27}$$

7.6 Structural Analysis

This section aims at providing a first order analysis that confirms that the structure of the GoPro stabilizer does not require further in-depth structural analysis. The arm will be made of T6061 aluminium. Referencing the ASM Material Data Sheet [MatWeb, 2016], the properties for this specific aluminium alloy are displayed in Table 7.6.

Property	Value
Ultimate Tensile Stress S_{uts}	310 MPa
Tensile Yield Stress S_y	276 MPa

Table 7.6: Structural Properties of Aluminium [MatWeb, 2016].

Through inspection, the estimated weakest part of the aluminium arm is located at the bend in the Pitch Swing Arm. This is because the second moment of inertia will be the lowest at this point due to it's low vertical height.

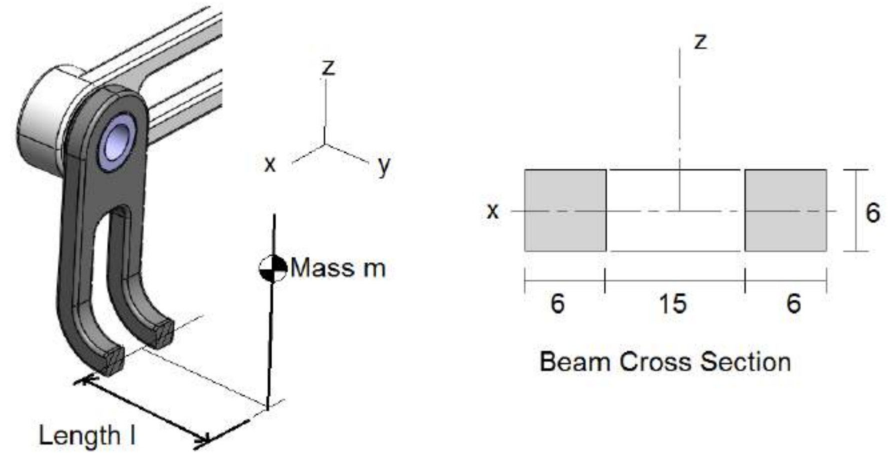


Figure 7.14: Pitch Swing Arm Cross Section

From section 3.1.4, the maximum acceleration in the z-direction is 49 m/s^2. From this, the following parameters are calculated:

1. The force on the arm is calculated by $F = ma_{max} = (0.43\text{kg})(49m/s^2) = 21.1\text{N}$.
2. The internal moment on the arm is calculated as $M = Fl = (21.1\text{N})(0.03\text{m}) = 0.633$ Nm.
3. The inertia is calculated as $I = 2\frac{1}{12}bh^3 = 2\frac{1}{12}(0.006)(0.006)^3 = 2.16\text{E-}10$ m^4.

Note that the factor 2 is due to the fact that there are two squares in the cross-section. The following equations are given by Mechanics of Materials [Hibbeler, 2014]. The maximum bending stress in the bar is calculated as follows:

$$\begin{aligned}\sigma_{bending,max} &= \frac{My}{I} \\ &= \frac{(0.633)(0.003)}{2.16\text{E-}10} \\ &= 8.78 \text{ MPa}\end{aligned} \tag{7.28}$$

The maximum shear stress in the bar is calculated in equation 7.29. In order to determine the shear stress, we first need the value of Q. This maximum value of Q is at the x-axis and is calculated as $Q_{max} = \bar{y}'A' = 2(0.0015)(0.006)(0.003) = 5.4\text{E-}8\ \text{m}^3$.

$$\begin{aligned}\sigma_{shear,max} &= \frac{VQ}{It} \\ &= \frac{(21.1\ \text{N})(5.4\text{E-}8\ \text{m}^3)}{(2.16\text{E-}10\ \text{m}^4)(0.006\ \text{m})} \\ &= 0.88\ \text{MPa}\end{aligned} \tag{7.29}$$

It is noticed that the bending stress is the dominating stress and will therefore be the only stress considered in this analysis.The safety factor against yielding is calculated as

$$\begin{aligned}n_f &= \frac{276\ \text{Mpa}}{8.78\ \text{MPa}} \\ &= 31.4\end{aligned} \tag{7.30}$$

This very high safety factor confirms that it is not necessary to perform a more in-depth structural analysis of the design.

Conclusion

This section set out to prove that it would not be necessary to do any further strength calculations on the GoPro stabilizer. This was evidenced by the high final safety factor against yielding of 31.4 at the assumed weakest point of the structure. Even if the assumed weakest point is indeed not the weakest point, the high safety factor provides the confidence that a weaker point will also be strong enough.

7.7 Low Frequency Reaction

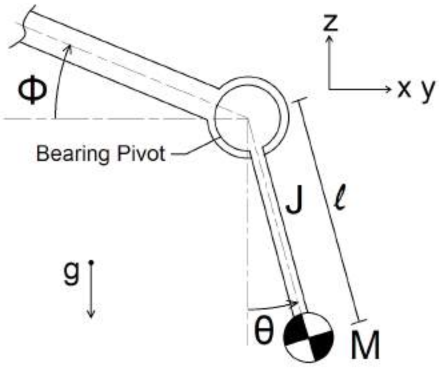


Figure 7.15: Free Body Diagram for Bearing Pivot

For both the roll and pitch reaction, we model the arm as a point mass m, distance to the centre of mass l with a predetermined inertia J as can be seen in Figure 7.15. The angle θ references the displacement of the roll or pitch arm and the angle ϕ is an input function that represents the user's movement. For this analytical solution, we assume that $\sin\theta \approx \theta$. The equation of motion becomes:

$$J\ddot{\theta} + B\dot{\theta} + mgl\theta = -B\dot{\phi(t)} = -B\omega A\cos(\omega t) \tag{7.31}$$

The total response of the system for forced vibration, assuming zero initial conditions, as defined by Mechanical Vibrations chapter 3.4 [Rao and Fah, 2011], is given as $\theta(t) = X\cos(\omega t - \psi)$. We are only interested in X since that represents the maximum angular displacement seen by the swim arm. X is determined as follows:

$$X = \frac{\delta_{st}}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}} \tag{7.32}$$

Where $r = \frac{\omega}{\omega_n}$, $\zeta = \frac{B}{2\sqrt{mglJ}}$ and $\delta_{st} = \frac{-B\omega A}{mgl}$.

