Integrated brake in hydraulic motor for winch applications

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Preface

This work is done for Bosch Rexroth AB and the content in this document reports what’s deemed possible to report without disclosing any technical solutions that may be subjects for patents.

Thank you for the valuable help Marcus Björling, Senior Lecturer at Luleå University of Technology and also the examiner of this project.

For providing this opportunity, helping out finding data and giving relevant input, thank you Joel Andersson, Anders Nordin, Andreas Wallgren and more at Bosch Rexroth’s engineering department, Roger Granström and Hans Sahlin at Bosch Rexroth’s sales department.
Abstract
The company named Bosch Rexroth, previously Hägglunds drives, in Mellansel, Sweden develops, produces and sells radial piston hydraulic motors for industrial use, some are used to drive winches. The winch application mostly needs a brake to function as a parking brake and emergency brake. The company offers brakes that are installed as external assemblies.

This thesis work is done to finding brake concept solutions that requires less space and doesn’t add as much to weight and cost and that could be better integrated in the application.

The steps to achieve this includes evaluating the needs, generating new brake concepts, refining these, evaluating the concepts found, refining the best concept and finally presenting the result.

The brake concept found to be the best of the set generated is a slightly altered drum brake that, as an example, is integrated into the winch drum and has the potential to have a shorter response time than concepts used today.
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1. Project introduction and methodology

This aim of this project is to develop a brake concept that can be integrated in a winch application. As seen in the project description in Figure 1 this could be by integrating the brake in the motor, but as this report reveals it could also be integrated in the winch drum. There it’ll function as a parking brake and as an emergency brake. The project was issued by Bosch Rexroth AB in Mellansel and is described as:

Figure 1: Thesis work description.
1. Project introduction and methodology

This introductory part of the report describes the phases of this project and describe the limitations of the scope.

1.1. Project phases
The project at hand is a concept development process and comprises different phases to achieve its goals in terms of concept properties. These goals are further mentioned in 3. *Need finding* as the project rolls on. Though in an early stage the phases can be described as steps to take to find and express these goals as properties that can be quantified and then pursued in the subsequent development. Here these phases are described.

1.1.1. Risk analysis
To prevent the possible consequences of risks from having too deteriorating effect on the project result a risk analysis is made. Here the foreseeable risks are listed and coupled with a level of severity as well as a level of plausibility. These levels multiplied gives a sense of priority. Note that these levels are determined with a great level of subjectivity which doesn’t call for high resolution, but rather a low one in order to provide a just comparison.

1.1.2. Need finding
Staff from Bosch Rexroth’s development department primarily stated a coarse description of the needs, but to increase the likelihood of concept success further a more thorough investigation is done. Other potential users within the company e.g. sales personnel, manufacturing and assembly personnel etc. is interviewed for their point of view to add needs previously missed.

All the technology of the products included in a customer application is looked into to provide knowledge about the constraints and possibilities for the function of the concept.

The needs are broken down along with the primarily stated ones, into as small constituents as possible to express the needs in a more fundamental way. This is done to enhance the inputs to the following concept development phase, allowing a more precise but possibly also wider part of the solution space to be taken into account.

1.1.3. Literature study and benchmarking
A study of the technology used within the company and by competitors, but also of technology deemed relevant for new ways to fill the function helps to get a view of the solution space. Benchmarking of previous products by the company and its competitors helps the decision making concerning continued work.

1.1.4. Specification of demands and decision making
After the need finding and benchmarking phase is done all demands should be clearly understood and possible to rank, quantify and limit. The listing of these is then done in chapter 6. *Specification of demands and decision making*, later functioning as a document for objectively evaluating which concept or concepts that are good enough for further development.

1.1.5. Concept generation
By looking at the problems and needs present and the available solution space, multiple brainstorming sessions are held. The concept ideas and ideas of specific parts of such
1. Project introduction and methodology

concepts generated during these sessions are then organized in such a way that the needs are satisfied.

1.1.6. Screening
The concepts are then either accepted or excluded in a rough screening process considering their properties soundness and a rough estimation on whether they’re achievable or not.

1.1.7. Concept refinement
The concepts that passes the screening are then modeled in a slightly higher level of detail and their properties are roughly calculated to produce a basis for objective comparison with both each other and with presently existing products.

1.1.8. Concept evaluation
The previously mentioned specification of demands described in chapter 6. Specification of demands and decision making is used for this and the properties of the concepts is objectively scored and documented. This reveals which concept is worthy of proceeding the work with.

1.1.9. Finishing work
As a final phase of the project the best concept or concepts are further modeled and calculation models for concept properties are developed for further work in subsequent projects dimensioning work.

If demonstration is deemed to be enhanced by building a simple prototype instead of just preparing a CAD-model, and if there’s any time and resources available to do so, that is done.

1.1.10. Report and presentation
This report is written in parallel to the project to reflect on project phases, investigation and development progress and success, findings and also difficulties, drawbacks and any changes in project scope.

