Master of Science

Part B: Construction & Design

Project: Wire Gear for Small Rotary Actuator

Contracting Authority: Kongsberg Space Systems

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i. Abstract

This Part B of this master thesis focuses on the translation of requirements to design - prototype. Features that are included in Part B are, idea generation, concepts, concept validation, concept ranking, design and verification/validation of the design/prototype. The main goal for this Part B is to achieve a good solution for the SRA and a stable prototype.

The result of Part B is a well working prototype, based on the input from Part A. Overall, all major requirements related to function is established and the prototype behaves like expected. This means that the test phase can start, and Part C can be created.

ii. Contents

iii. Figure List

iv. Table list

Abbreviations

AIT	-	Assembly, Integration and Test
APM	-	Antenna Pointing Mechanisms
CONOPS	-	Concept of Operations
CUS	-	Customer
ECSS	-	European Cooperation for Space Standardization
EREQ	-	Environmental Requirement
ESA	-	European Space Agency
FOS	-	Factor of safety
KDA	-	Kongsberg Defence and Aerospace
KSS	-	Kongsberg Space Systems
LEO	-	Low Earth Orbit
NORF	-	Norsk Fletteri
OP	-	Output pulley
OTSP	-	Off the shelf products
PDR	-	Preliminary Design Review
PWG	-	Pinion Worm Gear
QDA	-	Quick Design Analysis
QDR	-	Quick Design Review
REQ	-	Requirement
SAM	-	Supplier and manufacture
SRA	-	Small Rotary Actuator
SSM	-	Small Satellite Mechanisms
TBC	-	To Be Complied
TBD	-	To Be Determined
TREQ	-	Technical Requirement
VOA	-	Vebjørn Orre Aarud
WG-SRA	-	Wire Gear Small Rotary Actuator

1. Introduction

Part B of this master thesis is about translating the Part A into a physical prototype as seen in Figure 2. Consider Part A as the input of the Blackbox while Part B is what happening inside the Blackbox. Inside this Blackbox, the following governing process shown in Figure 1 is executed.

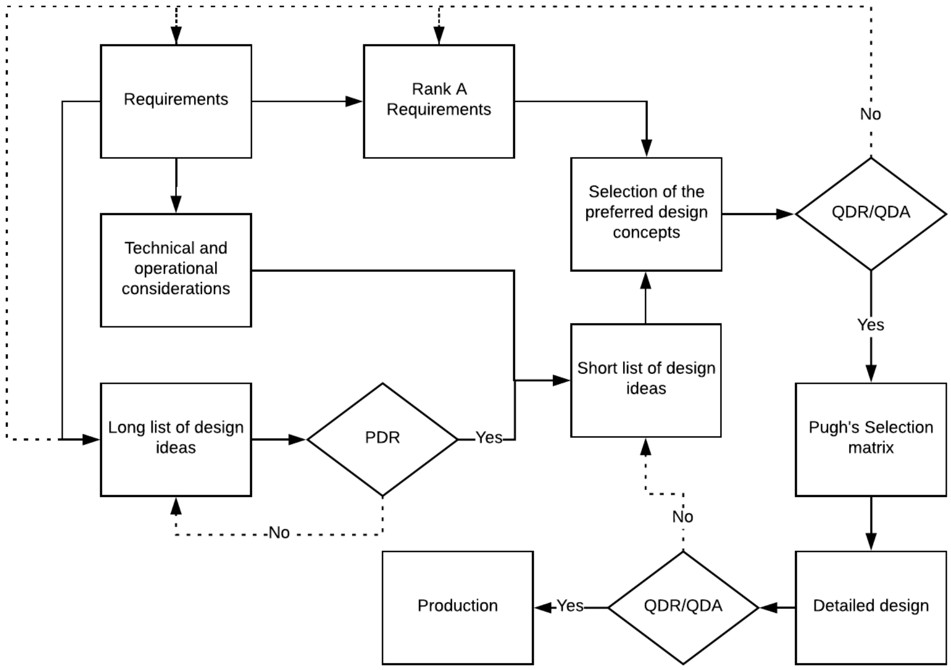

Figure 1: Construction and Design process

This process is first applied to selecting the SRA, with focus on wire layout. The bearing layout will follow the governing principal of SSM's solution discussed in selection 3.5. The second use of this process is applied to selecting the correct wire for the SRA.

Note that the SRA must be designed to drive suggested wires. The suggested wire from KSS is a circular cord with approx. diameter of 1.0 mm and a shoelay design with approx. thickness of 2.0 mm.

The final boxes, production, QDA/QDR and detailed design involve several important subjects. The WG-SRA is being detailed designed with respect to motor design and gear ratio, torque, control system, wire layout , wire pulleys, material selection and preload with needed calculations. And then checked with technical budget to validate our prediction during the last 3 stages. This stage finishes with genuine production drawings and 3D design in SolidWorks, and the product is sent for production when accepted.

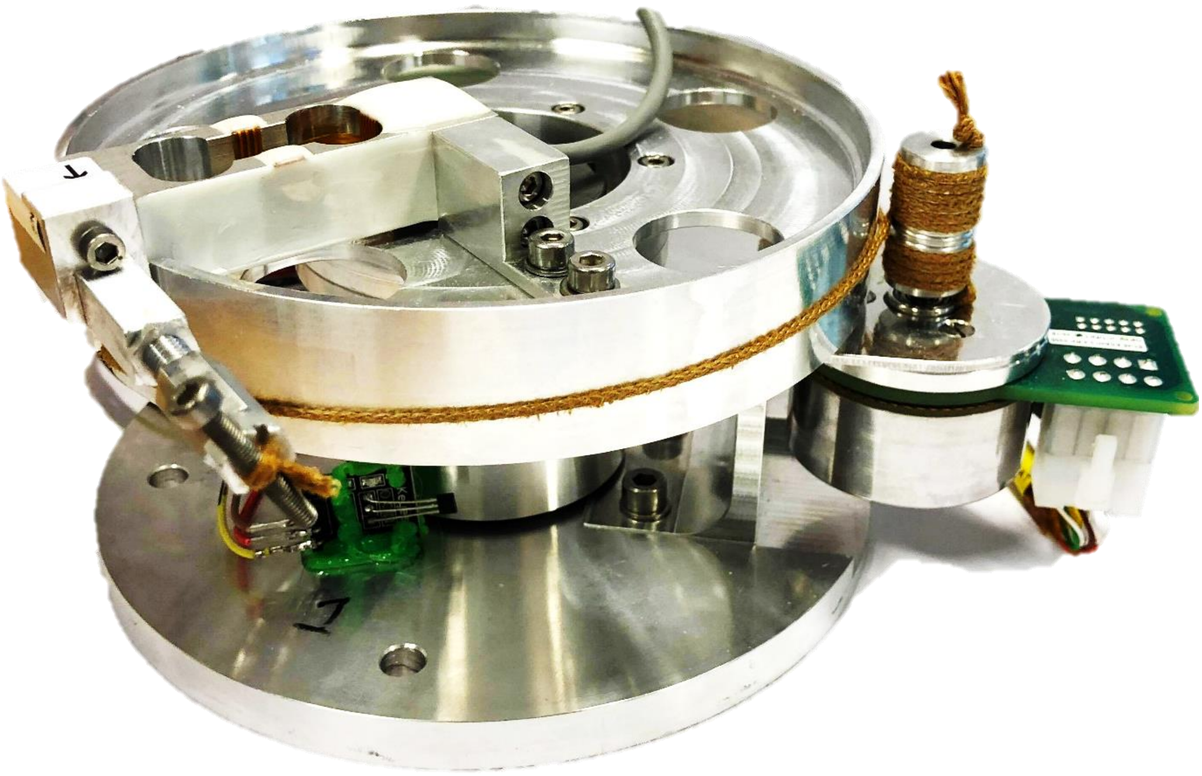

Figure 2: Resulting Prototype.

2. SRA Concept Study

This selection will systematically by following the process illustrated in selection 1, evolve from idea to selected concept.

2.1. Ideation

Based on appendix 8.1 the following short list of ideas are to be discussed (idea 2, 4, 6, 8, 10 and 12). The main goal for this idea presentation is to describe each idea and link it up to technical and operational concerns.

2.1.1. Idea 2

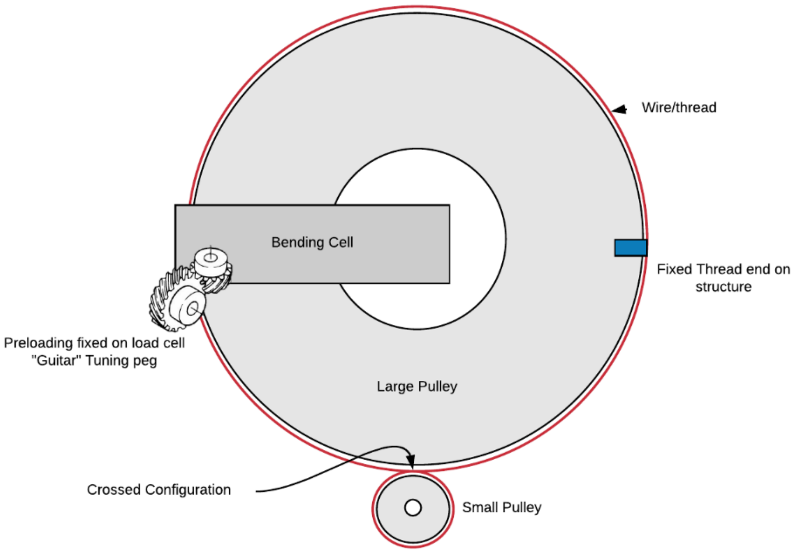

Figure 3: Idea 2.

Idea 2, shown in Figure 3 consist of five parts. A large pulley, a small pulley, a bending cell, guitar tuning peg and the wire. The wire is cut in half, wire A and wire B. Wire A is connected to the structure, at the blue point and follows counter clockwise to the small pulley where it is connected, fixed. Wire B is connected to a guitar tuning peg on the load cell working as the preload mechanism and follows clock wise to the small pully where it is connected, fixed.

This setup provides zero slip and high driveline stiffness. It may be some issues regarding the requirement related to the required angular travel. This because the integration of the guitar tuning peg uses some “space” of the travel lane needed by the wire. All major requirement may be fulfilled by this idea.

Table 1: Idea 2 Advantages and disadvantages.

Advantages	**Disadvantages**
• High adjustability regarding preload force. May obtain very accurate load scenarios • Well known tensioning mechanism. • Infinite travel length of preloading-wire, does not need to be pre-fitted.	• Complex preloading mechanism. • Space consuming – may interferes with other systems. • Low hold torque. • High integration cost.

2.1.2. Idea 4

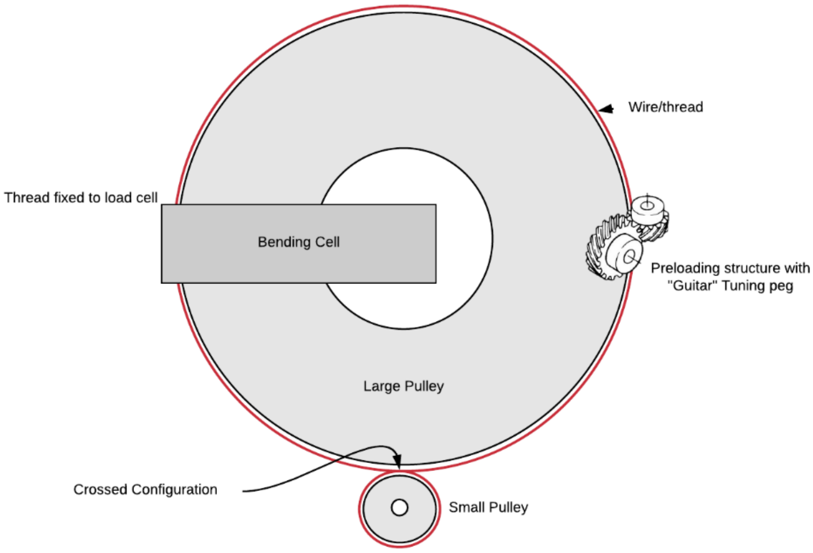

Figure 4: Idea 4.

Idea 4, shown in Figure 4 consist of five parts. A large pulley, a small pulley, a bending cell, guitar tuning peg and the wire. The wire is cut in half, wire A and wire B. Wire A is connected to the load cell follows clockwise to the small pulley where it is connected, fixed. Wire B is connected to a guitar tuning peg on the structure working as the preload mechanism and follows counter clock wise to the small pully where it is connected, fixed.

This setup provides zero slip and high driveline stiffness. The preloading mechanism is placed "inside" the large pully resulting in a space effective solution. All major requirement may be fulfilled by this idea.

Table 2: Idea 4 Advantages and disadvantages.

Advantages	**Disadvantages**
• High adjustability regarding preload force. May obtain very accurate load scenarios • Well known tensioning mechanism. • Infinite travel length of preloading-wire, does not need to be pre-fitted. • Good space management. • Allow full angular travel.	• Complex preloading mechanism. • Low hold torque. • High integration cost. • Hard to adjust if covered to manage UV beams etc. • Complex fabrication

2.1.3. Idea 6

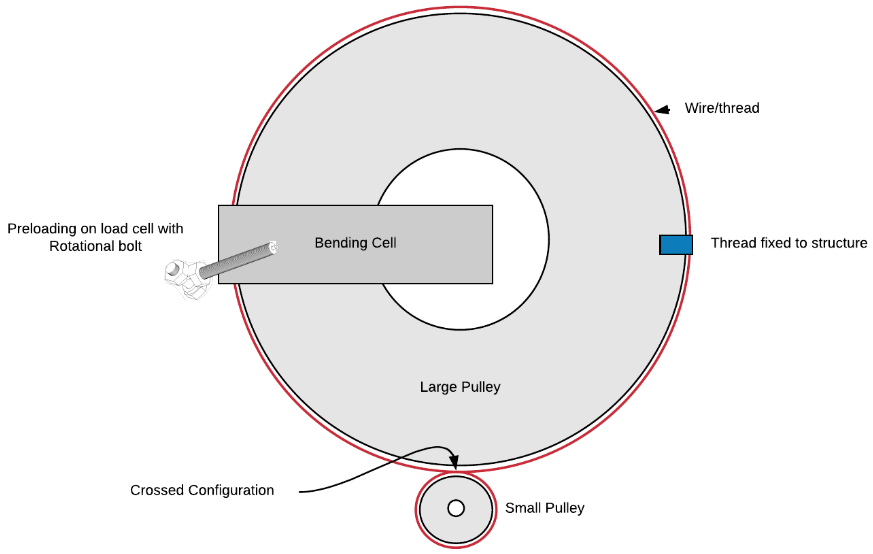

Figure 5: Idea 6.

Idea 6, shown in Figure 5 consist of five parts. A large pulley, a small pulley, a bending cell, rotational bolt mechanism and the wire. The wire is cut in half, wire A and wire B. Wire A is connected to the structure, at the blue box and follows counter clockwise to the small pulley where it is connected, fixed. Wire B is connected to the rotational bolt on the load cell as the preload mechanism and follows clockwise to the small pully where it is connected, fixed.

This setup provides zero slip and high driveline stiffness. The preload mechanism is a simplified version of the guitar tuning peg. This mechanism is slightly better at space usage, however not that accurate. This means that the interference with the pulley is lower than the guitar tuning peg mechanism. All major requirement may be fulfilled by this idea.

Table 3: Idea 6 Advantages and disadvantages.

Advantages	**Disadvantages**
• Good adjustability regarding preload force. May obtain accurate load scenarios • Well known tensioning mechanism. • Infinite travel length of preloading-wire, does not need to be pre-fitted. • Non-complex preloading mechanism • Low integration cost.	• May interfere with the large pully • Medium hold torque • Complex fabrication

2.1.4. Idea 8

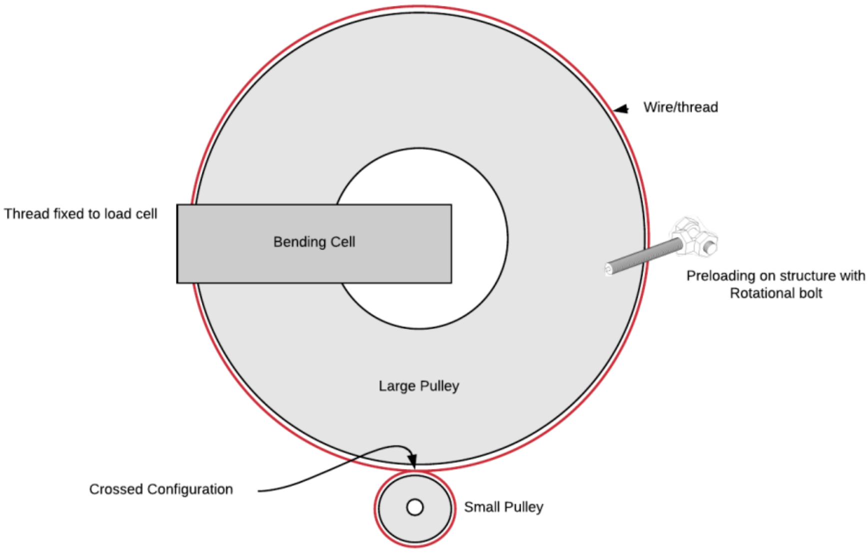

Figure 6: Idea 8.

Idea 8, shown in Figure 6 consist of five parts. A large pulley, a small pulley, a bending cell, rotational bolt mechanism and the wire. The wire is cut in half, wire A and wire B. Wire A is connected to the load cell, fixed and follows clockwise to the small pulley where it is connected, fixed. Wire B is connected to the rotational bolt on the structure as the preload mechanism and follows counter clockwise to the small pully where it is connected, fixed.

This setup provides zero slip and high driveline stiffness. The preload mechanism is a simplified version of the guitar tuning peg. This mechanism is very good at space usage for the same reasons and the guitar tuning peg fixed at the structure. However not that accurate as the guitar tuning peg. All major requirement may be fulfilled by this idea.

Table 4: Idea 8 Advantages and disadvantages.

Advantages	Disadvantages
• Good adjustability regarding preload force. May obtain accurate load scenarios • Well known tensioning mechanism. • Infinite travel length of preloading-wire, does not need to be pre-fitted. • Non-complex preloading mechanism. • Low integration cost.	• Medium hold torque. • Hard to adjust if covered to manage UV beams etc. • Complex fabrication

2.1.5. Idea 10

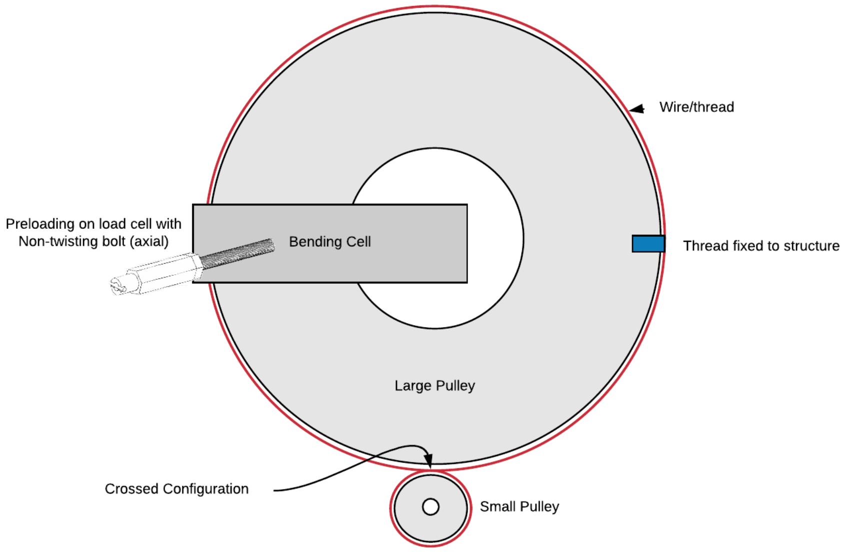

Figure 7: Idea 10.

Idea 10, shown in Figure 7 consist of five parts. A large pulley, a small pulley, a bending cell, non-twisting bolt mechanism and the wire. The wire is cut in half, wire A and wire B. Wire A is connected to the non-twisting bolt mechanism fixed to the load cell. Follows clockwise to the small pulley where it is connected, fixed. Wire B is connected to the structure and follows counter clockwise to the small pully where it is connected, fixed.

This setup provides zero slip and high driveline stiffness. The preload mechanism is very simple, working as a normal bolt. This mechanism is very good at space usage, since it is just a bolt, no extra features. The preloading mechanism may be very accurate depending on the pitch on the bolt. All major requirement may be fulfilled by this idea.

Table 5: Idea 10 Advantages and disadvantages.

Advantages	**Disadvantages**
• Very adjustability regarding preload force. • Well known tensioning mechanism. • Very good space usage. • Very simple. • Almost of the shelf – part. • Excellent holding torque. • Low integration cost.	• Fixed travel length of preloading- wire may be pre-fitted.

2.1.6. Idea 12

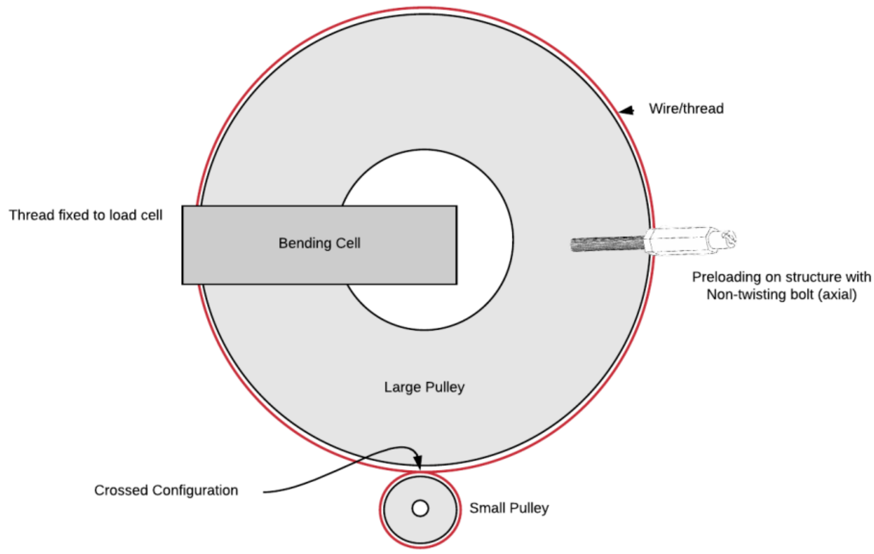

Figure 8: Idea 12.

Idea 12, shown in Figure 8 consist of five parts. A large pulley, a small pulley, a bending cell, non-twisting bolt mechanism and the wire. The wire is cut in half, wire A and wire B. Wire A is connected to the load cell, fixed and follows clockwise to the small pulley where it is connected, fixed. Wire B is connected to the preloading mechanism at the structure and follows counter clockwise to the small pully where it is connected, fixed.

This setup provides zero slip and high driveline stiffness. The preload mechanism is very simple, working as a normal bolt. This mechanism is very good at space usage, since it is just a bolt, no extra features. The preloading mechanism may be very accurate depending on the pitch on the bolt. If the bolt gets too "long" it may interfere with other systems. All major requirement may be fulfilled by this idea.

Table 6: Idea 12 Advantages and disadvantages.

Advantages	Disadvantages
• Very adjustability regarding preload force. • Well known tensioning mechanism. • Very good space usage. • Very simple. • Almost of the shelf – part. • Excellent holding torque. • Low integration cost.	• Fixed travel length of preloading- wire may be pre-fitted. • Hard to adjust if covered to manage UV beams etc. • Not easy to practical adjust. • May interfere with other system if demanded size is "too" big.

2.1.7. Idea conclusion

All these ideas may fulfill the requirement list. Idea 4, 8 and 12 have some issues if a casing is built around the structure to handle the requirement regarding UV beams. Idea 2,4,6,8 have some issues regarding hold torque, especially idea 2 and 4. While idea 10 demand all through holes in the load cell. In collaboration and discussion with KSS Idea 6,8 and 10 seem most feasible and are now considered concepts for this thesis.

2.2. Concepts

In the previous section idea 6,8 and 10 where selected as concept. Idea 6 is now renamed concept Jupiter, idea 8 to concept Mars and idea 10 to concept Pluto. This chapter will in more detail rate and investigate each concept and ending up with a selection matrix to select the best concept. The sizing in the pictures are 1:1, note that the pulleys diameter and height of the assembly may change. The design is kept fixed and the free variable is wire setup. This to choose concept based on wire setup and not "geometry".