Roll Case

For the roll case, the mass $m = 0.523kg$, $J = 0.0014kgm^2$, $B = 0.03Nms/rad$ and $l = 0.057m$. These values were obtained from the SolidWorks CAD model. See Appendix A.2 on how the damping B was obtained.

We let the input function $\phi(t) = 0.5\sin(t)$ as specified in section 3.1.3. Hence, $\omega = 1$ and $A = 0.5$. The natural frequency $\omega_n = \sqrt{\frac{J}{mgl}} = 0.07rad.s^{-1}$.

$$r = \frac{\omega}{\omega_n} = \frac{1}{0.07} = 14.28 \tag{7.33}$$

$$\zeta = \frac{B}{2\sqrt{mglJ}} = \frac{0.03}{2\sqrt{(0.523)(9.81)(0.057)(0.0014)}} = 0.74 \tag{7.34}$$

$$\delta_{st} = \frac{-B\omega A}{mgl} = \frac{(-0.03)(1)(0.5)}{(0.523)(9.81)(0.057)} = -0.051 \tag{7.35}$$

The magnitude of the amplitude X is determined using equation 7.32:

$$\begin{aligned} X &= \frac{\delta_{st}}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}} \\ &= \frac{|-0.051|}{\sqrt{(1-(14.28)^2)^2 + (2(0.74)(14.28))^2}} \\ &= 2.5\text{E-}4 \text{ rad} \end{aligned} \tag{7.36}$$

The demonstrated procedure is then repeated for the pitch case. Once again, the physical parameters such as the inertia J, mass m and length l are obtained from the SolidWorks CAD model. Table 7.7 summarizes these results.

	Roll Axis	Pitch Axis
Inertia J	0.0014 $kg.m^2$	0.0014 $kg.m^2$
Damping B	0.03 Nms/rad	0.03 Nms/rad
Mass m	0.523 kg	0.473 kg
Length l	0.057 m	0.057 m
Input $\phi(t)$	$0.5\sin(t)$ rad	$0.5\sin(t)$ rad
δ_{st}	-0.051 m	-0.057 m
ζ	0.74	0.78
Response Amplitude X	2.5E-4 rad	2.8E-4 rad

Table 7.7: Calculation Parameters and Final Response Amplitude

From table 7.7 it is evident that with a sinusoidal hand movement of 0.5 radians (28 degrees) will almost cause minimal disturbance to the roll and pitch arms respectively.

Non linear analysis

It must be noted that the assumptions that $\sin\theta \approx \theta$ and that damping is a linear function of rotational velocity are highly inaccurate. A Simulink model was developed that takes this non-linearity into account as well as a non-linear function for the damping coefficient. For brevity, this is omitted here and rather discussed in section A.3 of Appendix A.

Conclusion

This section determined the effect a user's sweeping hand motion would have on the low frequency movement of the GoPro stabilizer. Due to the low bearing viscous damping, it was shown that the response amplitude was much lower than required (0.00025 rad was obtained and 0.1 rad was desired). This fully satisfies the design requirements. An additional comment was made stating that a non-linear analysis could improve the accuracy of this section a lot and could be looked into for future designs.

7.8 Design Process Conclusion

A variety of design aspects were not documented. This is because of a variety of reasons such as:

1. The design is part of a CAD iteration that evolves over time but cannot be documented;
2. The designed parameter is not a fundamental aspect of the working of the GoPro stabilizer and would distract the reader from more important design procedures;
3. The design parameter can simply not be documented using mathematical engineering relations as it is a overall material choice or a geometrical consideration

Some of the design requirements were not directly designed towards in the aforementioned sections. However, these requirements were still considered and fulfilled. The following items were addressed:

1. **Waterproofing:** All the components of the GoPro stabilizer use either aluminium, stainless steel, plastic or other non-corroding materials. In this way, the design requirement that the device should be water resistant was met.
2. **Geometrical Considerations:** The final design takes the Field of View of the GoPro into account. Multiple CAD iterations were conducted on SolidWorks which resulted in the GoPro being able to rotate 45 degrees in any direction without being disturbed by the device structure.
3. **Ergonomics:** The handle was designed according to the design specifications. The length of the handle is 11cm and the diameter is 4.5cm which conforms to the design requirements.

In conclusion, all the design requirements were addressed and met. The result is a final design that, according to first order analysis and MATLAB simulations, should function as intended. Since all the design requirements were met, this inherently means that the user requirements are met.

Chapter 8

Additional Analyses

8.1 Manufacturing Analysis and Schedule

This manufacturing analysis aims at specifying the materials and manufacturing procedures that will be followed in order to manufacture all the components for the GoPro stabilizer.

Referencing the part names shown in Figure 7.1, Table 8.1 shows every part that requires manufacturing, the material which that part will be made of as well as the manufacturing methods that will be required to manufacture this part

	Part Description	Material	Manufacturing Procedure or Machinery
1.	Grip handle	aluminium	Lathe
2.	Roll spindle	plastic	injection moulding
3.	Bearing bolt	aluminium	machining
4.	Swing arm - pitch	aluminium	injection moulding
5.	Swing arm - roll	aluminium	injection moulding
6.	Cup - upper	plastic	injection moulding
7.	Cup - lower	plastic	injection moulding
8.	Internal pin	plastic	injection moulding
9.	Flexure upper	Isodamp ®C-1002	laser cutting
10.	Flexure lower	Isodamp ®C-1002	laser cutting
11.	Z-damping Flexure	Isodamp ®C-1002	laser cutting
12.	Ball cup	plastic	injection moulding
13.	Fastener	plastic	injection moulding

Table 8.1: Part materials and manufacturing procedures

Manufacturing schedules were devised for the Roll Arm and the Internal Pin. Both these parts were drawn and can be viewed in section 9. The manufacturing schedule details the machining process that needs to be done once each part has been cast.

8.1.1 Manufacturing Schedule of Roll Arm

The Roll Arm will be cast from aluminium using a centrifugal casting procedure. Once this part has been cast, it will need to be machined so that all the dimensional tolerances are met.

