Precision Dynamometer Brake for Gear Testing

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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.

1 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.

2 Background

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.

3 Conceptual Design - rheometer

3.0.1 Description

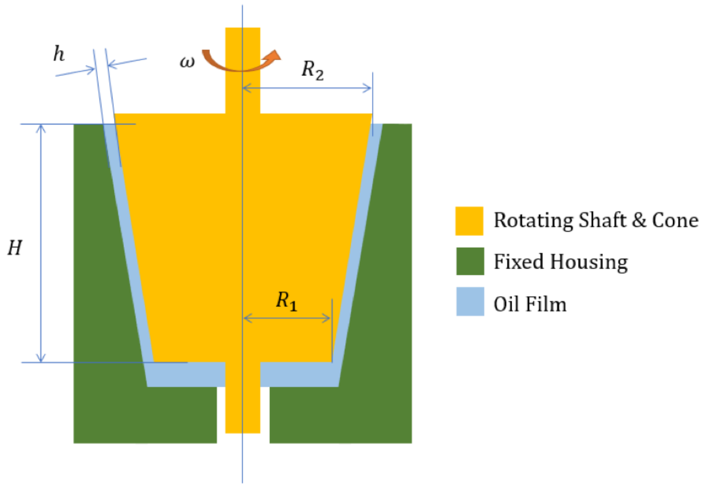

Figure 1. Conceptual concept 5 - rheometer

The idea was generated from the rheometer concept as shown in Figure 1. 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.

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

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:

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

4 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.

4.1 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 2. 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.

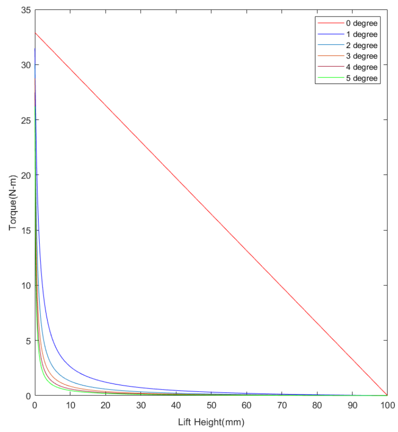

Figure 2. Variation of Torque-Lift height with cone angle

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 1. Table of design constraints

4.2 Final Design

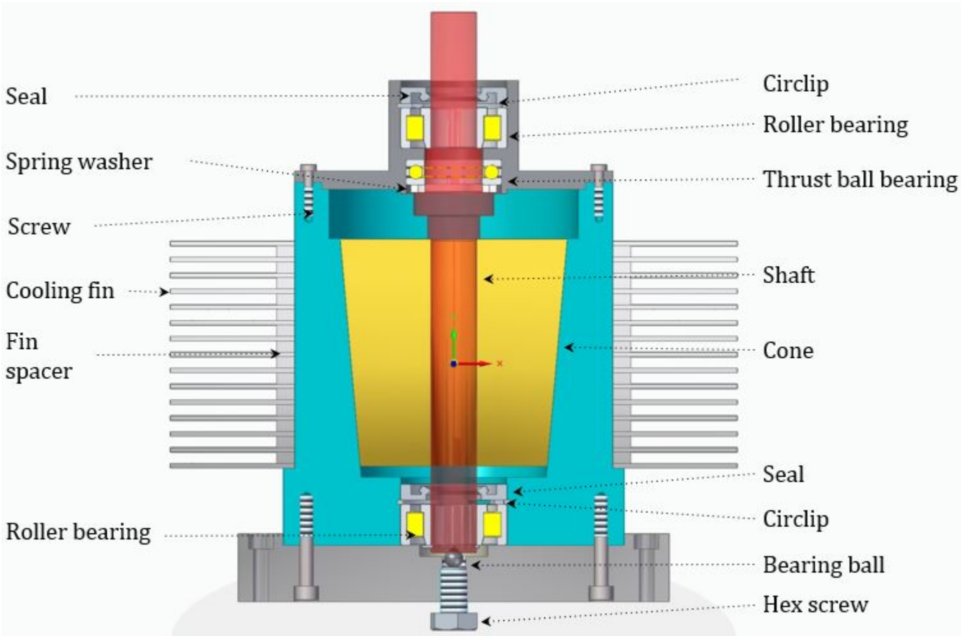

Figure 3. Final design

4.2.1 Design Overview

The final design concept is developed as shown in figure 3. 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.