1.2. Project scope
Since the result only is to be on a conceptual level many system property demands and inputs can be ignored or weighed lightly, eg. interface details and other high level detailed demands by customers and certifying organs.

What needs to be taken into account are more profound demands by customers and certifying organs concerning load capacity, environmental resistance, safety mechanisms, energy consumption, production cost, etc. that impacts the choice of concept type altogether.

At its minimum, the goal of this project is:

- To develop concepts for braking winch applications by e.g. either braking the motor or the winch drum. The brake shall be used as parking brake and emergency brake.
- To perform a benchmarking process to map the company’s brakes in comparison to competitors to clarify where the needs are.
- To demonstrate the function of the found concepts by means of eg. a CAD-model.
- To propose further work.
1. Project introduction and methodology

- To arrange a project presentation.

In the most successful scenario of the project some efforts can be added. These would be:

- To demonstrate the function of the concept with a simply built prototype.
- To present calculation models for the found concept/concepts to ease further work with dimensioning, validation and whatnot.
2. Risk analysis

Here the risk analysis matrix is presented. The severity and plausibility is set on a scale of one to three and the result being a number between one and nine. These risks are then coupled with actions for both preventive and damage limiting use. This matrix functions as a reminder of what to do in case a foreseeable problem occurs.

Table 1. The risk analysis matrix filled in.

<table>
<thead>
<tr>
<th>Risk</th>
<th>Plausibility</th>
<th>Severity</th>
<th>&lt;= Product of the two</th>
<th>Preventive actions</th>
<th>Damage limiting actions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Misunderstanding project goals</td>
<td>1</td>
<td>3</td>
<td>3</td>
<td>A thorough conversation with tutors and other staff at Bosch provides understanding of company values in terms of usable products.</td>
<td>Identify usable result and work with that.</td>
</tr>
<tr>
<td>No usable concepts generated.</td>
<td>2</td>
<td>3</td>
<td>6</td>
<td>Be careful when researching application functions and previous concepts and respective pros and cons. Pay attention to the benchmarking done.</td>
<td>Identify the problems of the concepts and the result that is usable. Propose improvements.</td>
</tr>
<tr>
<td>Calculations going wrong</td>
<td>2</td>
<td>2</td>
<td>4</td>
<td>Review work done by others, implement present knowledge in upcoming models. Consult experts in</td>
<td>-“-”</td>
</tr>
</tbody>
</table>

Table 1. The risk analysis matrix filled in.
2. Risk analysis

<table>
<thead>
<tr>
<th>Issue</th>
<th>CAD not working</th>
<th>Conflict</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Use in-programme tutorials and consult experienced personnel.</td>
<td>-&quot;-</td>
<td>Find out what caused the conflict and remedy by respectful discussion about this.</td>
</tr>
<tr>
<td>Tread carefully. Respect both others and own opinions. Argument objectively.</td>
<td>1 2 2</td>
<td></td>
</tr>
</tbody>
</table>
3. Need finding

Here follows the process of gaining insight into what functions that needs to be fulfilled.

3.1. Understanding the motors

The first step is to understand the motors so that the conditions are fully understood.

3.1.1. Working principles

The hydraulic motors produced at Bosch Rexroth in Mellansel uses rings with radial cam profiles on their inside coupled with rolls pressed radially outwards onto the ring’s cams by means of hydraulic pressure exerted on pistons holding the rolls as seen in Figure 2. By applying pressure as the roll travels over sections of the ring with increasing inner radius and relieving the pressure as the roll travels over sections of the ring with decreasing inner radius, a torque is driving the cylinder housing and the ring to rotate relative each other. By having multiple rolls and even more cams on the ring, the torque distributed over the full revolution can be kept even. The company’s earlier (and some still produced) motors has a fixed cylinder housing and a rotating cam ring on which the power output shaft was installed (such as their Viking and Marathon motors). Newer models however have their cylinder housing as power output shaft and their cam rings stationary (such as CA, CB, etc.) presenting variations in possible customer applications. By adopting technical development and by varying number of rings and rolls mounted in parallel, radiuses, various internal interfaces, output shaft interface, hydraulic connection interface etc., the company can deliver motors with a diversity in torque-speed characteristics to match whatever application that has a shaft to be turned.

3.1.2. Strengths

As said, the working principle is suitable to vary for different characteristics and in relation to its size, it can produce a very high output torque.
3. Need finding

3.1.3. Weaknesses
None of the existing models has an integrated brake. Installing a brake, should the customer need one, of any sort coaxial with the motor adds to application space requirements and weight as this part of the application has to be installed in a separate space with a separate housing. This also adds to cost and subsequently to customer price if the customer were to need one. This is why an integrated brake that has components in common with or more adjacent to the motor would be desirable.

As the motor is a hydraulic one it is also sensitive to heat. Excessive heat contribute to thinner oil with deteriorated lubricating properties during operation and also reduces oil life. In the long term a lack of lubrication shortens motor life as well.

