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Precision Dynamometer Brake for Gear Testing

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1 Abstract

Dynamometer is a common type of device that measures the force, torque or power in a system. Current dynamometer brakes have the problem of generating unstable torque during operation. This project aims to design a new dynamometer brake that provides braking with low torque ripple. The final design utilized viscous shear braking which was proven by Atlas Copco experiments to be capable of providing stable torque. To fix design parameters, the team performed fluid dynamic calculations and heat transfer simulations. Detailed design drawings were created and a product prototype was manufactured and tested. Though the final prototype performance was affected by precision limits in manufacturing, the model met several design requirements and showed potential of stable torque generation.

2 Acknowledgement

We would like to express our gratitude to Mattias Andre from Atlas Copco, who was the industrial supervisor providing us with all the support we needed during the whole project. His guidance on manufacturing and assembling was significantly useful that helped this project come true.

I take this opportunity to express our profound and sincere gratitude, indebtedness and affinity to my respected guides Prof. Kjell Andersson and Prof. Stefan Björklund, for their professional guidance, valuable advises, constructive criticism and immense help throughout the course of the project work.

We also thank Jan Stamer from production department for the help he has provided us with the manufacturing phase of our project.

Last but no least, we thank to the effort from all the teammates, and the hard work from each of us which made the project come to an end successfully.

Above all, we thank God Almighty without whose blessings our efforts would not have been a reality.

3 Introduction

The project task is to design a brake that can provide the consistent and smooth torque needed for torque measurements in gear testing.

The idea started with an attempt to find a solution to reduce the torque ripples or the variation of torque within the braking system of a dynamometer. One previous prototype based on viscous shearing made at Atlas Copco has has confirmed to give less ripples than the ones on the market now. This motivated the current project, which aims to research into the probability of a fluid based braking device and to design and test it.

Several ideas were generated based on providing counter torque by means of viscous shear. Ideas were evaluated based the complexity of analysis via analytical or numerical methods.

The final design should meet the following requirements:

a) Torque should be adjustable between 5 and 30 Nm.

b) Torque should be stable during long periods of braking at constant speed.

c) Torque should contain an absolute minimum of ripple (max 1%).

d) Possibility for forced cooling and circulation of the oil.

After proper study of different concepts, one best design was fixed for detail design and and simulation. The design was then developed more on different criterion's and calculations for the best results. A prototype was manufactured, assembled and tested at Atlas Copco.

The results from the design was compared with the expected results obtained through calculation. Conclusions were made on this in the final stage and also future works for the better results were suggested.

4 Project Planning

4.1 Scope

a) Concept development;

b) Detailed design of the components, selection of the materials and working fluid;

c) Manufacturing and assembling the prototype brake;

d) Basic functional tests and performance measurements.

e) Comparison of ripples from measurements from the prototype with conventional ripple measurements.

4.2 Requirement Specifications

4.2.1 Performance

a) Torque should be adjustable between 5 and 30Nm (desirable up to 50Nm);

b) Torque should be stable during long periods of braking at constant speed of 60 rpm;

c) Torque should contain an absolute minimum of noise/ripple (max 1%);

d) Possibility for forced cooling and circulation of the oil;

e) Cleaning, sealing and turbulence are also some factors to be considered.

4.2.2 Environment

a) Noise level: below 80-85 db. This is desired for the safe working environment of the personnel;

b) No corrosion from the working fluid. Desired to avoid any form of degradation of the materials used;

c) Environment friendly components. Desired to avoid dangerous products and risks following them;

d) Personnel Safety;

e) Use of non-hazardous materials. Desired for the health of personnel in contact with the prototype.

f) Avoid spillage of the working fluid used in the design.

4.2.3 Target Product Cost

Since the project aims at testing of a working prototype, cost related topics are discussed later during the mass production.

4.2.4 Testing

a) Testing carried out at the Atlas Copco in Sickla, Stockholm.

b) Torque should be adjustable between 5 and 30 Nm (desirable up to 50 Nm);

c) Torque should contain an absolute minimum of noise/ripple (max 1%).

4.2.5 Safety

a) Should comply with EU workplace standards on Physical, chemical and electrical safety;

b) Personnel safety should be taken into account;

c) Use of non-hazardous materials.

4.2.6 Legal

The product should meet the Consumer Protection Policy.

4.2.7 Documentation

a) Work Breakdown Structure (WBS);

b) Gantt Chart;

c) Requirement Specification Table;

d) Weekly and Termly Presentations;

e) Drawings and CAD models;

f) Prototype.

4.2.8 Materials

a) Materials selected were mostly aluminium and cutting steel. The oil selected for the design was Silicon oil of viscosity 3000 cps;

b) Non-hazardous and should comply with EU environmental standards.

4.2.9 Aesthetic and Appearance

Austere and Ergonomic design.

4.2.10 Installation

a) Easy installation;

b) Good modularity;

c) Effortless changing of working fluid.

4.2.11 Life in service

Assume 12 hours per day, 200 days per year, 2 year.

4.2.12 Time scale

a) Concept generation, final concept and final design by May 16;

b) Manufacturing and product development by December.

4.2.13 Processes

a) Will decide after the detailed design stage;

b) Size and weight;

c) Robust and compatible;

d) Good mobility and portability.

4.2.14 Company constraints

a) Torque ripple less than one percent;

b) Capable to measure torque in 5-30 Nm range;

c) Constant torque measurement with time.

4.2.15 Maintenance

Easy Maintenance.

4.2.16 Patent

Consider Patents with similar solutions and ideas.

4.3 GANTT-Chart

GANTT chart is a bar chart that defines a project schedule from start to finish. Tasks from the network diagram are taken and presented in a time schedule. This helps in the streamlined functioning of the project In the current project concept generation and evaluation is of primary importance and hence more time is allotted. Evaluation techniques like fluid and structural analysis is time consuming hence one month is set as its duration.

The chart that was used for the concept generation and evaluation phase is shown below in Figure 1.

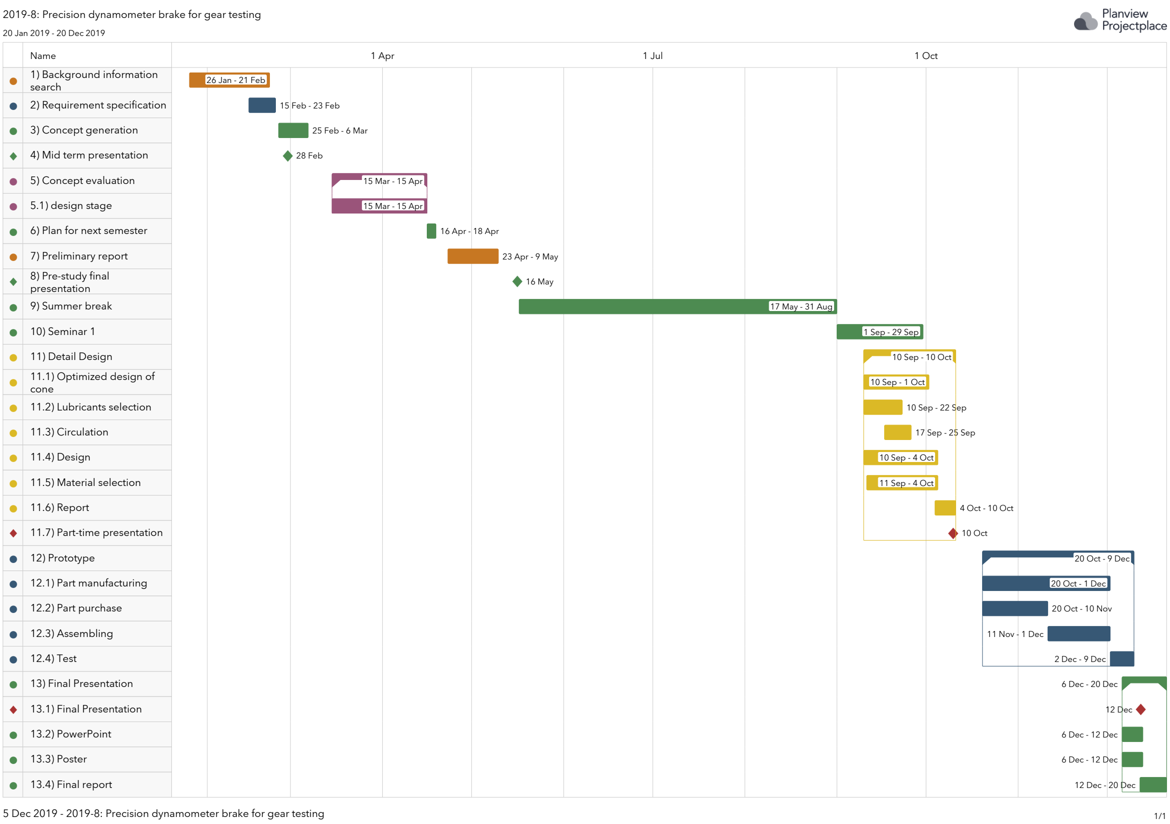


Figure 1: Gannt chart of the whole project

4.4 Work Breakdown

"A Work Breakdown Structure (WBS) is a hierarchical structure of things that the project will make or outcomes that it will deliver."

In other words, the Work Breakdown Structure defines the "what" of the project. Everything you need to accomplish in the project is displayed in a single, easy to understand chart. The purpose of this chart is to break down complex activities into smaller, more management constituents.

In our project the total work is separated into 5 sub-divisions:

a) Project management: Project decomposition and scheduling,

b) Concept selection: Concept generation and evaluation,

c) Design Phase: prototype design followed by detailed component design,

d) Product manufacturing: Cost assessment, manufacturing and assembly,

e) Product testing: Analysis of test methods, Testing and compilation of results.

The work breakdown structure used for the project has been shown below:

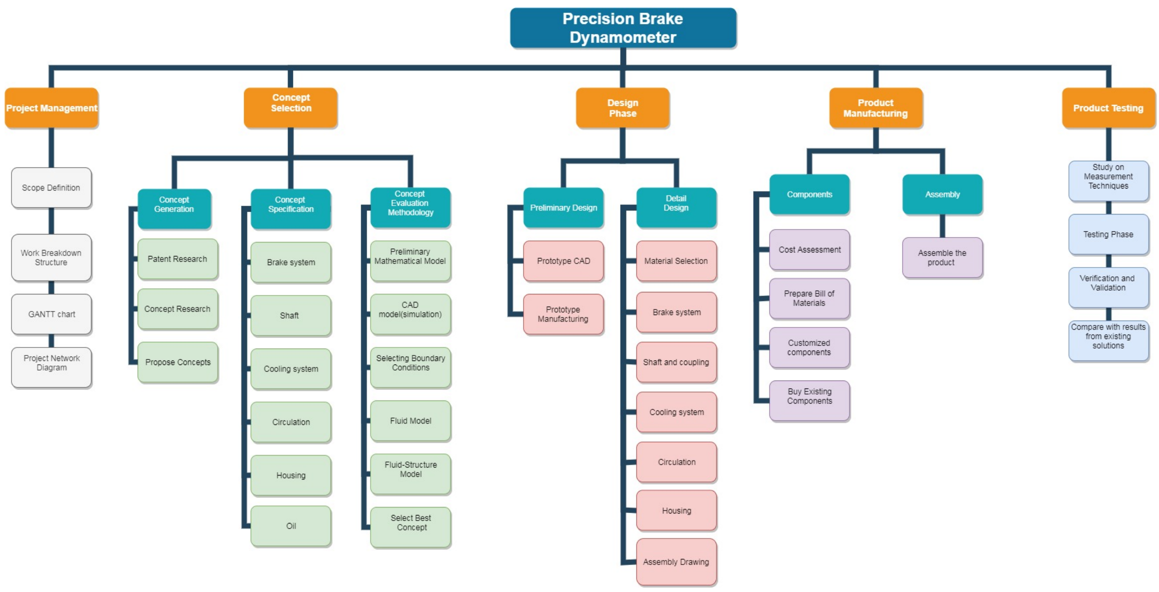


Figure 2: Work breakdown structure

5 Background

5.1 Existing Designs

The project started with the study of the model and results achieved from the existing design. The design consisted of a shaft and some discs which were attached to the shaft as shown in the figure. The discs with the shaft was immersed in an oil bath as seen. The shaft was first rotated using a nut runner with the help of two flexible couplings and a transducer. The measurements from the transducer showed better results on ripples as compared to the ripples found from the dynamometers in the present market. The setup was then changed from the nut runner to a flywheel with a fish string attached to pulleys and weight to provide a smooth input torque. This provided even more better results than expected. This showed the expectations and reach of the project for effective results if a proper design with calculations were done using the principle of viscous shear as the mode of braking.

5.2 Oil Shear Theory

Oil shear theory is based on newton's law of viscosity. It states that the shear stress between the adjacent fluid layers is proportional to the velocity gradient between the two layers. The equation below shows the relation where the parameters are shear stress , dynamic viscosity and the rate of shear deformation.

$\tau = \mu \frac{du}{dy}$

6 Conceptual Design

In this section, five different system designs are discussed. Concept 1 to 4 are all based on the principle of shear friction between discs while concept 5 utilizes the friction between static housing and rotating shaft. Torque is adjusted through the velocity difference between static discs and rotating ones in concept 1. Meanwhile, concept 2 and 3 varies the distance between discs with different types of machinery to change the torque. Concept 4 on the other hand controls the temperature so that the viscosity and torque could be changed. All the system concepts will be introduced in detail below.