Roll Arm Manufacturing Schedule				
Step	Task	Machine	Tooling	Measurement
1	Aluminium Cast part	n/a	n/a	n/a
2	Obtain finished casting	n/a	n/a	n/a
3	Debur rough edges	n/a	deburring tool	n/a
4	Positioning	Mill	Locator	n/a
5	Ream 21mm Dia Bearin Seat A	Mill	ø 21mm Reamer	n/a
6	Rotate Part			
7	Positioning	Mill	Locator	n/a
8	Ream 21mm Dia Bearin Seat A	Mill	ø 21mm Reamer	n/a
9	Check Concentricity	n/a	n/a	CMM
10	Positioning	Mill	Locator	n/a
11	Ream 21mm Dia Bearin Seat B	Mill	ø 21mm Reamer	n/a
12	Rotate Part			
13	Positioning	Mill	Locator	n/a
14	Ream 21mm Dia Bearin Seat B	Mill	ø 21mm Reamer	n/a
15	Check Concentricity	n/a	n/a	CMM

Table 8.2: Part materials and manufacturing procedures

Note: This component has been drawn and can be viewed in section 9.

8.1.2 Manufacturing Schedule of Internal Pin

The Internal Pin will be cast from aluminium using a die casting procedure. Once this part has been cast, it will need to be machined so that all the dimensional tolerances are met.

Internal Pin Manufacturing Schedule				
Step	Task	Machine	Tooling	Measurement
1	Obtain finished casting	n/a	n/a	n/a
2	Debur rough edges	n/a	deburring tool	n/a
3	Positioning	Mill	Locator	n/a
4	Drill 2.5mm Dia hole, depth 7.5mm	Mill	2.5 mm Drill	Depth
5	Tap M3, depth 6mm	n/a	M3 Tap	n/a
6	Positioning	Mill	Locator	n/a
7	Drill 5mm diameter hole, through all	Mill	5mm Drill	n/a

Table 8.3: Part materials and manufacturing procedures

Note: This component has been drawn and can be viewed in section 9.

8.2 Maintenance Analysis

This section details the projected maintenance that should be done on the GoPro stabilizer. Since the frequency of use of the GoPro stabilizer is dependent on the user, the maintenance cycle is specified in terms of hours used instead of a date-based maintenance scheme (as seen for most mechanical appliances).

8.2.1 Flexures

C-1002 ISODAMP Thermoplastic is a very resilient material but excessive exposure to sun can cause it to degrade and eventually cause brittle fracture [EARComposites, 2011]. E-A-R Specialty Composites recommends that the sun-exposed flexure be replaced after approximately 120 hours of use to ensure optimum performance.

Part	Allowable Hours of Use	Cost per part
Upper Flexure	120 hours	R18.28
Z-damping Flexure	120 hours	R15.46

Table 8.4: Flexure Maintenance Schedule

8.2.2 Bearings

Due to the low loads on the bearings, no excessive wear is expected on the bearing races and hence, no bearing replacement plan is made. However, in order to ensure that the bearings roll smoothly, a lubrication schedule has been developed. The type of lubrication (lube) and application frequency is dependent on the environment the device will be used in. The maintenance schedule is shown in Table 8.5.

Use Environment	Lubrication Cycle	Lubricant Type
Fully dry conditions	150 hours	dry wax-based
Mostly dry conditions	120 hours	dry teflon lube
Mostly wet conditions	50 hours	ceramic wet lube

Table 8.5: Bearing Maintenance Schedule

Note: The lubrication cycles and lubrication type is based on bicycle chain types. The information in the table was obtained from Finish Line Cycles's lubrication section [FinishLine.com, 2016].

8.2.3 Grip Replacement

The necessity for a grip replacement is largely a function of the user's grip strength and amount of perspiration when using the device. A maintenance schedule based on hourly use is therefore somewhat futile as it will largely depend on each user. However, it must be noted that the design has allowed for seamless grip replacement if the need were to arise to do so.

8.3 Cost Analysis

For this design, and generally for most design projects, the cost that leads up to the final product is important. This will be one of the key factors regarding whether or not the design actually gets implemented and used. A cost analysis was done to document the total expected cost of the design. However, it is important to note that this is a first order design and, hence, the costs could potentially be lowered considerably using internal manufacturing and production methods as well as supplier contracts that allow more affordable material rates.

8.3.1 Standard Components

The costs of standard components such as bearings, magnets and other miscellaneous items are listed in Table 8.6.

Component	Part No. / Details	Supplier	Unit Cost	Qty.	Total Cost
Yaw Bearing	SKF-61801-2RS1	SKF Bearings	R41.23	1	R41.23
Roll Bearing	SKF-6004	SKF Bearings	R30.71	2	R61.42
Pitch Bearing	SKF-6004	SKF Bearings	R30.71	2	R61.42
Grip	FHG-22 NPVC 1 in.	Gripworks.com	$1.56	1	R24.40
Magnet	D14-N52	K&J Magnetics	$0.23	4	R14.38
M3x0.5 Bolt		McMaster.com	$0.08	1	R1.25
Belleville		McMaster.com	$0.05	4	R3.13
Total Cost					R208.23

Table 8.6: Projected Costs for Standard Components

8.3.2 Manufactured Parts

Costs of parts that need to be manufactured specifically for this design are listed in Table 8.7. Note that it is assumed that manufacturing will be outsourced. The listed amounts therefore take the material and manufacturing into account.

Component	Material	Supplier	Unit Cost	Qty.	Total Cost
Roll Arm	Aluminium	Proto-Labs	$2.63	1	R41.13
Pitch Arm	Aluminium	Proto-Labs	$2.71	1	R42.38
Grip Handle	Aluminium	Alibaba	$1.78	1	R27.84
Cup - upper	High-density Polyethylene	Alibaba	$0.63	1	R9.85
Cup - lower	High-density Polyethylene	Alibaba	$0.77	1	R12.04
Counterweight	Steel	Hebei Inc.	$1.23	1	R19.23
Total Cost					R152.47

Table 8.7: Projected Costs for Manufactured Parts

8.3.3 Raw Materials

The flexures will need to be manufactured internally. The flexures are made of steel rings infused in C-1002 ISODAMP thermoplastic structures. The raw material costs are tabulated in Table 8.8.