Another drawback with being a hydraulic motor is the need to keep the oil free of flow restricting and abrasive particles that in obvious and similar ways also shortens motor life.

Both the need to keep the temperature low as well as the need to keep the oil free from particles are needs inflicting on the set of solutions when prospecting for alternatives for integrating a brake into the motor.

3.2. Understanding the previous and current brakes
Bosch Rexroth currently uses three different brakes suitable for different situations. The Viking motor with its outer rotor uses band brakes to brake the outer rotor. Multi disc brakes are used for the others where the rotor is the inner part.

3.2.1. The Viking band brake
The Viking motor has an outward facing rotor, which makes it suitable to brake with a band brake. The brake comprises two bands that can be turned in either rotational direction, see Figure 3. If the load is known to be applied in one direction only, both bands are turned that way to maximize the braking torque.

The bands are tightened by using a cylinder. The motor foundation is what holds the cylinders and the braking band from rotating with the motor.

Figure 3: Viking motor with band brakes. Note the different directions of the two bands.
3. Need finding

3.2.2. The MDA wet multi disc brake
With the entrance of the compact motor, CA, another brake had to be used. The MDA brake is mounted on the motor housing and is a wet multi disc brake. It is operated with a hydraulic pressure that keeps the brake disengaged. As the hydraulic pressure drops two mechanical bevel springs installed serially applies the brake. The brake is thus a fail safe brake. As the brake discs are submerged in fluid, the brake has power losses. This means that the brake is developing heat as the motor runs. The heat generated increases the surface temperature of the brake which calls for motor restrictions in order for the application to run within ATEX [8] environments. The hydraulic pressure applied to keep the brake disengaged also results in an axial force on the motor bearings, which is sorted out by adding an axial roller bearing seen in Figure 4.

3.2.3. The BICA dry multi disc brake
To solve this the BICA brake was developed. The BICA brake is a dry multi disc brake which thus doesn’t have as high drag losses. Hence the brake isn’t developing any heat which makes it more energy efficient and safer to use in ATEX [8] environments. This brake is also a fail safe brake that engages as the hydraulic pressure drops. In this brake the hydraulic pressure isn’t applying any axial forces on the motor bearings which further increases efficiency and eliminates the need for an axial roller bearing as seen in Figure 5. This brake is applied by multiple coil springs instead of the bevel springs used in the MDA.

3.3. Understanding the manufacturing methods
During a guided tour the current manufacturing and assembly methods were briefly presented. Most parts including the pistons, piston housing and motor housing are forged, lathed and milled. The cam ring steel is an ECR steel with a very low level of impurities, purchased in certain dimensions to ease manufacturing. The width and outer diameter are already set upon delivery and the steps taken in-house are cam profile milling, hardening and surface improvements. Piston rollers, springs, bearings and other standard components are bought.

Aside from the piston insertion during the piston block preassembly, most motor components are manually assembled in an axial direction at one single station with aid of a bearing press and other standard tools.
3. Need finding

The MDA and BICA brakes are preassembled before arriving the facility in Mellansel, while the band brakes are assembled in house. The brake assemblies are then installed at a station for accessory installment.
Due to the design of the MDA brake, both the brake and the motor must be partly disassembled prior to installation of the brake on the motors when done on site. This adds to manual labor and uncertainties concerning maximum brake torque.

3.4. Understanding the use of the applications
To gain more knowledge about customer applications and their needs, opportunities and restrictions, an interview was held with sales staff with long experience of the different market segments. During this interview information concerning where the hydraulic motor and its brakes are situated emerged. Both which applications they’re in and where in the application they’re placed. The load situations were also discussed both for motors and brakes. Typical operating cycles were also mentioned as well as maintenance actions. Distinction between different braking situations were sorted out. Information about the typical power take outs and application spaces was also dealt with.

The winches are typically used as an internal component in cranes for hoisting the different constituent structures of the crane or to wind the lifting wire with. These cranes are of varying size, lifting capacity and location around the working site. In cases where the load capacity demand for the crane is big, multiple winches are rather used than one big. This allows the use of thinner wires. A thinner wire allows bending in smaller radiuses without suffering from premature failure due to material fatigue as a wire of bigger diameter would have. For the application this means narrower winch drums, smaller motors and smaller brakes reducing the cost.

When placing cranes on board the ship the foundation space needs to be narrow to maximize cargo space. Rotation of the crane around the crane’s vertical axis takes place in the interface between the narrow crane foundation and the crane by means of a pinion/pinions driving a gear mounted on either side of the interface. The foundation is higher than the expected cargo height in order to make room for containers, movement of hatch coverings and so forth. Above this interface the crane increases in width and acts as protective housing for internal components such as winches for lowering and lifting the top beam of the crane and winding the lifting wire. In cases where the crane is situated on the dock, the space limitations aren’t as severe as on deck.

Electric power is mostly present on both ship and dock application which can be used to drive hydraulic pumps which in turn can drive the hydraulic motors. Pressurized air is seldom a readily available power source.