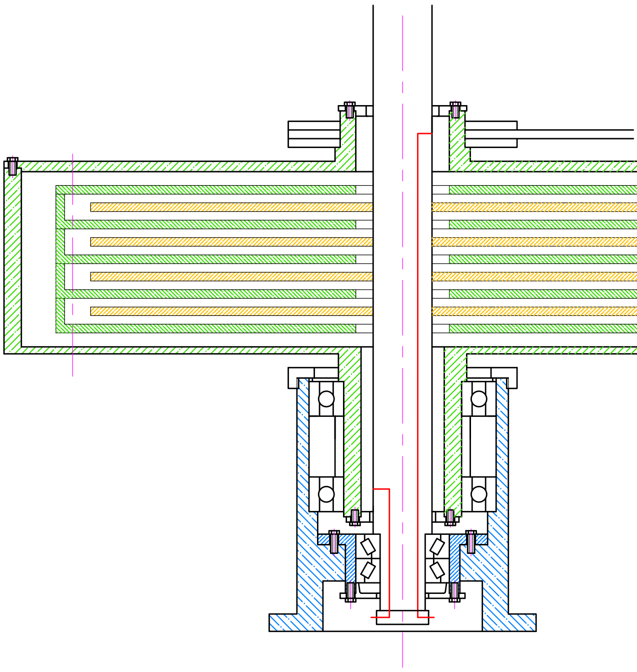


Figure 3: Conceptual concept 1 - changing velocity difference by rotating housing

6.1 Concept 1 - Changing velocity difference by rotating housing

6.1.1 Description

In this idea shown in Figure 3, there is a housing attached to a motor or belt transmission and it is free to rotate about its axis. The housing-disks are fixed to the housing using splines and allow it to rotate with the housing. The extended portion of the fixed disks are arranged axially in-order to maintain equal gap between the disks. The rotating disks are attached to the shaft using splines. These are also designed in similar fashion in order to prevent axial displacement and maintain equal film thickness between the disks.

The challenge in circulation is dealt by boring two channels in the shaft and using a rotary unit at the shaft end. The lubricant is pumped from the reservoir and is directed by the rotary unit to the inlet lubrication channel in the shaft. The lubricant enters near the top end of the housing and comes out through the bottom and into a second channel which is attached to the rotary flange and then pumped back to the reservoir.

6.1.2 Individual Evaluation

Torque variations can occur due to eccentricities in the shaft and rotating disks attached on the shaft. This leads to unbalanced vibrations in the longitudinal direction. In order to tackle this problem, we need to dampen the vibrations with the help of a highly viscous oil. In this case, the disks will have a varying angle of misalignment and in order to prevent this, hydrodynamic pressure should be built up in the region of minimum film thickness. But since the housing also rotates, this causes vibrations that can prevent the built-up of hydrodynamic pressure in the region.

In conclusion, there are such advantages as below:

a) Linear variation of torque achieved. Hence no need of extra calibration;

b) Rapid change in torque can be achieved.

Meanwhile, the disadvantages are:

a) It is a large system;

b) It does not have good self-aligning property. Hence requires proper centering in manufacturing;

c) Vibration occurring due to the rotation of the housing;

d) Ripple from the motor used to turn the housing and hence may not supply a constant torque;

e) The circulation in this system could be complicated.

6.2 Concept 2 - Changing film thickness with gear

6.2.1 Description

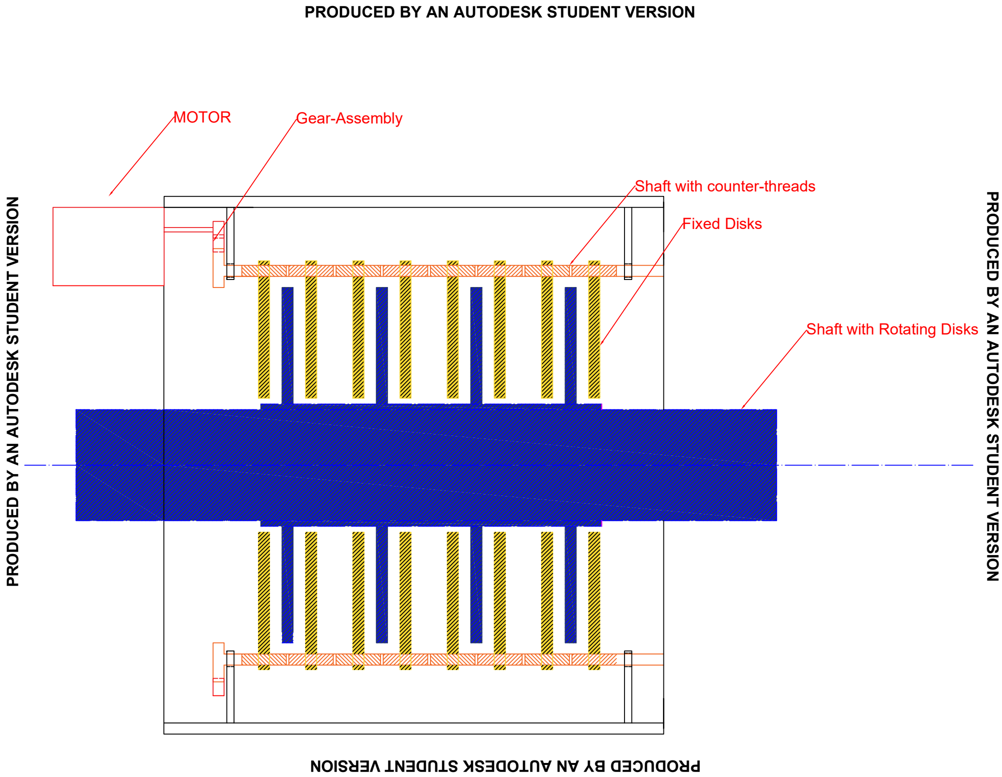


Figure 4: Conceptual concept 2 - changing film thickness with gear

For this concept (Figure 4) explore the possibilities of changing torque of the system by adjusting the film thickness. This is done by moving the rotating disks closer to the fixed ones and thereby increasing the shear rate of the working fluid.

In this idea we use the fixed disks attached to a shaft with counter threads at alternate intervals. This shaft is attached to the housing and is free to rotate. The fixed disks have threaded holes which mesh with the threads of the threaded shaft. Once the shaft is rotated, depending on the pitch of the thread and the rotation, the fixed disks move towards the rotating ones and thereby reducing the film thickness and increasing the torque.

In order to eliminate the use of multiple motors, a gear assembly is set up which theoretically moves entire disks, axially. Here input from a shaft is given to the input gear which rotates the ring gear, which in turn rotates the four output gears attached to the four threaded shafts.

6.2.2 Individual Evaluation

For circulation, since we have a fixed housing, we pump the liquid into the housing directly without the need for any external devices like rotary flange. This makes this idea easier to implement when compared to the previous ones.

Torque variation can occur due to eccentricities in the rotating disks. This can occur because of manufacturing errors. This causes a vibration which results in varying angles of misalignment. In case of such a vibration the hydrodynamic pressure built-up is limited to a single wedge region. Eccentricities for each disk must be balanced in order to prevent ripple.

Torque variation in this case is by moving the input gear by an angle -so a larger input gear is preferred for easiness to control the angle. The angle of rotation is linked to the film thickness between the disks. Since multiple gears are meshing, there can be meshing errors like backlash which can affect the torque calibration.

In conclusion, there are such advantages as below:

a) Simple concept as no complicated circulation setup is required because of rotating housing.

b) Precise lead screws are available to control motion.

c) Rapid change in torque can be achieved.

Meanwhile, the disadvantages are:

a) Lesser number of shear pairs per length of the shaft within the housing when compared to the idea with rotating housing

b) Manufacturing errors can cause backlash in the gears leading to misalignment of the fixed disks. This can lead to extra forces and an unpredictable torque.

c) Due to large contact area in the threads, it can cause abrasive wear which can affect the linear travel. Constant lubrication is required

d) Non-linear torque variation with a sharp torque drop.

6.3 Concept 3 - Changing film thickness by varying control pressure

6.3.1 Description

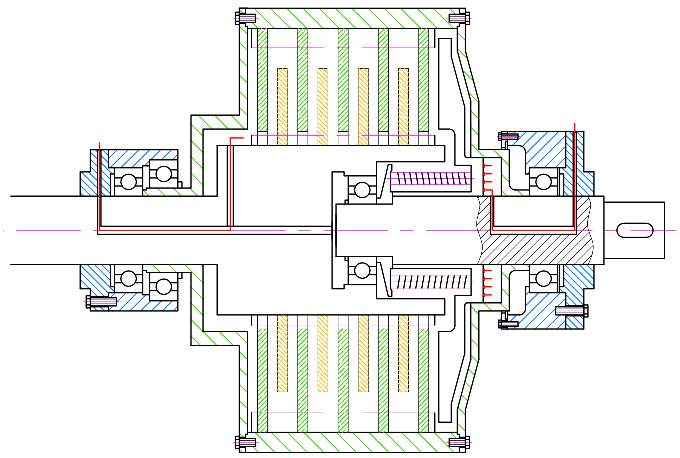


Figure 5: Conceptual concept 3 - changing film thickness by varying control pressure

For this concept (Figure 5) explore the possibilities of changing torque of the system by adjusting the film thickness. This is done by moving the rotating disks closer to the fixed ones and thereby increasing the shear rate of the working fluid.

In this idea, the fixed discs are mounted at the housing with spline and the rotating connect to the shaft with spline too. Therefore, all the discs can move in the axial direction. The loads applied to the piston are oil pressure and spring force. The system oil is pumped into the system through the left inlet, which causes the shear force.

The control oil enters the system through the right inlet so that the piston is pressed to the left. As a result, the discs would move together and the oil film thickness decreases, so the torque goes down.

If the pressure of control oil declines, the piston is move backward. Therefore, the oil film thickness would increase and torque goes up.

6.3.2 Individual Evaluation

The design concept is more complex than all the other ideas, which involved more parts and components with limited tolerance. All kinds of manufacturing tolerance can cause transmission error and vibration. If the shaft components can not be installed well, the system would have strong vibration, which causes torque ripple.

For circulation, since we have a fixed housing, we pump the liquid into the housing directly without the need for any external devices like rotary unit. But there are two different inlet in this system, in other word, two circulation circuits are needed, including 2 pumps and 2 valves.

Torque ripple can occur due to manufacturing and assembly errors. These error causes not only vibration but also misalignment of the shaft and discs.

In conclusion, there are such advantages as below:

a) It is easy to control the pressure;

b) The torque adjustment can be accuracy due to high-accurate valves.

Meanwhile, the disadvantages are:

a) It is a more complex system, even the most complex, in which there are more parts and components;

b) The parts need specific manufacture finishing;

c) The torque adjustment is not sensitive to the pressure, which means when the pressure changes a lot but the torque may not change that much.

6.4 Concept 4 - controlling temperature

6.4.1 Description

As shown in Figure 6, this was one of the easiest model concepts that was developed. Even though it is smaller and easier in design the method of achieving the result we are looking for in the project seem to be quite hard compared with the other concepts or ideas. This has to deal a lot more with the control theory than with the machine design and is mostly out of our scope. Research was done on the idea regarding the temperature build up inside the brake system and tried doing analysis in Ansys software to see how hard it is to achieve the result that is needed. It was not able to find much of the results because we think that this field is too vague and almost impossible to achieve. The insulation methods in this concept also seemed to be very complex since this only depends on the varying the temperature to control the torque in the system. Other disadvantage that we found from this idea was that it needs more time

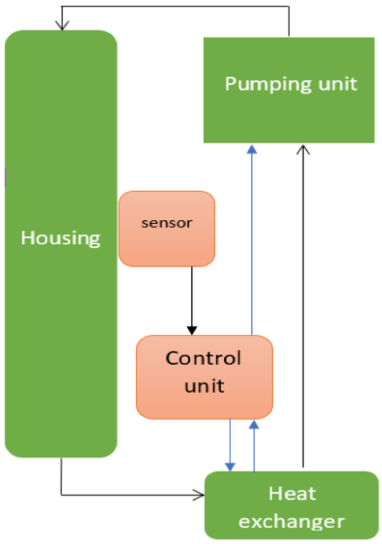


Figure 6: Conceptual concept 4 - controlling temperature

to achieve the torque needed than the other ideas because the time needed to build up the temperature using the heat exchangers are more in this case since this only depends on the variation in temperature. Some of the advantages found from this idea was that it is much simpler in the terms of manufacturing and working because it doesn't have any heavy components. We only have an extra heat exchanger compared to the prototype we had as a basis on which we worked with.

The system consists of a rotating shaft with discs attached to it similar to the prototype and a hosing where it will be placed in. The housing will also have discs attached to it which causes a shear when the system works and provide the brake in the dynamometer. The extra components included in the system are the sensors which will be connected inside the housing which will measure the temperature inside the system since the whole system works on the concept of controlling the temperature to vary the torque. The sensors measure the readings and sends the data to the control unit and then this data is used in the heat exchanger to vary the temperature to keep it constant using the flow through the heat exchanger into the system using a pumping unit.