The following criteria's that is investigated for each concept are shown in Table 7. Each concept will obtain a score of 1-5 where 5 is the best performance/solution and 1 very weak performance/solution. The score of each criterion is evaluated against to each concept. This score will be used in the Pugh's selection matrix in Table 12.

Table 7: Criteria's.

Criteria:	**Unit:**	**Explanations:**
Preload hold torque:	Nm	How good is the hold torque of the preloading mechanism? How good is the hold torque over time?
Preload adjustability:	mm	How adjustable is the preload mechanism in relation to hold torque over time?
Integration complexity:	-	How complex is it to implement the preload mechanism?
Maintenance:	-	How easy is the preload, wire and system to maintain? Can the wire easily be changed? Can the load cell easily be changed?
System complexity:	-	How complex is the final system, regarding failure mechanisms?
Wire fitting:	-	Is the wire easy to fit and adjust before preloading?
Supplier cost:	€	Cost of purchased and manufactured parts from external suppliers?
AIT cost:	€	In relation to system complexity, is the system easy to produce and assemble?

2.2.1. Concept Jupiter (idea 6)

In Figure 9 we see the evolution of idea 6 to concept. Table 8 shows feature discussion, rating and means related to this concept.

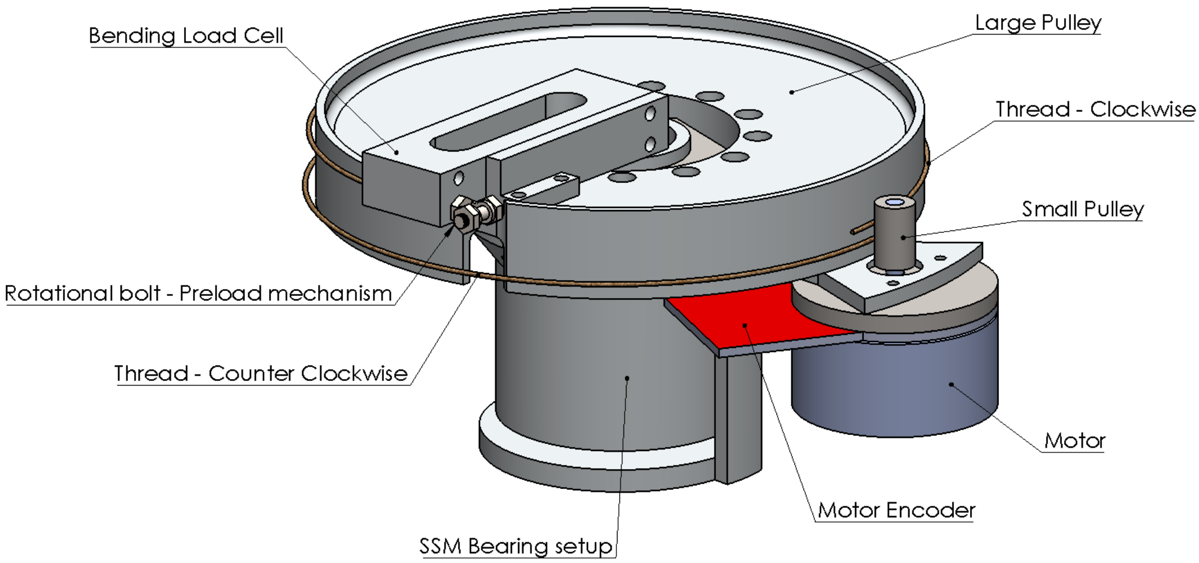

Figure 9: Concept Jupiter Overview.

Table 8: Concept Jupiter Criteria investigation.

Criteria:	**Score:**	**Reasons:**
Preload hold torque:	3/5	This preload mechanism uses friction and clamping force to hold the tension from the wire. For future iterations regarding vibration analysis, thermal analysis and more this solution may not compile to hold the hold torque over time. Because of this the hold torque gets an evaluation score of 3, meaning that it is above average and seems feasible.
Preload adjustability:	5/5	The preload mechanism work as a drum with a wire, giving it very good possibilities to hit detailed load scenarios/spectra's. Over time the preload can easily be adjusted by turning the bolt. Because of this the adjustability of the preload mechanism gets an evaluation of 5, meaning that it is very good.
Integration complexity:	3/5	The bolt itself is a complex part, assumed to be machined and not welded by parts. Welding is not normally accepted in space solutions. This is therefore a custom part and not an of the shelf solution. However, to integrate this solution to the system is very easy. Because of this the integration complexity gets an evaluation score of 3, meaning that it is above average and seems feasible.
Maintenance:	4/5	The mounting place of the preload mechanism provides good accessibility, even when shielded for UV beams. The wire configuration is also easy accessible when shielded. Everything may be accessed by removing one "shield" and not the hole shield structure. The only drawback is limited space for tools regarding adjusting the wire tensioning without moving too much shielding parts. Because of this maintenance gets an evaluation of 4, meaning that it is good.

System complexity:	4/5	The system over all is simple, the only drawback is the small issues regarding the preloading hold torque and integration complexity. The systems seem stable and feasible regarding failure mechanism. Because of this system complexity obtain an evaluation of 4 meaning that it is good.
Wire fitting:	5/5	The solution provides excellent wire fitting, no pre-cutting or pre-fitting is needed. Just fix the wire in one end and pull it around the bolt and start turning. Because of this wire fitting gets and evaluation of 5, meaning that it is very good.
Supplier cost:	2/5	Since the preloading mechanism is considered a custom part in addition to the other subsystems the cost is likely to increase. Because of this supplier cost gets an evaluation of 2, meaning that is below average.
AIT cost:	2/5	Because of the drawback regarding the custom preload mechanism the AIT cost is evaluated to 2, meaning that it is below average.

2.2.2. Concept Mars (idea 8)

In Figure 10 we see the evolution of idea 6 to concept. Table 9 shows feature discussion, rating and means related to this concept.

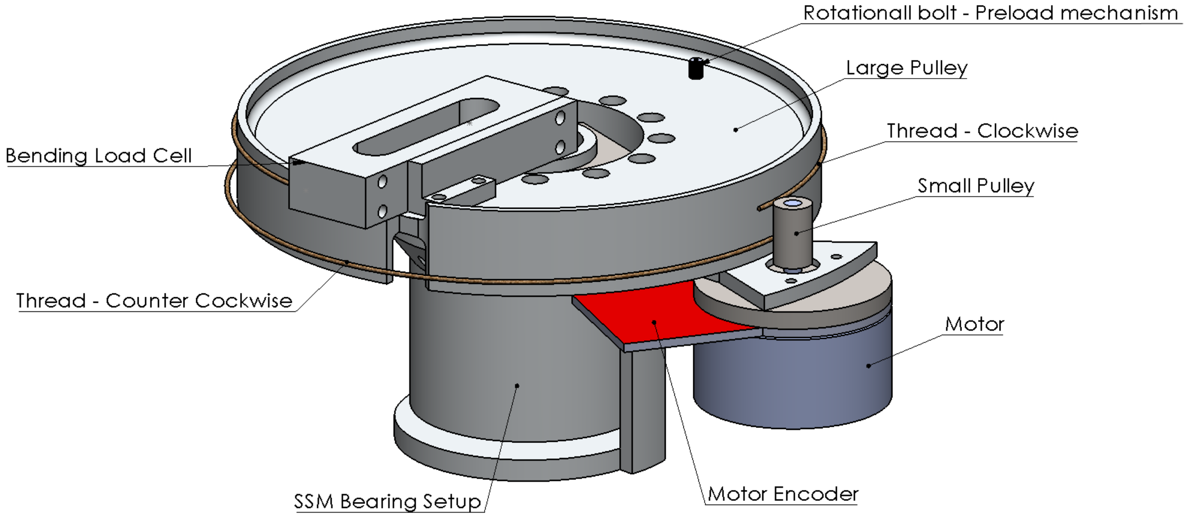

Figure 10: Concept Mars Overview.

Table 9: Concept Mars Criteria investigation

Criteria:	**Score:**	**Reasons:**
Preload hold torque:	3/5	Same as in Concept Jupiter.
Preload adjustability:	5/5	Same as in Concept Jupiter.
Integration complexity:	2/5	The bolt itself is a complex part, assumed to be machined and not welded by parts. Welding is not normally accepted in space solutions. This is therefore a custom part and not an of the shelf solution. In addition to integrate this solution to the system is more complex than Concept Jupiter because it interferes with other subsystems, the disk. Because of this the integration complexity gets an evaluation score of 2, meaning that it is below average.
Maintenance:	1/5	The mounting place of the preload mechanism poor accessibility, when shielded for UV beams. Because of this maintenance gets an evaluation of 1, meaning that it is very bad.
System complexity:	3/5	The system over all is simple, the only drawback is the issues regarding the preloading hold torque, integration complexity and maintenance. The systems seem stable and feasible regarding failure mechanism. Because of this system complexity obtain an evaluation of 3 meaning that it is above average.
Wire fitting:	5/5	Same as in Concept Jupiter.
Supplier cost:	2/5	Same as in Concept Jupiter.
AIT cost:	2/5	Same as in Concept Jupiter.

2.2.3. Concept Pluto (idea 10)

In Figure 11 we see the evolution of idea 6 to concept. Table 10 shows feature discussion, rating and means related to this concept.

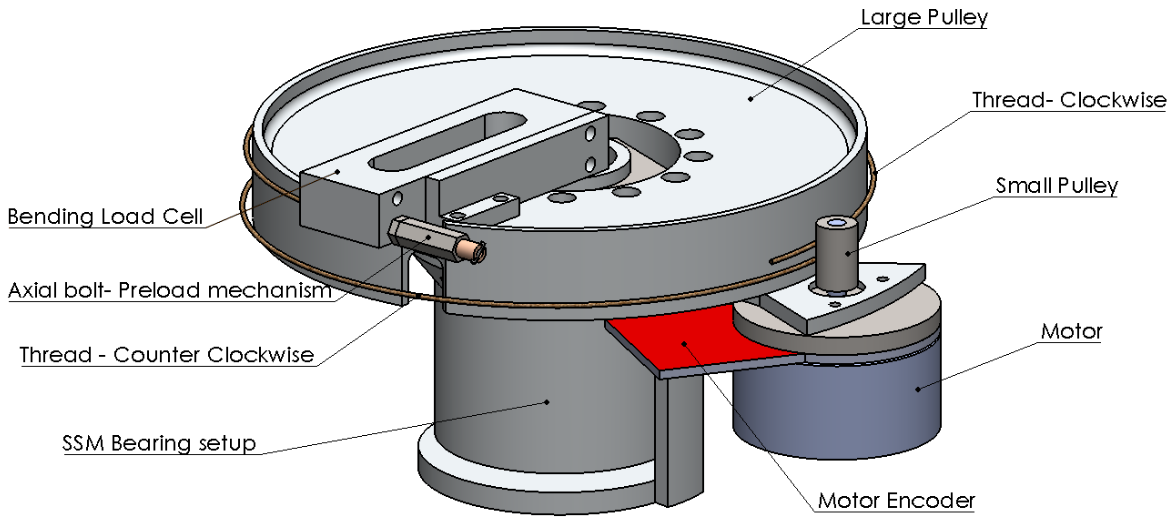

Figure 11: Concept Pluto Overview.

Table 10: Concept Pluto Criteria investigation

Criteria:	**Score:**	**Reasons:**
Preload hold torque:	5/5	This preload mechanism uses friction and clamping force to hold the tension from the wire in an axial direction. The friction is between the wires in the bolt and additional clamping force from a nut is applied to hold the position. Because of this the hold torque gets an evaluation score of 5, meaning that it is very good.
Preload adjustability:	5/5	The preload mechanism work as a simple bolt with a gliding pin. A bolt can be obtained with different pitch giving it very good possibilities to hit detailed load scenarios/spectra's. Over time the preload can easily be adjusted by turning the bolt. Because of this the adjustability of the preload mechanism gets an evaluation of 5, meaning that it is very good.
Integration complexity:	4/5	The bolt itself is a of the shelf part, assumed to be light machined. This is therefore a of the shelf part. This mechanism is easy to integrate. Because of this the integration complexity gets an evaluation score of 4, meaning that it is good.
Maintenance:	4/5	The mounting place of the preload mechanism provides good accessibility, even when shielded for UV beams. The wire configuration is also easy accessible when shielded. Everything may be accessed by removing one "shield" and not the hole shield structure. The only drawback is limited space for tools regarding adjusting the wire tensioning without moving too much shielding parts. Because of this maintenance gets an evaluation of 4, meaning that it is good.
System complexity:	5/5	The system over all is simple. The systems seem stable and feasible regarding failure mechanism. Because of this system complexity obtain an evaluation of 5 meaning that it is very good.
Wire fitting:	3/5	The solution provides average wire fitting, pre-cutting or pre-fitting may be needed. Just fix the wire in one end and pull it around the bolt and cut

		to estimated length. start turning. Because of this wire fitting gets and evaluation of 3, meaning that it is above average.
Supplier cost:	4/5	Since the preloading mechanism is considered a of the shelf part. It's cheaper to integrate. No additional cost. Because of this supplier cost gets an evaluation of 4, meaning that is good.
AIT cost:	4/5	Because of the advantage of the shelf parts regarding the preload mechanism the AIT cost is evaluated to 4, meaning that it is good.

2.2.4. Concept Selection

In this selection, the investigation will be displayed against each other in relation to the most important criteria. The weighting of the criteria's is described in the table below. The table is verified with KSS.

Note that criteria's such as mass, pointing error, life time and other important features is excluded. This because the concept is to be selected around the driveline. Pointing error and lifetime will be investigated on the winning concept and can be found in their respectively budget in selection 5. This is an iterative process.

Table 11: Weighting of criteria's.

Criteria:	**Unit:**	**Weight:**	**Explanations:**
Preload hold torque:	Nm	20%	Highly weighted. Hold torque is very important to keep the required tension in the wire. This affect the system stability. Slack wire equals increased pointing error and directly violates several requirements.
Preload adjustability:	mm	7.5%	Medium to low weighted. Preload adjustability is important to the system but does not interfere with system stability.
Integration complexity:	-	10%	Medium weighted. Integration complexity is important to the system but does not interfere with system stability.
Maintenance:	-	22.5%	Highly weighted. Maintenance is very important to keep the required tension in the wire. This affect the system stability. Slack wire equals increased pointing error and directly violates several requirements. It's feasible to believe that the wire must be adjusted several times during lift time testing.
System complexity:	-	10%	Medium weighted. System complexity is important to the system but does not interfere with system stability.
Wire fitting:	-	10%	Medium weighted. Wire fitting is important to the system, and it can simplify the preloading process. However, this does not affect the system stability.
Supplier cost:	€	10%	Medium to high weighted. Supplier cost does not affect the stability of the system but is a large requirement regarding total cost.
AIT cost:	€	10%	Medium to high weighted. AIT cost does not affect the stability of the system but is a large requirement regarding total cost.

Table 12: Pugh Selection matrix.

Pugh's Concept Selection Matrix

			Concept alternatives		
Criteria:	**Unit:**	**Weight:**	**Concept Jupiter**	**Concept Mars**	**Concept Pluto**
Preload hold torque	Nm	20 %	3	3	5
Preload adjustability	mm	7,5 %	5	5	5
Integration complexity	-	10 %	3	2	4
Maintenance	-	22,5 %	4	1	4
System complexity	-	10 %	4	3	5
Wire fitting	-	10 %	5	5	3
Supplier cost	€	10,0 %	2	2	4
AIT	€	10,0 %	2	2	4
Sum		**100 %**	**3,475**	**2,6**	**4,275**

2.2.5. Concept conclusion

The preload hold torque and maintenance of the wire driveline is very important, both for endurance under lifetime test and system stability. Because of this the concepts are fairly depended on this criteria's.

Concept Jupiter obtain a total concept score of 3.475. It loses score due to the preload mechanism and cost criteria's. Concept Mars obtained a concept score of 2.6 and is the "worst" concept. It loses the same scores in the same criteria's as concept Jupiter but in addition have poor maintaining abilities. Concept Pluto is overall better than the other concepts with only a small drawback regarding wire fitting.

Concept Pluto is the winning concept in this Pugh's selection with a score of 4.275. The result was presented to KSS and verified.

3. SRA Detailed Design

In this selection Concept Pluto are to be redesigned. This redesign is needed to implement and verify/validate each sub system. In other words, concept Pluto is to be integrated with the rest of the subsystems. Figure 12 shows how to implement the concept into a detailed prototype design. This phase will end with a QDR or QDA and production drawings. Each segment is to be designed in detail and implemented accordingly to their given requirement.

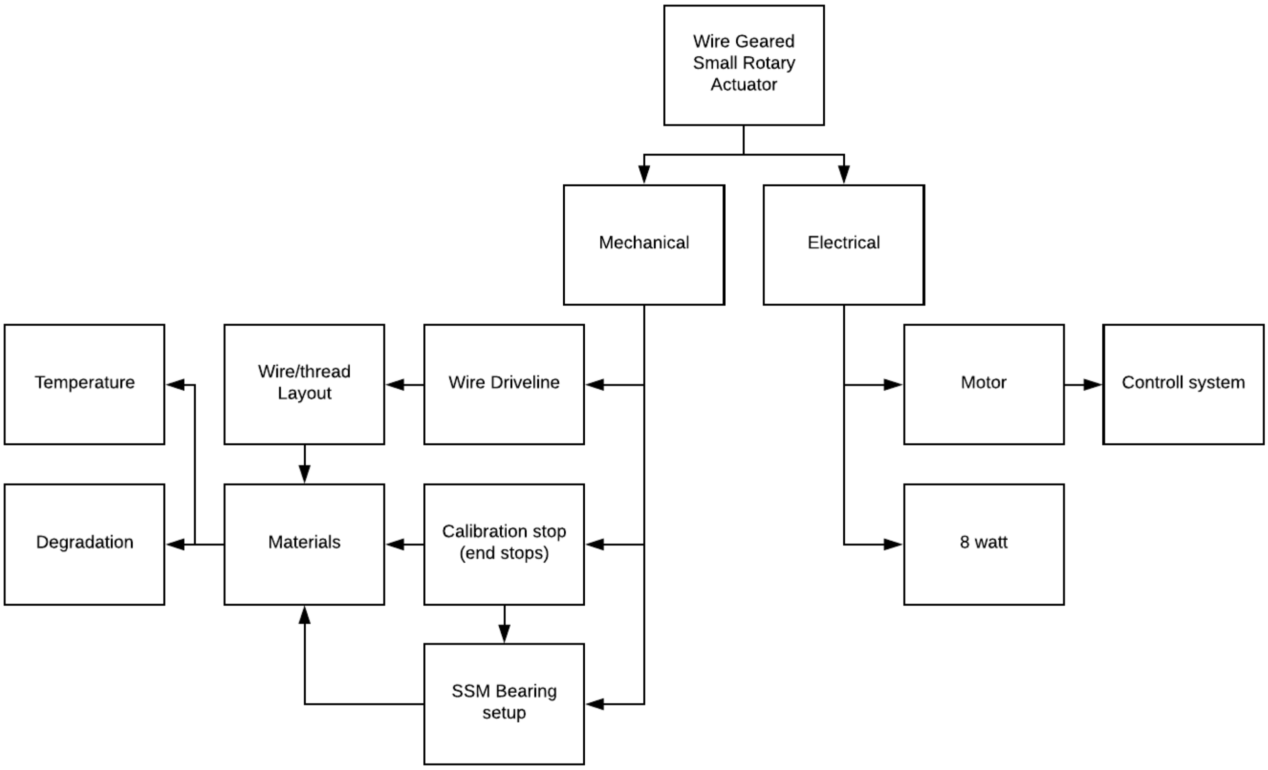

Figure 12: Implementation of concept into prototype.

3.1. Motor design and gear ratio

The pre-selected motor for the SRA is a brushless direct current motor. The preselection process of the motor was done by SSM and verified through KSS in relation to the development of this new WG-SRA. The Motor Maxon EC 45 flat 70W was selected by SSM since smaller motors are less efficient and therefore does not meet performance requirement. This must be investigated also for this thesis, to verify that the motor can deliver the right amount of torque without using too much power.

Figure 13 shows the relation of output torque on the driveshaft on the motor versus consumed power. TREQ-5 assume that the motor shall use less than 8 Watt. By reading the figure, the motor produces approx. 140 mNm at 8 Watt.

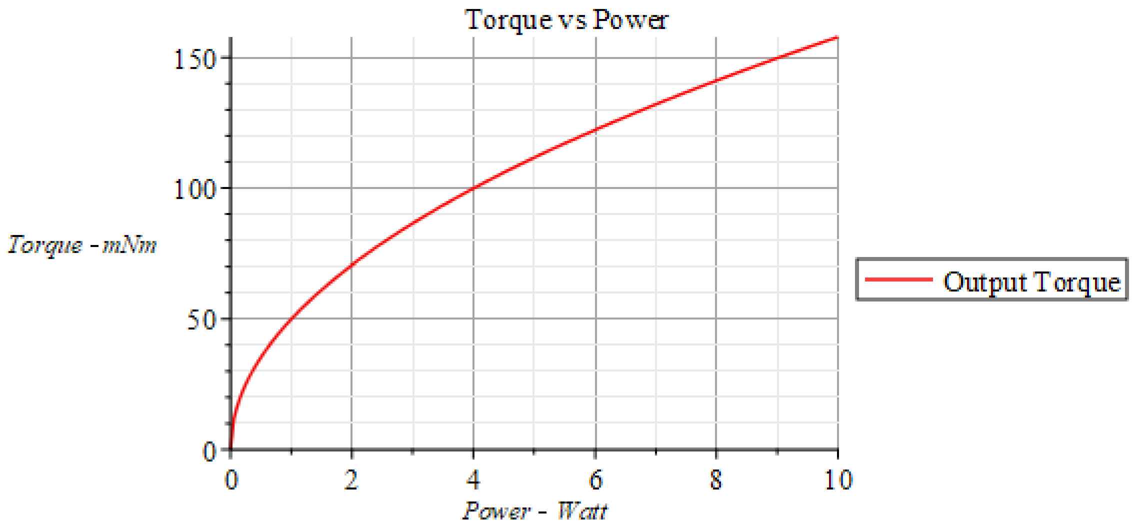

Figure 13: Torque vs Power.

Figure 13 is made by the following formulas:

$$T = Ik_t \tag{1}$$

where T= motor torque (mNm), I= motor current (A) and k_t is motor torque constant (mNm/A).

The current is found by:

$$I = \sqrt{\frac{P}{R}} \tag{2}$$

where P is the power (Watt) and R the winding resistance (Ohm).

The value for k_t is found to 131 mNm/A and the value for R is found to 6.89 Ohm by [1] motor number 402687. The value P is free – but limited to 8 W in this system.

From the bearing calculation in Part A, the friction torque in the bearing was estimated to 0.1728 Nm, approx. 172.8 mNm. This means that just to drive the system a gearing ratio must be establish.

The following factors must be adjusted for when designing the gear ratio:

- The motor produces at max 140 mNm at 8 Watt (TREQ-5)
- The friction momentum in one bearing is estimated to 172.8 mNm (TREQ-12)
- The demanded performance of 90 deg/s top speed (TREQ-8)
- The demanded performance of top speed within 0.5 seconds (TREQ-9)
- ECSS motorization standards (TREQ-16) in relation to output motor torque.

And the following processes are to be executed:

1. Using the concept design to estimate required torque due to inertia.
2. Using Solidworks motion to simulate the case with representative loads.
3. Read out output and adjust accordingly to ECSS.
4. Estimate needed gear ratio.

Figure 14 shows the needed torque to drive the system with only the inertia from the concept drawing and gravity working. Note that there is no external torque applied to the mechanism. The only torque contribution is from the system is caused by its own inertia.