The lowering and raising of ramps on either ships or docks for loading ships are also done with hydraulic motor winches with the wires approaching the ramp from above. A high angle between the ramp and the wire produces lower wire forces due to trigonometry, causing the wire outlet to be placed highly with reference to the ship deck. In order to prolong wire fatigue life the winch drum is placed coincident with the wire outlet to avoid unnecessary bending that otherwise would add to cyclic loading of the wire.
Since the water and thus the ship is in constant motion with respect to the dock, the winch is actively regulating the force put on the ground or deck by the ramp by lifting and lowering as needed.
3. Need finding

When winding fishing nets on trawlers or towing boats with tuggers, winches are used. Safety demands are then somewhat lowered due to the nature of the activity with lesser safety consequences associated with snapping a wire. If only considering the brakes, a strong brake can be disadvantageous as unexpected heavy tugs can damage gear; it can even flip the boat if unfortunately arranged.

In dredging works the ladder on which the cutterhead are mounted are positioned with winches, both in elevation and rotation.

All so far mentioned applications are mainly used at sea or port in a marine and very corrosive environment, requiring the application to be well sealed or corrosion persistent. As marine environment also hosts life, applications stationed at sea needs to be environmentally friendly. This leads to ever increasing requirements on materials and oils to be biologically decomposable and non-toxic.

Winches are also used in industrial applications, foremost as belt tensioner in conveyor band applications as an alternative to weight stacks that for bigger sized conveyors would be too space demanding. A winch is then placed at the non-drive side of the band, pulling the tail end pulley to provide belt tension to maintain drive pulley traction on the belt.

The motors are often driven by hydraulic pumps driven by electric motors delivered by Bosch Rexroth, but can be driven by any pressurized flow of hydraulic fluid available. This also applies for brakes with the only difference being the pressure applied. Also, the brakes doesn’t demand the high flux of fluid as the motors does. The motors typically operates at 350 bar whereas the brakes are sufficiently pressurized at only 15 bar. Higher pressure contributes to higher axial forces leading to more bearing wear and is therefore not wanted. The brakes MDA and BICA are failsafe brakes, which means that the brakes are applied by spring force at hydraulic pressure relief and vice versa for the band brakes that the Viking motors are suited with.

When dimensioning a winch the load is the first thing to consider, these typically reaches between 5 and 150 tons but could be even higher. This gives the necessary thickness of the wire and how many of them that are needed. The thickness of the wire gives the required radius of the winch drum to avoid excessive bending of the wire. The radius of the drum combined with the load gives the torque demanded of the motor, which then with a safety factor gives the required braking torque. The brake properties are carefully regulated by a diversity of certifying organs spread across the globe, and since Bosch Rexroth has customers across the globe most of these organs are relevant to the company products. Luckily, most have overlapping and similar demands.

When in use, winch applications typically has to prevent load drops occurring at movement initiation as the parking brake opens and the hydraulic motor engages and vice versa at the end of the movement. This is handled by applying hydraulic pressure to the motor prior to brake disengagement to relieve brake strain before opening. Upon completion the brakes are applied prior to motor pressure relieving. This is done automatically by the operating system.

Winch applications meant to carry personnel need to have two independent emergency brake systems according to DNVGL [15] certification. Braking with the motor or parking brake are not valid as emergency brakes.
3. Need finding

The brakes needs to be able to stop a falling load within 0.5 s. The faster the brake applies, the lesser the energy is that the brake needs to handle, and the lesser the force is on the rest of the equipment. If the brake was to be applied later and the energy and forces involved were to increase, the braking would have to be more gentle not to snap the lifting wire or equipment. This results in a greater stopping distance and thus an increased risk of human and/or equipment injury.

In order to avoid excessive wear on pumps, motors and brakes the brake has to open within one second at the start of operation.

In case of emergency braking, the brake needs to be designed so that the brake manually can be gently disengaged in order to hoist the load down.

Typically brake calipers, discs, lamellae packages etc. is dimensioned big enough to deal with a worst case scenario load and then, if deemed necessary, the actuating forces are decreased to make the brake characteristics milder.

Customers need to maintain their brakes, so other valuable properties of brakes are to be easily understood, inspected, disassembled and assembled. At the same time the whole application is appreciated if it is compact and not intruding on space used for other purposes. After a certain number of emergency brakings, the brake needs to be inspected or rebuilt. Very robust and capable brakes can be inspected and rebuilt more seldom, also adding to customer values.

During the interview the importance of a low cost for marine applications was emphasized over and over. One way of avoiding cost is if the motor’s bearing can act as the bearing of the application and thus eliminate the need for extra components that otherwise would’ve added to the price. The Viking motor is an example on how the motor eliminates the need for extra components as the rotor, radially being the outer part, directly can be bolted onto the winch drum without the need for any adapter. The band brakes used with this motor are installed around the rotor without the need for any additional drum or disc as being the case with BICA and MDA brakes.