The time required to change the constant temperature within the system is more in this case because of the time required to heat up the liquid. We also tried to plot a graph between the torque and temperature in the system and it was studied that we have an exponential decrease in the torque with respect to the temperature which is actually an advantage. The Reynolds equation was used here for the plot.

6.4.2 Individual Evaluation

The chances of torque ripple will be less in this case for the fact that the system consists of least number of components. So the chances of vibration are less in this case. The chances of ripples can be due to the fact that the discs used are here flat and hence the chances of misalignment are more.

In conclusion, there are such advantages as below:

a) It is almost the simplest model of all the ideas;

b) Less torque drop with respect to temperature.

Meanwhile, the disadvantages are:

a) Insulation methods can be difficult;

b) Rapid change in torque is not possible.

6.5 Concept 5 - rheometer

6.5.1 Description

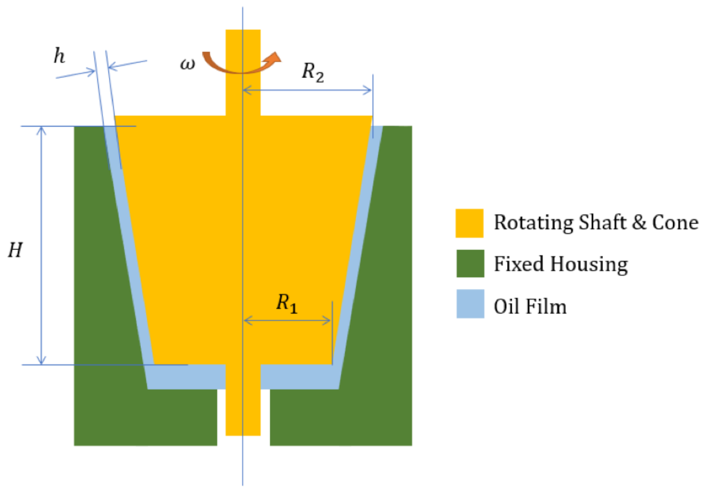


Figure 7: Conceptual concept 5 - rheometer

The idea was generated from the rheometer concept as shown in Figure 7. The design consists of a shaft and a cone attached together known as the rotor and a stationary housing. The very thin gap between the rotor and the housing is filled with high viscous oil which provides the braking for the system design. The range of torque is achieved by lifting the rotor from the bottom thereby varying the shear area in contact and the resulting torque in the system. The cone shape is proved to provide a range of torque with the least movement of the rotor. H shows the height of the cone in the design, h shows the film thickness between the rotor and the housing with a viscosity μ. R_1 shows the lower radius of the cone and R_2 shows the upper radius. The rotor is rotated at an angular velocity ω .

The torque formulation for this concept is derived as follows:

From the newton's law of viscosity:

$$\tau = \mu \frac{\partial u}{\partial y} = \frac{\mu \omega r}{h} \tag{1}$$

The shear force is found in terms of cone angle θ.

$$dF = \tau dA = \frac{\omega}{h}\left(2 \times \frac{dr}{sin(\theta)}\right) \tag{2}$$

The net torque is obtained by multiplying the equation above with r and then integrating.

$$T = \int_{R1}^{R2} \frac{\mu r \omega}{h} \frac{2\pi r}{sin(\theta)} r dr \tag{3}$$

For the dimensions given for the concept, angle θ can be defined as given below.

$$sin(\theta) = frac(R_2 - R_1)H \tag{4}$$

Integrating and simplifying the equation, we get:

$$T = \frac{\pi \mu \omega \left(R_2^2 + R_1^2\right)}{2h}(R_1 + R_2)\ \sqrt{H^2 + (R_2 - R_1)} \tag{5}$$

6.5.2 Individual Evaluation

The challenge in circulation is dealt by immersing the body in a fluid bath and this allows fluid flow into the cone when it lifts. The temperature inside can be maintained by precisely measuring the average temperature of the tank and cooling using a heat-exchanger.

In this idea torque ripple can occur due to improper centering of the shaft and the cone. This can cause dynamic variation in misalignment angle, like the vibration behavior in other ideas. But due to the geometry, this helps in hydrodynamic pressure built-up in more regions when compared to other ideas. (see fig 15). But in this case, there is possibility of ripple due to oil whirl and whip. Proper design needs to be done to avoid such variations in torque.

In conclusion, there are such advantages as below:

a) It is a simple and moderately sized system;

b) It has a better self-aligning property;

c) The torque-drop is less when compared to other height adjustment ideas;

d) Rapid change in torque can be achieved.

Meanwhile, the disadvantages are:

a) Torque ripple can occur due to oil-whirl and oil-whip;

b) Torque inversely proportional to lift height.

6.6 Concepts Comparison

After all the concepts were generated and successfully studied, they were all put together into the Pugh's evaluation chart and graded to get the best concept from the five of the ideas. The grading scale was taken from 1 to 5 with 1 the poor, 3 the average and 5 the best. The design requirements were chosen as the parameters to grade each idea which are:

a) Size;

b) Complexity;

c) Ease in Torque variation;

d) Less Torque ripples in the system;.

6.6.1 Size

Concept 1 was given a grade of 1 since it has the most number of components among all the ideas generated.

Concept 2 and 3 were graded 2 since they are a little bit better than idea 1.1 in terms of size.

Concept 4 has the smallest size since it has the least number of components in the system and therefore a grade of 4 was given.

Concept 5 got a grade of 3 since the size stays between the first three ideas and the forth one.

Grading the complexity was also similar to the size factor where the number of components and the ease of installation and less maintenance were considered.

		Concepts				
	Weight	**1**	**2**	**3**	**4**	**5**
Size	3	1	2	2	4	3
Complexity	4	2	3	1	3	4
Torque Control	4	5	2	2	2	4
Less torque ripple	5	2	3	3	4	4
Sum without weight		10	10	8	13	15
Sum with weight		**41**	**41**	**33**	**52**	**61**

Table 1: Pugh's Evaluation chart

6.6.2 Ease in torque variation

This factor shows the easiness or how quick you can change the torque value from one to a new value. Since the first method were the housing was rotated along with the shaft showed a linear relationship for the torque and the speed it was graded the maximum which is 5.

The next two methods show an inverse relation for the torque hence the grade 2 and the fourth method which only depends on the temperature proved that it takes time to change the temperature of the working fluid and this will in turn slow down the easiness to achieve a new torque. The last method or the rheometer one even though didn't show a linear relation it was found that by adjusting the height of the cone we could achieve a good control on the torque.

6.6.3 Less torque ripples

This factor was too vague to grade so we tried to find other reasons that causes the ripples like the vibration in the system and the misalignment.

Since the first four ideas uses the flat discs, the chances of misalignment are more and the fifth idea where we use the cone has the property of self-alignment and thus gets the highest score there.

6.6.4 Pugh's Evaluation Chart

From all the studies and deep evaluation on all the ideas, a decision was made to fix on the idea 4 which is the rheometer concept of using a cone and housing and immersing them whole in the oil bath. This idea has proven to show the most efficiency in all the parameters we were designing for and also the Pugh's evaluation chart also showed a maximum score for this idea. The chances of torque ripples are believed to be the less in this case and also the torque can be rapidly changed by adjusting the height of the cone using the precision screws.

A good thorough study is to be done for the whole system design in the future for the manufacturing process.

7 Development of Detail Design

A final system concept and drawings are the end target of the detailed design phase. Once the concept of the conical rheometer is selected, several iterations of system concept are analysed for their feasibility. The following parameters are used to judge different concepts:

a) Manufacturing constraints: The system concept has to envision the simplest of manufacturing methods for most of the components, save for the conical surface which will require grinding for precise tolerances. Turning, Milling by Water-jet, 3-D printing are resources easily available within KTH. Grinding machines are not available within KTH and require assistance from Atlas Copco;

b) Cost: The system concept should have components of reasonable price;

c) Modularity: Standard parts should be used wherever possible to allow for easy replacement of parts;

d) Control of torque: It should be able to control the torque in a precise and accurate manner.This can be only achieved by the precise control of a lifting mechanism for the cone;

e) Easiness to assemble and disassemble: A quick assembly and disassembly process is useful to save time for testing.

7.1 Iterations

7.1.1 System Concept 1 - Hollowed Rotor

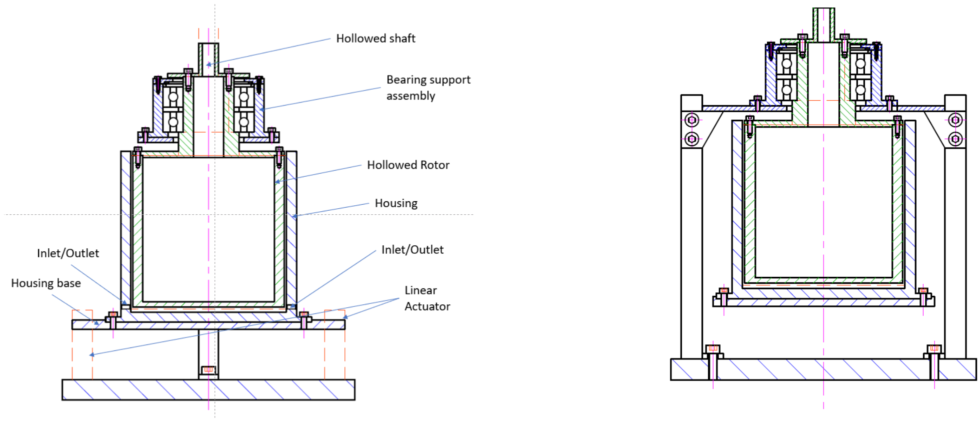
The main parts of the concept are rotor-shaft sub-assembly, housing, a support structure assembly that holds bearings and a linear actuator setup for changing the vertical position of the housing.

The linear actuator setup has to pull down the housing by 80 mm to achieve the full torque range, provided the film thickness of the lubricant is 15 microns. This is because, unlike a conical rotor, dropping the cylindrical housing doesn't decrease the film thickness.It only decreases the shear area of the lubricant. But it removes the non-linearity of torque vs lift height curve, making torque control easier.

a) The rotor-shaft assembly (in green hatches-see figure 8) is has three parts. The input shaft is attached to a part with bearing seats. This part is attached to the top of the hollowed cylindrical rotor.

b) The support structure (in narrow blue hatches-see figure 8) holds two ball bearings that help in taking radial loads and control misalignment of the rotor shaft. This support structure is attached to an outside structure for support (not shown).

c) The housing is attached to a base which is attached to two linear actuators (wide blue hatches-see figure 8).This will lift the base and housing , resulting in a control of torque.



(a) Front View

(b) Side View

Figure 8: Draft of concept 1 - Hollowed Rotor

In this concept, the input torque is given to the rotor shaft and the braking is provided by the film between the rotor and housing. The working fluid is pumped through the inlet,circulates throughout the gap and exits through the outlet. The cooling of the circulating oil will be done outside the system, using a heat exchanger. The torque will be adjusted by the linear actuator setup which is attached to the housing base.

a) Manufacturing: There are a lot of parts to manufacture. Precise manufacturing of the rotor and housing are required.Shaft boring can be costly and may not be feasible.Indents for positioning the actuators have to be made precisely, else it will cause misalignments.

b) Cost: It would require an external cooling system with pumps, pipes and a heat exchanger. It would also require two linear actuators.

c) Modularity: Good design with several replaceable parts.

d) Control of torque: If manufacturing problems are sorted out, it would be able to achieve a precise control of torque depending on the linear actuator. The bearing arrangement of having two bearings close to each other is not ideal for limiting misalignments. Since there are two linear actuators, their precise positioning is required to prevent misalignements.

e) Easiness of assembly : Easy to assemble due to good modularity.Lack of any housing-rotor connection makes it hard to assemble them concentrically.

7.1.2 System Concept 2- Hollowed Open Rotor

This system concept is similar to the Hollowed rotor concept explained in section 7.1.1. The main difference here is that the cylindrical rotor is attached to the end of the shaft using bolts. The inside of the rotor is open unlike the previous iteration. The parts of the concept can be seen in figure 9 and have been explained in section 7.1.1. This concept also has slightly lesser mass due to lack of top covering.

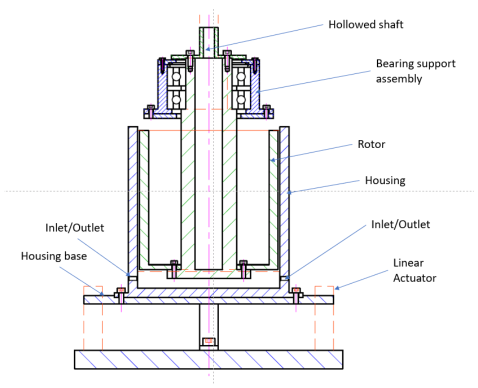
a) Manufacturing: Precise Manufacturing of the cylindrical rotor and housing is necessary. Shaft boring can be costly and may not be feasible. Indents for positioning the actuators have to be made precisely, else it will cause misalignments.

b) Cost: It would require an external cooling system with pumps, pipes and a heat exchanger. It would also require two linear actuators.