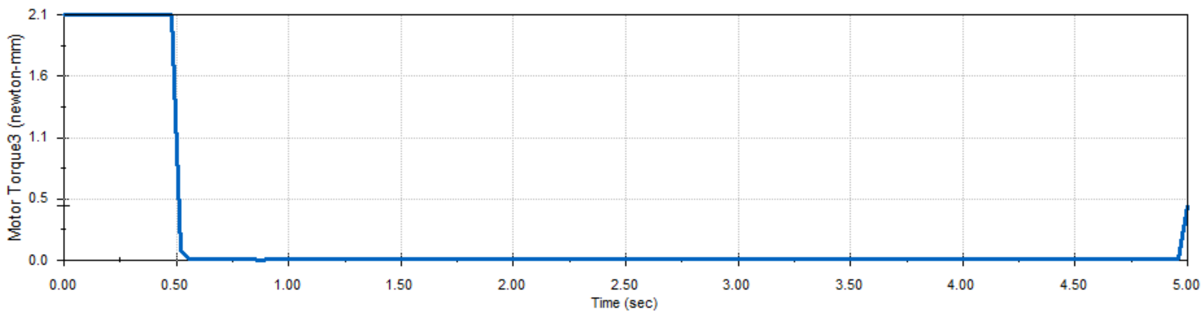

Figure 14: Torque due to gravity and inertia only.

Figure 15 shows the needed torque to drive the system when gravity, inertia and friction force from one bearing is applied.

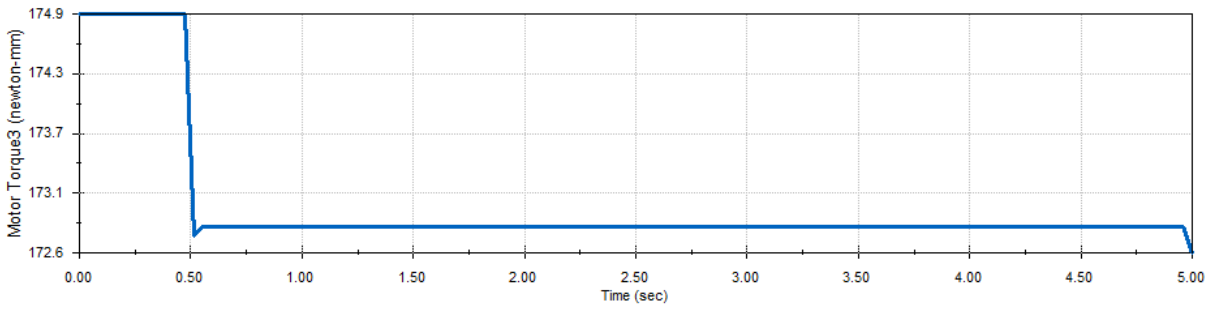

Figure 15: Torque due to gravity, inertia and friction.

Figure 16 shows the resulting performance of the system.

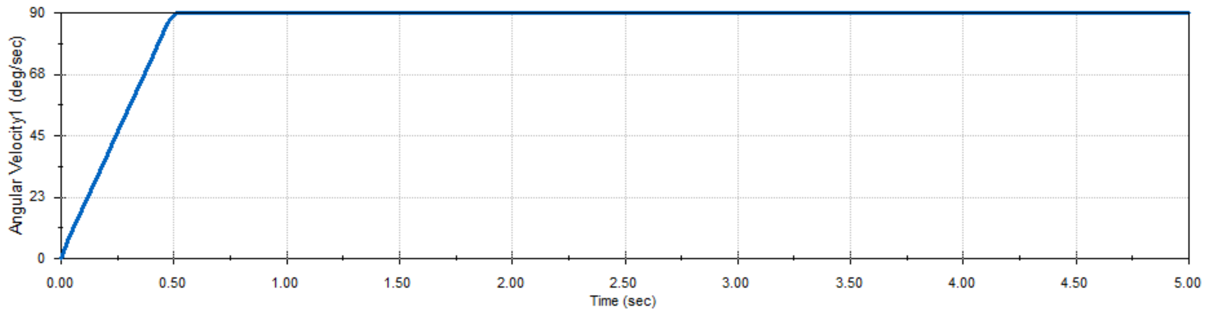

Figure 16: Velocity results with respect to applied torque.

Based on this it requires 2.1 mNm to drive the system without friction and 175 mNm to drive the system with friction from one bearing. Accordingly, to ESA the following torque must be implemented with several uncertainty factors. [2]. Note that some factors are dropped, and the measured factors are used, this is discussed with KSS.

Table 13:ESA uncertainty factors [2].

Component of resistance:	**Symbol:**	**Value:**	**Resource:**	**Theoretical Factor:**	**Measured Factor:**
Inertia	I	2,1 mNm	Solidworks motion study	1,1	1,1
Motor mag. Losses	H	15 mNm	KSS	1,5	1,2
Friction	F	345.6 mNm	Solidworks motion study (both bearings)	3	1,5

This gives the following minimum actuation torque.

$$T_{\mathrm{min}} = 2(1{,}1 \cdot \mathrm{I} + 1{,}5 \cdot \mathrm{H} + 3 \cdot \mathrm{F})$$

$$T_{\mathrm{min}} = 2(1{,}1 \cdot 2{,}1 + 1{,}2 \cdot 15 + 1{,}5 \cdot 345.6) = 1077.42 \text{ mNm} \tag{3}$$

Recall that the motor produce 140 mNm at 8 Watt. The gear ratio can be found by dividing minimum actuation torque by produced torque.

$$\text{Minimum gear ratio} = \frac{T_{\mathrm{min}}}{T_{\mathrm{motor}}} = \frac{1077.42}{140} = 7.695 \tag{4}$$

Thus, the system must have a minimum gear ratio of 1:7.7 to deliver correct performance within the power spectra of the motor.

It is favorable with a small gear ratio due to failure, shorter wire for increased stiffness and better mass consumption. A gear ratio of 1:10 is therefore selected for the WG-SRA based on these calculations and discussions with KSS. With the following spec, the motor must produce 107,4 mNm with this setup, which is soft verified with respect to requirement TREQ-5, 8, 9, 12 and ESA standards TREQ-16.

This segment is to be "hard" verified through a torque and power budget when designed, later in this thesis.

3.2. Control system and end stops

To control the WG-SRA two systems are used. Figure 17 shows the first system. This system uses a of the shelf controller unit from Maxon motor named ESCON 50/5. This is a speed and current controller specially made for the selected motor with a given software. The controller is used in the following way. A deacceleration and acceleration is set to obtain a good sequence. In addition, a limit of speed is set. The deacceleration is set to one million rounds per minute per second and acceleration set to 2.5 rounds per minute per second. The top speed limit is set to 5 rounds per minute. This is not to verify the performance requirement but to ensure a good test sequence.

Two digital inputs are used to send start and stop signals to the controller which is discussed in the code section below.

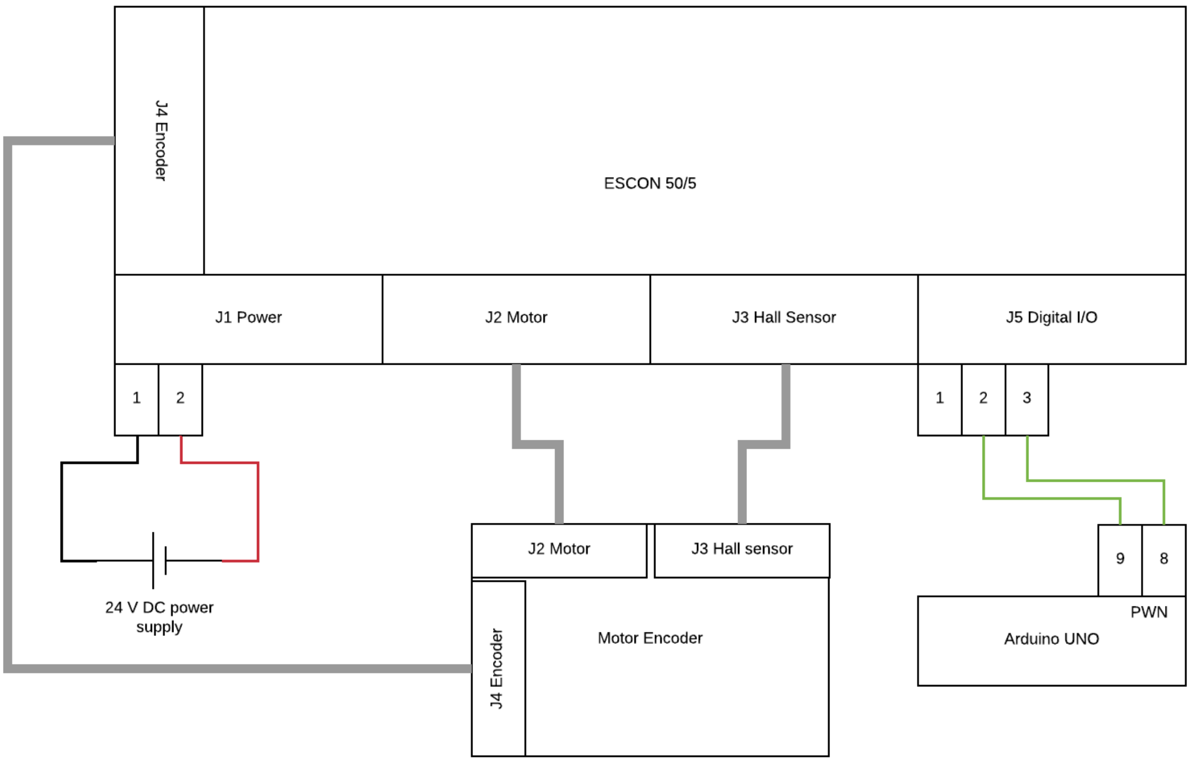

Figure 17: Motor controller.

Figure 18 shows the second system. This system uses two magnetic hall effect sensors to position the system with the following program in addition to an Arduino UNO. At first the system was designed digitally by reading if the sensor is “on” or “off”. However due to noise or unspecified errors, these signals were not stable and the WG-SRA didn’t behave correctly in practice. The second system is designed analog and defines an area for the value pulses such as, its “on” when the reading value is below 50 and “off” when the reading value is above 50. This works as a filter and the WG-SRA works perfectly. In addition, the Arduino needed external power supply to drive the system.

Table 14: Basic outline of code.

Is/If start key is pressed, and hall effect sensor 1 is trigged and hall effect sensor 2 is not trigged, send signal to ESCON controller and move clockwise until hall effect sensor 2 is triggered. Is start key still triggered and is hall effect sensor 1 not trigged? If so send signal to ESCON controller and move counter clockwise until hall effect sensor 1 is triggered. If not do nothing. Repeat.

The complete code can be found in appendix 8.2.

This code provides not steering and does not comply with any requirement but is made so the system obtain a desired sequence for the life time test in vacuum. The hall effect sensor also work as end stops because the code is depended on these inputs. If the inputs are not correct the code does nothing. Example, we may not start the test rig if the hall effect sensor is not triggered in the right way.

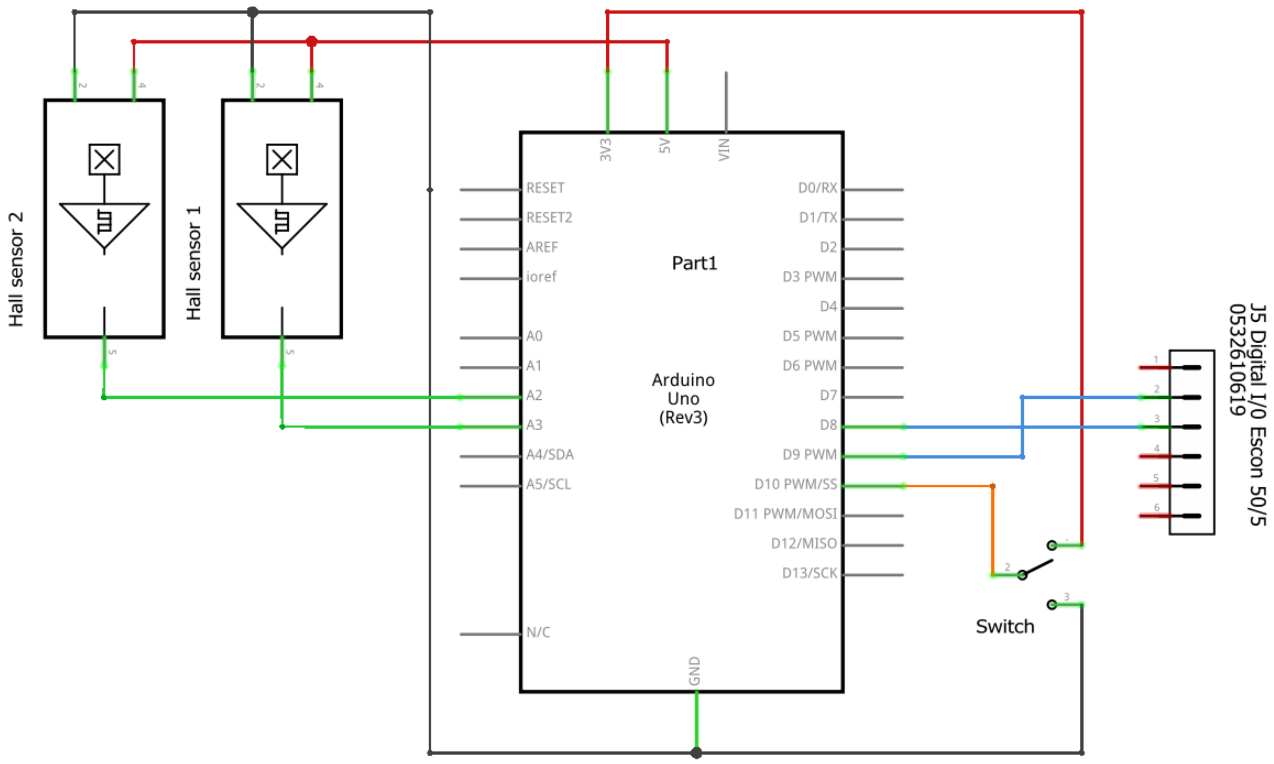

Figure 18: Arduino control system.

3.3. Wire and wire Layout design

The preselected wire to drive the system is made of PBO, also known as Zylon. Zylon is a poly(*p*-phenylene-2,6-benzobisoxazole fiber with excellent thermal properties, low mass per length and high strength [3]. This wire was preselected due to result in previous test at KSS by Kodyna.

Zylon wire is extremely hard to obtain within the size of 0.25-1 mm diameter and the following wire configurations and layout is obtained.

Table 15: Obtained wires.

Wire Id:	**Fiber type:**	**Manufacture /source:**	**Remarks:**
MALO	Zylon PBO High Modulus	Brand: Marlow Ropes Color: Dark Gold Made in England? Sold by Marlow	Diameter: 1.0 mm Tensile Modulus (fiber): TBC Braid: 12 strands, braided – pitch (lay) to work with the yarn twist. “Armorcoated” with Polyurethane
NORF-1	Zylon PBO High Modulus HM 1640 dtex	Brand: Toyobo fibers Color: TBC Made in Norway by Norsk-fletteri Sold by Toyobo	Diameter: 1.0 mm Tensile Modulus (fiber): 1650 cN/dtex Braid: 8 strands, braided – TBC Fiber from Toyobo and braided service from Norsk-fletteri
NORF-2	Zylon PBO High Modulus HM 1640 dtex	Brand: Toyobo fibers Color: TBC Made in Norway by Norsk-fletteri Sold by Toyobo	Size: 2.0 by 0.5 mm Tensile Modulus (fiber): 1650 cN/dtex Braid: TBC strands, hollow double braided (shoelace braid) Fiber from Toyobo and braided service from Norsk-fletteri
NORF-3	Zylon PBO Normal Modulus AS 1670 dtex	Brand: Toyobo fibers Color: TBC Made in Norway by Norsk-fletteri Sold by Toyobo	Diameter: 1.0 mm Tensile Modulus (fiber): 1140 cN/dtex Braid: 8 strands, braided – TBC Fiber from Toyobo and braided service from Norsk-fletteri
NORF-4	Zylon PBO Normal Modulus AS 1670 dtex	Brand: Toyobo fibers Color: TBC Made in Norway by Norsk-fletteri Sold by Toyobo	Size: 2.0 by 0.5 mm Tensile Modulus (fiber): 1140 cN/dtex Braid: TBC strands, hollow double braided (shoelace braid) Fiber from Toyobo and braided service from Norsk-fletteri

Because of limited time, only the circular wires can be tested in this iteration in relation to lifetime. All wires will be investigated in relation to friction and tension.

3.4. Wire lead and pulleys

Based on the Kodyna report, the large pulley should be free, no tracks or grooves to guide the wire. The small pulley should be a helical gear with a profile to fit the wire and to guide the wire.

The most important feature is surface roughness for both pulleys. Roughness is measured by arithmetical mean deviation of the assessed profile. A Rough surface may decrease the lifetime of the wire. ISO standard N5 is the suggested standard for surface finish for the pulleys and approved by KSS.

Figure 19 shows the pinion worm gear (PWG) designed for the WG-SRA. Requirement TREQ-7 states that the output sweep shall be minimum $\pm 190°$ and suggested gear ratio is 1:10. This means that the PWG must turn 10 times to achieve one round on the output pulley.

A QDR and QDA was executed and discussed with KSS to obtain the diameter sizing for the PWG. The outcome of this analysis is that the PWG shall have a diameter of 14mm. By setting this limit the groove can be designed. Thus, groove depth is set to 1,5mm to ensure good connection between the PWG and the 1 mm diameter wire.

Recall that $360°$ angular displacement on the PWG results in $36°$ displacement on the output pulley, the pitch and turn can be designed.

To find maximum angular displacement that satisfy TREQ-7 on the PWG number of turns is calculated. Recall that one turn $360°$ on the PWG equals $36°$ on the output. Thus, following relation can be applied

$$\text{desiered output angular displacement} \\ = \text{ output angular displacement of one turn times "}n\text{" turns}$$

$$\pm 190° = 36° n$$

$$n = \pm \frac{190°}{36°} = \pm 5{,}27 \text{ turns}$$

This implies that the needed turns are $\pm 5{,}27$ ending up with a total of 10,55 turns. Too adjust for connection and compliance a total of 12 turns seems feasible. For the pitch recall that the groove size is 1,5mm and a flange size of 0.25mm seems feasible accordingly to separation the wires.

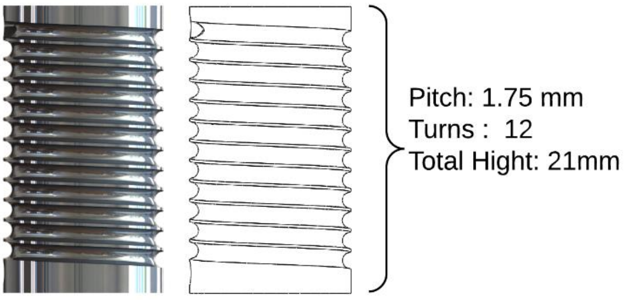

Figure 19: Pinion Worm Gear.

Figure 20 shows the output pully (OP). This pully is highly dependent on the PWG regarding sizing. Because of the diameter size of 14mm in the PWM the OP must have a diameter of 140mm, same height.

Note, no groves just smooth roughness finish as stated above this because, experience indicates that a "free" wire is better for lifetime.

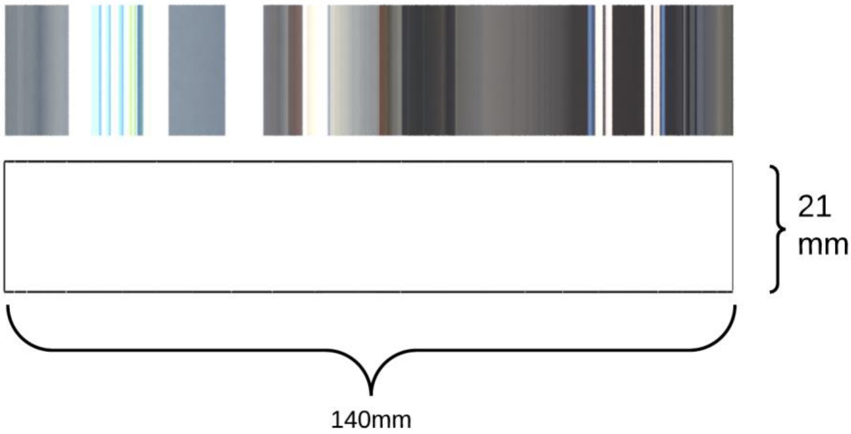

Figure 20: Output pully.

Figure 21 shows the total wire layout combined with the pullies.

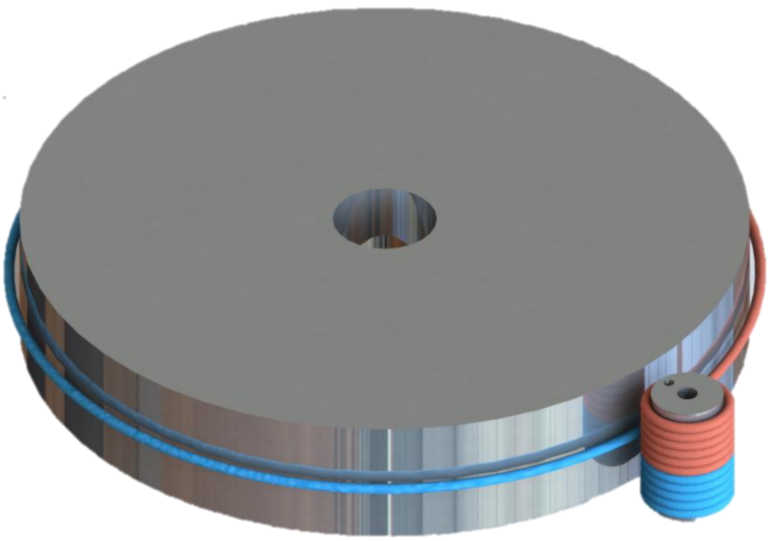

Figure 21: Combined PWG and OP.

3.5. Bearing House implementation

As discussed in ideation and concept selection. SSM bearing setup are to be used. The principle of this setup is that, two deep groove ball bearings are preloaded and fixed in a specific method to obtain an angular contact direction, often seen in angular contact bearings. Figure 22 shows the difference between angular contact bearing and deep groove ball bearing. The deep groove ball bearing has radial contact angles, which means that the bearing may not absorb large amount of axial force – the axial force may dislocate the bearing. However angular contact ball bearings can observe both axial and radial force. This is much more favorable in our case. Angular contact bearings cannot be preloaded as wanted due to the prefixed contact angle and because of this, deep groove ball bearings are used and preloaded to given, vibration standard as discussed in part A and to obtain the right contact angle.

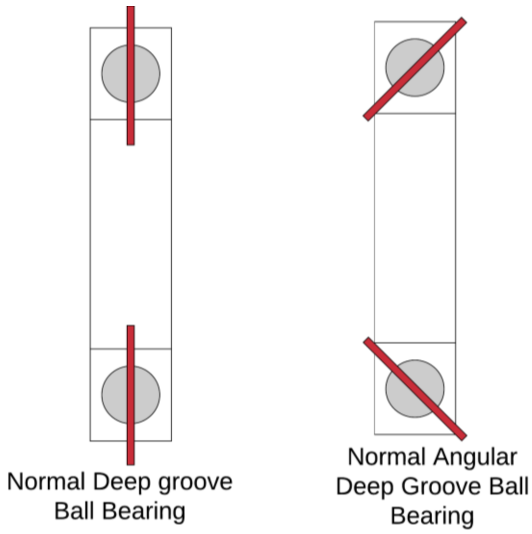

Figure 22: Contact angle bearings.

Figure 23 shows the total setup, where the green lines represent the contact angles – force distribution.

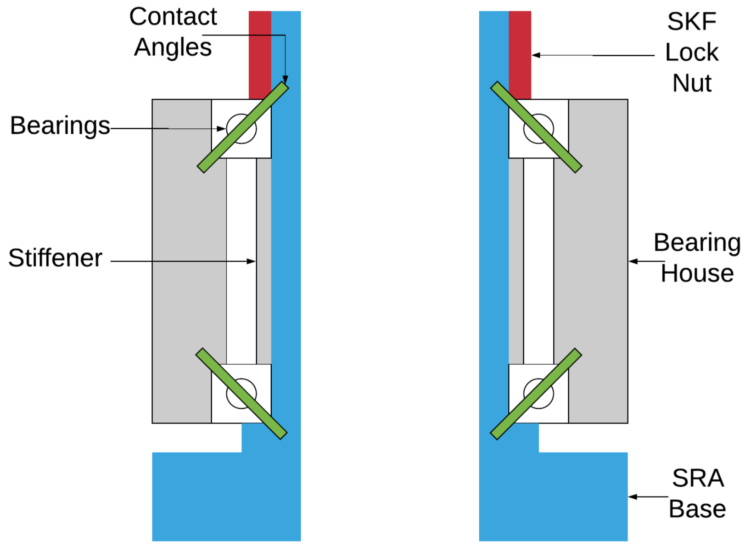

Figure 23: Bearing setup.