3.5. Needs found

In the project description, during the process of understanding the motors (section 3.1 Understanding the motors.) and the brakes (section 3.2. Understanding the previous and current brakes.), during conversation with the company (section 3.3. Understanding the manufacturing methods and section 3.4. Understanding the use of the applications), during reading certification documents [14][15] and while reviewing previous work done [1][2] a number of needs have emerged. Some of these are quite vital for the applications while others may be interpreted as advantages adding to customer value. These needs must be reflected upon, sorted and ranked to later form the specification of demands and aid the decision making.

The brake needs to

- Be able to operate over the whole speed interval of the motor.
- Be able to exert a braking torque 1.8 times greater than the maximum torque of the motor.
- Bring the load to a halt within 0.5 s.
- Deal with the heat generated from braking.
3. Need finding

- Be cheap to produce.
- Be resistant to marine environment.
- In combination with the motor, be able to deal with load drops that occur during operation.
- Not affect the motor life.
- Be non-intrusive on working space.
- Be easily inspected.
- Be easily serviced and rebuilt.
- Disengage within one second.
- Have a long life-span.
- Be compatible with relevant power supplies.
- Be energy efficient.
- Have predictable, even and suitable brake characteristics.
- Be easily installed.
- Be highly reliable.
- Be easily adjusted.
- Be simple.
- Be readily modularized.
- Work well with product architecture.
- Be easily engaged
- Be easily disengaged
- Have an emergency release function with suitable characteristics.
- Be environmentally friendly.
- Have an indicator showing brake engagement.
- Be able to deal with radial and axial forces
- Preferably acts directly on the winch drum.

Here the same properties are sorted into needs that are either critical and beneficial.

The brake has to:

- Be able to exert a braking torque 1.8 times greater than the maximum torque of the motor.
- Bring the load to a halt within 0.5 s as an emergency brake.
- Hold the load still as a parking brake.
- Deal with the heat generated from braking.
- Be resistant to marine environment.
- In combination with the motor, be able to deal with load drops that occur during operation.
- Operate without affecting motor life.
- Disengage within one second.
- Be compatible with relevant power supplies.
- Have suitable brake characteristics.
- Be highly reliable.
- Have an emergency release function with suitable characteristics.
- Not be easily interfered with.
- Function well, even with humidity, oil or other contaminations entering the brake.
- Spring-loaded braking pads or discs shall be loaded by pressure springs.
3. Need finding

- Checking wear of brake pads or discs shall be possible without dismantling the braking unit.
- Self-blocking brakes are only admissible for stowage or idle positions.

It is of additional value if the brake:

- Is cheap to produce.
- Be non-intrusive on working space.
- Be easily inspected.
- Be easily serviced and rebuilt.
- Have a long life span.
- Be easily installed.
- Be easily adjusted.
- Be simple.
- Be readily modularized.
- Be environmentally friendly.
- Be able to operate over the whole speed interval of the motor.
- Isn’t deteriorating the winch’s energy efficiency.
- Preferably acts directly on the winch drum.
4. Competitor study and benchmarking

In studying competitors to compare the brakes available on the market with brakes used by Bosch Rexroth, brake data is gathered and visualized in the following graphs. These are providing an understanding as to what has to be achieved with the concepts that are developed in this project.

In cases with brake calipers installed on brake discs, the assumption is made that the disc mean radius is 0.4 m. The caliper dimensions decides how many calipers are used on this brake disc. For more precise data on how much torque each brake can deliver, another evaluation is needed.
Figure 6. The braking torque delivered by some brakes (brakes currently used by Hägglunds in yellow and other brakes in black) and the braking torque demanded for use with Hägglunds hydraulic motors (red). M, D and Q is for 1, 2 and 4 (Mono-, Dual- & Quad-) pistons/caliper respectively where applicable. The brake torque is presented on the vertical axis and brake models are scattered along the horizontal one.

In Figure 6 it can be seen that there’s quite good availability of brakes for lower levels of torque while the bigger motors doesn’t have that many matching brakes.
4. Competitor study and benchmarking

Figure 7. Axial dimension including service space if stated by the manufacturer for some brakes. Brakes currently used by Hägglunds in yellow, others in black. The axial dimension is presented along the vertical axis and the brake models are scattered along the horizontal one.

Figure 7 tells us that the brakes used by the company is among the brakes with the smallest axial dimensions. The other brakes found are of bigger axial dimensions but the smaller ones are rarer.
Figure 8. Radial dimension including service space if stated by the manufacturer for some brakes. Brakes currently used by Hägglunds in yellow, others in black. The radial dimension is presented along the vertical axis and the brake models are scattered along the horizontal one.