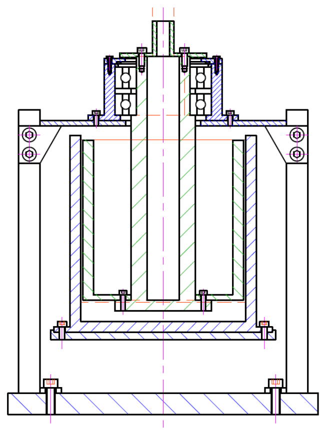
c) Modularity: Good modularity with multiple replaceable parts.

d) Control of Torque: Same as in section 7.1.1.

e) Easiness of assembly: Easier assembly of cylindrical rotor, compared to concept in section 7.1.1. Lack of any housing-rotor connection makes it hard to assemble them concentrically.



(a) Front View



(b) Side View

Figure 9: Draft of concept 2 - Hollowed Open Rotor

7.1.3 System Concept 3- Closed Rotor with internal circulation

The main feature of this concept is a rotor partially dipped in a the housing. The housing is longer than the rotor and encompasses a fixed quantity of lubricant. The lubricant can be filled in through the inlet/outlet holes in the housing. As the housing is moved vertically, the area of lubricant oil in shear changes, thus resulting in a change of possible braking torque. However, this could build up some vacuum that could be detrimental for its motion. Hence, internal circulation pipes are attached to the top and bottom of the rotor (see figure 10) to allow easy flow during the vertical motion cycles. This concept eliminates the need for continuous pumping of lubricant during the lifting cycles. However, cooling is achieved by external convection by fins attached to the housing(not shown in figure).

From previous iterations it was realized that some connection between the rotor and housing is necessary for an easy assembly. This issue was solved by having the rotor extend longer than necessary and then having a linear bearing to attach with the housing as seen in figure 10.

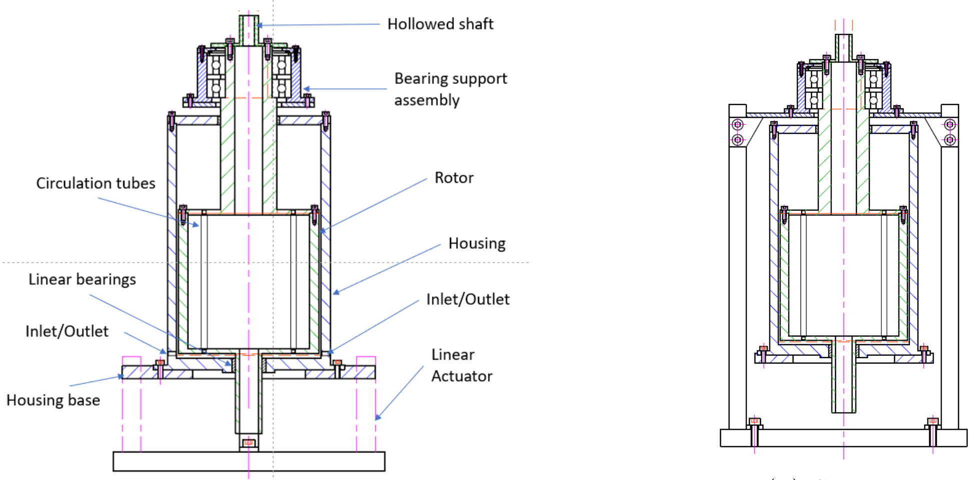
a) Manufacturing: Precise Manufacturing of the cylindrical rotor and housing is necessary. Shaft boring can be costly and may not be feasible. Indents for positioning the actuators have to be made precisely, else it will cause misalignment.

b) Cost: Cheaper than previous iterations, since pumps and heat exchangers are not required.

c) Modularity: Good modularity with multiple replaceable parts.

d) Control of Torque: Good control of torque expected. However, more time will be needed for the oil to settle after lifting due to the lack of a pump.

e) Easiness of assembly: Easier assembly of cylindrical rotor, compared to other sections. Easy to maintain concentricity of shaft and rotor during assembly



(a) Front View

(b) Side View

Figure 10: Draft of concept 3 - Closed Rotor with internal circulation

7.2 Design constraints

Before implementing the final design, parameters of the torque formulation must either be limited to a range that must be further explored or must be fixed entirely. Concepts of the iterations have also been further modified to reduce its cost.

a) **Cone angle**: In the previous iterations of concepts, a cylindrical rotor is shown. This is because a linear relation exist between torque and lift height. However, maintaining the alignment of the rotor and housing throughout the lift cycle is hard to achieve. Thus a new angle of cone is required.

   The dependence of the cone angle on the torque-lift height characteristics is seen in figure 11. As the cone angle increases, the torque drop with lift height increases. It can be seen that even a single degree increase in angle can drastically alter the torque characteristics. Lower angles are also harder to manufacture and therefore an angle of 5 degrees was selected as the cone angle. It was found that the the cone needed to lift 2mm to cover the entire range of torque from 5 Nm to 30 Nm.

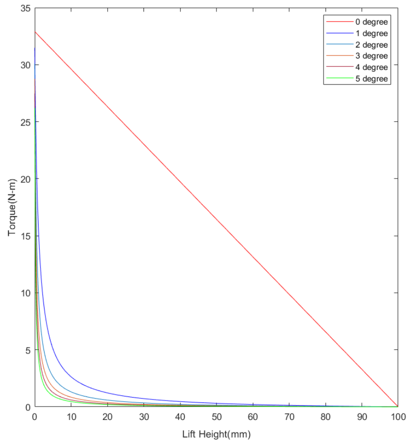


Figure 11: Variation of Torque-Lift height with cone angle

b) **Dimensions of the rotor**: To determine the optimum dimensions of the cone, a parameter connecting the overall dimensions to the torque variation needs to be formulated. The height and the outer diameter of the cone (largest diameter) are chosen as the overall dimensions. Since the angle is already fixed, only the overall height of the rotor can be varied.

   A design space of (100 * 100 * 100) mm was chosen for the design. Larger the surface area of the cone larger is the maximum torque it can brake. A 1:1 ratio of height and diameter of the cone was fixed.

c) **Film thickness**: The torque varies inversely with film thickness.Since other parameters are fixed, the selection of film thickness depends on viscosity and manufacturing constraints. Initially, a film thickness of 15 microns was chosen, but later it was found to be unfeasible. This is because, during the course of the final design it was found that the roller bearings used had an internal radial clearance value of 25-40 microns. Thus this value was updated to 50 microns.A higher viscosity oil is then needed to be used to obtain the required torque.

d) **Lubricant Viscosity**: Once the film thickness was decided, the problem becomes one-dimensional; The lubricant oil should have enough viscosity to give 30 Nm torque. It was found that 3000 mPa-s of viscosity was enough to get the required torque.

e) **Cost**: Circulation of lubricant oil was not possible as pumps could not be afforded. Thus conductive/convective cooling methods were used. Linear actuators also could not be afforded, therefore manual solutions for torque control were explored for the final concept.

Table below shows the final design constraints:

Sl no	Parameter	Value
1	Cone Angle	5 degree Celsius
2	Height:Diameter	1:1 (100 mm,100 mm)
3	Viscosity of lubricant	3750 mPas
4	Film Thickness	50 microns

Table 2: Table of design constraints

7.3 Thermal Simulations

In the final design it was decided to use aluminium fins on the housing for cooling purposes. The dimensions of the fins are decided by thermal simulations.

In all the concepts, a 30 Nm torque acts on a thin film of fluid at 60 rpm. This results in a large amount of heat generated for a tiny volume of fluid (30 ml). The temperature rise of the lubricant has to be controlled, else the viscosity of the lubricant decreases rapidly with rise in temperature. This, in turn reduces the maximum braking torque of the system.

1. **Calculation of heat generated**: The heat generated for a steady state operation is calculated given below.In this calculation, the angle of cone is neglected and a cylindrical shape is assumed for the rotor and the housing.

   Assumptions

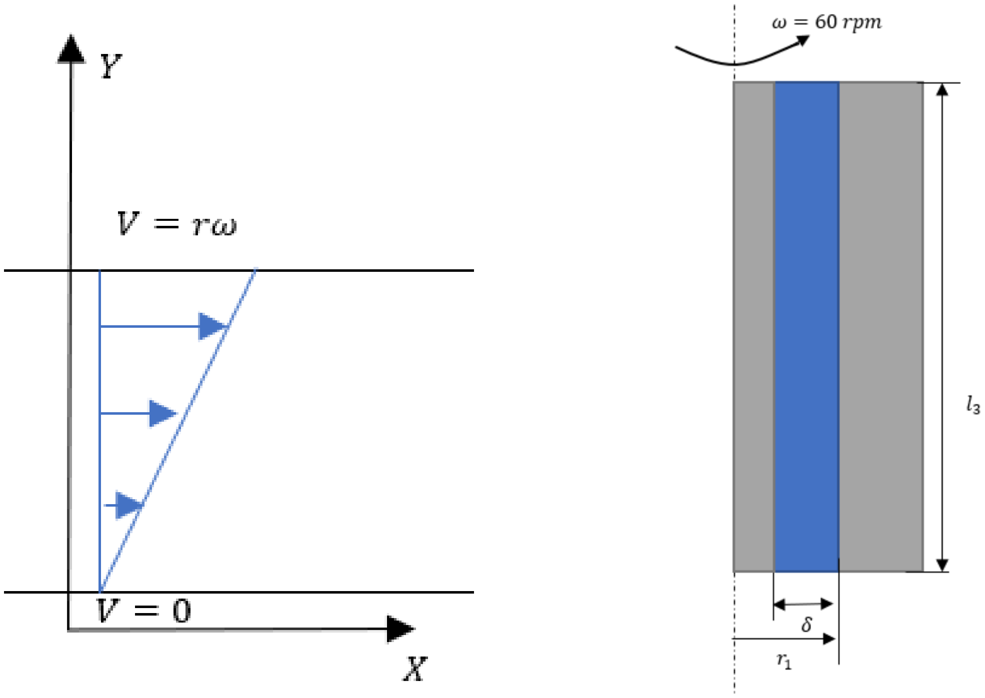
   - The oil flow can be assumed to be laminar flow between two parallel plates.
   - A linear velocity gradient is assumed for the flow.
   - Body forces such as gravity are negligible.
   - Heat transfer rate to housing or shaft has to be assumed to be zero to obtain the other.

   Parameters

   - Initial temperature: t_1= 25 degree Celsius.
   - Outer radius of film; r_1 = 50.05 mm.
   - Height of film: l_3= 100 mm.
   - Film thickness: δ_1=0.05mm.
   - Thermal conductivity of oil: k= 0.15 W/(mK)[1].

- Viscosity of oil: ν= 3750 mPa-s

The boundary conditions and problem parameters are visualized in the figure 12 below:



(a) Velocity profile

(b) Problem parameters

Figure 12: Boundary conditions and parameters

Here we shall first consider the continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y} = 0 \tag{6}$$

We can assume that the velocity is not varying in the x-direction due to any lack of pressure / shape pushing it in that direction. Therefore:

$$\frac{\partial u}{\partial x} = 0, \ \frac{\partial P}{\partial x} = 0 \tag{7}$$

The momentum equation in x-direction:

$$\rho \left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} \right) = \mu\frac{\partial^2 u}{\partial y^2} - \frac{\partial P}{\partial x} \tag{8}$$

Since the velocity gradient is linear :

$$\frac{\partial^2 u}{\partial y^2} = 0 \tag{9}$$

After applying boundary conditions in figure 12:

$$\rightarrow \ u = C_1 y + C_2 = \frac{V}{\delta} y \tag{10}$$

Here we assume that the two parallel plats(or both cone and housing) are isothermal. Therefore the energy equation is reduced to equation 11 after applying the boundary conditions:

$$k\frac{\partial^2 T}{\partial y^2} + \mu \left(\frac{\partial u}{\partial y}\right)^2 = k\frac{\partial^2 T}{\partial y^2} + \mu \left(\frac{V}{y}\right)^2 = 0 \tag{11}$$

Integration of equation 11 gives:

$$T = -\frac{\mu}{\lambda}\left(\frac{V}{\delta}\right)^2 \frac{y^2}{2} + C_3 r + C_4 \tag{12}$$

An assumption is made at this stage that the initial temperature of the cone is 25 degree Celsius.And at the surface of the housing, there is no heat flux

$$y = \delta, \quad -\lambda \frac{dT}{dy}\bigg|_{y=\delta} = 0 \tag{13}$$

Applying the boundary conditions to equation 12, we get the temperature distribution across the film:

$$T = t_1 + \frac{\mu V^2}{\lambda \delta}\left(y - \frac{y^2}{2\delta}\right) \tag{14}$$

The heat transfer rate to the housing is obtained by differentiating equation14 w.r.t to the y axis and multiplying with the area and thermal conductivity of the oil.

$$\phi = -kA\frac{dT}{dy}\bigg|_{y=0} = -kA\frac{\mu V^2}{\lambda \delta}\left(1 - \frac{0}{\delta}\right) = -A\frac{\mu V^2}{\delta}\left(1 - \frac{0}{\delta}\right) \tag{15}$$

$$V = r_1 \omega = 0.050 \times \frac{2\pi \ \times 60}{60} = 0.3142 \ m/s \tag{16}$$

Substituting the parameters we get:

$$A = 2\pi r_1 l_3 = 2\pi \ \times 0.05 \times 0.100 = 0.03142 \ m^2 \tag{17}$$

Therefore from equation 15, heat power going to the housing is:

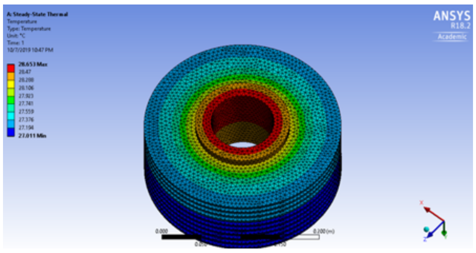
$$0.03142 \times \frac{3.75 \times 0.3142^2}{0.00005} = 232.64 \ W \tag{18}$$

2. **Fin Design and numerical study** Three major designs of fins were considered, namely circumferential fins, radial fins and pin-fins. The thickness/radius of fins, number of fins, length of the fins and spacing between the fins are subject to study. Limitations due to manufacturing are kept on the dimensions. The final aim is to obtain a fin design with a lubricant temperature of $<$50 degrees Celsius at steady state conditions.