3.6. Material selection

Material selected for this design is highly dependent on the technical requirements and ECSS-Q-ST-70 and ECSS-Q-70-71a standards. CES material selection is applied to each case in relation to the technical requirements and validated in the standard.

Global limits for all cases are shown in Table 16.

Table 16: Limits for material selection

Origin:	Limit:
EREQ-1	Thermal Expansion coefficient not larger than 25 µstrain/°C
EREQ-2	Service temperature range of -25°C and +65°C
EREQ-4	Excellent UV radiation properties
EREQ-7	Good fresh water corrosion properties
TREQ-3	Set as axis on table - free
TREQ-6	Set as axis on table - free

Figure 24 shows the general material groups as the output of the applied limits. These groups are composites, metals and alloys. The following materials which seems feasible regarding price and density are wrought magnesium alloys, cast magnesium alloys, age-hardening wrought Al-alloys and cast Al-alloys.

The larges drawback of magnesium is that it does not serve well in non-protected areas and it's not suggested by ECSS-Q-70-71a in exposed environment, however extremely suitable for internal and protective areas. The cheapest and relative light weight suggestion is therefore Al-alloys. In the next compliance against design structure is discussed.

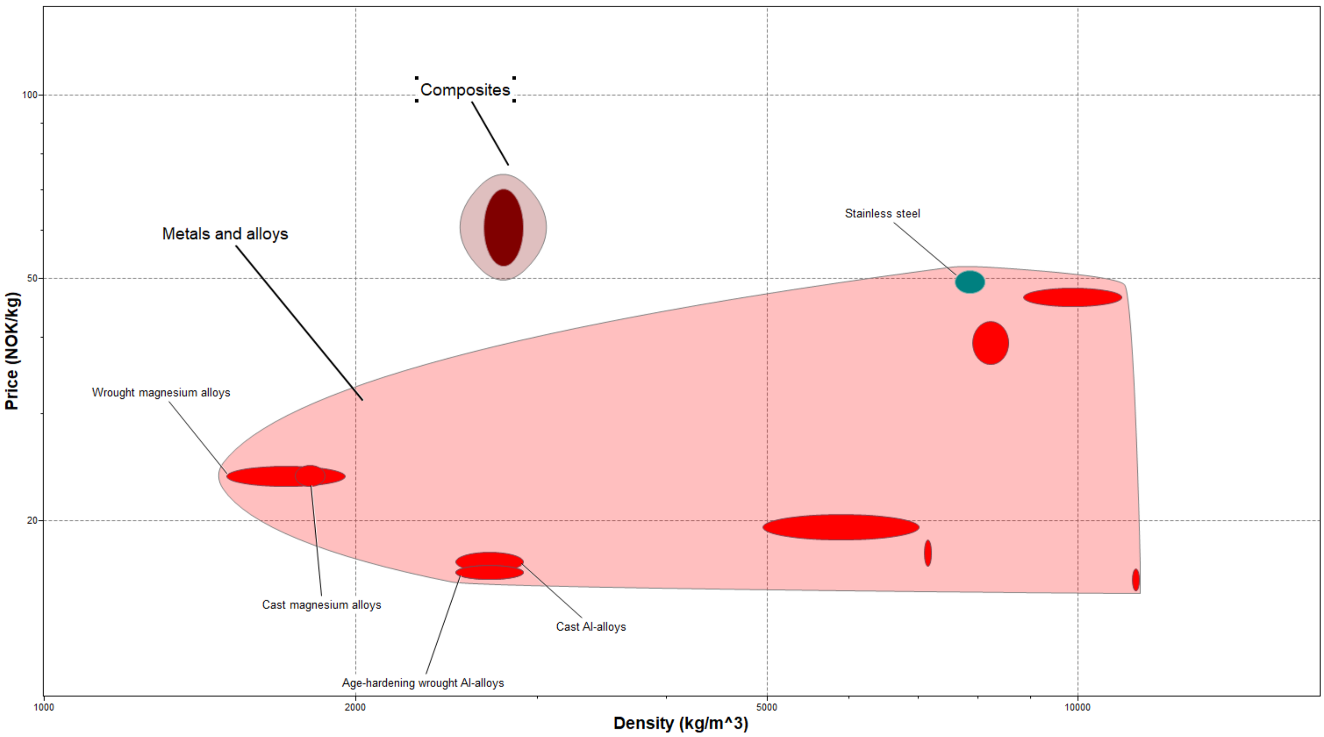

Figure 24: Material groups.

3.6.1. Bearings & Fastener

Bearings are not traditionally made from AL-alloys due to the load spectra bearings operate in. It's not favorable to have plastic deformation in bearing due to loss of integrity. By following the same limits as stated earlier the only options is composite or stainless steel. Composite and hybrid bearings are relative complex and are limited in size and applications. However stainless steel is very common in bearings. W61906 which is the selected SKF bearing for our WG-SRA is ready, of the shelf in stainless steel.

The main advantages of stainless steel are, that it suitable for very high temperatures, up to 250℃, obtain a protective film at 12% chromium content and is not affected by LEO environments. From ECSS-Q-70-71a material list, AISI316L seems like a perfect choice due to its low-cost range, good space experience, easy to obtain, excellent resistant in LEO environment and high stress corrosion.

Note that Stainless steel in contact with Al-alloys cannot be used in non-controlled environment. In this case the environment is controlled by the engineer and accepted.

Additionally, stainless steel is commonly used for fastener accordingly to ECSS-Q-70-71a. Table 17 shows the properties of AISI316L.

Table 17: AISI 316L.

Nature:	**Typical value:**	**Remarks:**
Specific Gravity	8	@Room Temperature
Ultimate Tensile Strength	560 MPa	@Room Temperature
Proof Stress (0.2%)	290 MPa	@Room Temperature
Elongation at Break	50%	@Room Temperature
Development status	Commercial	@Room Temperature
Cost Range	Very Low	@Room Temperature
Corrosion	Excellent in LEO	@Room Temperature
Stress Corrosion	High Resistance	@Room Temperature
Contact Corrosion with Al-alloys	Accepted	@Room Temperature

3.6.2. Structure

The Structure of the WG-SRA is exposed to LEO environment and should have materials accordingly. By structure everything except bearings, motor, fastener and wire are identified.

Contact stress and contact corrosion between al-alloys is very good and can be used without restrictions. Al-alloys is the primary choice of structural parts for space solution and its stable in vacuum and does not degraded in LEO environment. This accordingly to ECSS-Q-70-71a.

Because of this Al-alloys is selected as the materials for the structural parts of the WG-SRA.

From ECSS-Q-70-71a material list, ISO Al 99.5 seems like a perfect alloy due to its low-cost range, good space experience, easy to obtain, excellent resistant in LEO environment and high stress corrosion. Additionality it can be chromated for harsher environments.

Table 18 shows the properties of ISO Al 99.5.

Table 18: ISO Al 99.5.

Nature:	**Typical value:**	**Remarks:**
Specific Gravity	2.27	@Room Temperature
Ultimate Tensile Strength	75 – 146 MPa	@Room Temperature
Proof Stress (0.2%)	55-133 MPa	@Room Temperature
Elongation at Break	25%	@Room Temperature
Development status	Commercial	@Room Temperature
Cost Range	Very Low	@Room Temperature
Corrosion	Excellent in LEO	@Room Temperature
Stress Corrosion	High Resistance	@Room Temperature
Contact Corrosion same material	Excellent	@Room Temperature

4. Wire investigation

4.1. Preload

Figure 25 shows the force overview in the wire driveline. Figure 25a shows the global forces. T is the torque input from the motor while $10T$ is the output torque on the OP.

This because:

$$T = \frac{d}{2}(F_1 - F_2),$$

where F_1 is the tight side and F_2 is the slack side,

and

$$10T = \frac{D}{2}(F_1 - F_2),$$

because $\mathrm{D} = 10\mathrm{d}$.

Figure 25b shows the working forces in the wire related to the input torque without pretension or other external forces. Figure 25c shows the working forces in the wire related to the input torque with pretension.

The purpose of preloading the wire is so that no slack can be measured at worst case scenario. If slack happens, it may cause major pointing errors and interfere with the systems integrity making it unusable.

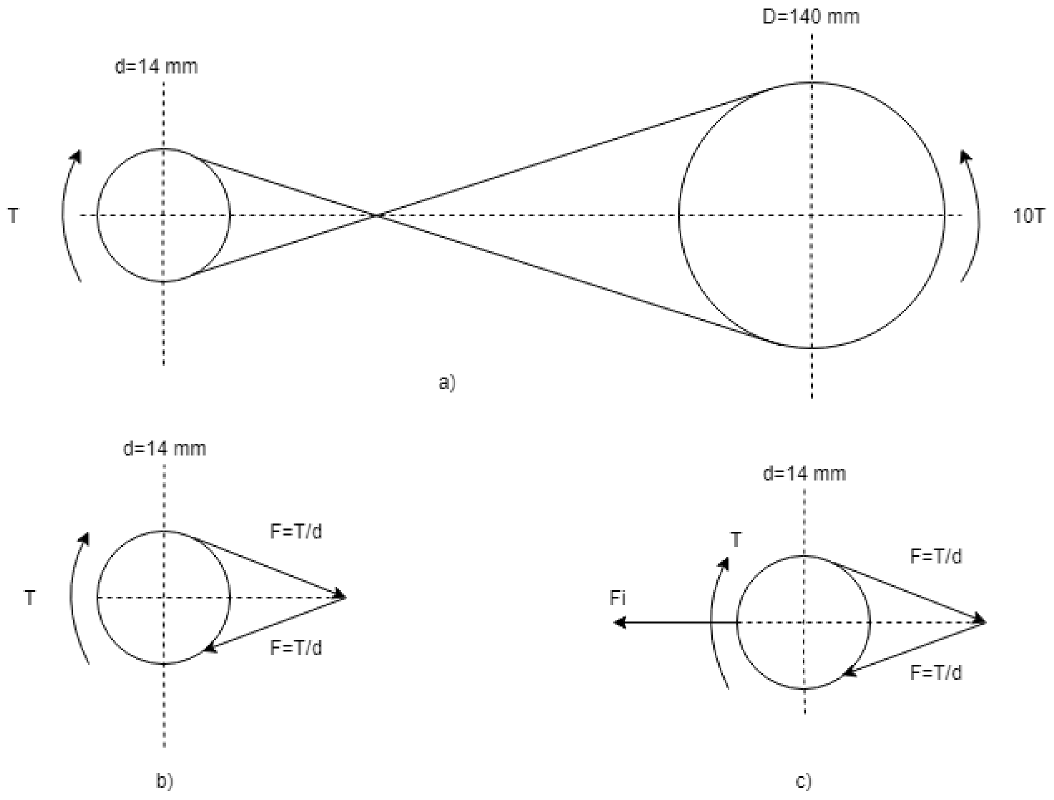

Figure 25: Force overview preload.

4.1.1. Worst case scenario – maximum torque (start and stop)

Assume that the system is at hold and receive a commando from the satellite to rotate 100 deg. Assume that the system produces maximum power accordingly to performance requirements and execute the command. We know that in this moment T is at max and we have the following force scenario.

Force due to torque

$$F-(-F)=\frac{T}{d/2}\leftrightarrow 2F=\frac{2T}{d}\leftrightarrow F=\frac{T}{d}=\frac{110\text{ mNm}}{14\text{ mm}}\cong 8\text{ N} \tag{5}$$

This implies that with no preload the force is ± 8 N. This means that we have one tight side (8N) and one slack side (-8N, this because it takes 8 N to remove all slack). To counter this, the slack side and the tight-side must have positive numbers. In other words, the preload is correct when the tension is above zero on both sides.

Thus, by checking that the forces in the system are above the larges observed forces, the correct preload can be found.

$$\sum F_{\text{system}} > 8\,.$$

Hence

$$\sum F_{\text{system}} > (F_1 - F_2) + F_{\text{p}}\,,$$

where F_1 is the tight side of 8N, F_2 the slack side of 8 N and F_{P} preload.

$$\sum F_{\text{system}} > (8-8) + F_{\text{p}}$$

$$\sum F_{\text{system}} > F_{\text{p}}$$

Thus

$$F_p > 8 \tag{6}$$

This means that when the wire is preloaded with more than 8 N the tension axially will be above zero meaning no slack. A factor of 2 is used as uncertainty. This because wires normally “lives” and need to be broken in. This was very clear during the tension tests.

Actual preload is therefore suggested to be 16N.

Note that forces related to angular movement, friction and more is excluded because of simplification reasons and relevancy.

4.2. Test

As discussed in selection 3.3, several Zylon wires was obtained. To gain information about the custom wires and the standard issued wire from Marlow. Two tests are executed. The first test is a friction test. This test is to gain information on how tough the wire is to adhesive friction against aluminum over time. Also, to see how light bending affect the life-time of the wire. The second test is to obtain properties of the wires. Properties can easily be found for the fiber, but not for a specific weaving. Important factors such as creep at maximum load, stiffness and tensile strength. Creep is highly important because of the unwanted pointing error in relation with preload. Figure 26 shows the friction test rig.

Figure 26: Friction test rig.

4.2.1. Friction test setup and results

Figure 27 shows a schematic over the test rig. A motor with an angular velocity *n* rotates with a gearbox and contact rod. This allows the motor to turn infinite degrees resulting in a linear motion for the wire as illustrated. Because of this the wire "glides" over the aluminum rod, and wear against, and friction can be investigated. The spring is needed to allow this motion, and the system is light preloaded – no slack. A hall sensor is used to count numbers of cycles, one round on the motor equals one cycle. This hall sensor is connected to an Arduino with a code to execute this act.

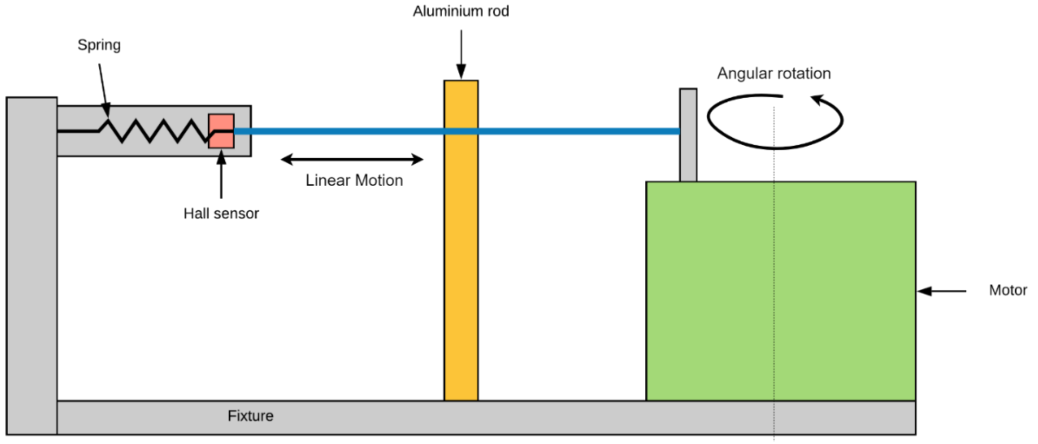

Figure 27: Test rig schematics

Table 19 shows the results from the friction test. Figure 28 and Figure 29 on the next page shows the wire before and after the test. In Figure 28 a very strong gold color is appealing, and the wire is "loosely" and beautiful braided. Additionally, no "debris" can be seen. In Figure 29 a miss color has happened, mostly due to aluminum dust from the rod. The wire is also more compact, and the wire has some "debris" – or wear.

Table 19: Friction test results

Wire:	Number of cycles:	Comment:	Result:
NORF-1	1271100	Stopped	Feasible
NORF-2	-	Not started	TBD
NORF-3	-	Not started	TBD
NORF-4	-	Not started	TBD
MALO	-	Not started	TBD

Due to limited time, NORF-1 is the only wire that was tested for friction. The result is promising because the wire shows only superficial damage. However, during this test, the wire is not preloaded correctly. I think this might be wrong and other results will occur if preloaded correctly.

That in mind, I continue, and I will use this wire for life time endurance test. This test will include much more, such as correct preload.

Figure 28: Before

Figure 29: After

4.2.2. Tensile strength – Elongation setup and results

The wires were tested at Norut Narvik by Øystein Kleven. The fixture for mounting the wire was not perfect but mounted equally for all specimens. Because of this all wires broke due to rough edges around the mount. However, this does not compromise the test because of the “real” situation. If mounted on the SRA sharp edges, knots and more must be investigated in detail to counter every weak point. I mainly did this test to compare my wires against the wires used in the Kodyna report from KSS. I cannot show the Kodyna report, however the wires I obtained seems to have a higher tensile strength, but not as good in elongation. I think this is because the NORF wires is custom braided, no specific was given on how “tight” the braid should be. The Marlow wire is slightly better.

I still choose to use this data because it contains the uncertainty related to the real life and is equal for all specimens. The most interesting results are presented in the figures below.

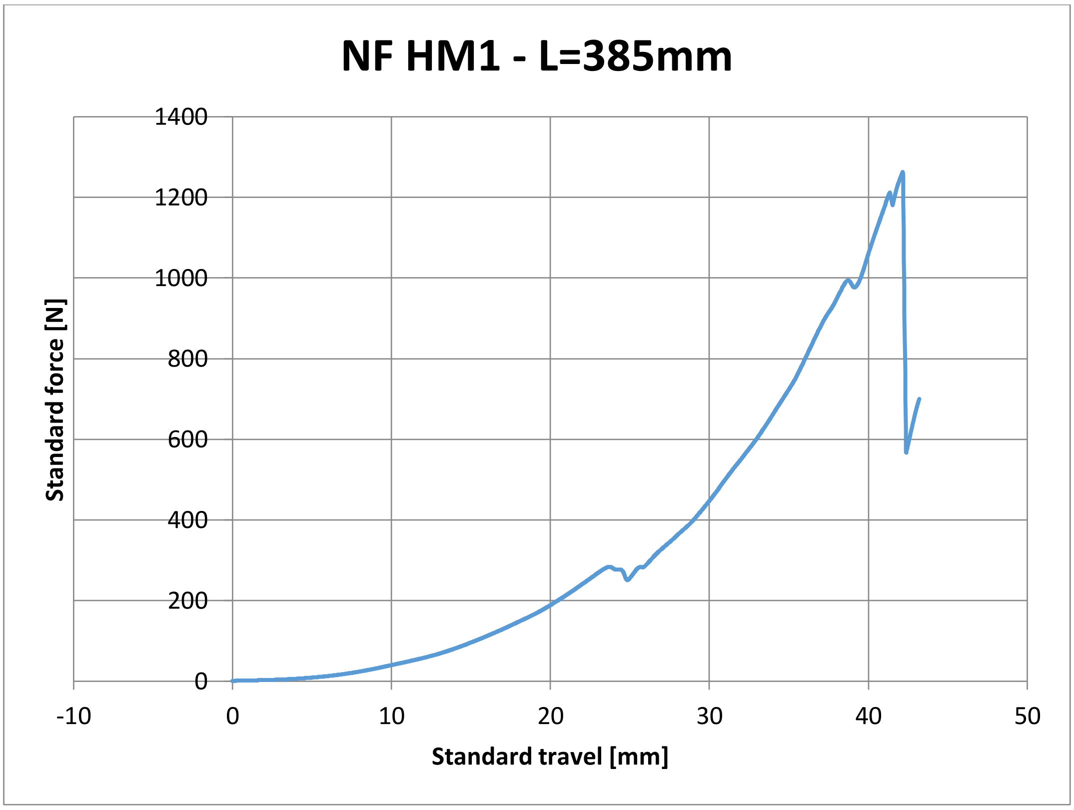

Figure 30: Norsk Fletteri High modulus circular wire 1mm dimeter

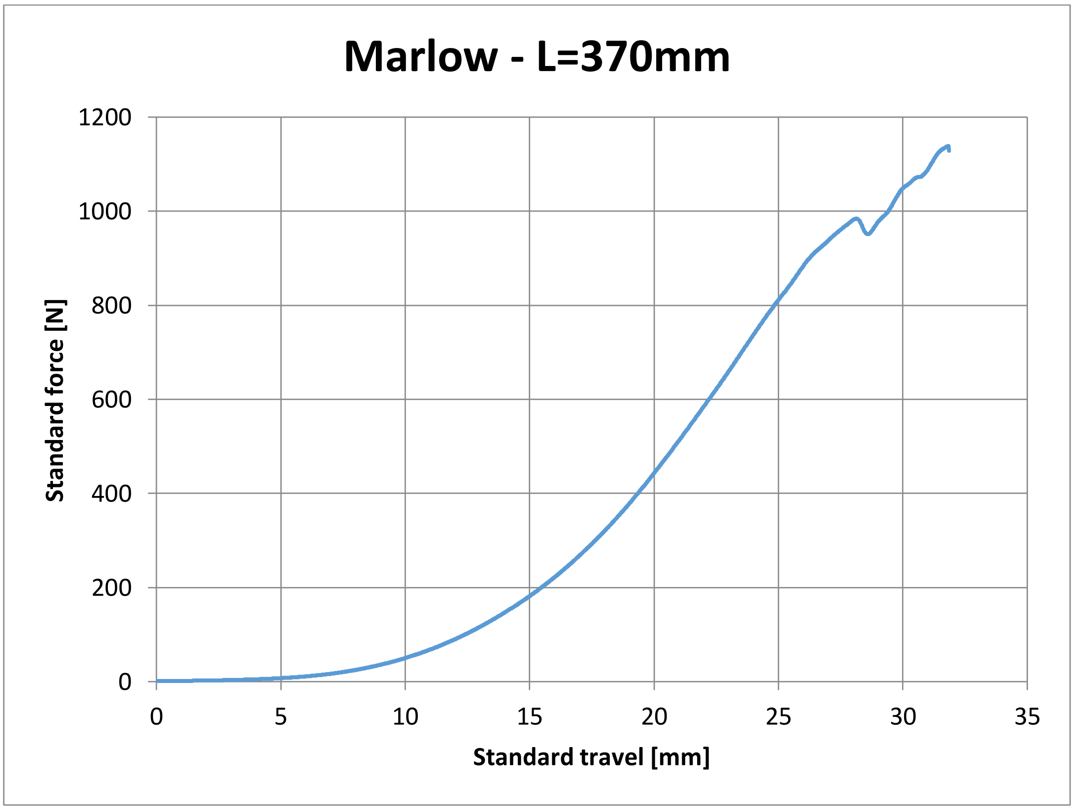

Figure 31:Marlow High modulus circular wire 1mm dimeter

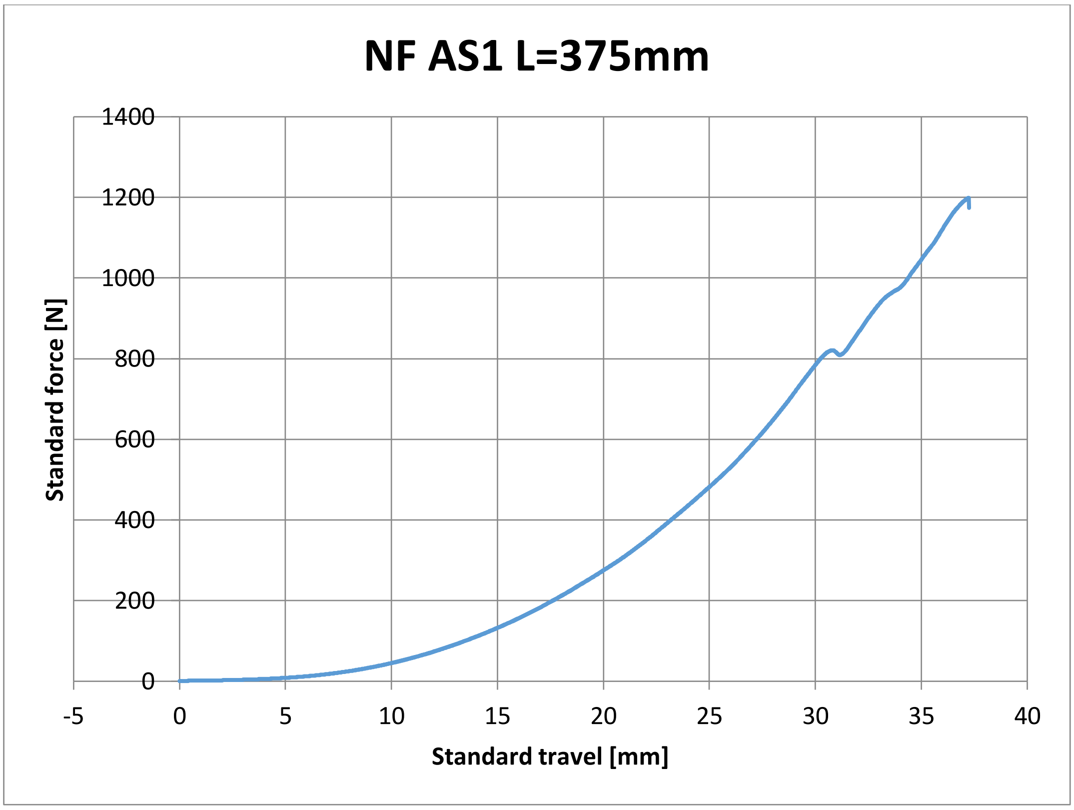

Figure 32: Norsk Fletteri Normal modulus circular wire 1mm dimeter

5. Technical budgets

5.1. Mass Budget

Requirement TREQ-3 defines that the WG-SRA should have a mass of less than 1.0kg. This is an AB requirement meaning that this requirement must be fulfilled for the system to be stable. This requirement is also marked as TAR meaning it can be verified trough test, analysis or review.