As in the case with axial dimension, Figure 8 tells us that many brakes with larger dimensions exist. Finding brakes that are radially smaller than those already used can be hard. Note that the brakes listed in Figure 7 and Figure 8 are the brakes found during this search with a given dimension. Many more brakes were found as seen in Figure 6, but many of them came without dimensions.
5. Literature study

Brakes in general have a number of functions to fulfill. These are to dissipate the kinetic energy of the components braked, store or transmit the energy absorbed and hold the braked components fixed [3][4][5]. Stopping often is a crucial safety consideration requiring the brake to be designed and dimensioned to do so even under misfortunate conditions [3], e.g. partial brake system failure, power failure etc. Often this is achieved and even prompted to be achieved by means of redundancy [14][15]. Brakes can also work over a transmission. Using a transmission between the brake and the axle to brake permits torque- and speed converting and thus the use of components dimensioned to deal with different situations in terms of torque and speed.

Braking can be achieved in numerous ways; mechanically, hydraulically, electrically etc. and there is variation within all of these ways. [3][4][5][16][17]

5.1. Mechanical brakes

The mechanical ways of braking in cases including rotational motion often comprises a rotating body attached to the element braked and a braking body attached to the carrying structure. Upon contact between these two kinetic energy is transformed into heat as friction between the two restricts the movement and deforms the bodies and any material in between, i.e. third body wear particles, lubricating fluids etc.[3][4][5][6][17]. The contact is optimized for friction, longevity [3] and predictability [6] as seen fit by e.g. variation in materials choice[3], contact surface area[4], material volume [6], applied normal force [4] and whatnot. The torque generated is optimized geometrically or by varying the force applied. These variations and methods of actuation also have influence on torque controllability [4].

5.1.1. Disc brakes

One brake configuration is disc brakes in which multiple discs or a single disc is installed in such a way that it rotates dependent on the rotation of the axle braked. One or many calipers are mounted floating about each disc, suspended onto the carrying structure so that it cannot move peripherally or radially. In some applications the calipers are completely fixed and cannot move even in axial direction with respect to the disc [5]. These calipers are equipped with frictional pads, one for each side of the disc, that the caliper squeezes onto the disc and thus provides friction. This squeezing force can be applied mechanically, hydraulically, pneumatically, electrically or by a combination of these in series or in parallel[5][7].

5.1.2. Drum brakes

Drum brakes or shoe brakes instead applies the braking bodies or shoes in a radial direction. In these cases the disc is replaced with a drum in which the braking bodies are installed. Upon actuation the braking bodies are pressed onto the drum and thus providing a frictional force[5]. As the braking pads are conform with the drum, restricted from peripheral movement and suspended on one end peripherally, the actuation implies that the shoe rotates around its suspension and an interesting phenomenon occurs. When the drum rotates in one direction the braking is amplified by drum torque and vice versa in the other direction. This provides a design opportunity in how to install the braking pads, either the brake can work better in one direction or it can work as good in any direction as in the other. This opportunity also comes with a drawback in controllability[3].
5. Literature study

5.1.3. Cone brakes
There are brakes that have a conical drum and the braking body is another cone applied axially into the conical drum; cone brakes. Such brakes have a geometrical advantage in that the applied force is amplified by the cone inclination due to trigonometrical conditions. As the pressure is deforming both bodies during braking, disengagement after rotational stop is counteracted by the material stress built up by actuation. The contact pressure is the same as during dynamic braking but as the static friction is a greater force than the dynamic friction, a greater disengagement force is required to overcome this retaining force[3].

5.1.4. Multi-disc brakes
Multi-disc brakes comprising discs where every other disc is dependently rotating with the axle and the rest of the discs are suspended by the carrying structure is another way of braking. These are actuated by making the discs slide axially towards each other providing braking torque. The number of discs used in such brakes is a key parameter in dimensioning for torque as the torque transmitted is proportional to the number of disc interfaces for a given set of actuating force, radius and material properties[3]. The fluid between the discs is having great influence on maximum torque transmittable and actuation characteristics. Choosing air as fluid provides a nearly drag-free brake while it’s disengaged[9] while different oils provide a smoother actuation, allow slip [12] and a more effective cooling[3].

5.1.5. Hydraulic braking and hydraulic brakes
There’s different ways to brake hydraulically, one of these ways is regenerative braking using the driving motor, another way is to have a pump especially designated for braking and then there’s brakes based on varying rheology [7]. The hydraulic ways of brake actuation aren’t discussed here.

5.1.5.1. Braking with the driving motor
To brake hydraulically driven units the motor itself could be used to resist movement[7]. The pressure built up by the braking motor could either be stored for later use, let out on any hydraulic grid or dissipated as seen fit.

5.1.5.2. Breaking with a pump
The principal of this is to use the rotating energy to drive a pump raising the pressure on a hydraulic flow over a flow restriction, thus converting the rotating energy into heat[7].

5.1.5.3. ER- & MR brakes
Then there’s brakes and clutches functioning with changed rheology within a fluid acting between a body dependently rotating with the axle and another one suspended by the carrying structure distributed axially, radially or in any other way. The change in the fluid done by applying a magnetic field on magnetorheological (MR) fluid[11]; by applying an electric potential on an electrorheological (ER) fluid[16] or in any other way restricts shearing and thus relative rotational movement of the rotating bodies. Due to the high rate of change with respect to time of the electric potential or the magnetic field and thus the fluid viscosity, these types of clutches and brakes have very short response times. [11][16]
5. Literature study

5.1.6. Electric braking and electric brakes
In cases where electric motors are used they can also be used as generators creating a braking torque[7]. As in the case with hydraulic braking, the electric charge can be either stored for later use, let out on any electrical grid or dissipated as seen fit.