Material	Supplier	Cost	Qty.	Total Cost
ISODAMP C-1002	E-A-R Comp.	\$0.12/cm^3	15.1 cm^3	R28.34
1mm Stainless Steel Sheet	Corr-line	R0.54 / cm^2	10 cm^2	R5.40
Total Cost				R33.74

Table 8.8: Projected Costs for Raw Materials

8.3.4 Labour Costs

The labour for the manufacturing of the flexures will require laser cutting and bending. The projected costs are tabulated in Table 8.9.

Operation	Hourly Rate	Estimated Labour Time	Total Cost
Laser Cutting	R100/hour	0.5 hours	R50
Bending	R100/hour	0.25 hours	R25
Total Cost			R75

Table 8.9: Projected Labour Costs

Note: Shipping delivery was taken into account where necessary. Additionally, the US\$-ZAR exchange rate was taken as R15.64/\$1.00

Important Consideration: The material and manufacturing costs are naturally dependent on the supplier. However, as will be discussed in section 8.4, the social and financial impact of outsourcing could overturn the financial benefits of cheaper outsourced manufacturing. This analysis documents the most financially appealing option but further investigation should be done by the client when the part is actually manufactured.

8.3.5 Total Cost

The total projected cost for the GoPro stabilizer is **R469.44**. However, it must be reiterated that this is just a first order cost analysis and needs to be considered as a rough estimate rather than an exact value.

8.4 Impact of Design

It is important to consider the impact that the design could have on a variety of non-technical aspects. This section aims at delving into these aspects in order to obtain a better understanding of the possible impacts this design could have.

Social Impact

As mentioned in section 8.1, the manufacturing of parts could be done in countries other than South Africa. The parts that require aluminium casting could be manufactured in China, India or the USA. This is because the cost of manufacturing in these countries is significantly cheaper than doing it locally.

One could consider manufacturing the parts locally at a company like Wahl Manufacturing (which is based in Johannesburg). This could have a positive social impact on South Africa's manufacturing industry. However, the client must carefully weigh the social benefits of local manufacturing compared to the financial benefits of outsourced manufacturing.

Environmental Impact

The design utilizes materials such as aluminium, thermoplastics and other high strength plastics. These materials usually go through intense mining and forming procedures. This has a significant impact on the environment and should be taken into account by the client.

Legal Impact

The design couples with the GoPro Hero 4 action camera which is a relatively expensive electronic device. Even though it has been designed to hold the GoPro camera very securely, it must be noted that some users could manage to damage their GoPro cameras with improper use of the design. It is recommended that this be taken into account and that the client who plans on selling this designed device considers a legal clause that removes him from the user were the GoPro to get get damaged while using the device.

Impact on Health and Safety

The user running with this device could potentially get distracted by it and injure themselves. This is possibly something that must be made clear to the end-user by the client.

Chapter 9

Drawings

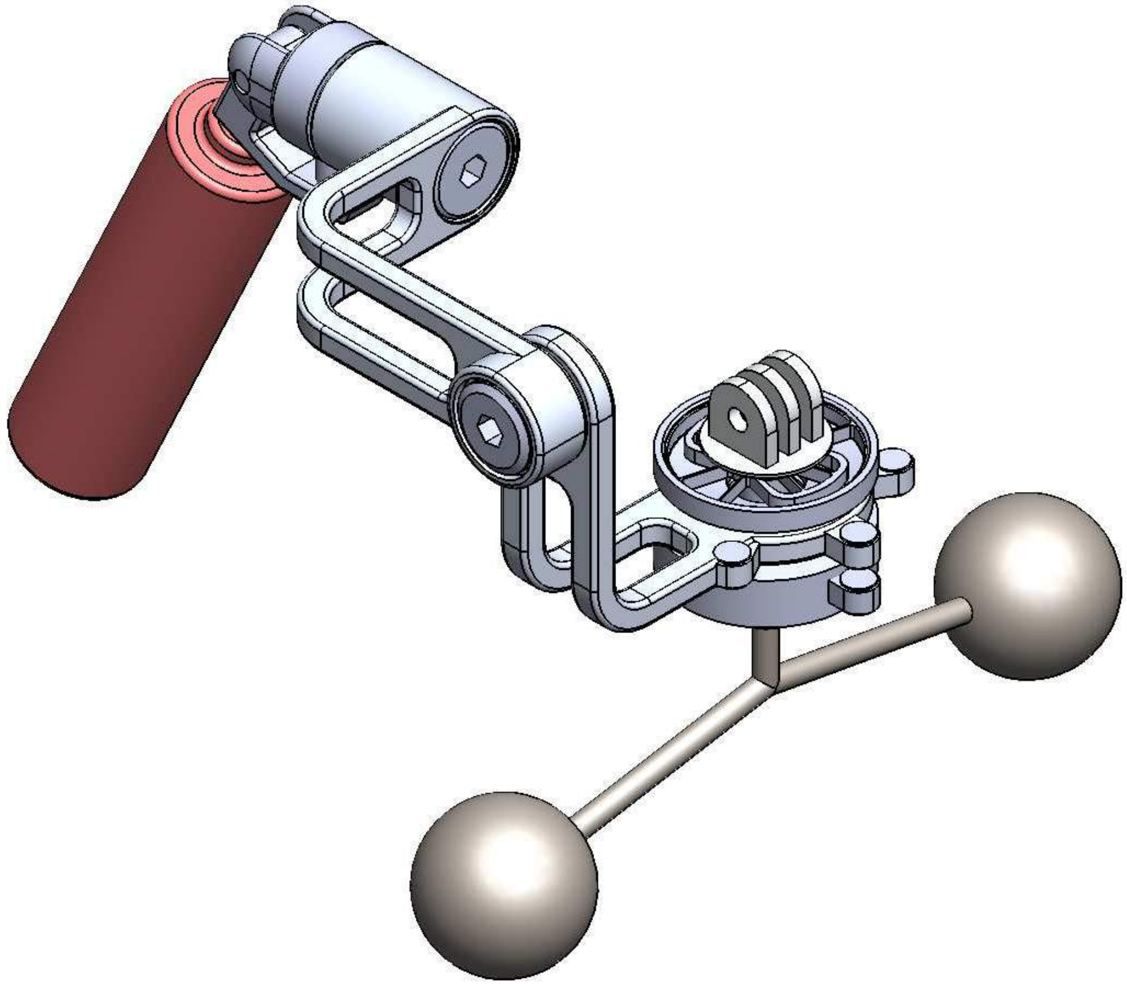
This section entails six technical drawings of the final design.