To verify this requirement, Solidworks mass properties, Solidworks materials and datasheets of OTSP is applied and used. Table 20 shows the mass distribution of the SRA prepared for test.

Table 20: Mass budget.

Mass below bearings:			
Specimen:	**Source:**	**Mass including test specimens:**	**Mass actual WG-SRA:**
Base	Solidworks	470g	230g
Bearing house	Solidworks	105g	105g
Bearings	SKF data sheet	96g	96g
Lock Nut	SKF data sheet	70g	70g
Stiffener	Solidworks	55g	55g
Motor bracket	Solidworks	55g	55g
Motor	Maxon Motor data sheet	160g	160g
Bolts	Solidworks	109g	109g
Total mass below bearings:		1120g	771g
Mass above bearings:			
Disk	Solidworks	193g	193g
Loadcell bracket	Solidworks	30g	0
Load Cell	HBM data sheet	70g	0
Preload mechanism	Solidworks	65g	30
Total mass above bearings:		358g	223g
Total mass		**1478g**	**994g**

From Table 20 the total mass for the SRA including test equipment is 1478g and without 994g. Note that the base significantly reduces mass. This is because it is over dimensioned to simplify mounting and adjustability regarding testing. Normally can this part be implemented inside or included in the satellite.

Because of this TREQ-3 is verified and approved trough review. The final prototype is going to be weighed and hard verified in part C.

5.2. Strength Budget

Accordingly, to ECSS-E-ST-32-10C a FOS for structural groups for satellites in the beginning phase should be 1.2. A FEM analysis within the load spectra discussed in part A and the masses obtained in the selection above is executed with the following results. Note that this analysis does not complies with random vibration fatigue, only static with variable G-force and assume infinite stiffness.

The following mesh is used for the analysis:

Table 21: Mesh.

Mesh type	Solid Mesh
Mesher used	Standard Mesh
Jacobian Points	29 points
Element Size	3.7 mm
Tolerance	0.2 mm
Mesh Quality	High
% of disoriented elements	0

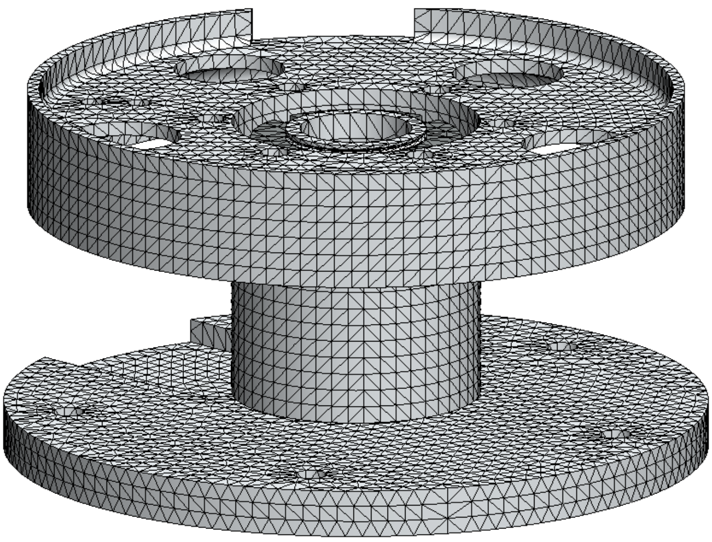

Figure 33: Mesh on structure.

The following fixtures and loads is used for the analysis:

Table 22: Loads and Fixtures.

Forces:	**Value**
Preload force	207 N
External load	15N
G_{rms} variable	58 m/s^2
Fixtures:	
Fixed – Fixed -Fixed	0 mm

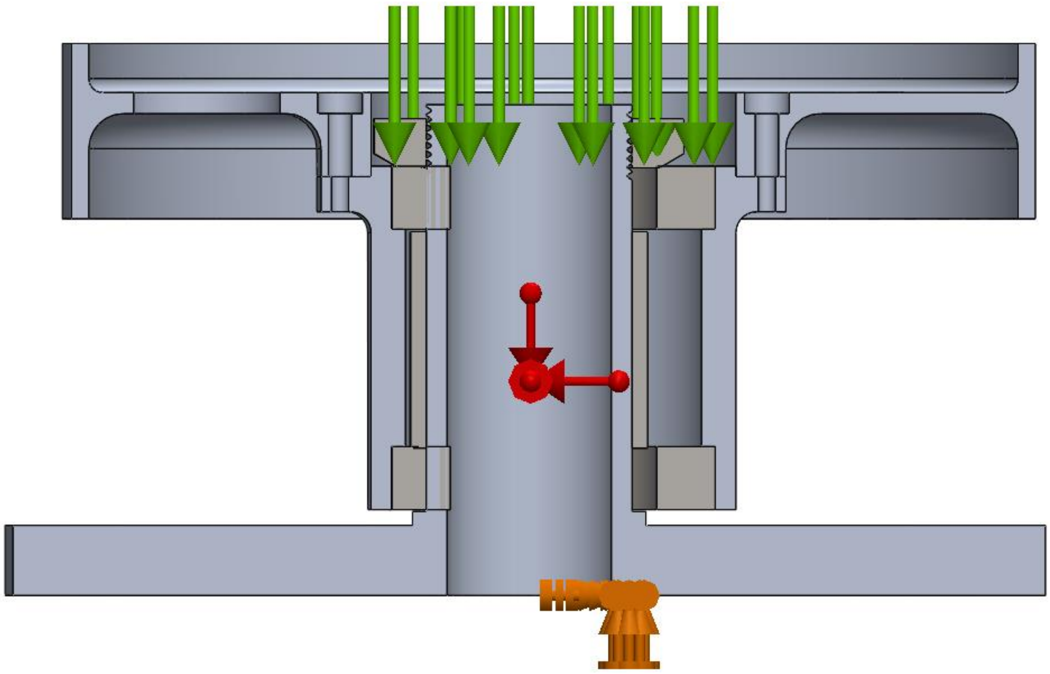

Figure 34: Forces and fixtures on model.

The green arrows in Figure 34 represent the loads both external and preload. The red arrows represent the variable G_{rms} and the orange, the fixed. Note that the hole bottom plate is fixed in all xyz- directions.

With this setup following loads and FOS is obtained for the model. Note that for aluminum proof stress 0.2 % is 55-133 MPa and for stainless steel 290 MPa. This means that maximum load with a FOS of 1.2 is 44-106.4 MPa for aluminum parts and 232 MPa for stainless steel parts.

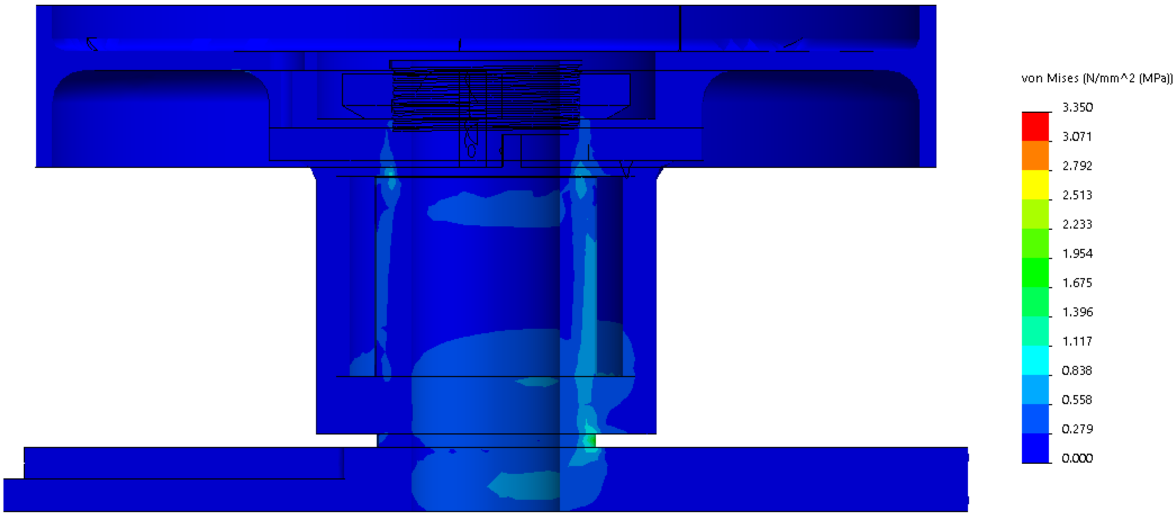

Figure 35: Resulting Stresses.

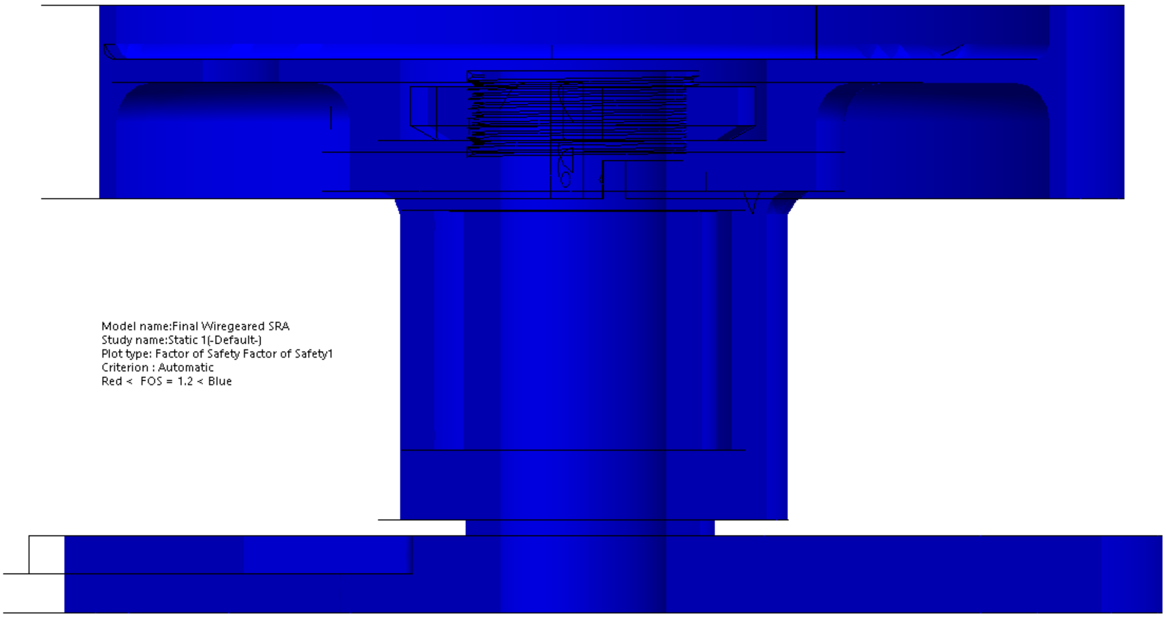

Figure 36: Resulting FOS.

Thus, Figure 35 shows that maximum stress obtained is 3.350 MPa between bearing and stiffener. These components are made from stainless steel and complies with desired FOS. It's a feasible result because all forces should go through the bearing house and its preload. Figure 36 verifies that the whole structure is within the desired FOS with Solidworks material appliance and calculation.

By this I conclude that the structure keeps its integrity and complies with the desired requirement. However, I suggest a random vibration fatigue force analysis for an upcoming iteration.

5.3. Torque Budget

Requirement TREQ-12 defines that the driveline must comply with the friction in the WG-SRA. This friction was estimated to 345,6 mNm for both bearings. This is an AA requirement meaning that this requirement must be fulfilled for the system to be stable. This is also a TA requirement meaning that it can be verified trough analysis or test. Note that the total angular momentum to drive the system accordingly to ECSS is 1077.5 mNm. Following figure shows performance graphs and input values.

A random virtual sequence is send to the WG-SRA with the following inputs.

Table 23: Sequence input.

Time (seconds):	Value (velocity) at pinion:	Comment:
0.0 seconds	0.00 deg/s	Starts at zero.
0.5 seconds	900.00 deg/s	Accelerate.
2.0 seconds	900.00 deg/s	Holding velocity.
3.0 seconds	0.00 deg/s	Deaccelerating to stop.
4.5 seconds	900.00 deg/s	Accelerating.
5.0 seconds	0.00 deg/s	Deaccelerating to stop.
6.0 seconds	0.00 deg/s	Hold stop.

Figure 37 shows the output torque distribution, the motor produces at max 108 mNm at full acceleration. TREQ-5 is therefore verified with a FOS of $\frac{T_{\text{motor}}}{T_{\text{measured}}} = \frac{140}{108} = 1.29$.

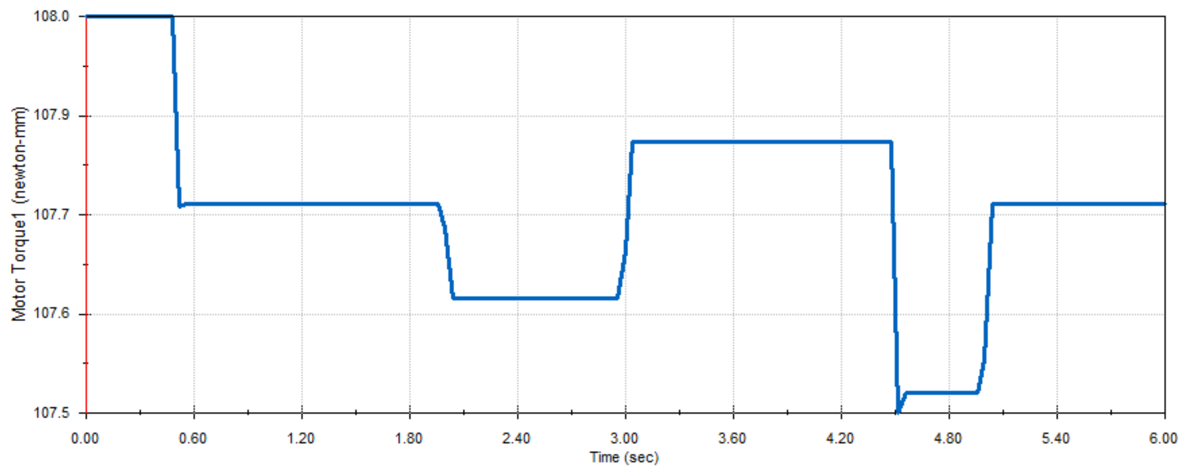

Figure 37: Resulting Motor Torque.

5.4. Performance Budget

Requirements TREQ-8 and 9 define that the driveline shall obtain a max angular velocity of 90 deg/s and be able to acceleration to 90 deg/s within 0.5 second. These requirements are AB, meaning that these requirements must be fulfilled for the system to be stable. These requirements are also marked AR meaning that it can be verified trough analysis or review. Following figures shows the resulting performance.

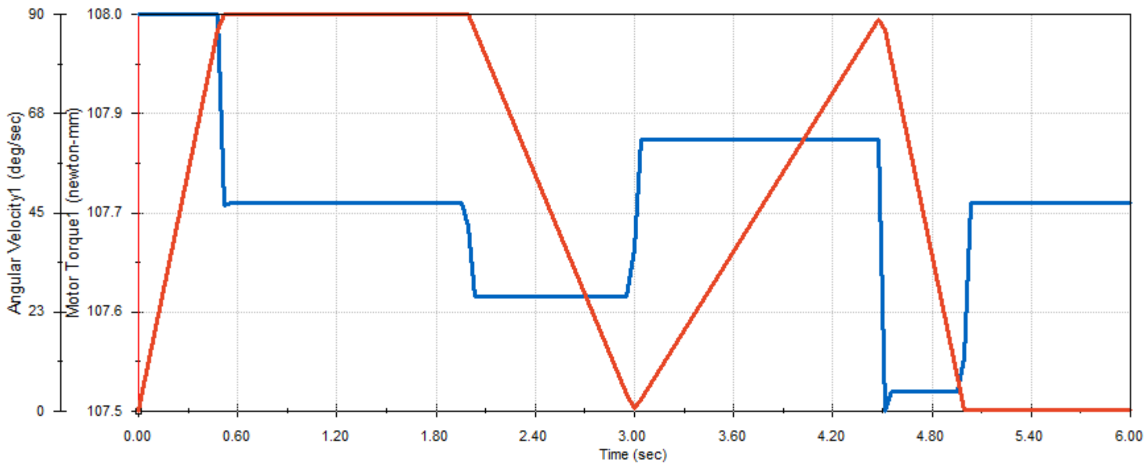

Figure 38 Resulting performance (red line) vs allowable motor torque (blue line).

By using the same input parameters for the motor and obtaining the same torque graph, performance can be measured on the output shaft. The blue line is the same torque distribution as in selection 5.3 and the red line resulting performance.

At 0.5 seconds the velocity of the output shaft is 90.00 deg/s as required. Torque drops but velocity is kept stable. Because of this result TREQ-8 and 9 is verified.

5.5. Pointing Budget

Requirement TREQ-2 defines that the output shaft of the WG-SRA should have a pointing error, or angular displacement error of less than 0.02 deg. This is an AB requirement meaning that this requirement must be met to obtain a stable system. Also, this requirement is marked TAR meaning that it can be verified trough test, analysis or review. The following error is considered.

5.5.1. Error due to position sensor and control system

Assuming that the WG-SRA is driven by a code much like the one used in the SSM project, with direct input and output on the motor. The following applies.

The encoder used on the motor has 4096 steps per revolution.

$$\frac{360 \text{ Degrees}}{4096 \text{ Steps}} = 0.08789 \text{ degrees/steps.} \tag{7}$$

This implies that one step is 0.08789 degrees. With respect to gear ratio we have the following pointing error due to encoder:

$$\frac{0.08789 \text{ degrees}}{10} = 0.008789 \text{ degrees.} \tag{8}$$

Assuming that control resolutions is 2000 steps per pole pair, with 8 pairs much like the code used in the SSM project. Then the following applies.

$$\frac{360 \text{ Degrees}}{8 \text{ pairs} \cdot 2000 \text{ Steps}} = 0.0225 \text{ degrees.} \tag{9}$$

This implies that the resolution generates 0.0225 degrees error. With respect to gear ratio total pointing error due to resolutions is

$$\frac{0.0225}{10} = 0.00225 \text{ degrees.} \tag{10}$$

Total pointing error due to position sensor and control system is therefore the sum of equation 8 and 9.

$$0.008789 \text{ degrees} + 0.00225 \text{ degrees} = 0.0110 \text{ degrees.}$$

5.5.2. Error due to wire stiffness

Toyobo states the following *"For ZYLON® HM, non-recoverable strain after 100 hours under 50% of breaking load (Safety factor (SF)=2) is less than 0.03%."* Assuming that the system is has infinite stiffens – direct drive with the same load spectrum as Toyobo states. And a factor of 0.0003 is applied to represent increased length in the wire. The following error is observed

$$360 \text{ degrees} \cdot 0.0003 = 0.108 \text{ degrees.} \tag{11}$$

With respect to gear ratio:

$$\frac{0.108 \text{ degrees}}{10} = 0.0108 \text{ degrees.} \tag{12}$$

Thus, total pointing error

$$0.0110 \text{ degrees } + 0.0108 \text{ degrees} = 0.0218 \text{ degrees.}$$

5.6. R&D Cost Budget

A budget of TBD was given to this project, Table 24 shows the development cost excluding labor hours.

Table 24: R&D Budget.

Description:	Development type:	Cost EURO:	Comment:
Motor with encoder and hall sensor	External Part	535.00	
Arduino KIT	External Part	100.00	
Maxon Motor controller	External Part	300.00	
Load cell	External Part	200.00	
Load cell instrument	External Part	840.00	
SKF Lock Nut	External Part	44.00	Estimated based on SSM budget.
SKF Bearings	External Part	130.00	Estimated based on SSM budget.
Production of 3d model	Manufactured	2000.00	Estimated based on SSM budget.
Miscellaneous	N/A	700.00	This post includes wires and wire required equipment, motor for friction test-rig and more.
Total R&D cost excluding labor:		**4849.00 Euro**	

5.7. Production Cost Budget

Requirement TREQ-9 defines that the finished prototype should cost less than 5,000 euro to mass produce. This is an AC requirement meaning that the requirement must be fulfilled for the system to be stable. Also, this requirement is marked R, meaning that it can be verified trough review. Following table shows the production budget.

Table 25: Production cost budget.

Description:	Cost EURO:	Comment:
Parts & construction	3400.00	Based on R&D cost excluding labor and reduced by 35% because of mass quantity. The number 35% is based on normal quantity discount.
Labor	1500.00	Assuming one worker needs 15 effective hours from drawing to product with an hour cost of 100euro. The hour price is estimated on normal project price hours and the hour spend is an “calculated” guess.
Total Production cost	**4900.00 EURO**	

6. Prototype Status

6.1. Proof of Concept

The proof of concept is made based on the dynamical outline of the 3d design in Solidworks. The first "prototype" is used to verify the dynamical performance intentionally wanted. Example, Solidworks does not comply with rope, chains or "free" movement parts and the "prototype" verify this aspect instead. Based on this and with the 3d model, drawings and production of the prototype can start.

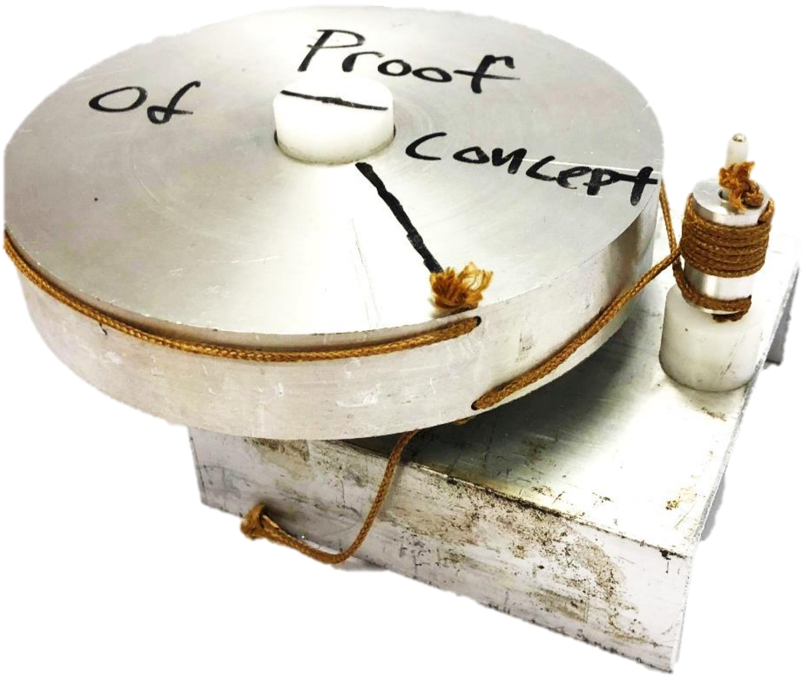

Figure 39: Proof of concept.

6.2. Prototype

Figure 40 shows the finished prototype fixed with wire and ready for testing and inspection. Thoughts and engineering statements are discussed in selection 6.3.