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 4 .

Figure 4. Tail-Stock center hole

4.2.2 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. But this can be a cause of ripple because when the roller is in contact with the flanges, it can introduce sliding due to stick-slip.

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.

4.2.3 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:

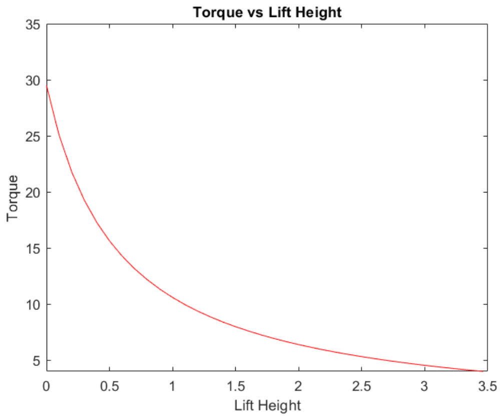

Figure 5. Torque-lift height characteristics

5 Testing

Tests were performed to verify if the designed dynamometer brake has met initial specifications. 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.

5.1 Nut Runner Setup

5.1.1 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) as shown in Figure 6. 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 6. Setup with nut runner

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.

5.1.2 Results

Average torque value is plotted against angular velocity in Figure 7. 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.

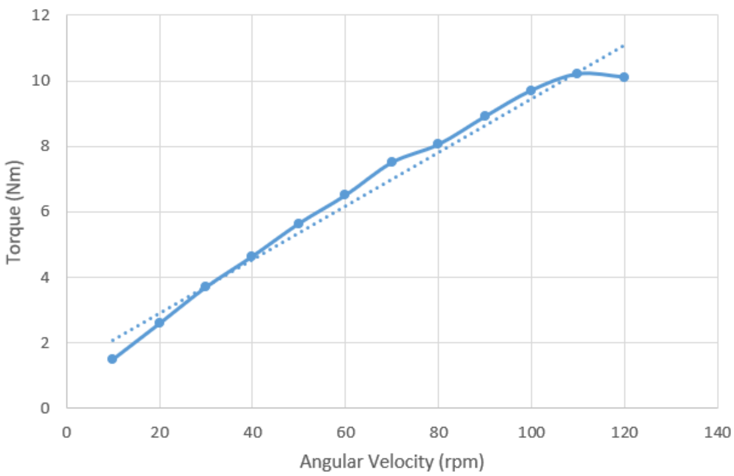

Figure 7. Torque change with respect to angular velocity with linear approximation

5.2 Flywheel Setup

5.2.1 Setup Description

The second set up uses a flywheel to rotate the rotor as shown in Figure 8. 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.

5.2.2 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, which is then connected to the computer through USB. Dewesoft X3 SP9 (release-191204) was used as software for recording data from the transducer.

5.2.3 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 9. 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.