Four CAD and two hand drawings were made of the final design. The drawings that were drawn with the SolidWorks CAD software are:

1. Isometric Assembly
2. Assembly Drawing
3. Roll Arm
4. Internal Pin

Two hand drawn drawings were made. These are:

1. Cup - upper
2. Grip Handle



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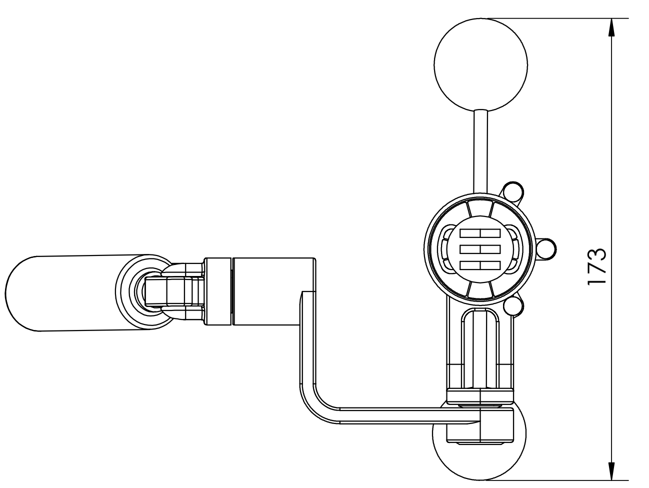

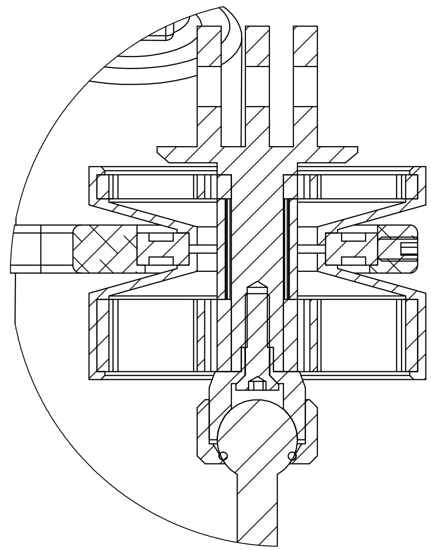

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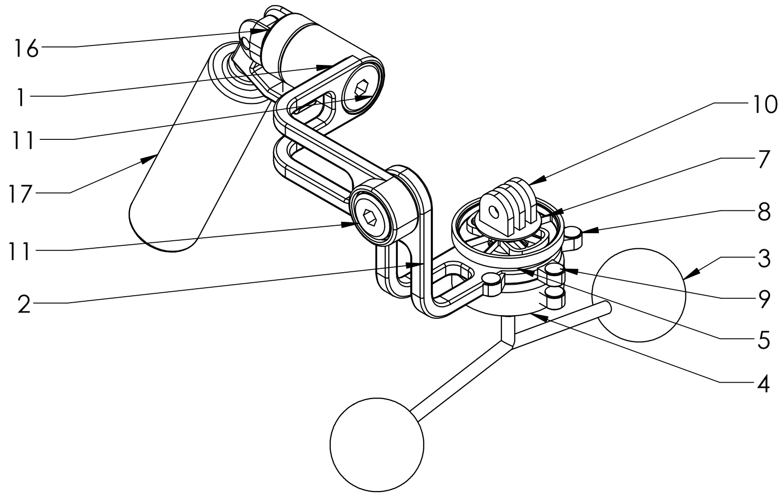