No large error is observed, initial function testing is started and finished, new wiring added with respect to test chamber started and finished. The prototype is ready for vacuum function test and vacuum life time test.

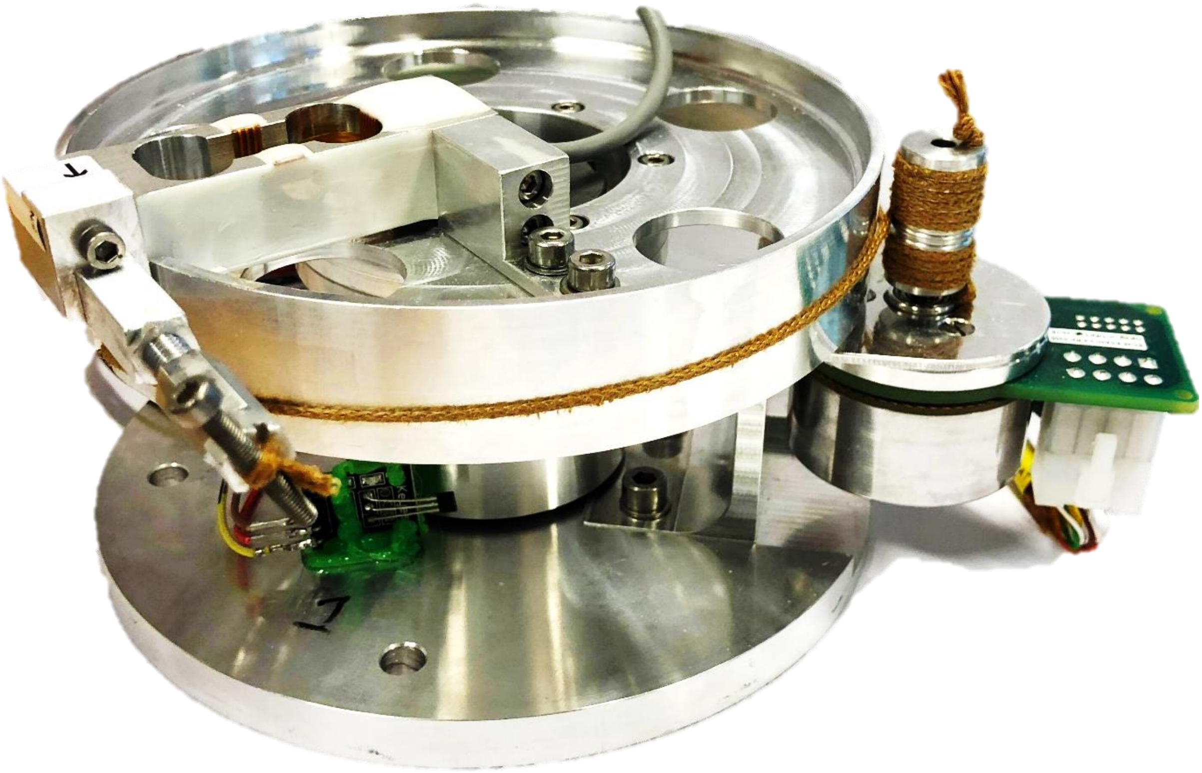

Figure 40: Prototype.

6.3. Remarks

Some improvement points are found. Figure 41 shows the location where the wires are connected to the output pulley. These points have some guide tracks for the wire to ensure that the wire is not exposed to rough edges, or steep angles. This track seems a bit too small as the wire is very close to the edge. Because of this the tracks should be redesigned to be 100% sure so that this cannot compromise the integrity of the wire. Figure 42 shows the angle of the wire between the load cell and the guide track. The track should be designed in a way so that the wire is not exposed to any sharp edges. Right now, the design is slightly off and the angle of the wire a bit to sharp. This results in "connection" between the wire and unwanted "metal". Because of this the guide track should be redesigned to match the angle 100%. Figure 43 shows the motor and motor bracket. The forces on the motor is low due to equilibrium, however KSS suggested that the motor bracket should be reinforced with some ribs on the side to keep maximum stiffness. In addition, on the pully, one wire had to be skipped to ensure that the wires are not in contact, this is a small design error and hard to verify without testing because of different wires and styles. Because of this the pitch should be increased slightly.

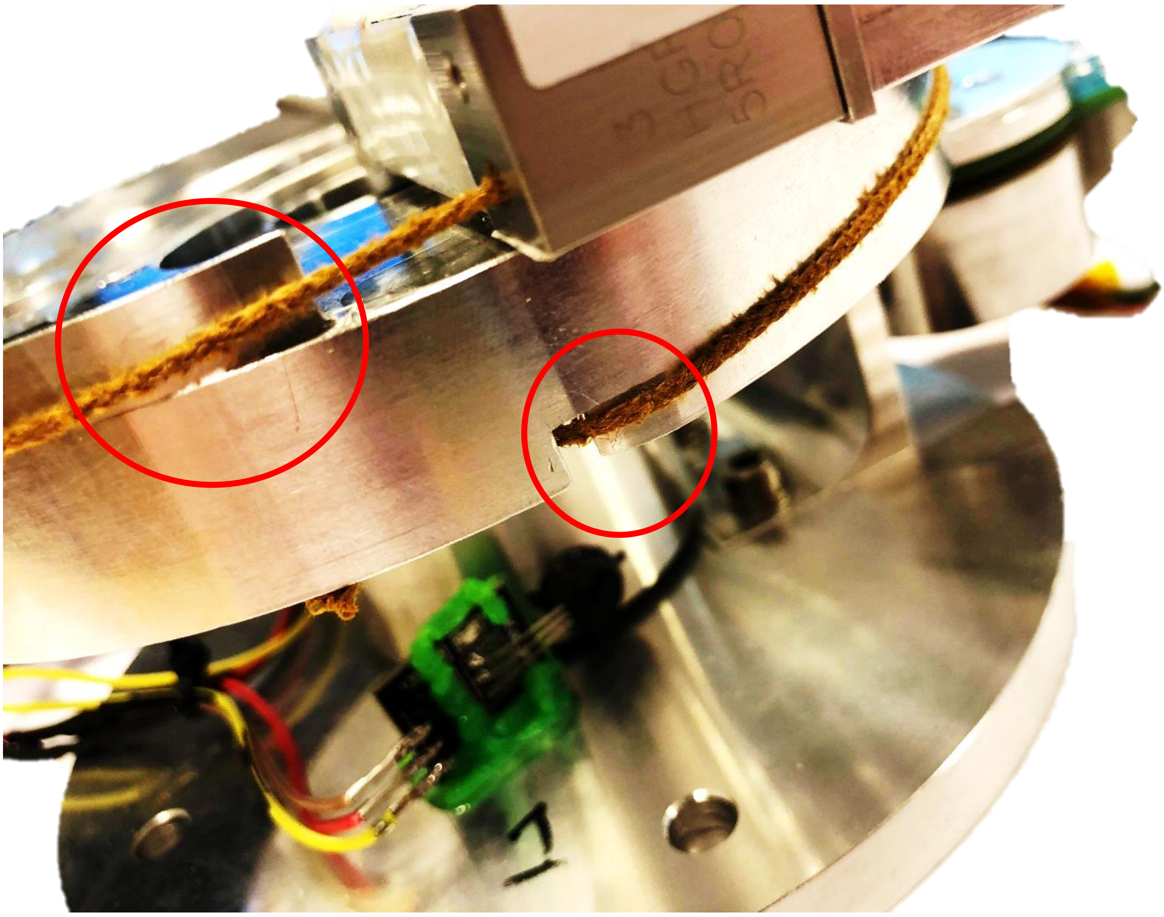

Figure 41: Wire guide tracks.

Figure 42: Wire drive angle.

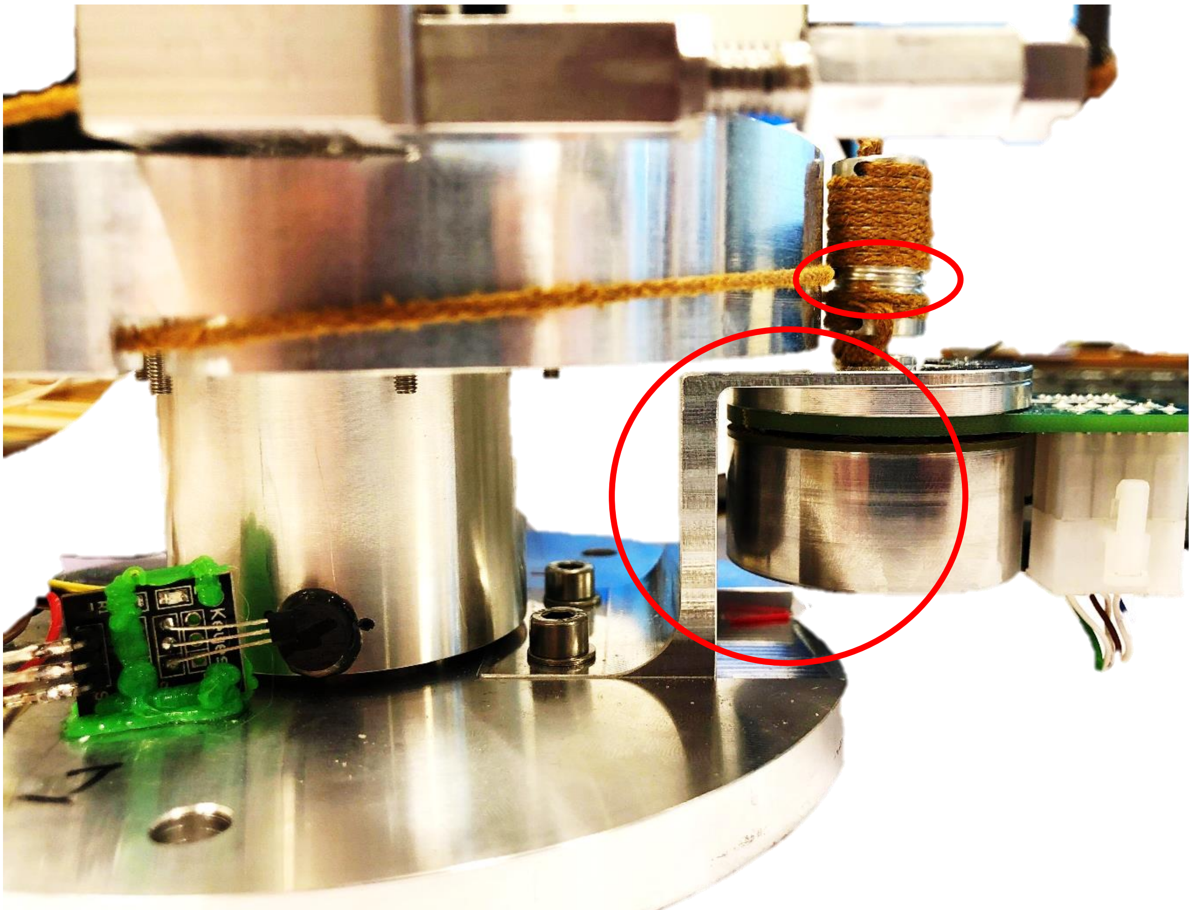

Figure 43: Motor and motor bracket.

7. References

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8. Appendices

8.1. Means, Features and ideation

The first step in ideation is to generate features that may be or can be implemented in the design. Table 26 shows the generated features and means.

Table 26: Features/Means

Features	Means					
Preload mechanism	3 "Guitar" tuning peg	1 Rotational bolt tensioner	~~Expanding feature~~	2 Non-twisting bolt tensioner (axial)	~~Twisting bolt tensioner~~	---
Load Cell Configuration	2 Compression/ tension	~~Washer~~	1 Bending	~~On motor~~	---	---
Wire Configuration	Parallel type crossed thread Configuration	~~Parallel type open thread configuration~~	~~Right angle type crossed thread Configuration~~	~~Right angle open thread configuration~~	---	---
Mounting	1 Fixed/on load cell with tensioner	~~Fixed/Fixed~~	2 Tensioner/load cell	---	---	---
Support	SSM Bearing setup	~~Bolt Through setup~~	---	---	---	---

Based on Table 26 a long list of ideas was generated and illustrated in Table 27. Accordingly, to the process shown in Figure 1 a preliminary design review was executed with the following results:

- Idea 1, 5 and 9 was instantly removed since the configuration is non-executable.
- Idea 2,4,6,7,8,10,11,12 seems feasible but have medium to high integration cost, meaning that the solution may work but is "high" cost to implement. Idea 2,4,6,8,10 and 12 are selected as the short list of ideas based on design meeting with KSS.

Table 27: Long List Ideas.

						Preliminary Design Review	
	Preload mechanism	**Load Cell Configuration**	**Wire Configuration**	**Mounting**	**Support**	**Integration Cost**	**Physical Stability**
Idea 1	3 "Guitar" tuning peg	2 Compression/ tension	Parallel type crossed thread Configuration	1 Fixed/on load cell with tensioner	SSM Bearing setup	3+2+1=6	Not Compliance
Idea 2	3 "Guitar" tuning peg	1 Bending	Parallel type crossed thread Configuration	1 Fixed/on load cell with tensioner	SSM Bearing setup	3+1+1=5	Compliance
Idea 3	3 "Guitar" tuning peg	2 Compression/ tension	Parallel type crossed thread Configuration	2 Tensioner/load cell	SSM Bearing setup	3+2+2=7	Compliance
Idea 4	3 "Guitar" tuning peg	1 Bending	Parallel type crossed thread Configuration	2 Tensioner/load cell	SSM Bearing setup	3+1+2=6	Compliance
Idea 5	1 Rotational bolt tensioner	2 Compression/ tension	Parallel type crossed thread Configuration	1 Fixed/on load cell with tensioner	SSM Bearing setup	1+2+1=4	Not Compliance
Idea 6	1 Rotational bolt tensioner	1 Bending	Parallel type crossed thread Configuration	1 Fixed/on load cell with tensioner	SSM Bearing setup	1+1+1=3	Compliance
Idea 7	1 Rotational bolt tensioner	2 Compression/ tension	Parallel type crossed thread Configuration	2 Tensioner/load cell	SSM Bearing setup	1+2+2=5	Compliance
Idea 8	1 Rotational bolt tensioner	1 Bending	Parallel type crossed thread Configuration	2 Tensioner/load cell	SSM Bearing setup	1+1+2=4	Compliance
Idea 9	2 Non-twisting bolt tensioner (axial)	2 Compression/ tension	Parallel type crossed thread Configuration	1 Fixed/on load cell with tensioner	SSM Bearing setup	2+2+1=5	Not Compliance
Idea 10	2 Non-twisting bolt tensioner (axial)	1 Bending	Parallel type crossed thread Configuration	1 Fixed/on load cell with tensioner	SSM Bearing setup	2+1+1=4	Compliance
Idea 11	2 Non-twisting bolt tensioner (axial)	2 Compression/ tension	Parallel type crossed thread Configuration	2 Tensioner/load cell	SSM Bearing setup	2+2+2=6	Compliance
Idea 12	2 Non-twisting bolt tensioner	1 Bending	Parallel type crossed thread Configuration	2 Tensioner/load cell	SSM Bearing setup	2+1+2=5	Compliance

8.1.1. Preload Mechanisms

The first feature is the preloading mechanism of the wire. Suggested solutions on how to preload are "Guitar" tuning peg, rotational bolt, expanding feature, axial bolt and twisting bolt tensioner.

Figure 44: Tuning peg [9].

- The guitar tuning peg mechanism uses a worm gear that drives an axel. This solution is compact, high versatility and can be operated easy. The drawback of this features is that it consists of several parts and seen as a "complex" mechanism, in addition a hold torque has to be added. Based on this guitar tuning peg obtain an implementation score of 3, meaning that it is the "hardest" to implement.
- A rotational bolt tensioner mechanism is slickly just a bolt rotating, pulling the wire in, much like a winch. This solution is very compact and hold torque can easily be managed. The drawback of this solution is that it has low versatility. This is the simplest solution and based on these inputs rotational bolt tensioner obtains an implementation score of 1, meaning that it is the "easiest" to implement.

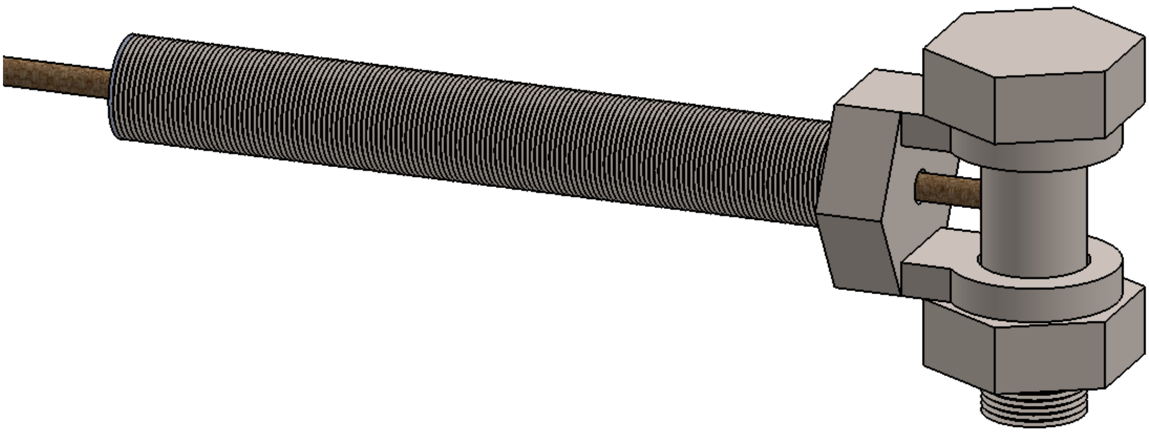


Figure 45: Rotational Bolt.

- An expanding feature can be used to tensioning the wire, however this will not compile with gear ratio and therefore excluded instantly.
- An axial bolt tensioning mechanism is much like the rotational bolt mechanism however instead of spin in the wire, the bolt extends in length, axially. This solution is compact, medium versatility and easy to operate. The drawback of this mechanism is the friction between a "gliding" bolt and the bolt. If the friction between the gliding bolt and bolt is too high, the wire will rotate with the bolt, which is not acceptable. Based on this implementation score of 2 is added, meaning that it is "ok" to implement.

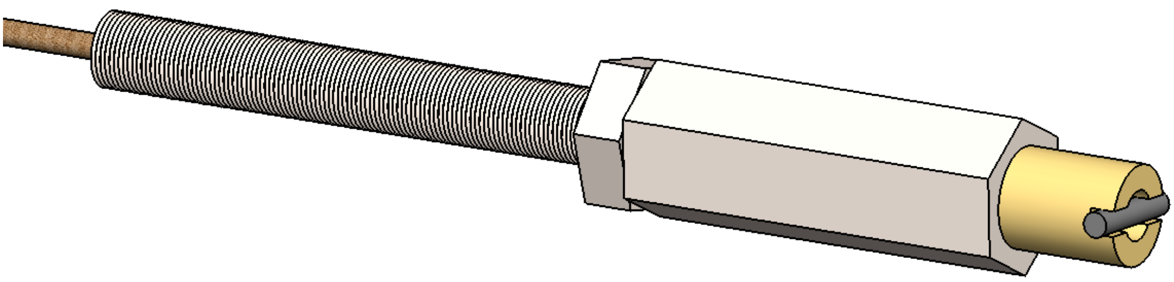


Figure 46: Axial Bolt.

- The last means of the preloading mechanism is an axial bolt tensioning mechanism much like the one above, however without a gilding bolt. This causes the wire to twist. This is not acceptable and therefore excluded instantly.

8.1.2. Load Cell Configuration

The second feature is the load cell configuration. Suggested solutions on which load cell to implement are tensioning/compress cell, washer cell, bending cell and on motor cell.

- Tensile/compress cell is a load cell measuring strain force or compression force. This cell is available in the load spectra and can be implemented. This cell is not complied with preloading mechanism mounted on cell. Therefore, the configuration must be adjusted accordingly. Because of this tensile/compress cell obtain an implementation score of 2, meaning that it is “ok” to implement.

- Washer cell is a load cell measuring compression force. This kind of load cell is not available within the load spectra, and therefore excluded instantly.

- Bending cell, is a load cell measuring the momentum and recalculate to strain force. This kind of cell is available in the load spectra and can be implemented. This cell is highly completable with the preloading mechanisms, assuming mounting holes goes all through. Because of this, bending cell obtain an implementation score of 1, meaning that it is the “easiest” to implement.

- On motor cell the last configuration is to place a load cell on the motor, however this feature increases the wire length and decrease the stiffness of the system. By discussion with KSS this configuration was instantly excluded because of this.

Figure 47: Bending cell [7].

Figure 48: Tensile/compressive force cell [8].

8.1.3. Wire Configuration

The third feature is the wire configuration. Suggested solutions on what wire configuration to implement are parallel crossed wire type configuration, parallel open wire type configuration, right angle type crossed wire type configuration and right-angle type open wire type configuration.

- Parallel crossed wire type configuration is a configuration where the wire is crossed resulting in opposite motion of the large pully versus the small pully. This setup also has large contact arc compared to open configuration. The pullies are placed in parallel.

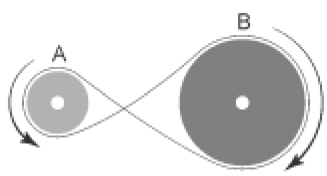


Figure 49: Crossed Type [10].

- Parallel open wire type configuration is a configuration where the wire is not crossed resulting in rotation in the same direction. This setup has lover contact arc than crossed belt configuration. The pullies are placed in parallel. Because of less contact arc and discussion with KSS this configuration is instantly excluded.

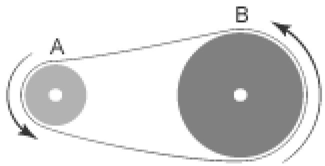


Figure 50: Open Type [10].

- Right angle type crossed wire type configuration is a configuration where the wire is crossed resulting in opposite motion of the large pully versus the small pully. This setup also has large contact arc compared to open configuration. The pullies are placed in a right angle. Because of less contact arc and discussion with KSS this configuration is instantly excluded.

- Right-angle type open wire type configuration is a configuration where the wire is not crossed resulting in rotation in the same direction. This setup has lover contact arc than crossed belt configuration. The pullies are placed in a right angle. Because of less contact arc and discussion with KSS this configuration is instantly excluded.

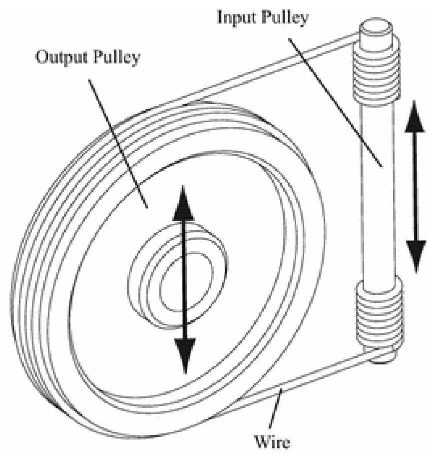


Figure 52: Right Angle Type [11].

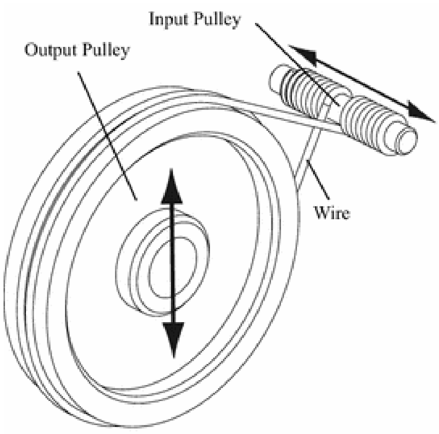


Figure 51: Parallel Type [11].

8.1.4. Mounting

- The fixed/on load cell mounting configuration is a configuration where one end of the wire is fixed to the base pulley and the other end is fixed to the load cell integrated with the preloading mechanism
- The fixed/tensioner configuration is a configuration where one end is fixed to the load cell and the other end is fixed to the base pulley integrated with the preloading mechanism.

8.1.5. Support

- The SSM bearing setup was instantly selected by KSS.