5.1.7. Cooling
The cooling of the brakes is crucial since the braking power often is very high and the frictional heat isn’t applied uniformly but on certain areas. Brakes often reaches as high temperatures as 700°C and temporarily on the friction interfaces even higher. While being of steel, this comes with a risk for phase changes and stresses leading to failure. In some applications fire hazard and fluid degradation are also limiting factors. [3][5]

5.1.7.1. Failure modes due to heat
Some failure modes are presented here. Note that these can occur in combination.

5.1.7.1.1. Braking surface material loss
As a mechanical friction brake steel surface heats up rapidly beyond austenitization temperatures only to then be quenched by the rest of the material, martensite is formed. Further use of the by then brittle brake can fracture the material surface leading to a higher wear rate and deteriorated brake characteristics. [3]

5.1.7.1.2. Brake disc cracking
Another risk of rapid heating is that the disc material on the disc peripheral area temporarily heats up more than the disc center. This means that the added circumference of the brake is distribute on the less enlarged radius, making waves with axial displacement on the friction surface. The increased tension cracks the disc and produces vibrations during further braking that deteriorates brake pad life and brake characteristics. [3]

5.1.7.1.3. Fading
The brake provides a torque by means of a friction force between the friction material in the brake lining and the disc/drum material. The force generated also needs to be dealt with by the lining resin. As the brake pad is subjected to high temperatures the resin fails and the shear strength of the material decreases and is thus more easily sheared away. When this happens the braking torque is reduced for a given force applied. [5]

5.1.7.1.4. Glazing
The opposite problem to fading; glazing, occurs when the brake lining isn’t subjected to enough heat. While braking mildly under low temperatures, the wear particles aren’t disposed of and instead clogs the friction surface, deteriorating the contact between liner and disc/drum. This could be interpreted as a too light usage of too large friction surface area. [5]

5.1.7.2. Some general cooling solutions
In order to deal with all the problems that comes with excessive heat, the source of heat must be cooled. This can be achieved in numerous ways. The most
common ways relies on either conductive or convective mechanisms done with various convective fluids or conducting materials.

Brakes used only for holding the structure fixed generally do not generate any heat as everything has come to a stop before the brake is engaged, and therefore does not have to be cooled.

If using the brake as an emergency brake that seldom is used and thus is having sufficient time to cool down even without design features promoting energy dissipation, energy can be stored within the structure material.

When using a brake more frequently so that the heat generated from each braking isn’t fully dissipated before the next braking occurs measures have to be taken, and what else is there besides enhancing dissipation if storing isn't enough?

5.1.7.2.1. Conductive cooling
In structures comprising conductive materials and masses big enough to store the heat generated without reaching application critical temperatures, conduction cooling can distribute any heat generated. As it is simple this method has a wide use, often combined with other cooling mechanisms to dissipate the energy stored. [5]

5.1.7.2.2. Self-convection air cooling
By leading the heat away from its origin by either directly exposing the source to a fluid flow such as air or any other suitable gas, or by conductively leading the heat to a secondary cooling by convection, the heat generated can be dissipated. As this also is a simple and reliable way to cool devices and sometimes also sufficient, it is widely used, also in brake cooling. If a design promotes convection cooling, savings could be done as the parts involved doesn’t have to store as much energy as in a case with less convection cooling. These savings might be in terms of material mass used, space utilization and energy efficiency, depending on application. [18]

5.1.7.2.3. Forced-convection air cooling
If self-convection isn’t enough, the convective mass flow could be forced to higher flow rates by means of e.g. a fan to further increase the heat transferred. This is cost driving but highly effective and frequently used. As such systems mostly are relying on readily accessible low density fluids such as air and most flow driving solutions are of light weight and small volumes they are a solution to many problems. [18]

5.1.7.2.4. Self-convection liquid cooling
As with air, liquid mass flow can also be driven by self convection. While adding a fluid such as a water-based coolant to the system enhances the thermal energy distribution, it also increases the material mass and thus the thermal energy capacity, lowering the application temperature. Dissipation can also be enhanced by increasing interface area for the subsequent convection to the surroundings as when using an external cooler. Experimenting with a phase changing liquid and capillary effects, Kim & Kim [13] found that if replacing conductive fins for self convective air cooling,
5. Literature study

Liquid cooling can be less space demanding and still a cooling enhancement. [18]

5.1.7.2.5. Forced-convection liquid cooling
To further increase the heat dissipation rate from those of self-convection liquid cooling systems, a pump can be added. Similarly to forced-convection air cooling it is cost driving and not as reliable as a passive system, but highly effective and allows for a decrease in coolant volume, application space usage and weight for a given heat flux. [18]

5.1.7.2.6. Electrical cooling
Peltier found a way of electrically forcing conduction. Peltier elements can be used for cooling heat sources by applying an electric current over the element in thermoelectric coolers (TEC), or they can be used as a thermoelectric generator (TEG) if a heat flux is applied over it. However, the technology still struggles with low efficiencies and isn't considered more in this work. [19]

5.1.8. Materials
In designing brakes of the mechanical kind, much interest is placed in choice of material, especially considering linings and multi-disc brake lamellas. The choice of material has an influence on frictional behavior and also states what forces can be applied.