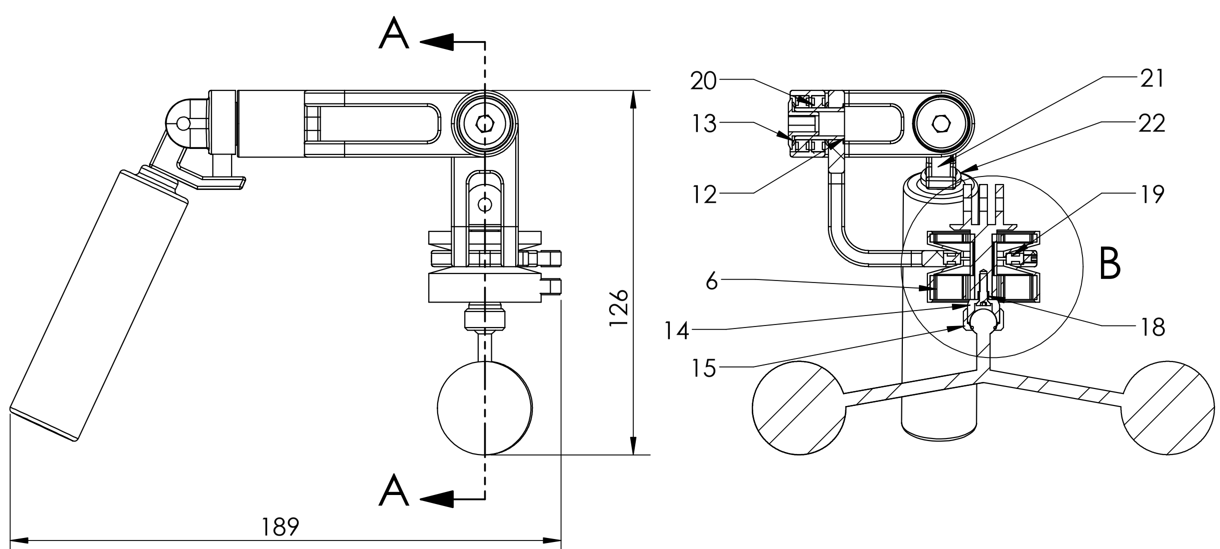

SECTION A-A

22	1	Handle Pivot	Aluminium	
21	1	Z-damping Elastomer	Thermoplastic	EAR-C-1002
20	4	Bearing - pitch and roll	Stainless Steel	SKF-6004
19	1	Bearing - yaw	Stainless Steel	SKF-61801-2RS1
18	1	M3x0.5mm Bolt	Steel	
17	1	Grip	Aluminium	
16	1	Handle Mount	Aluminium	
15	1	Counterweight Screw	Aluminium	
14	1	Counterweight Plug	Plastic	
13	4	Belleville Washer	Steel	
12	1	Pitch Plug	Aluminium	
11	1	Pitch Screw	Aluminium	
10	1	Internal Pin	Steel	
9	1	Magnet - S	Neobydnym	
8	3	Magnet - N	Neobydnym	
7	1	flexure - upper	Thermoplastic	EAR-C-1002
6	1	flexure - lower	Thermoplastic	EAR-C-1002
5	1	Cup - upper	Plastic	
4	1	Cup - lower	Plastic	
3	1	Counterweight	Steel	
2	1	Swing Arm - pitch	Aluminium	
1	1	Swing Arm - roll	Aluminium	
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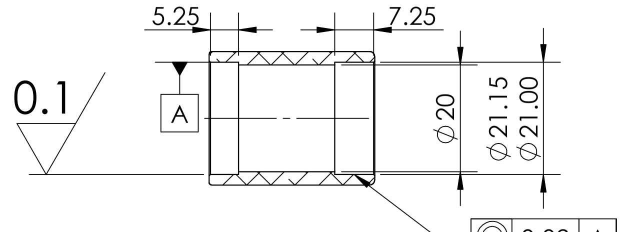
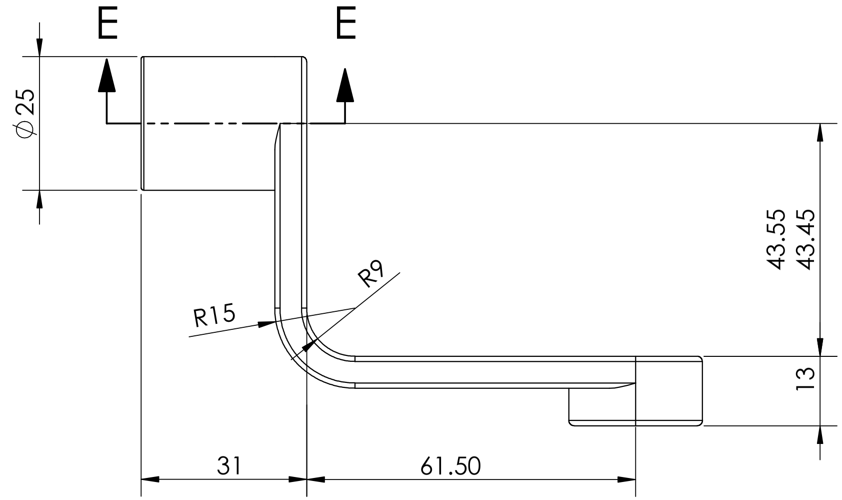
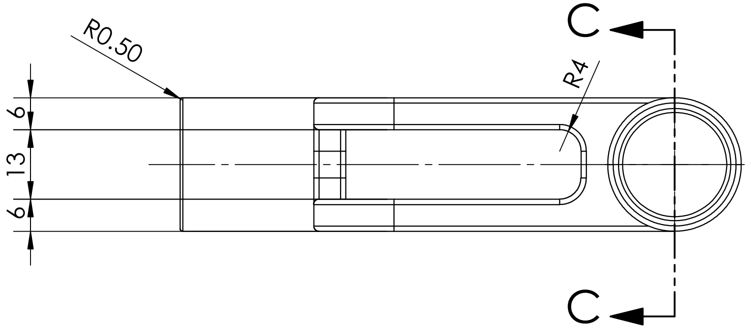
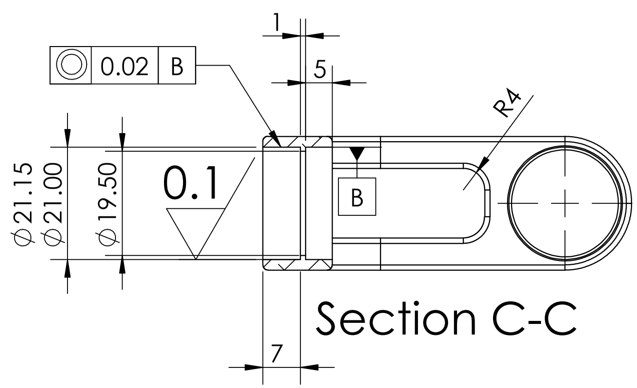


SECTION E-E







Section C-C

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STUDENTE NO / STUDENTE NR: 13015142 | RIGTING / DISCIPLINE: Mech | TEKENING NR / DRAWING NR: Roll_arm_1