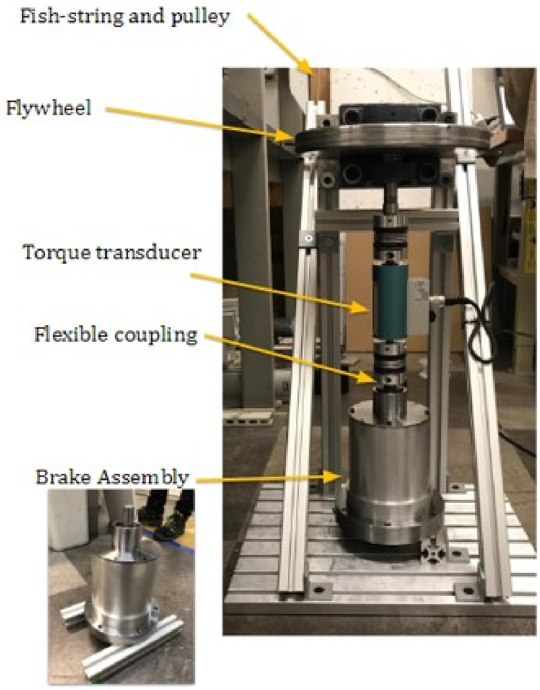

Figure 8. Setup with flywheel

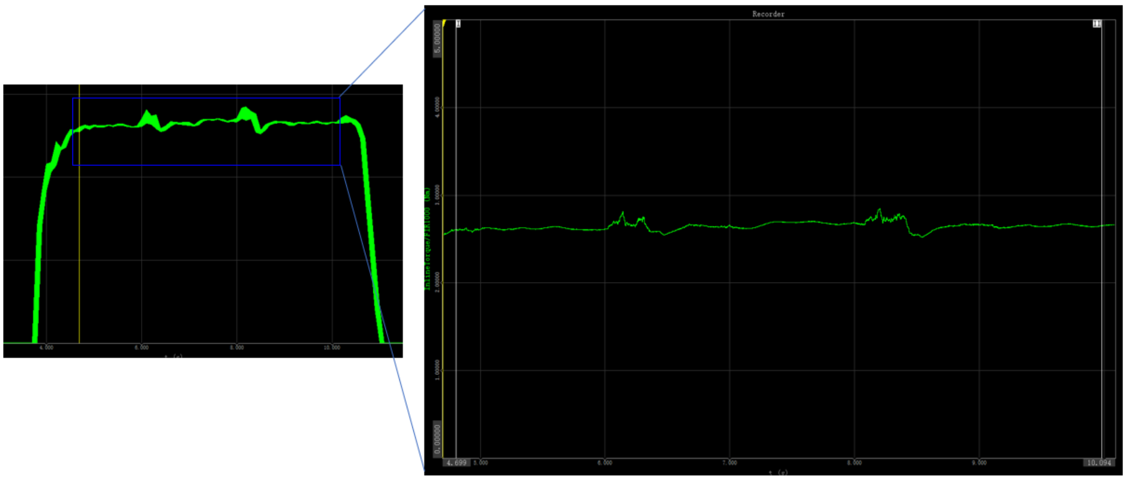

Figure 9. Torque measured at low lift height with runout

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 10). In this case the torque ripple is around 2.2%.

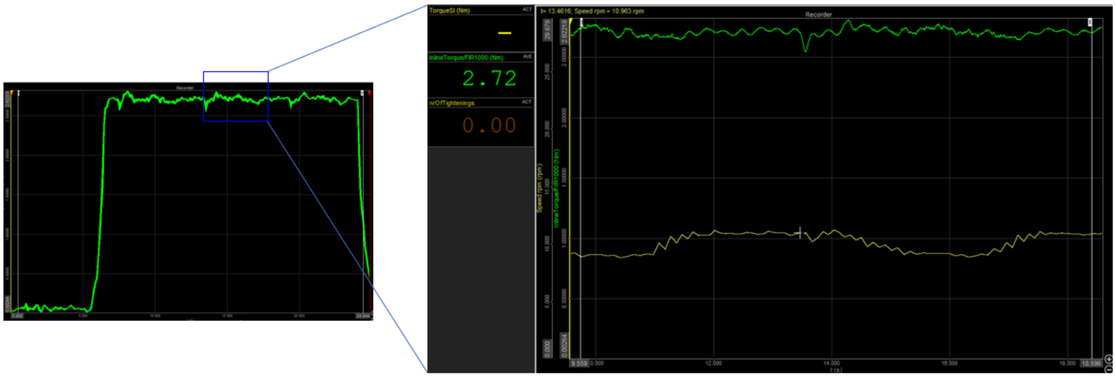

Figure 10. Torque measured at 0mm lift height, 2kg weight and 2.94 input torque

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 11).