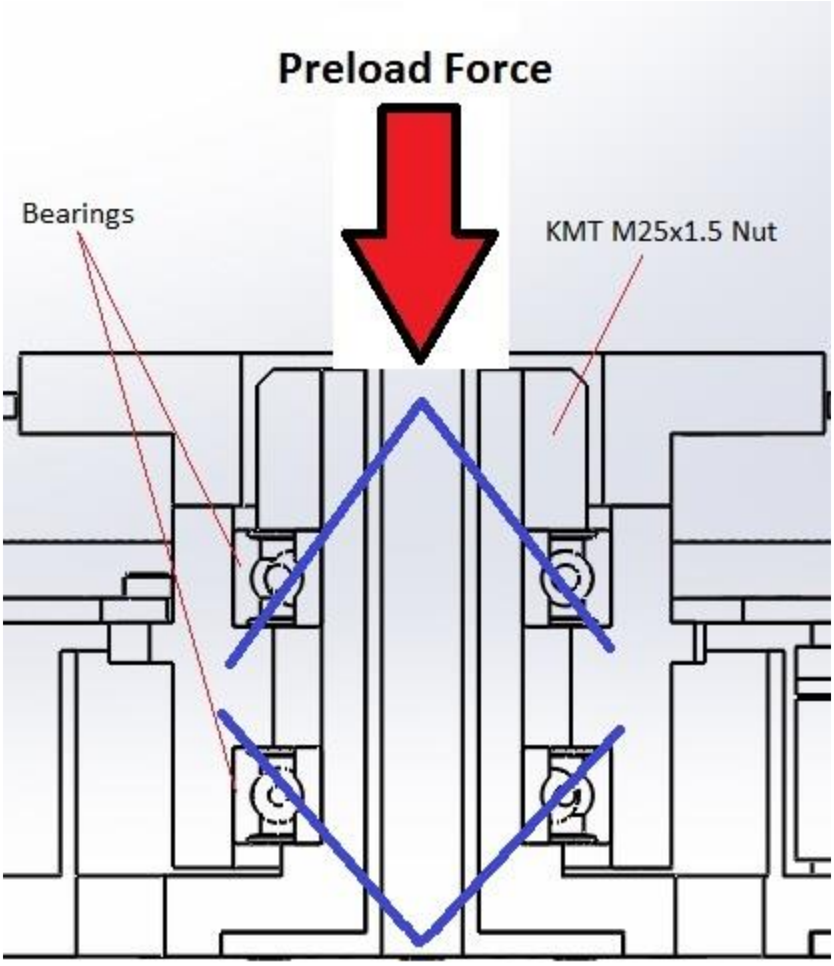
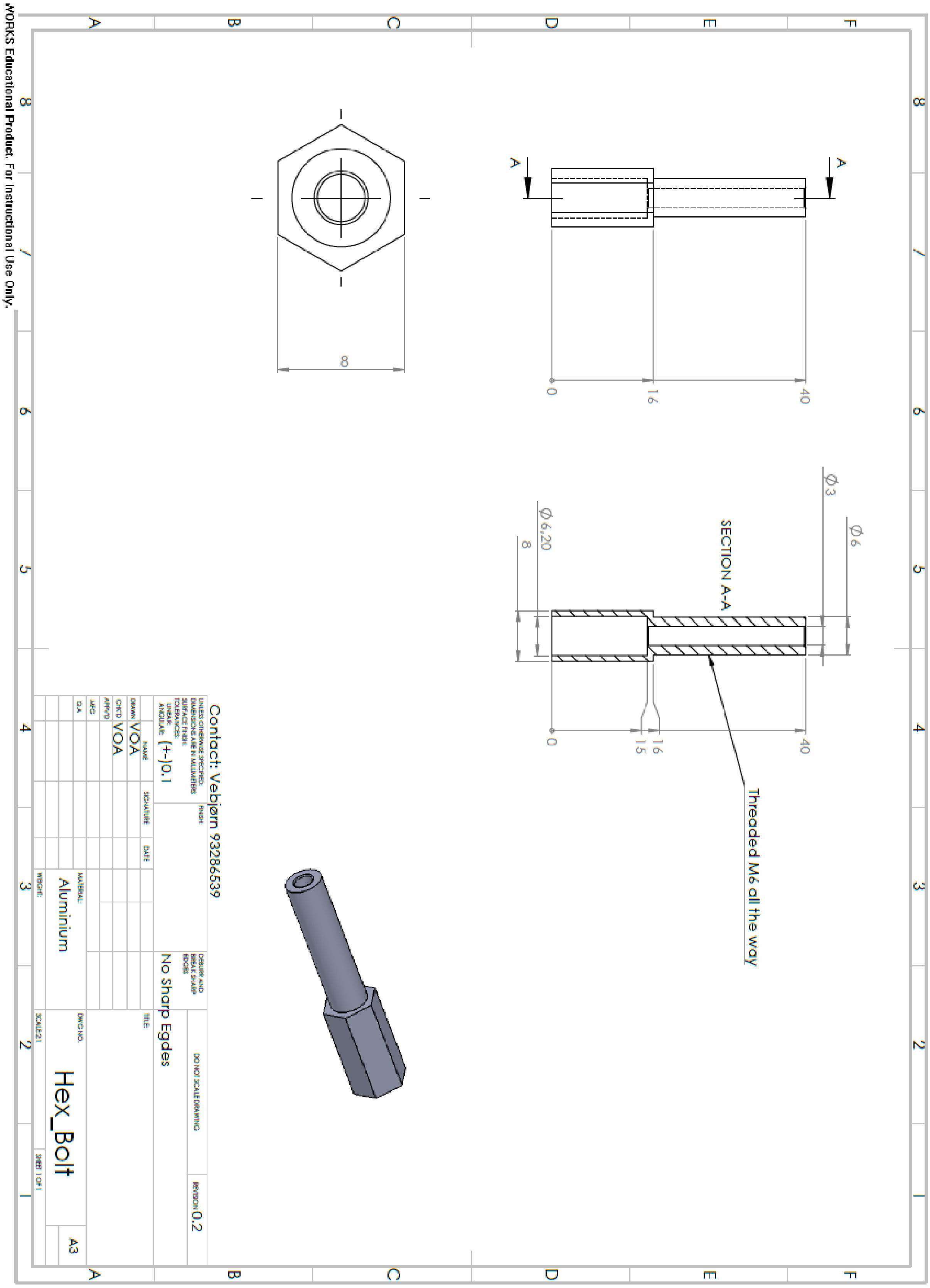


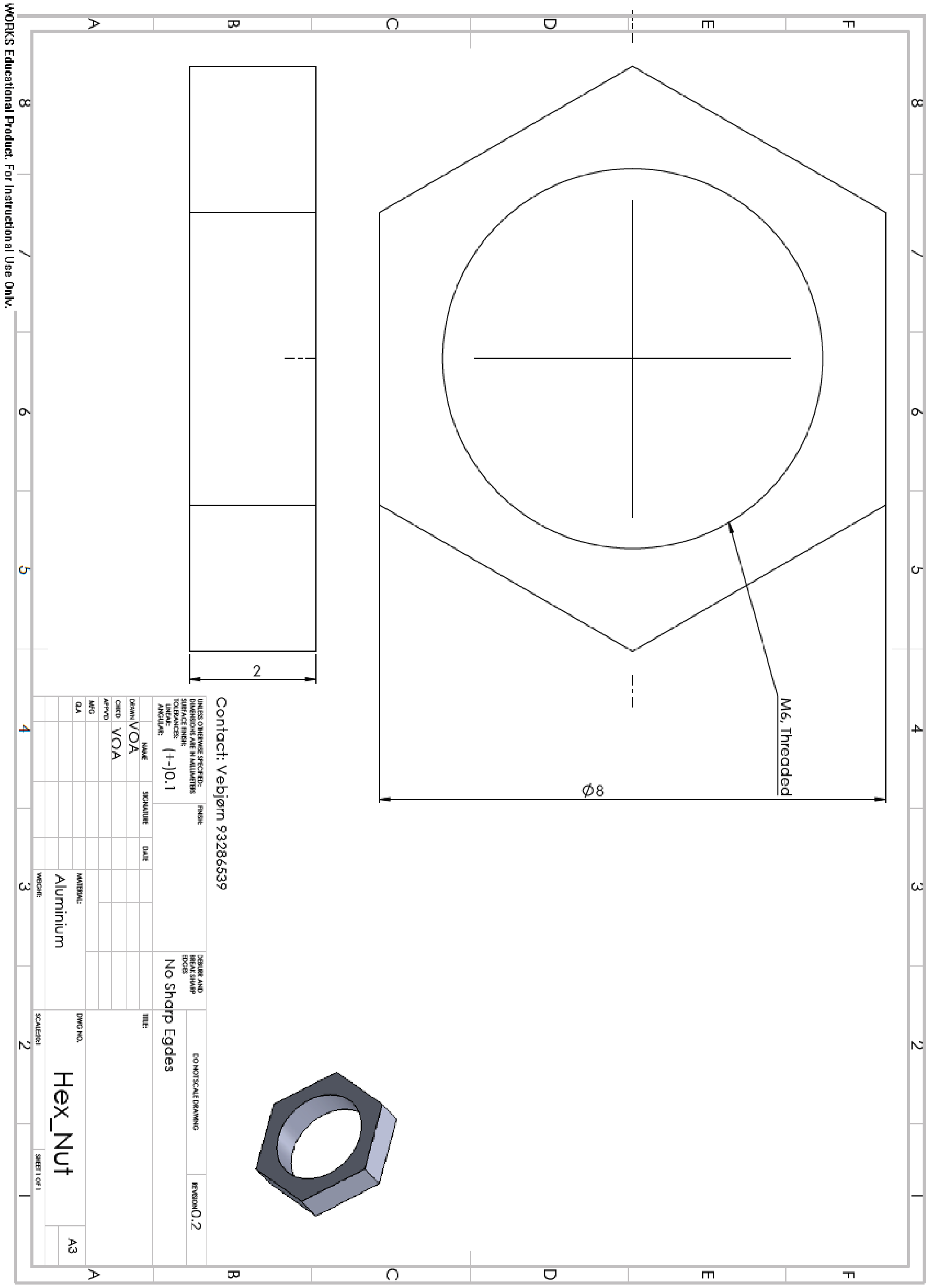
Figure 53: SSM bearing setup.

8.2. Motor code

```
 const int CCW=9;
 const int CW=8;
 const int trigger=10;
 const int hall1=A3;
 const int hall2=A2;

 void setup() {
  pinMode(CCW,OUTPUT);
  pinMode(CW,OUTPUT);
  pinMode(trigger,INPUT);
  pinMode(hall1,INPUT);
  pinMode(hall2,INPUT);

 }

 void loop() {

if(((analogRead(hall1))< 50) && (analogRead(hall2)>300) && (digitalRead(trigger)== HIGH)){
 digitalWrite(CCW,LOW);
 digitalWrite(CW,HIGH);
}
if(((analogRead(hall1))> 300) && (analogRead(hall2)<50) && ((digitalRead(trigger)== HIGH)){
 digitalWrite(CW,LOW);
 digitalWrite(CCW,HIGH);
}
stop_1();
 }

 void stop_1 (){
 if ((digitalRead(trigger)== LOW)) {
   digitalWrite(CW, LOW);
   digitalWrite(CCW, LOW);
 }
 }
```

8.3. 2D Drawings





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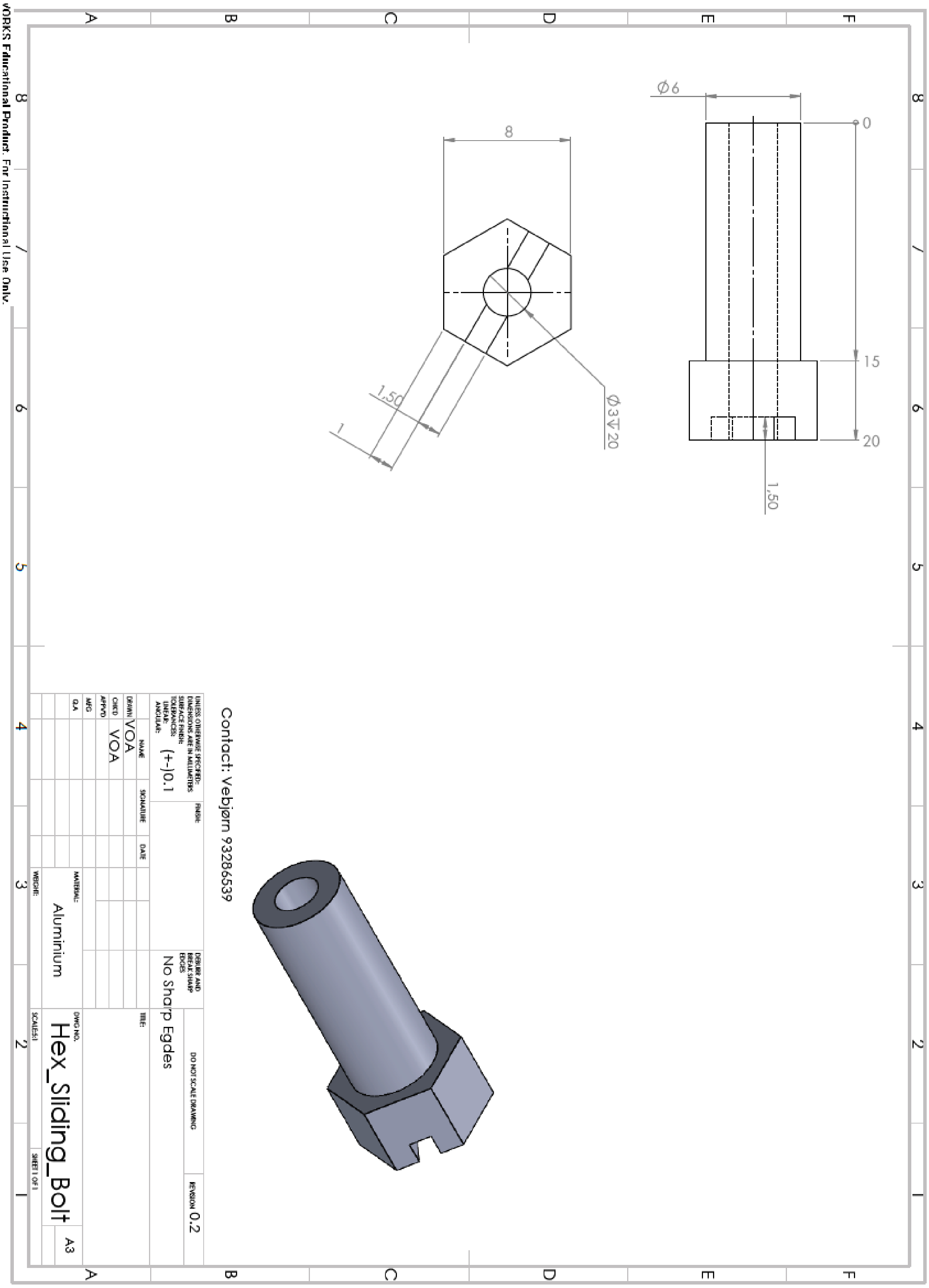
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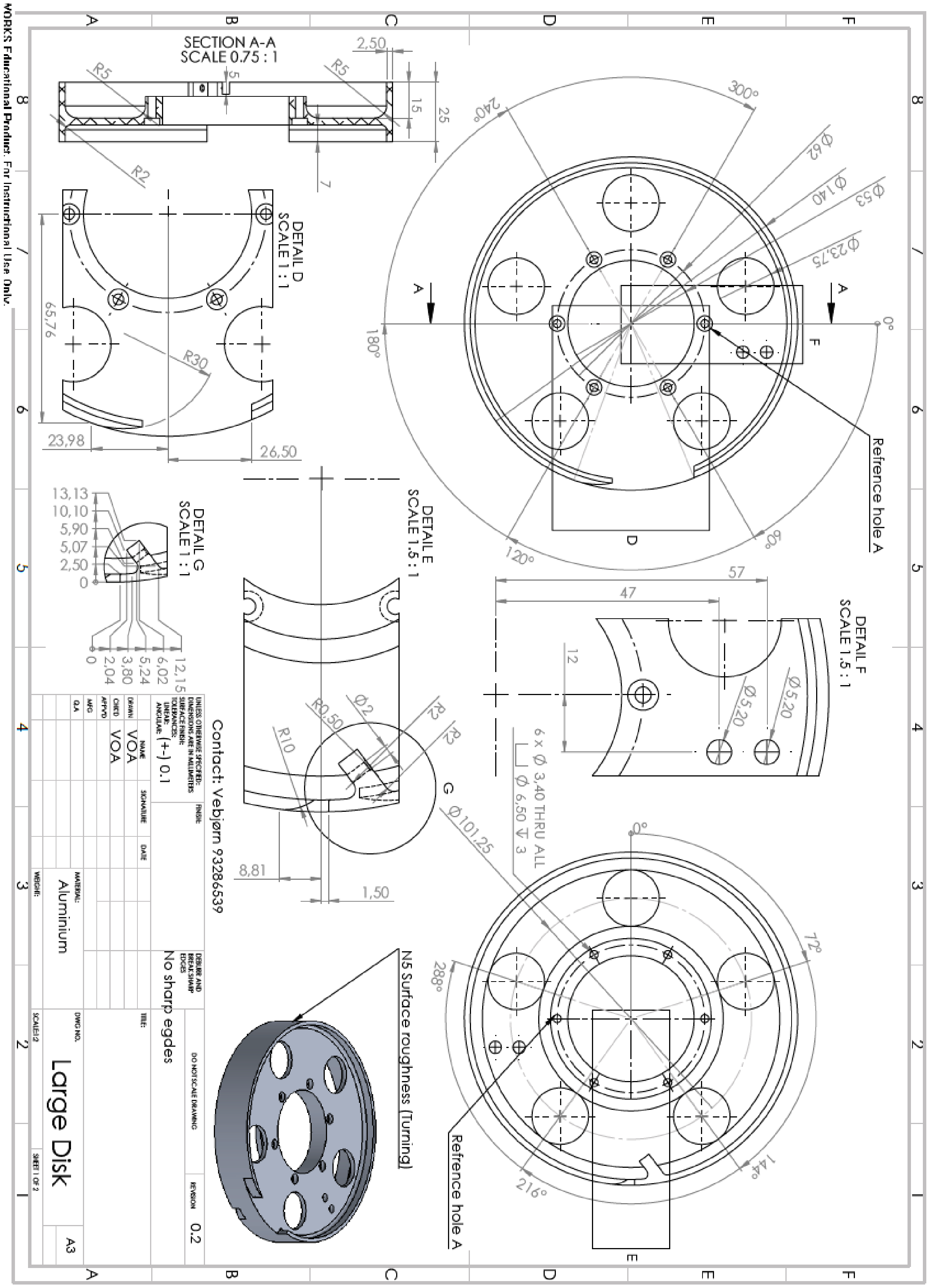
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MATERIAL: Aluminium
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No sharp egdes
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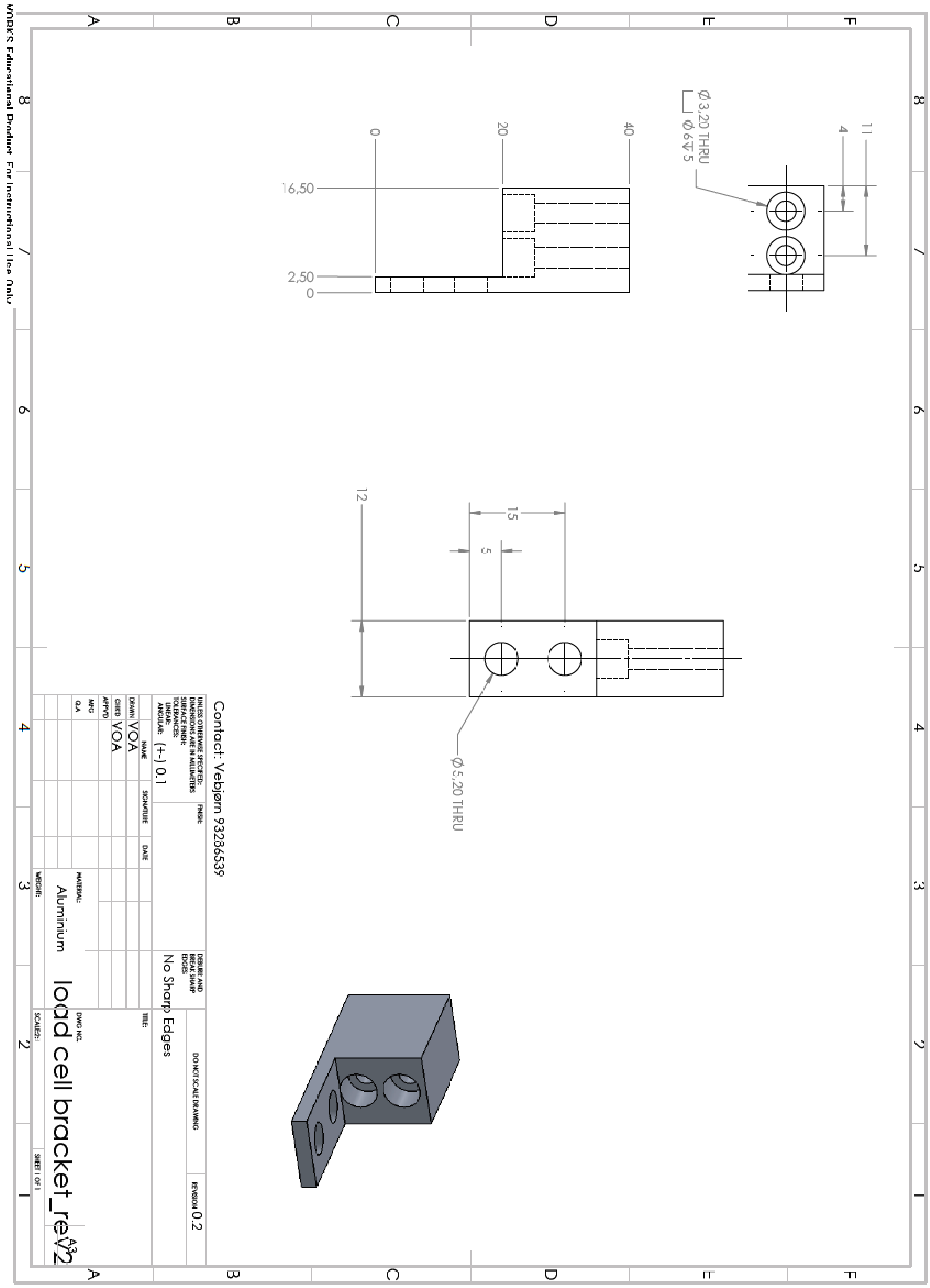
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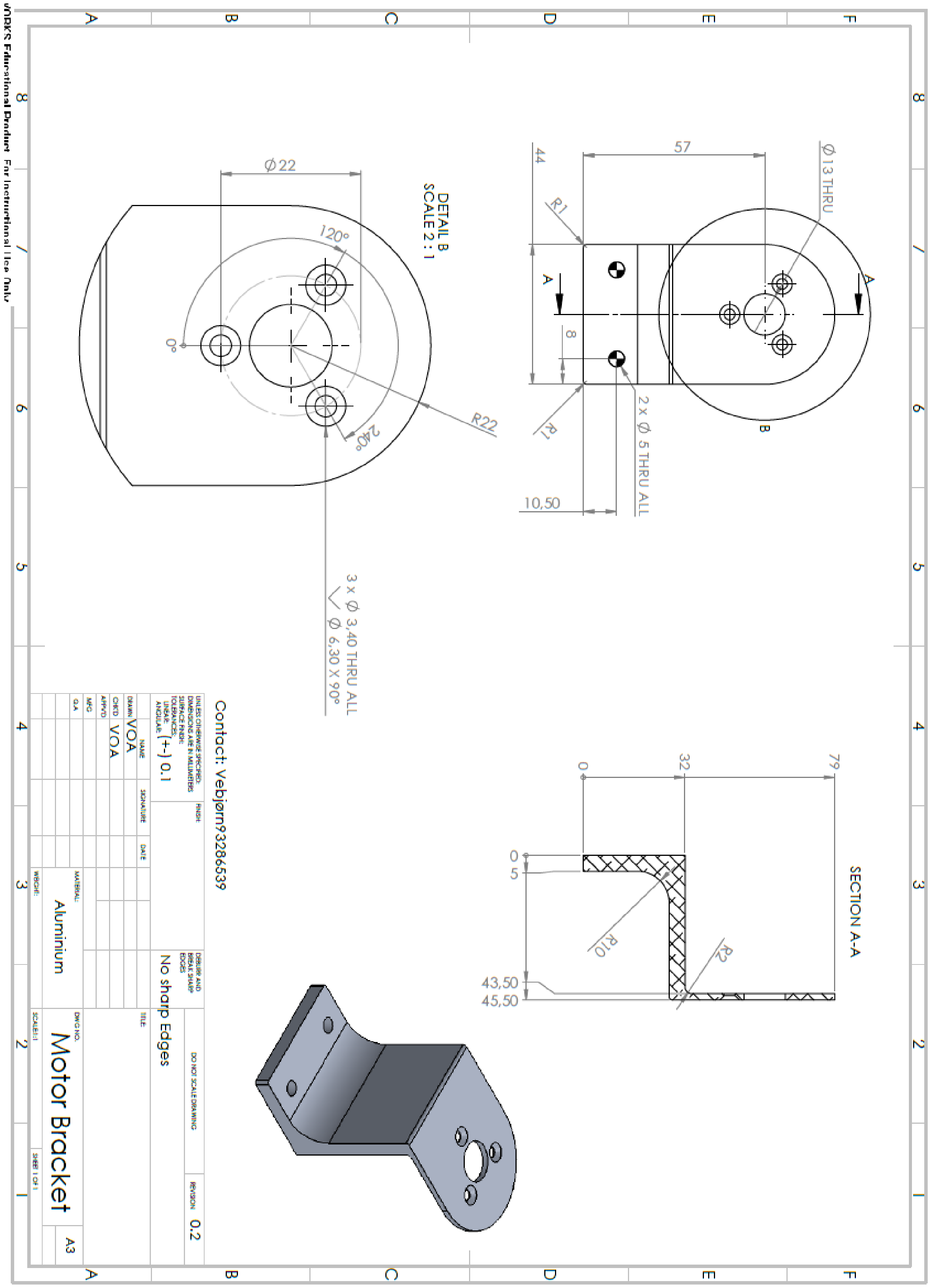
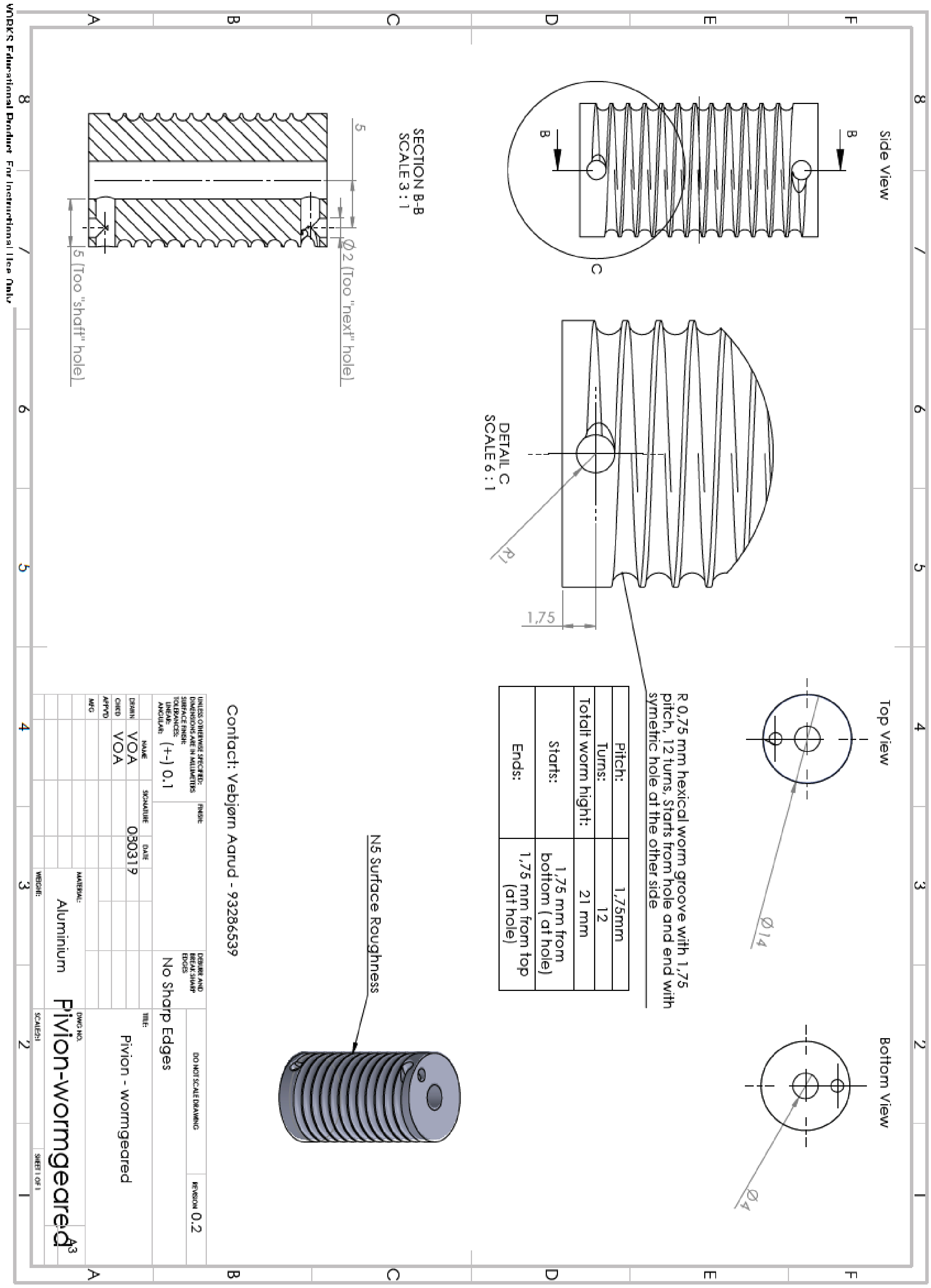
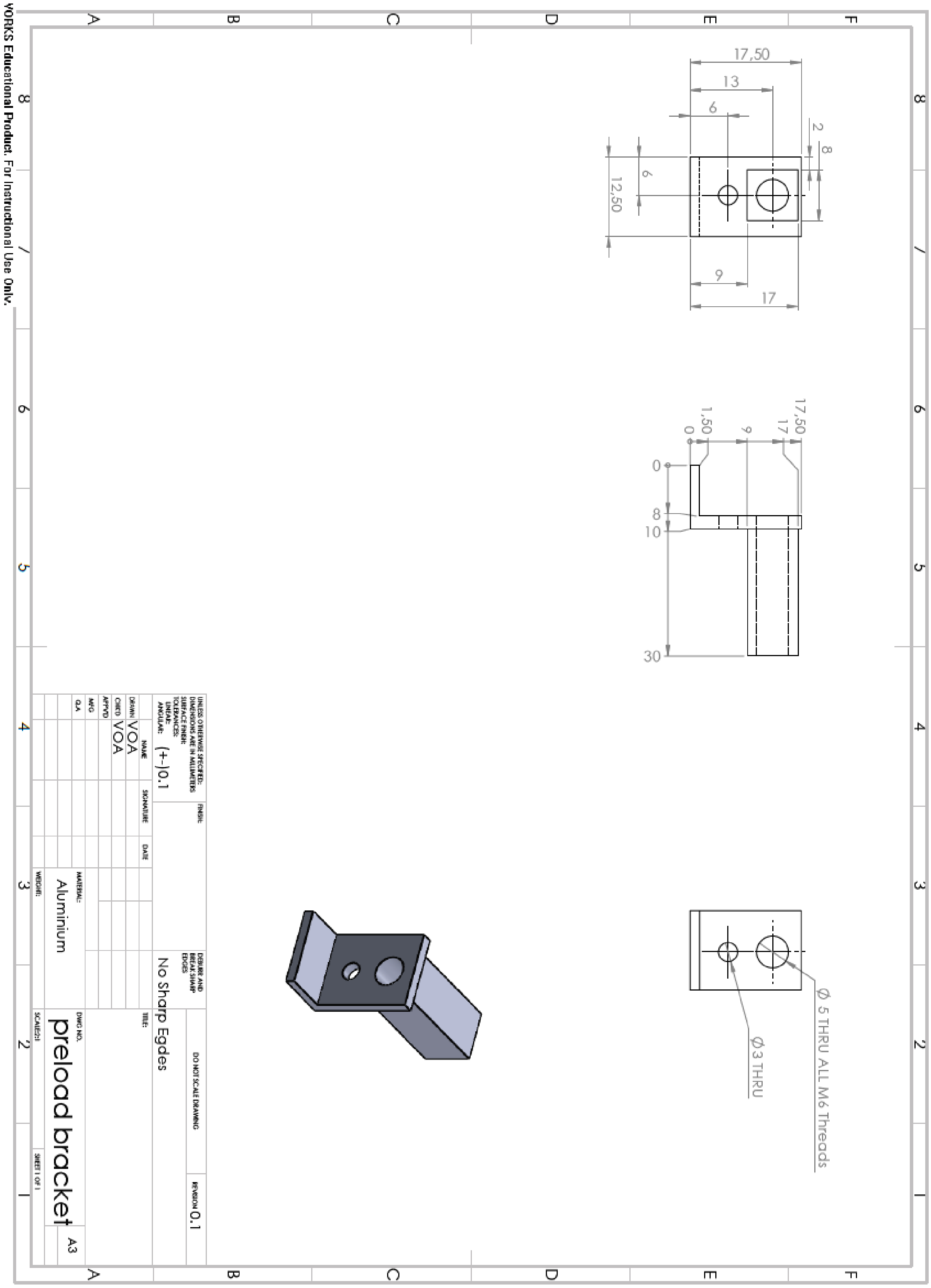
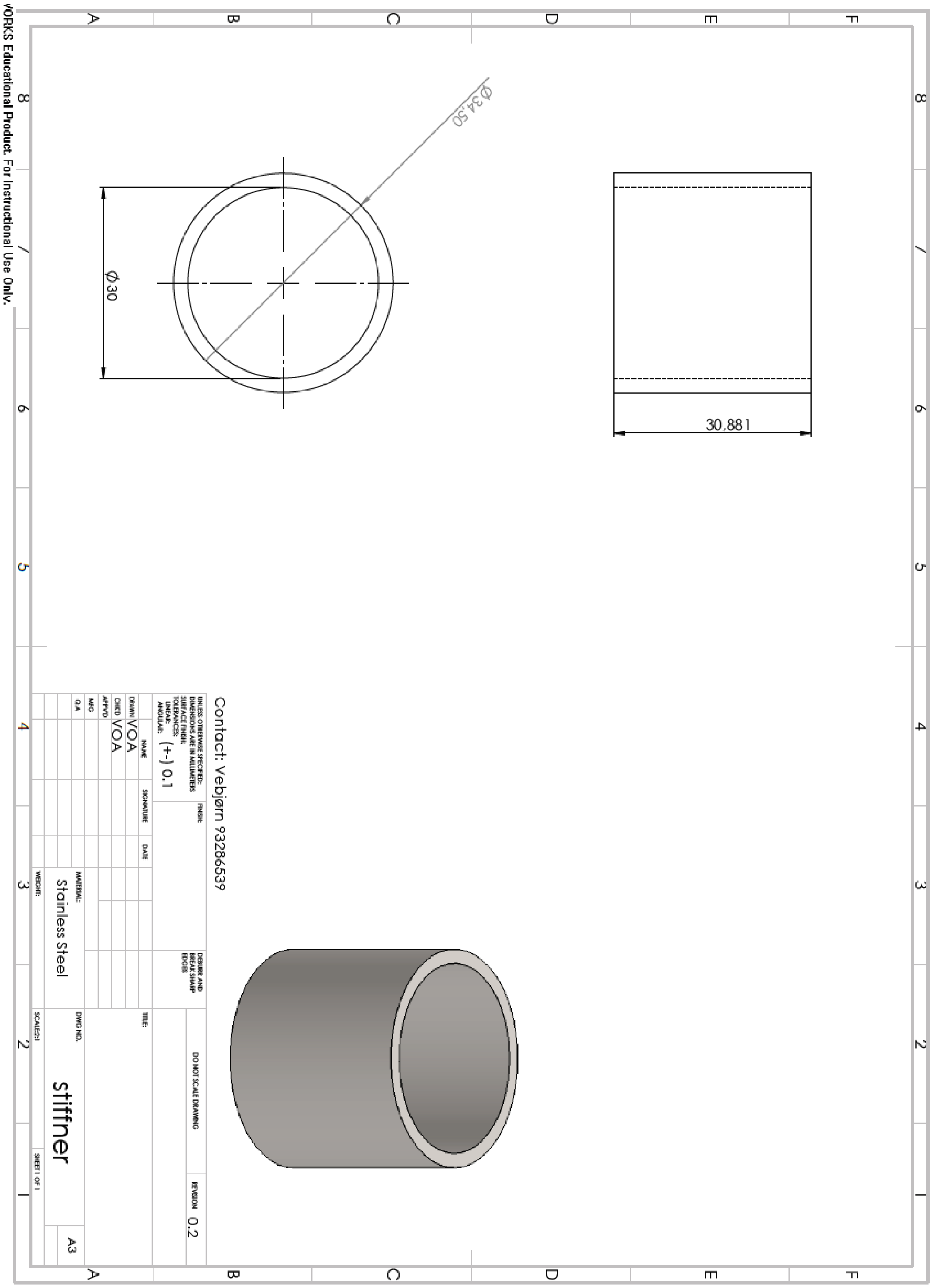
VOA
MATERIAL:
Aluminium
load cell bracket_rev2
SHEET 1 OF 1