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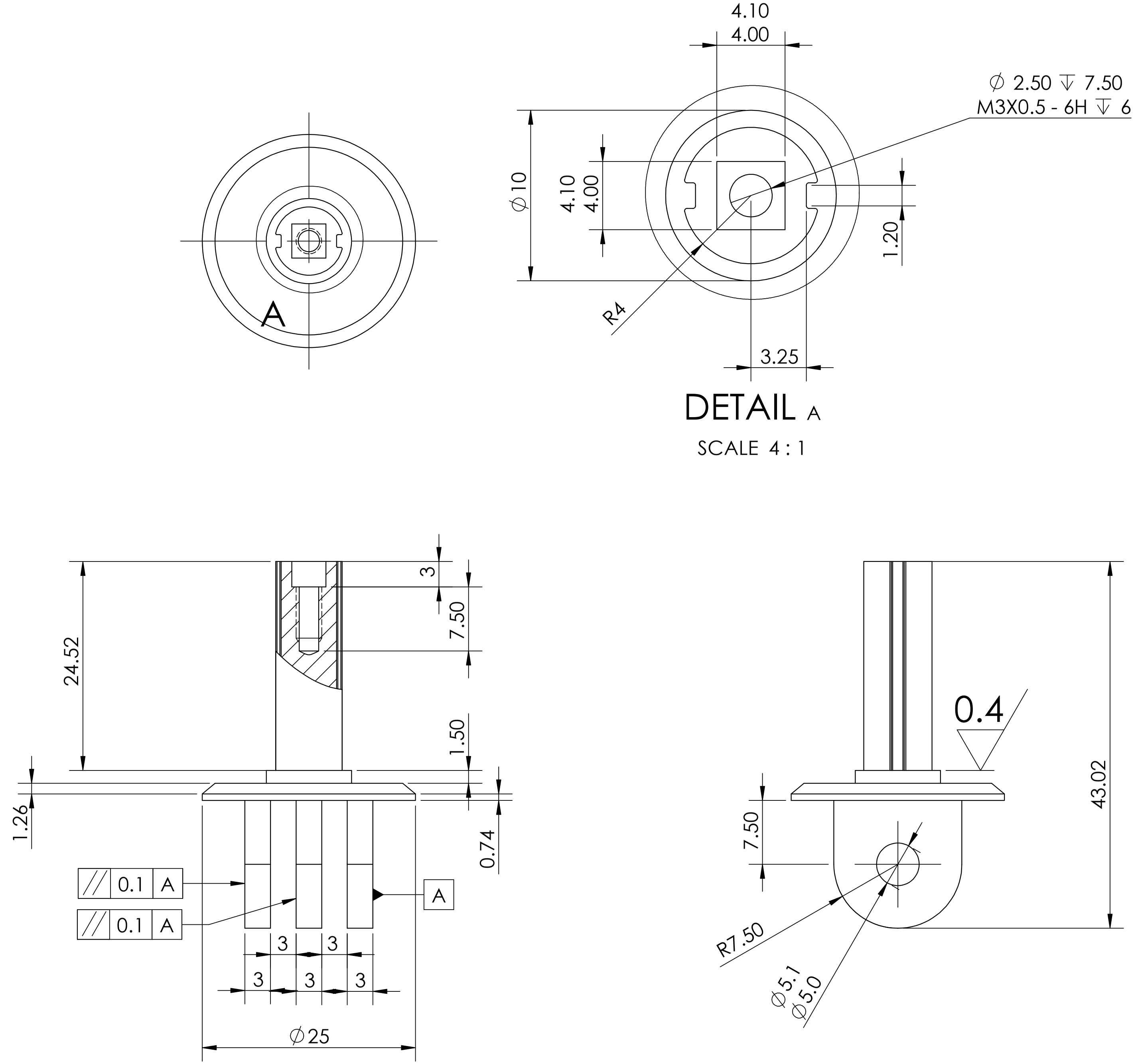

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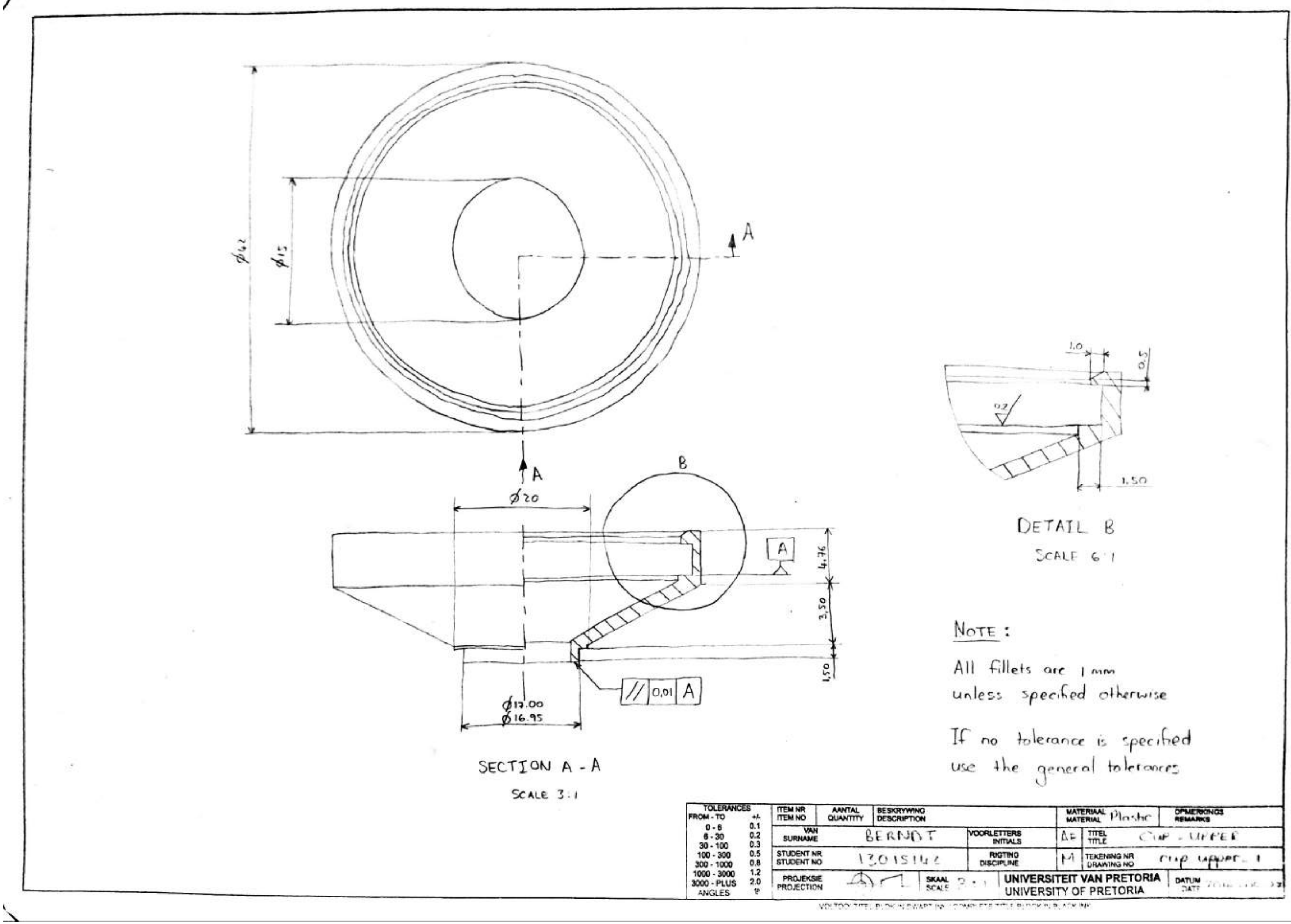