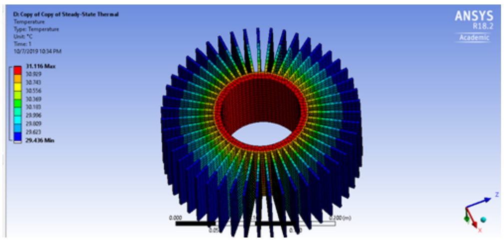
   (a) A part having a cylindrical housing and fins are made in SolidEdge and then exported to ANSYS for steady state thermal analysis.

   (b) On the inside surface of the housing the heat power and its direction is specified as outwards.

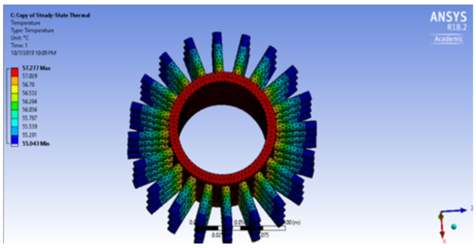
   (c) On the outer surface, a convection boundary condition is specified . The convection coefficient is found using the formulation for Nusselt number for flow around a perfect cylinder (Churchill-Chu formulation)[2]. When the velocity of the air outside is 3 m/s, the convection coefficient is found to be 18.65 W/(m^2 K). The results of the simulation are shown below:



(a) Circumferential fins



(b) Radial fins



(c) Pin fins

Figure 13: Temperature contours for different fin designs

It can be seen from figure 13 that the circumferential fins perform better than the others. At steady state conditions, there is only 3.6 degree Celsius increase in the lubricant temperature for this design. The final dimensions of the fin design is given in the appendix figure 41.

7.4 Final Design

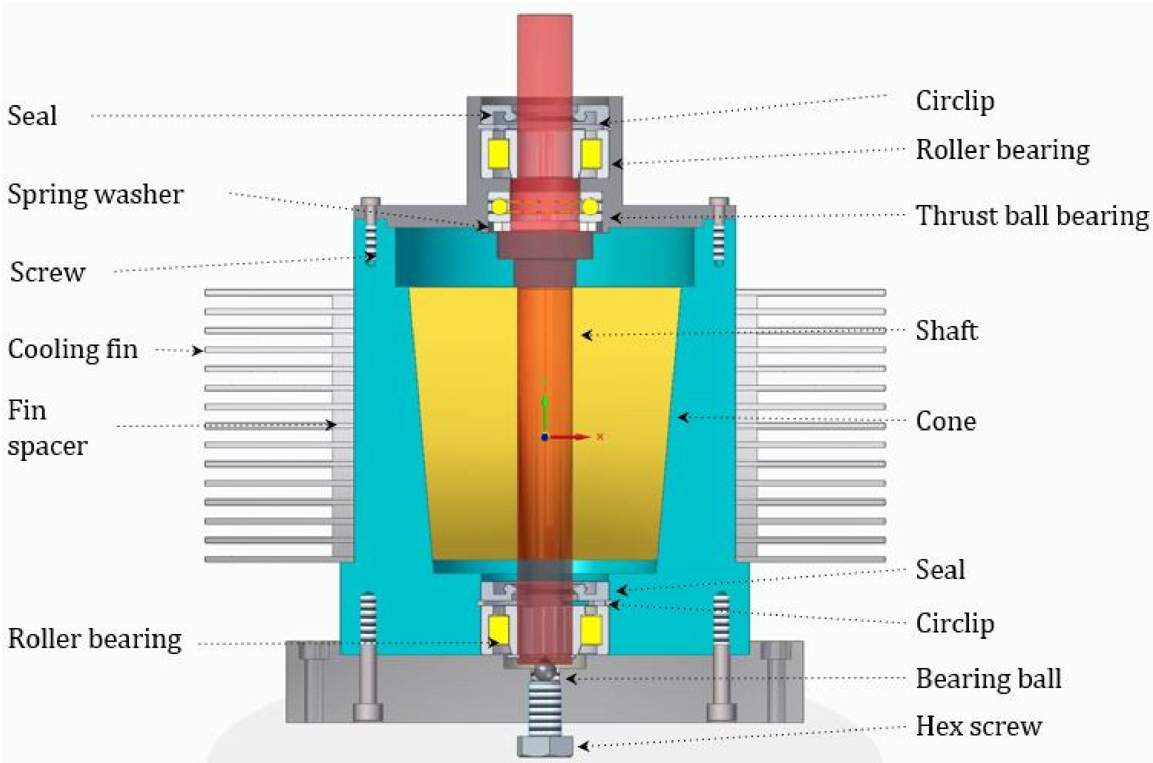


Figure 14: Final design

Design Overview

The final design concept is developed as shown in figure 14. The main parts are housing, housing cap, housing base, rotor shaft and cone.

The cone is attached to the rotor shaft via adhesive bonding. There is a step made on the shaft for proper assembly of the cone. The cone as well as the housing surface has to be grinded together (centerless grinding) to get the required surface runout (see Figure 39).

The housing is mounted on a base and also has threads for attaching the cap on top. It mimics the shape of the conical rotor and helps to maintain a constant film thickness across the area under shear. Outside of the housing, circumferential fins are fitted with a sliding fit and separated by spacers. The base of the housing has a roller bearing (NJ 2204 ECP) with an inner ring flange. It also has a shaft circlip to prevent slippage of the inner ring during lifting of the rotor.

The housing cap has a roller bearing (NJ 2204 ECP) with a flange on the inner ring. The flange prevents the cone from falling into the housing in the absence of the hex screw, supporting it at the bottom. The circlip locates the outer race of the roller bearing on one side and supports the seal on the other. The shaft seal protects the bearing and the lubricant inside the housing from contamination. There is also a thrust ball bearing with a spring washer attached to the housing cap. The spring washer rests on a shaft step.

The base has screw mounting holes for attachment to the housing. A threaded hole of 1.25mm pitch and 12mm diameter is made in the middle to assemble the set screw. It also has threaded holes for mounting it to a support base. The tip of the set screw is flat and it has a bearing ball placed on top of it. The bearing ball is kept in its place by wedging it in the tail-stock machining hole of the shaft. The tail-stock center hole is seen in figure 15.

Figure 15: Tail-Stock center hole

Part Selection

1. Roller bearings(NJ 2204 ECP): The shaft diameter was set as 25mm. This was to accommodate the coupling that connects the shaft to the transducer. No radial loads could be identified during the design, so a bearing that fits the diameter was taken. NJ series bearings were preferred due to the flanges that prevent the entire shaft-rotor assembly from falling into the housing.

2. Thrust bearings(51105) : The main purpose of the thrust bearing is to take the axial spring washer force during the lifting cycle. No sources of forces are expected to exist during normal operation.

3. Wave spring washers: Two wave spring washers are used in series to provide enough thrust force to push the cone down following the release of the set screw

from the bottom. 28mm internal diameter wave spring washers are used. Each one of them could provide 68N at full compression of 2mm. Thus when the cone is at full height it could exert a force of 136N [3].

Working

The input torque comes from the shaft and is transferred to the conical rotor through the adhesive connection. The braking torque is provided by shearing of the silicone oil between the housing and the cone. The maximum torque that can be provided by the brake unit is adjusted by lifting the rotor and shaft together. This is done by rotating the set screw. To bring the rotor down to a specific position, the screw is lowered and this allows the thrust from the spring washers to push the cone down to another position where the shaft can engage again with the roller on the set screw. The final torque curve that will be obtained from this design is shown below:

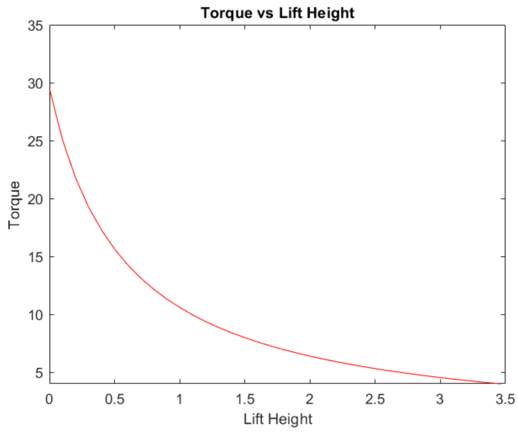


Figure 16: Torque-lift height characteristics

7.5 Manufacturing

Based on detail design, we manufactured the prototype for testing purposes. The following section describes manufacturing performed at Atlas Copco workshop, KTH prototype center and further processing during test. Assembly of the prototype is also discussed in details.

All main parts of the prototype were manufactured in the Atlas Copco workshop, including cap, housing, base, cone and shaft. The shaft, cap and base were manufactured with steel while the cone and housing were manufactured with aluminum to allow for further grinding. Parts manufactured in the Atlas Copco workshop are shown in Figure 17. All parts were turned on a lathe then screw slots were drilled at proper positions.

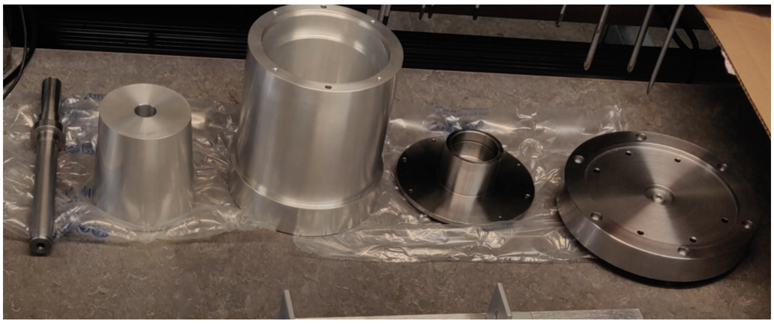


Figure 17: Parts manufactured at Atlas Copco workshop (from left to right: shaft, cone, housing, cap, base)

In order to reduce runout, the cone and housing were later grinded by filling in grinding paste between the gap and rotating the cone with the nutrunner. The nutrunner ran at 60 rpm for about 20 minutes. This grinding was later proved to be insufficient and further grinding is needed to reduce runout to an acceptable level.

Fins and the spacers were manufactured at KTH Royal Institute of Technology in the prototype center using water jet cutting and were later turned on lathe in the production department. Fins and spacers were manufactured with aluminum.

For assembly, the cone and shaft were bonded using adhesive. Hydraulic press was used where press fits occur in the design. When filling silicon oil in gap between housing and cone, the team first filled bottom of the housing with silicon oil then slowly pressed the cone into place. The cone was set at position with 1mm film thickness for about 10 minutes to allow air bubbles in the oil to come out. However, this method was not used for every time the device was assembled due to limited time. Therefore, possible air gap and uneven distribution of silicon oil could still be source of torque ripple of the brake.

8 Testing

Tests were performed to verify if the designed dynamometer brake has met initial specifications: first, if the braking torque could be adjusted between 5-30 Nm and second, if the brake provides smooth torque with ripples within 1%.

The following section describes two setups the team utilized to measure different aspects of the device. The nut-runner setup focuses on torque variation of the device with respect to angular velocity, while the flywheel setup provides more information on torque ripple. For each setup, description, data acquisition and corresponding results will be discussed in details.

Torque of the brake is controlled through lift height, which is adjusted through rotating the hex screw at the bottom. Each step of the rotation is 60 degree. For example, the second level of the lift height is 0.83mm, achieved by rotating the screw in 4 steps.

All tests are performed indoor at room temperature.

8.1 Nutrunner Setup

8.1.1 Setup Description

This set up uses nutrunner as source of input torque. The nutrunner on top and brake at bottom are connected with two flexible joints and a torque transducer (Kistler Dual-Range Torque Sensor Type 4503B, see Figure 18) as shown in Figure 19. This setup aims to measure relationship between braking torque and angular velocity, since angular velocity of the nutrunner can be easily controlled through a control box. However, it doesn't give accurate measurement on torque ripple since the torque transducer itself generates much torque ripple due to the planetary gear trains inside the nutrunner.

Figure 18: Kistler Dual-Range Torque Sensor Type 4503B

Figure 19: Setup with nut runner

8.1.2 Data Acquisition

Torque generated by the brake is obtained through display on the nutrunner control box. A relatively low lift height was selected for measurement. Torque was measured over 10 rotations at angular velocity from 10-120rpm with 10rpm interval. Figure 20 shows the twelve measurement results obtained from the above measurement.