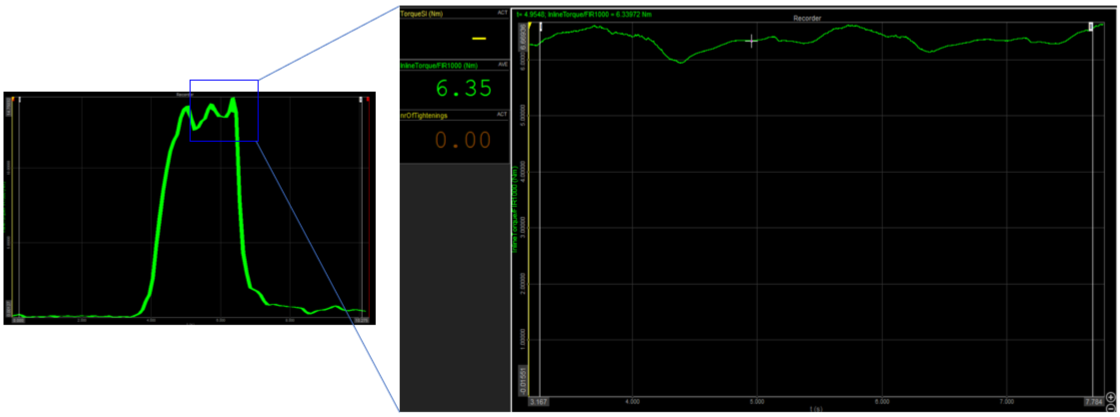

Figure 11. Torque measured at 0mm lift height, 4.5kg weight and 6.615 input torque

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

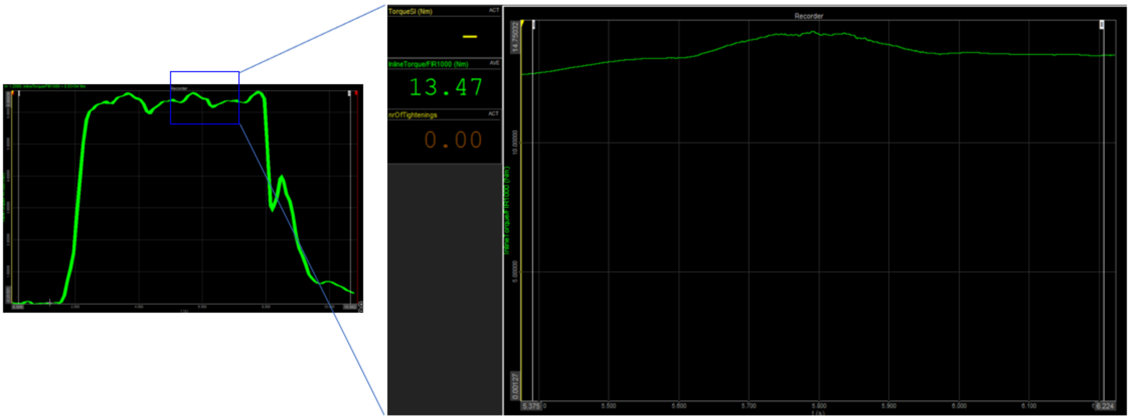

Figure 12. Torque measured at 0mm lift height, 10kg weight and 14.7 input torque

A summary of torque measurements is presented in Figure 13.

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 13. Results - Mean values

In Figure 14, 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%.

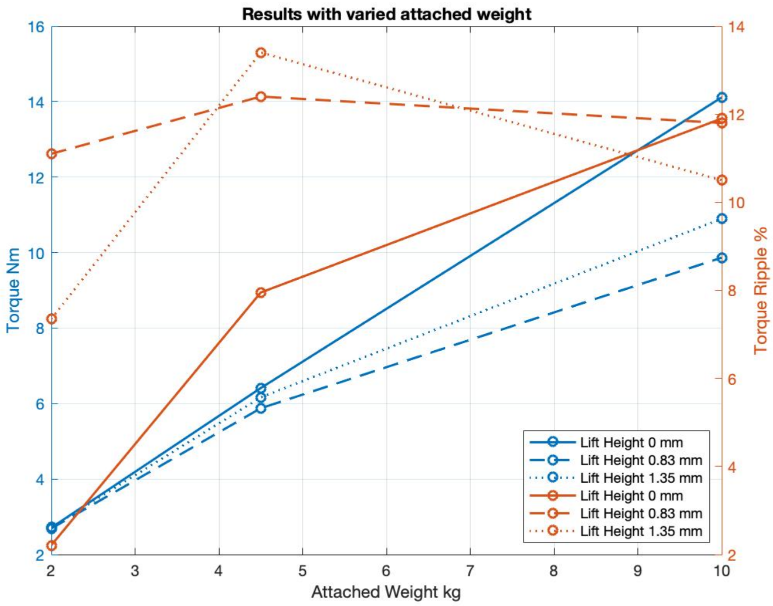

Figure 14. Result plotting varying attached weight

Same data is plotted in a different way in in Figure 15. 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.