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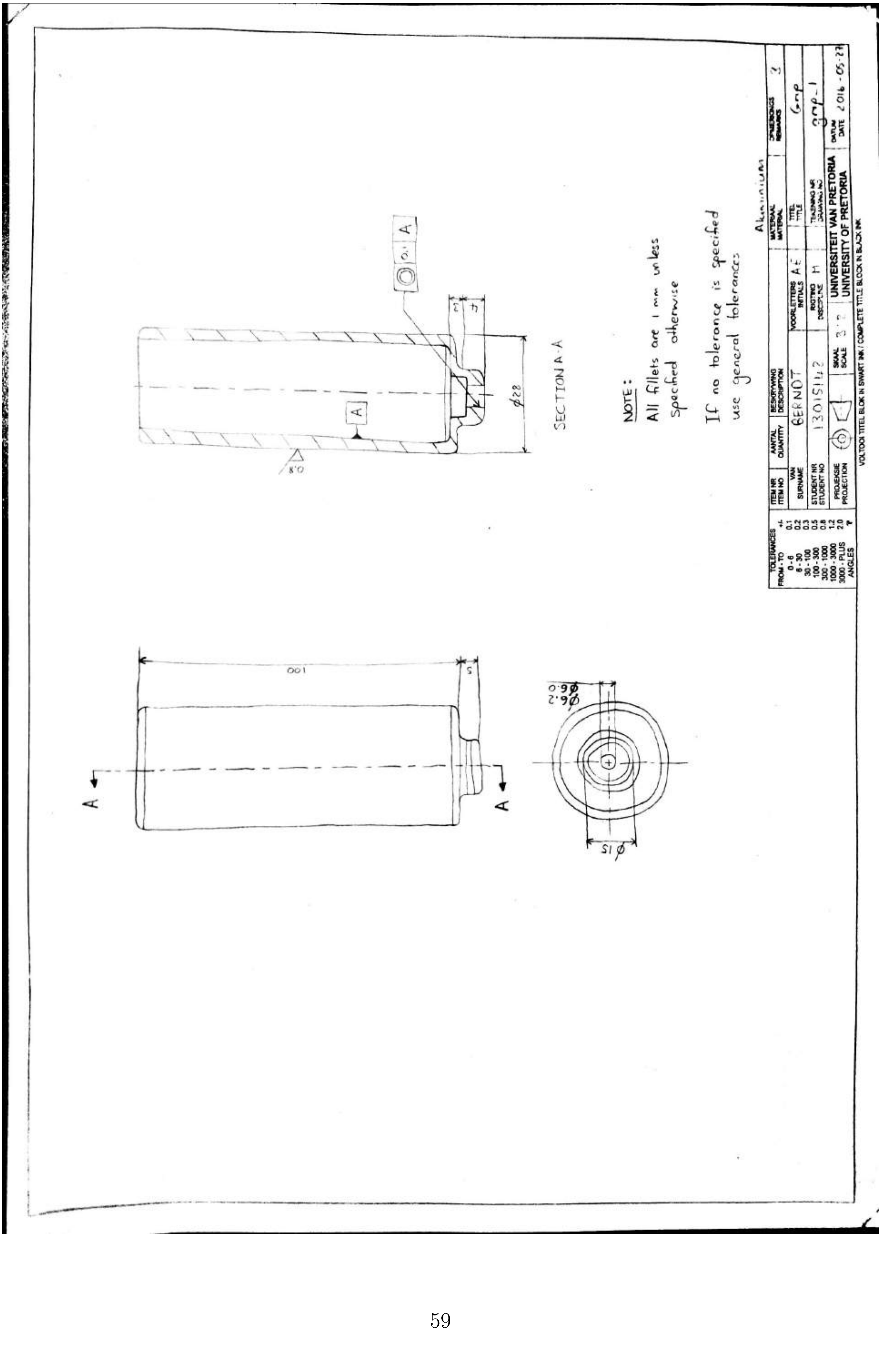
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Chapter 10

Conclusions and Recommendations

10.1 Conclusion

This design project entailed the design of a solution to a user's basic requirements: that a GoPro camera be stabilized for good quality footage during running based activities. This main requirement was dissected using a tried-and-tested design process involving a literature study, functional analysis, conceptual design phase as well as translation of the user's requirements into design specifications.

The literature study revealed some very interesting vibration-damping methods and materials that were eventually implemented in the final design. The functional analysis revealed the core aspects that the design should focus on and the design specifications supplied these core aspects with measurable criteria such that the success of the design could eventually be quantified. The conceptual design phase revealed some very interesting potential solutions and an in-depth selection process was used to select the final concept.

Multiple CAD revisions were made until a final, optimum design was obtained. The detailed engineering design covered the major dynamics response of the final concept. This process included multiple variable optimization (for the flexure design) as well as non-linear dynamics analyses. Detailed analyses were also done on manufacturing procedures, projected costs and the social, environmental, legal, heath and safety related impact the design could have potentially have on society.

In conclusion, the GoPro stabilizer is an elegant piece of hardware that will definitely work if implemented and significantly improve GoPro footage.

10.2 Recommendations

A number of recommendations can be made regarding the design project. They are as follows:

1. **FEA analysis of the flexure:**
   Due to time constraints, an FEA analysis was not conducted on the high frequency absorbing flexure. The flexure was designed using simple beam theory and analytical, first-hand calculations. It is recommended, however, the an FEA analysis be conducted on the Flexure to determine its exact stiffness values. Factors such as notch sensitivity could highly affect the obtained results and should be investigated further.

2. **Bearing viscous damping characterisation:**
   Bearing viscous damping is very difficult to quantify analytically and it is usually easier just to implement a simple test to measure this parameter. However, due to a lack of funding, the actual bearing viscous damping could not be determined and an engineering approximation was made. It is recommended that the non-linear relationship between viscous damping torque and rotational velocity and position be determined for a more accurate design. Further details are mentioned in Appendix A.2.

3. **Non-linear Analysis of Pendulum Effect:**
   The pendulum is a highly non-linear dynamics problem. This design incorporated an approximate solution by assuming small angles when determining the low frequency reaction to an input disturbance. It is recommended that this non-linear analysis be used for the low-frequency reaction instead. A portion of this procedure has already been done and can be found in Appendix A.3.

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