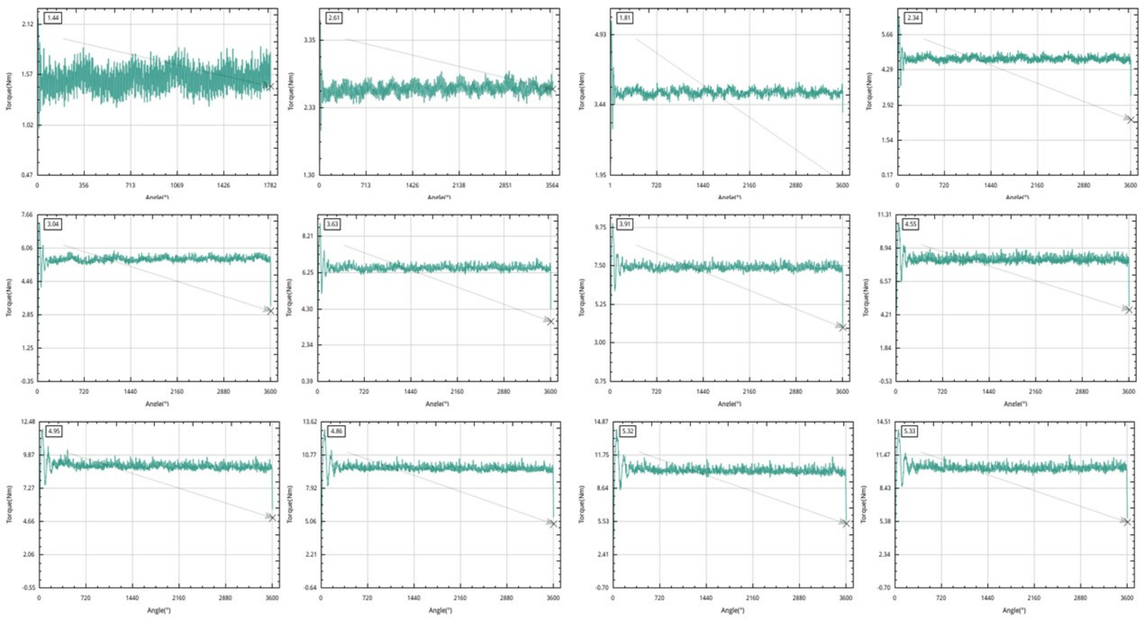


Figure 20: Torque at different angular velocity on nutrunner setup (angular velocity corresponds to 10rpm, 20rpm, 30rpm, 40rpm, 50rpm, 60rpm, 70rpm, 80rpm, 90rpm, 100rpm, 110rpm, 120rpm from left to right and top to bottom respectively)

8.1.3 Results

Average torque value is plotted against angular velocity in Figure 21. As shown in graph, the braking torque increases linearly with respect to angular velocity within range of 10-110rpm, which fits the mathematical model the team created for torque variation within oil film. However, torque at 120rpm stays at roughly the same level as that of 110rpm. The team suspects that this is due to shear thinning, an effect that often occurs in high viscosity fluids whose viscosity decreases under shear strain.

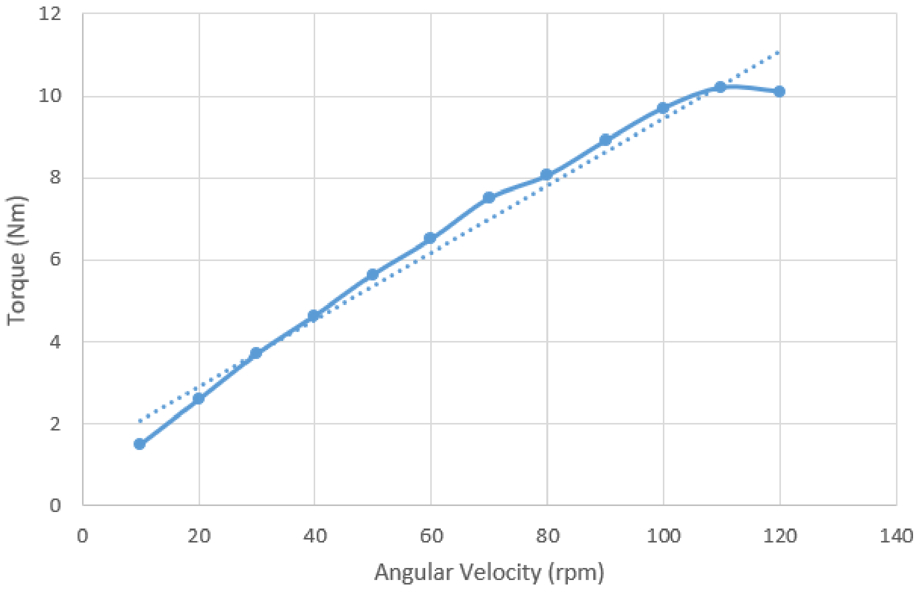


Figure 21: Torque change with respect to angular velocity with linear approximation

8.2 Flywheel Setup

8.2.1 Setup Description

The second set up uses a flywheel to rotate the rotor as shown in Figure 22. One end of a fishing wire is attached to the flywheel. The fishing wire then goes through a pulley on frame of the brake and a pulley on the ceiling. The other end of the fishing wire is connected to a weight, which varied between 2 kg, 4.5 kg and 10 kg.

This setup has the advantage of lower input torque ripple due to the flywheel. However, since input torque could only be provided when the weight falls from the ceiling, it only remains for a short amount of time. Angular velocity of the rotor is also hard to control through this setup.

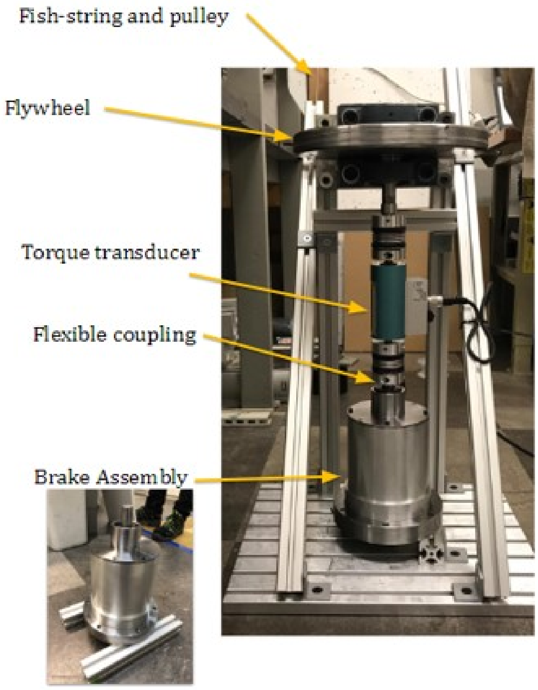


Figure 22: Setup with flywheel

8.2.2 Data Acquisition

Torque, rotating angle and angular velocity are measured through the Kistler torque transducer mentioned above. The transducer is connected to the DEWE-43A DAQ box (see Figure 23), which is then connected to the computer through USB. Dewesoft X3 SP9 (release-191204) was used as software for recording data from the transducer.

Figure 23: DEWE-43A DAQ box

8.2.3 Results

The team performed initial testing with the cone set at a relatively low lift height and attached it to the 2 kg weight which provides 2.94 Nm input torque. Results are shown in Figure 24. When zoomed in, it can be observed that smooth torque occurred at several locations. Torque ripple at these locations is 1.13%, which is close to the requirement. However when including periodical peaks during operation, the torque ripple rises to over 10% . The peak values occurs every 360 degrees of rotation, and thus the team concludes that these peak values were mainly due to contact between cone and housing.

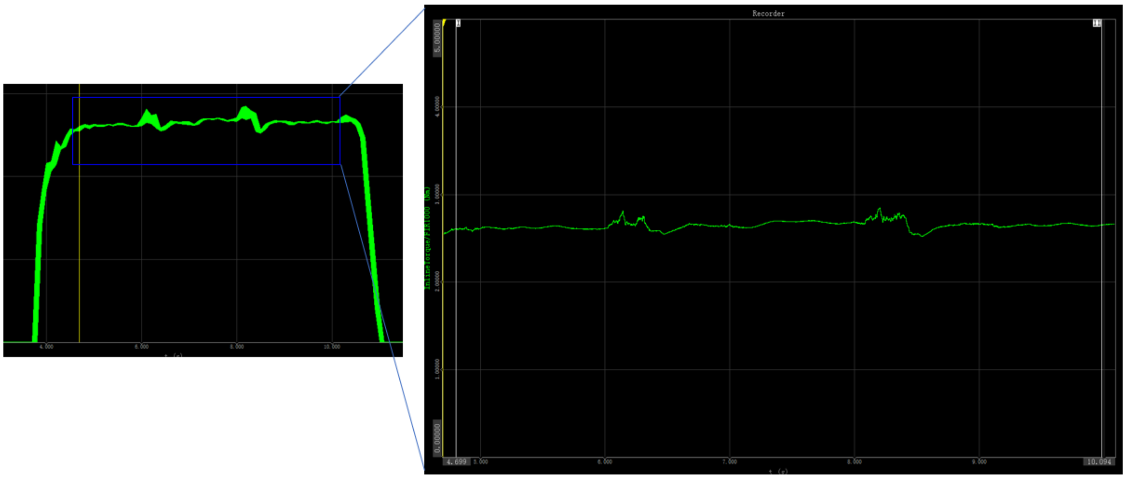


Figure 24: Torque measured at low lift height with runout

A factorial analysis with different lift height and different attached weight was conducted. The setup is at zero left height position with the 2 kg weight attached to the fly wheel (see Figure 25). In this case the torque ripple is around 2.2%.

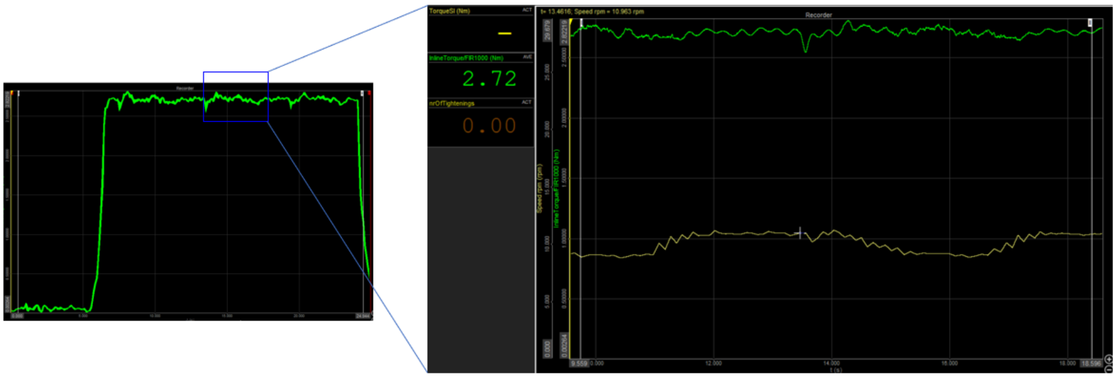


Figure 25: Torque measured at 0mm lift height, 2kg weight and 2.94 input torque

Then the weight was changed to 4.5 kg, giving an input torque of 6.6 Nm, and the torque ripple was found to be around 8% (see Figure 26).

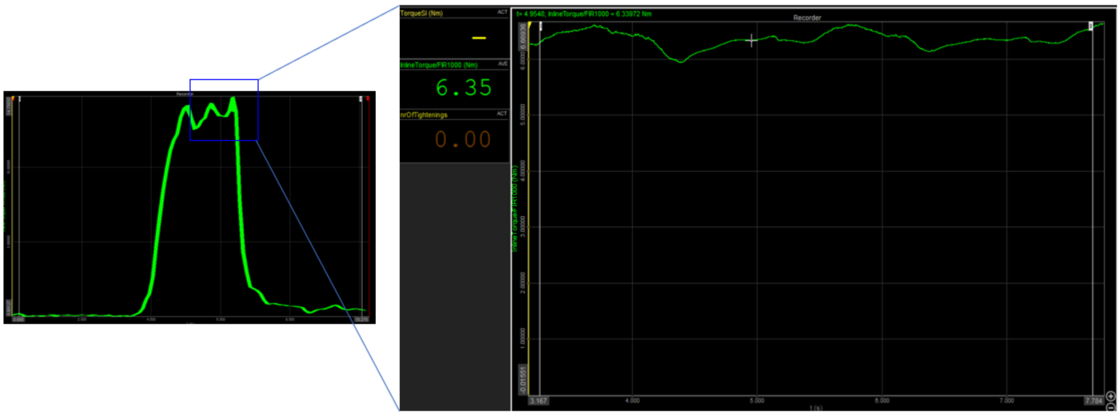


Figure 26: Torque measured at 0mm lift height, 4.5kg weight and 6.615 input torque

In the next stage when the weight was changed to 10 kg. The torque ripple rise to around 12% (see Figure 27).