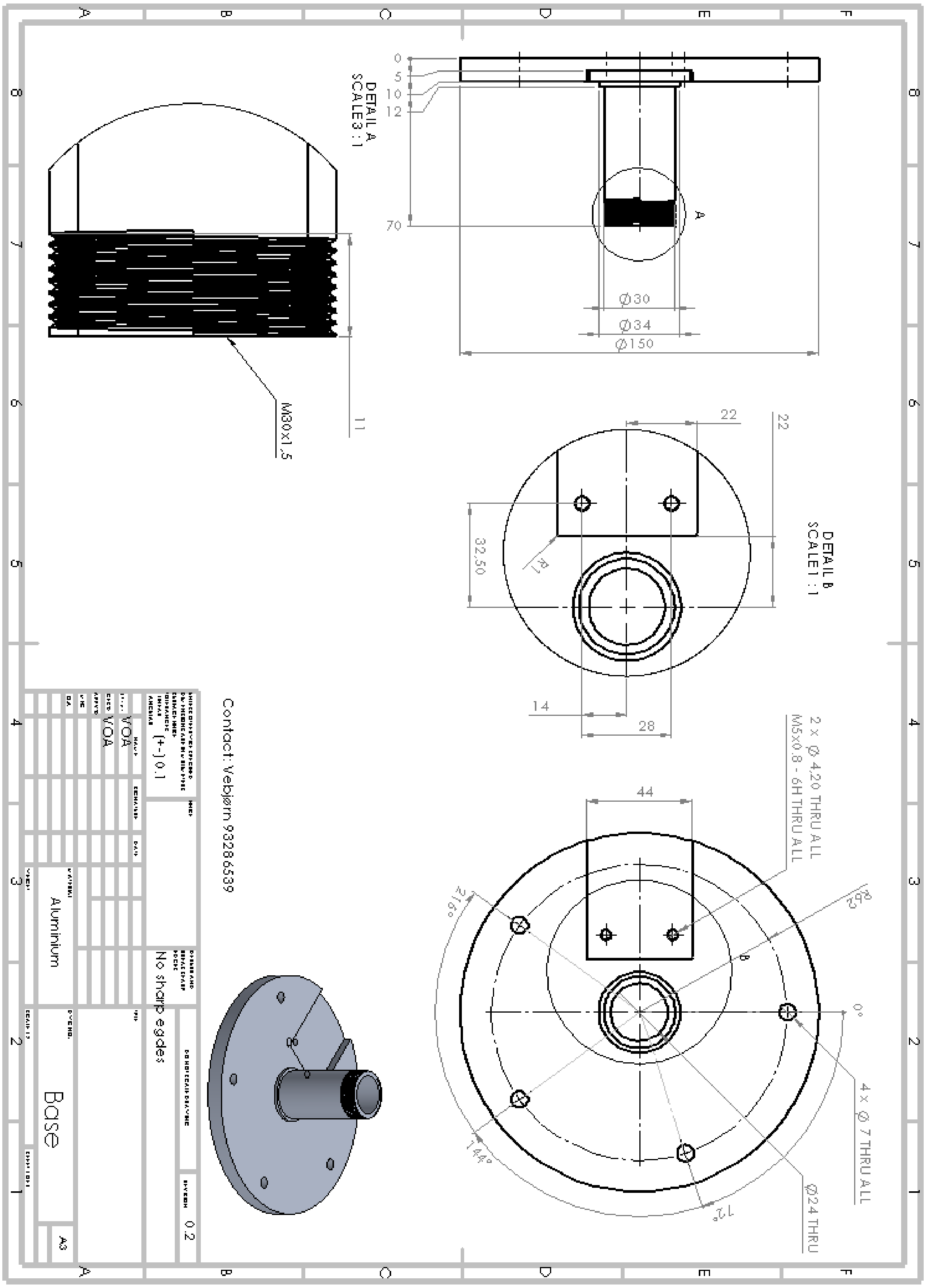

DETAIL B
SCALE 2 : 1
Ø22
120°
0°
240°
R22
3 x Ø 3,40 THRU ALL
⌵ Ø 6,30 X 90°
57
44
Ø13 THRU
R1
A
A
8
R1
2 x Ø 5 THRU ALL
10,50
B
SECTION A-A
0
32
79
0
5
43,50
45,50
R10
R2
Contact: Vebjørn93286539
(+-) 0.1
VOA
VOA
No sharp Edges
DO NOT SCALE DRAWING
0.2
Aluminium
Motor Bracket
A3

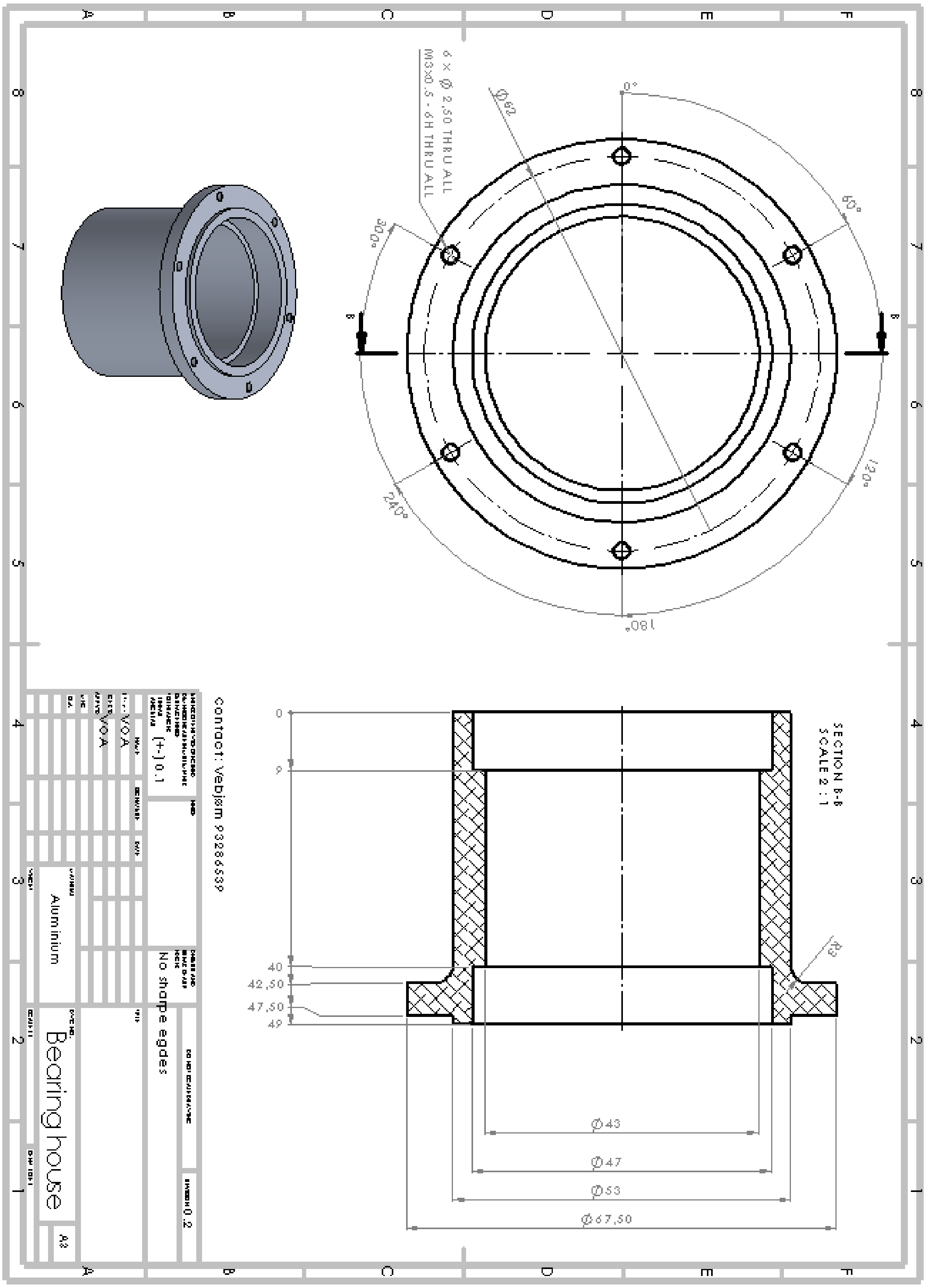

Side View
Top View
Bottom View
SECTION B-B
SCALE 3 : 1
DETAIL C
SCALE 6 : 1
Ø2 (Too "next" hole)
5 (Too "shaft" hole)
1,75
R1
Ø14
Ø4
R 0,75 mm hexical worm groove with 1,75 pitch, 12 turns. Starts from hole and end with symetric hole at the other side
Pitch: 1,75mm
Turns: 12
Totalt worm hight: 21 mm
Starts: 1,75 mm from bottom (at hole)
Ends: 1,75 mm from top (at hole)
N5 Surface Roughness
Contact: Vebjørn Aarud - 93286539
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS
TOLERANCES: (+-) 0.1
DRAWN VOA 030319
CHK'D VOA
No Sharp Edges
DO NOT SCALE DRAWING
REVISION 0.2
TITLE: Pivion - wormgeared
MATERIAL: Aluminium
DWG NO. Pivion-wormgeared
A3



VORKS Educational Product. For Instructional Use Only.
17,50
13
6
2
8
6
12,50
9
17
1,50
0
9
17
17,50
0
8
10
30
Ø 5 THRU ALL M6 Threads
Ø3 THRU
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
(+-)0.1
FINISH:
DEBURR AND
BREAK SHARP
EDGES
DO NOT SCALE DRAWING
REVISION 0.1
NAME
SIGNATURE
DATE
DRAWN VOA
CHK'D VOA
APPV'D
MFG
Q.A
MATERIAL:
Aluminium
TITLE:
No Sharp Egdes
DWG NO.
preload bracket
A3
WEIGHT:
SCALE:2:1
SHEET 1 OF 1


ORKS Educational Product. For Instructional Use Only.
Ø34,50
Ø30
30,881
Contact: Vebjørn 93286539
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
(+-) 0.1
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
0.2
NAME
SIGNATURE
DATE
DRAWN
VOA
CHK'D
VOA
APPV'D
MFG
Q.A
MATERIAL:
Stainless Steel
WEIGHT:
TITLE:
DWG NO.
stiffner
A3
SCALE:2:1
SHEET 1 OF 1


DETAIL A
SCALE 3 : 1
M30x1.5
Ø30
Ø34
Ø150
DETAIL B
SCALE 1 : 1
2 x Ø 4.20 THRU ALL
M5x0.8 - 6H THRU ALL
4 x Ø 7 THRU ALL
Ø24 THRU
Contact: Vebjørn 93286539
No sharp egdes
Aluminium
Base
A3


6 × Ø 2,50 THRU ALL
M3x0.5 - 6H THRU ALL
Ø62
0°
60°
120°
180°
240°
300°
SECTION B-B
SCALE 2 : 1
0
9
40
42,50
47,50
49
R3
Ø43
Ø47
Ø53
Ø67,50
Contact: vebjsm 93286539
(+-) 0.1
VOA
VOA
No sharpe egdes
Aluminium
Bearing house
0.2
A3