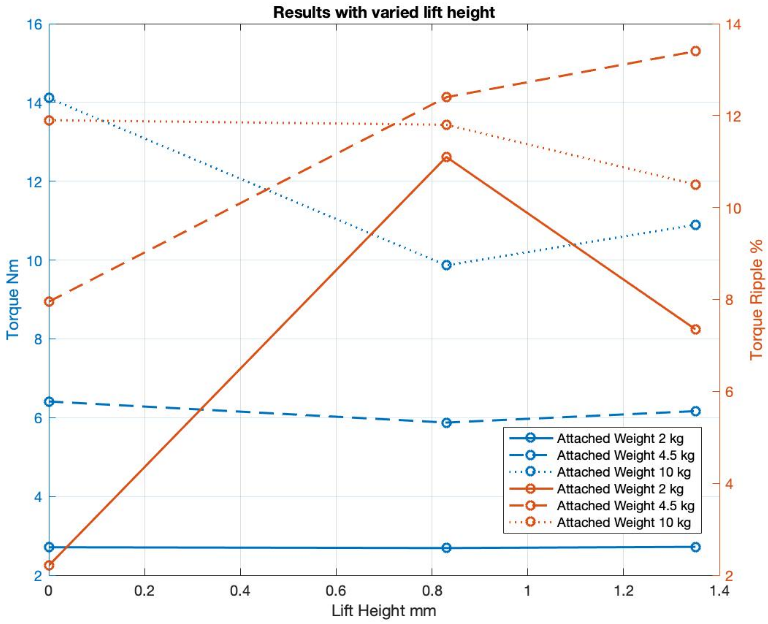

Figure 15. Result plotting with varied lift height of cone

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 16. 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 16. Level of factorial designed test

From the calculation in Figure 17, 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.

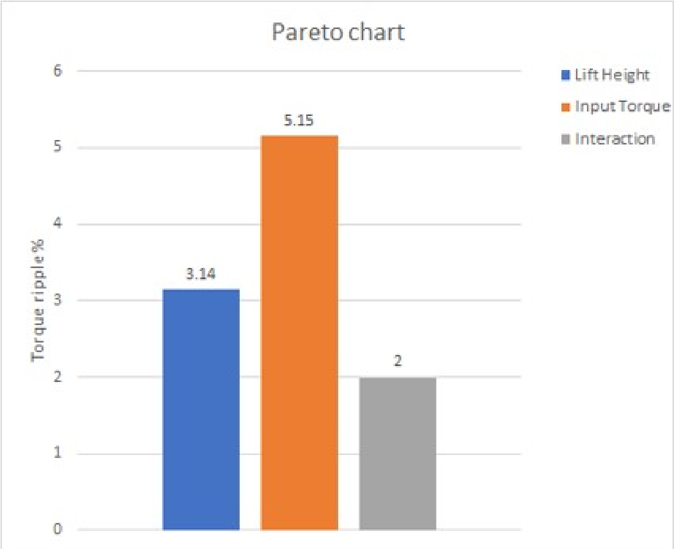

Figure 17. Factorial designed test

6 Conclusions and Future Work

6.1 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.

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. Runout of the cone was confirmed through measurement with dial, which shows runout on the cone could go up to 110 µm at certian points, which is larger than the designed film thickness.

The team identified the following reasons for torque ripple:

a) Runout of cone and shaft

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

c) Uneven distribution of silicon oil between cone and housing

d) Silicon oil in bottom roller bearing

e) Elasticity of fishing wire resulting in unstable input torque

6.2 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

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