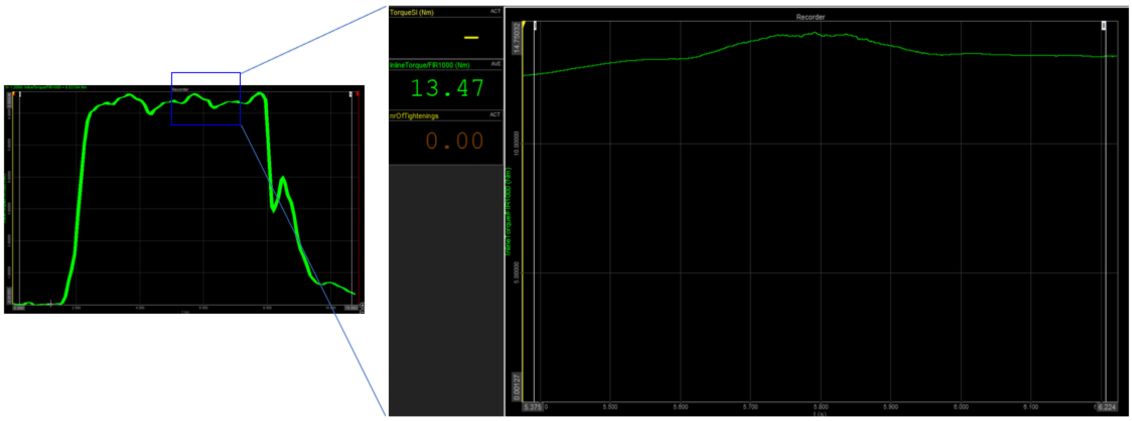


Figure 27: Torque measured at 0mm lift height, 10kg weight and 14.7 input torque

A summary of torque measurements is presented in Figure 28.

attached weight	2 kg	2 kg	4.5 kg	4.5 kg	10 kg	10 kg
Lift height	torque	torque ripple	torque	torque ripple	torque	torque ripple
0mm	2.71	2.21	6.41	7.95	14.11	11.9
0.83 mm (60 degree*4)	2.69	11.1	5.875	12.4	9.865	11.8
1.35mm (60 degree*6.5)	2.72	7.35	6.165	13.4	10.895	10.5

Figure 28: Results - Mean values

In Figure 29, the blue lines represent the torque and the red lines represent torque ripple. Same attached weight results in same angular velocity of the shaft. Focusing on each curve with the same lift height, the proportional relationship between the torque and the angular velocity can be observed. The torque ripple cannot be controlled and peak value is around 16%.

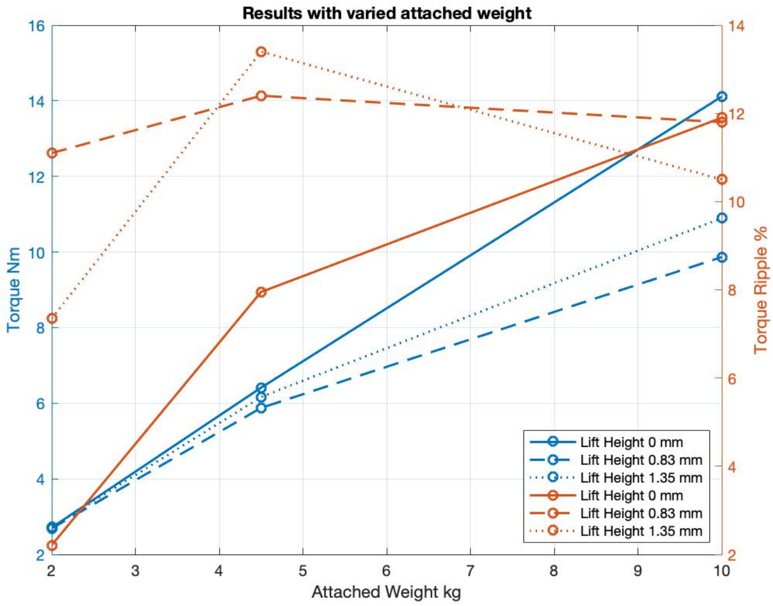


Figure 29: Result plotting varying attached weight

Same data is plotted in a different way in in Figure 30. The blue lines show the torque and the red ones show torque ripple. One of the curves show trend close to inverse proportional relationship between torque and lift height, which is expected through calculations.

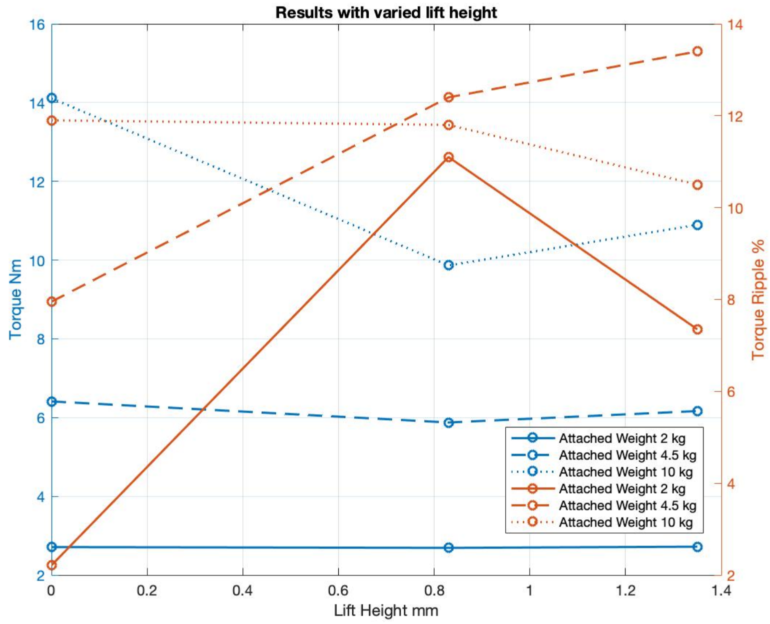


Figure 30: Result plotting with varied lift height of cone

A factorial analysis with two factors (attached weight and lift height of the cone) at two levels (low and high) resulting in 4 different parameter combinations of the nominal low and high values is shown in Figure 31. The calculated effects for the mean value tell us how the means value of the factor affect the result, which is level setting.

Level	Attached weight	Lift height
-1	2 kg	0mm
1	10 kg	1.35mm (390 degree)

Figure 31: Level of factorial designed test

From the calculation in Figure 32, the influence of the torque ripple from the negative weight level to the positive weight level is 5.15, while the difference of torque ripple with different lift height levels is 3.14. This value is lower than that of weight and interaction effect is even lower.

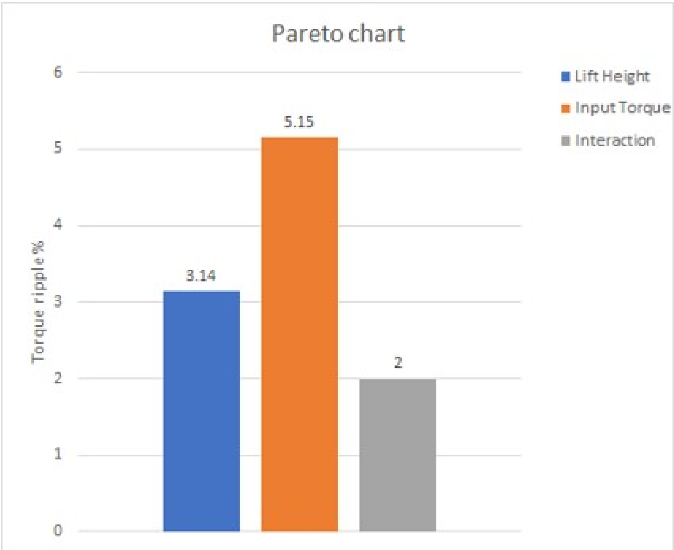


Figure 32: Factorial designed test

9 Conclusions and Future Work

9.1 Conclusions

The designed dynamometer brake partially meets the first design requirement of variable torque. Torque could be adjusted continuously roughly within range of 2-10 Nm by changing lift height or angular velocity. The team concludes that the final design couldn't reach the ideal torque range of 5-30 Nm mainly due to shear thinning of the high viscosity silicon oil. This effect was confirmed by test results from the nutrunner setup shown in Figure 21.

Though the final design cannot provide constant smooth torque at all lift height and angular velocity, it has the potential of generating braking torque with low torque ripple since torque ripple within 1% has been observed under several test conditions. The unstable torque performance is mainly due to limitations on manufacturing precision and assembly, and more specifically runout and air bubbles in the silicon oil film.

In order to confirm source of the torque ripple, further measurements on the cone and shaft to quantify runout were conducted. As shown in Figure 33, the team fixed the shaft and measured maximum runout at different locations using a radial dial. Results for the measurement are shown in Figure 34. It can be observed that runout at several locations exceeds the designed runout of 20 µm, and runout on the cone exceeds the designed oil film thickness of 50 µm. Therefore, it can be concluded that the cone is likely to touch the housing during operation, thus generating torque ripple at those contacts.

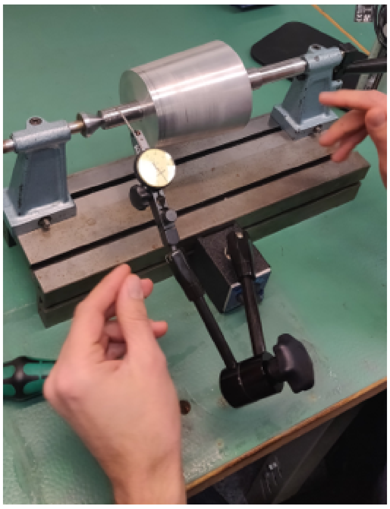


Figure 33: Measuring runout of shaft and cone

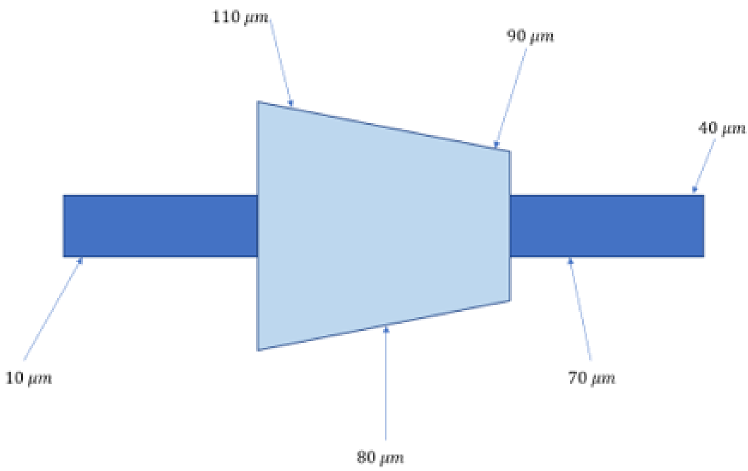


Figure 34: Maximum runout at different locations on shaft and cone

Besides runout, the team also identified the following reasons for torque ripple:

a) Stick slip due to seals on top and bottom of the shaft

b) Uneven distribution of silicon oil between cone and housing

c) Silicon oil in bottom roller bearing

d) Elasticity of fishing wire resulting in unstable input torque

9.2 Future Work

For further improvement of the design, the team gives the following suggestions on future work:

1) Grind the cone and housing more to reduce torque ripple due to runout
2) Reduce seals from the design and find other ways to prevent oil leakage to reduce stick slip
3) When filling silicon oil into the thin gap, start with a larger film thickness and slowly press in the cone; allow enough time for settling before testing so that the oil is distributed evenly and air bubbles come out of the oil film
4) Implement design for assembly so that the device could be taken apart and assembled easier
5) Change location of set screw or utilize actuators for lift height variation in order to control lift height more accurately
6) Monitor temperature of the oil film to determine effect of increased oil temperature on torque generated
7) Run brake over long period of time to determine stability time of the device

References

[1] *Whacker Silicone Fluids AK-Product Catalogue*

[2] Çengel, Y., Ghajar, A., Kanoglu, M. (2015). *Heat and mass transfer : Fundamentals and applications* (5th ed. in SI units. ed.). New York: McGraw-Hill.

[3] Wave Spring Catalogue-lesjoforsab AB. *https://www.lesjoforsab.com/strip-components/138-139*$_e n_i$*d957.pdf*

[4] Kistler Instruction Manual Dual-Range Torque Sensor Type 4503B *https://www.mts.ch/public/upload/assets/925/BA*$_4$*503B*$_E$*N.pdf*

Appendices

A Drawings

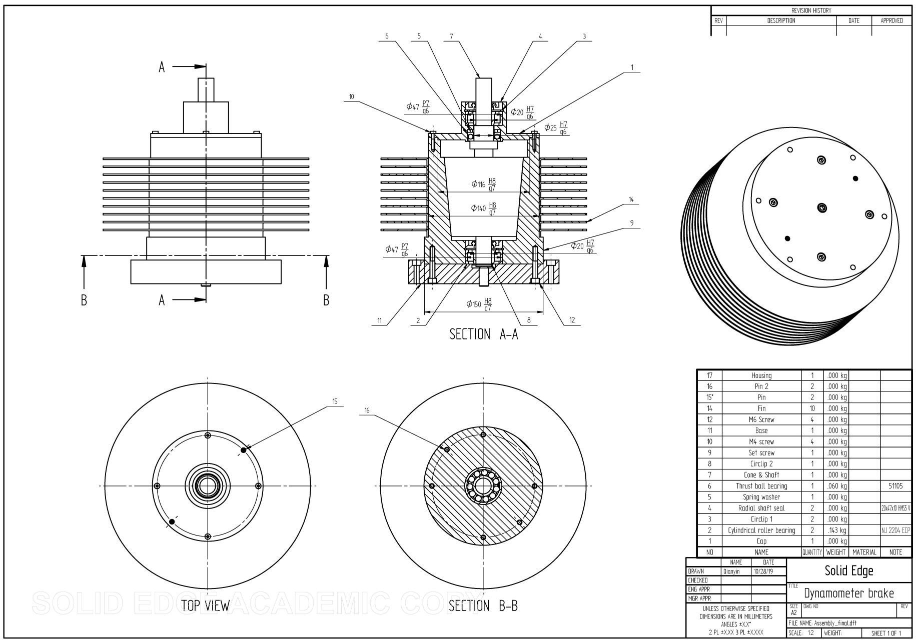


Figure 35: Assembly Draft

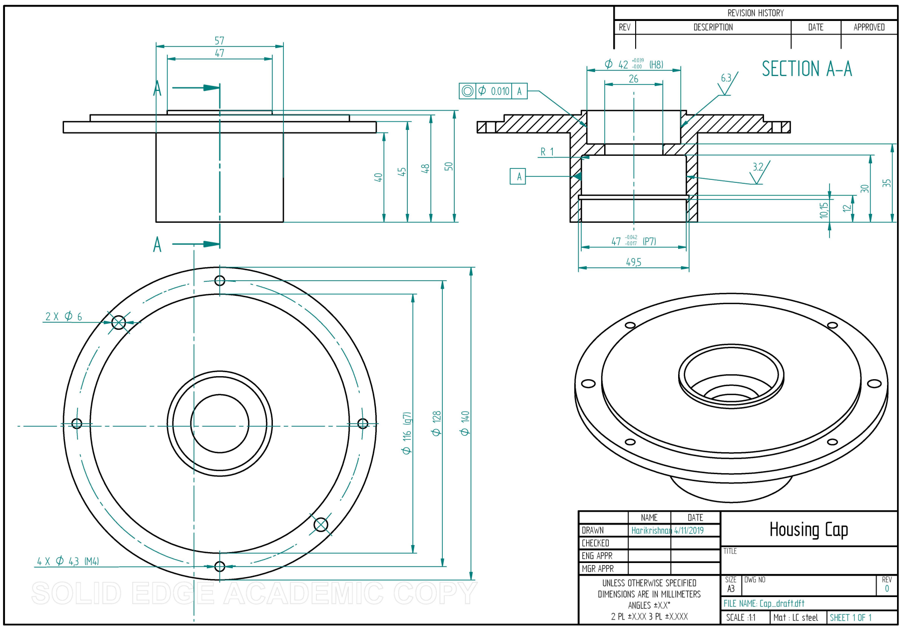


Figure 36: Draft of Housing Cap

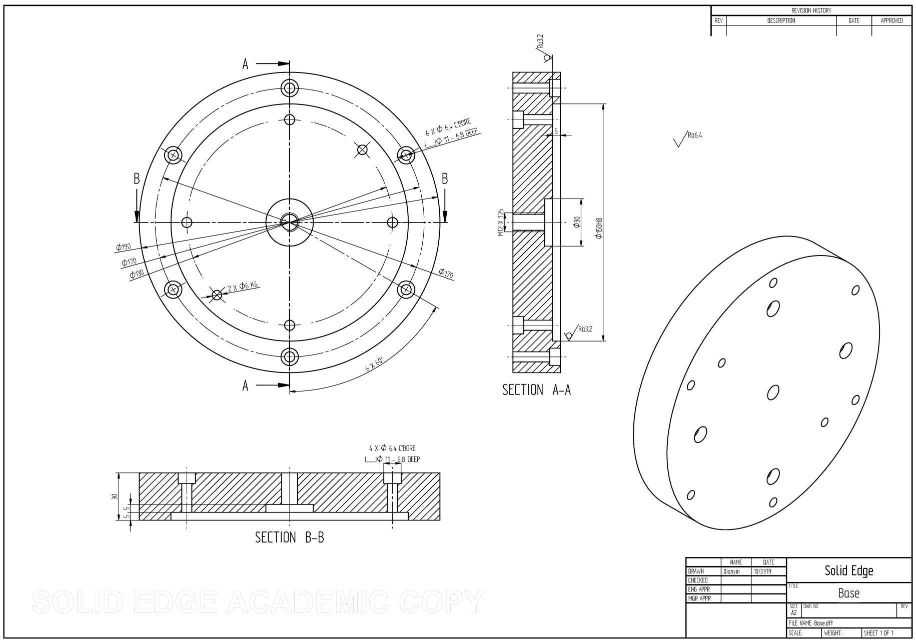


Figure 37: Draft of Housing Base

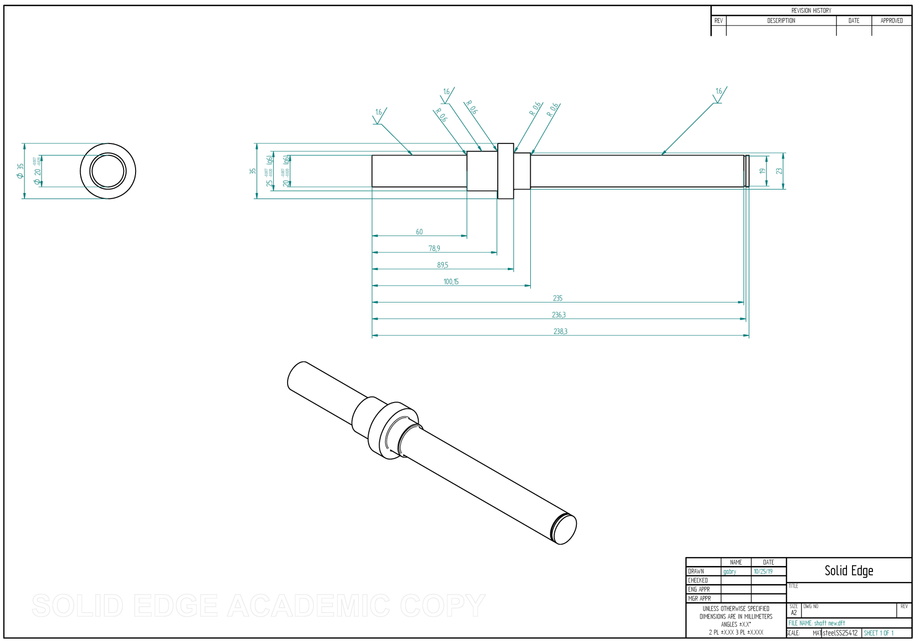


Figure 38: Draft of Rotor Shaft

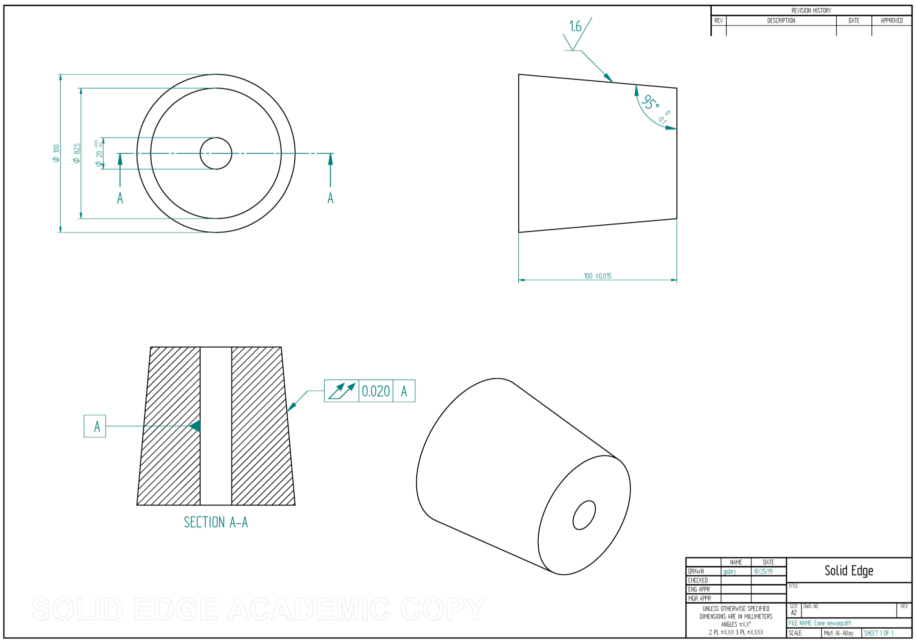


Figure 39: Draft of Cone

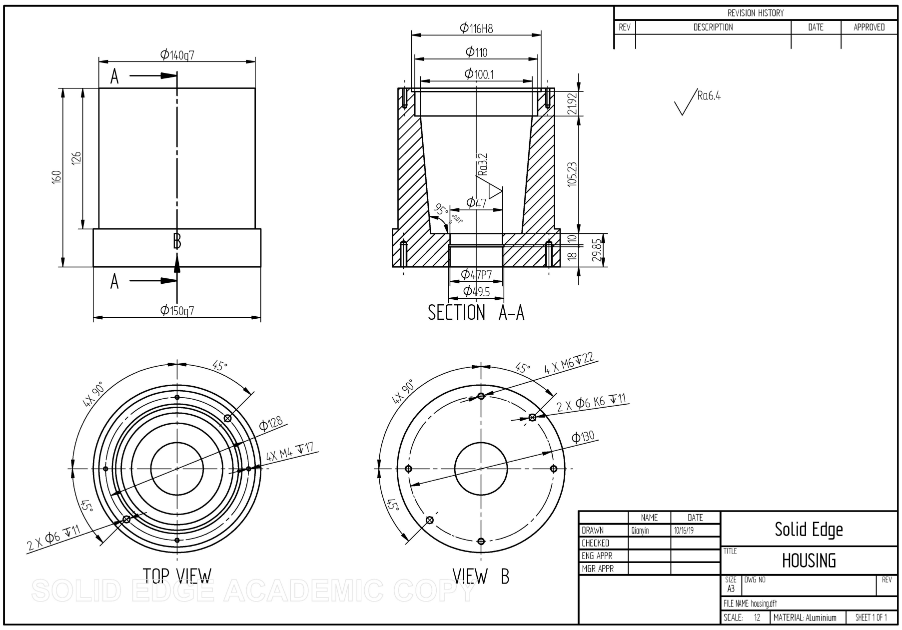


Figure 40: Draft of Housing

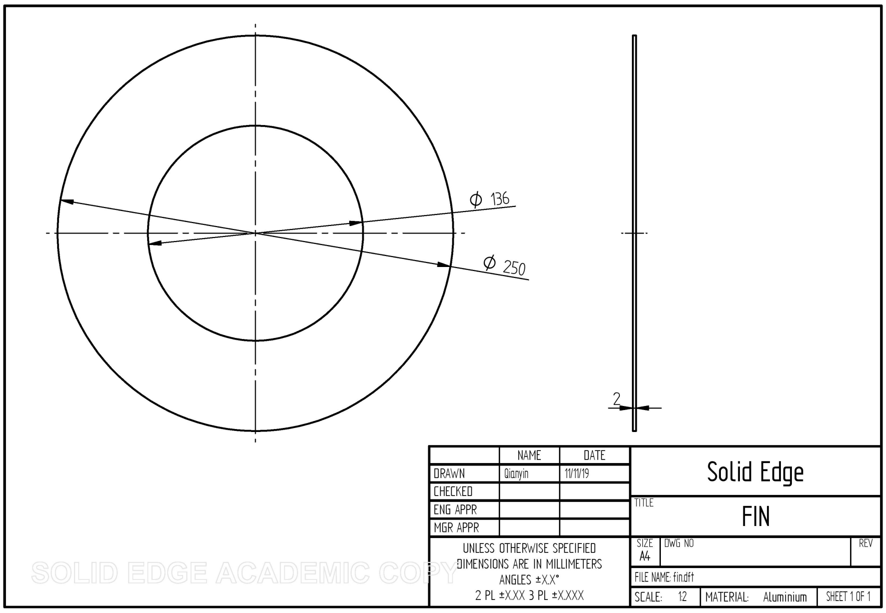


Figure 41: Draft of Fin

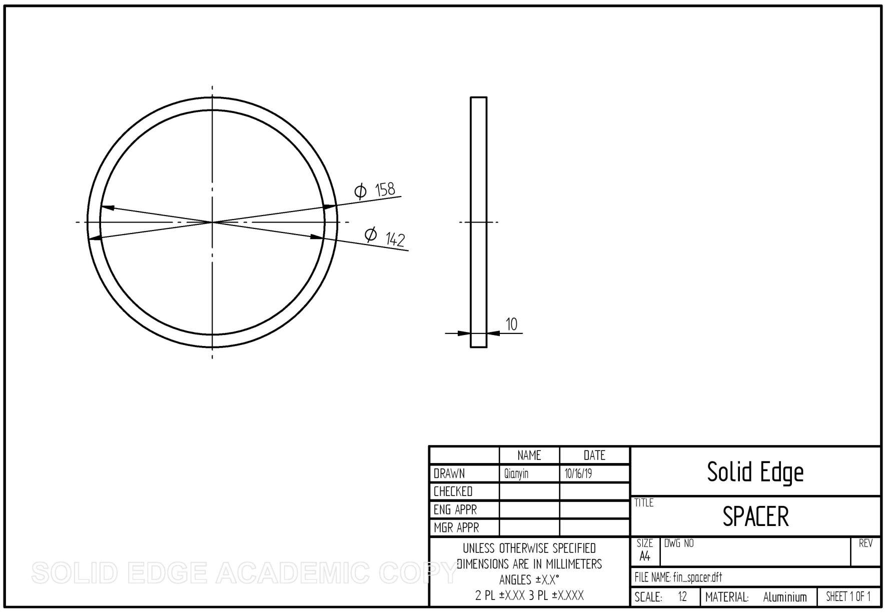


Figure 42: Draft of Fin Spacer