5.1.8.1. Brake linings
Teflon (PTFE and TFE) are materials that exhibits low coefficients of friction and could be used up to $260^\circ C$. By adding reinforcing fibers such friction materials are made more stress durable and also more abrasive, increasing the friction coefficient. These are suitable for clutches limiting transmitted torque in different applications. Their frictional behavior is seems independent of sliding speed and could deal with application pressures of up to $7 \text{ MPa}$. The coefficient of friction then varies between 0.2 and 0.5 with different fillers. The highest coefficient of friction is seen with ZrO$_2$.

Mineral based friction materials may be used in either wet or dry conditions, their coefficient of friction ranges between 0.1 and 0.6. They can operate in temperatures at up to $600^\circ C$ and can deal with pressures of up to $2.41 \text{ MPa}$. This makes the material suitable for applications where high torque is needed with low engagement force. Due to the fact that some of the resins used in these materials are hazardous any debris must be collected and removed.

Sintered metal materials can provide good frictional performance with high coefficients of friction, even at high temperatures and while fading. A proprietary material, HF-61 shows a coefficient of friction at approximately 0.6 during fade at up to $600^\circ C$.

In Dry clutch control for automotive applications [20], P. J. Dolcini et al. speaks of the technologically feasible static coefficient of friction as being 0.4.

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5.1.8.2. Discs and drums
Maleque et al. [21] speaks of several brake disc materials:

- Cast iron, a material that is cheap and paves way for easily manufactured disc brake that performs well.
- Titanium alloys, a more expensive material that promotes weight reduction.
- Aluminum metal matrix composites, AMC, which is aluminum based materials with added ceramic reinforcements to enhance material strength and wear resistance.

When comparing the materials, Maleque et al. finds that cast iron performs well considering friction, cost, weight etc.
6. Specification of demand and decision making

Here the combined specification of demands and decision matrix are presented so that the reader can understand both the algorithm used to compare the found concepts and the importance of each parameter mentioned earlier in this document.

### 6.1. About the matrix

![Image of matrix]

*Figure 9: Specification of demands and decision matrix. Line 3, 11, 21, 25, 32 and 37 are main topics, the other lines are subtopics. Column D specifies the weight to distribute within each main topic. Columns E and F specifies the internal relation among the subtopics. Column G specifies the actual weight of each subtopic. Columns H and J is where the score for each concept are placed. Columns I and K is where the result emerges. Line 43 sums the results.*
6. Specification of demand and decision making

The demands listed in Figure 9 are sorted into a number of main topics. These are weighted against each other in a way so that the ratio between the weight numbers in corresponding row and in column C: Weight to distribute all makes sense from a customer and company value perspective. The higher the number is the more important the corresponding topic is. These main topics are the divided into more specific topics which in a similar way are compared and weighted. The ratio between the numbers in column D: Internal weight within each group should, again, make sense from a customer and company value perspective. The individual weights in column D are then evaluated as a share of the sum of all weights of all subtopics under that main topic. The share of the weight is presented on corresponding rows in column E: Weight share. This multiplied with the weight of the corresponding main topic is presented in column F: Weight.

As the concepts are evaluated by scoring each of these subtopics in corresponding rows in column G: Score concept 1, the score is multiplied with the corresponding Weight from column F and divided by the maximum score, 5 in this case, which is producing a result in column H: Result concept 1. A concept with the highest score thus has a result column identical to column F: Weight.

More concepts are evaluated in the same way in column I and forth. The concept result can be read and compared for each evaluated concept as either a total result of all main topics in row 42, as a result on a specific main topic in the rows 2, 10, 19, 23, 27, 31 and 36 or even a specific subtopic on all other rows.

This produces a specification of demands and a decision matrix where the main topics and subtopics are distinguished and kept apart so that the internal weighting among the subtopics in one group and the number of them doesn’t inflict on the inter main topic weight ratio, and the same goes for how the inter main topic weight ratio inflicts on the subtopics. In other words adding another subtopic under a main topic doesn’t lessen the value of subtopics in another group, as would be the case without this distinction.

6.2. About the topics
Here the main topics of the specification of demands and decision matrix are presented, they and all their respective subtopics beneath are explained. The numbers starting all headlines are the cell ID in the Specification of demands excel-file.

6.2.1. B3: Brake torque characteristics
As being a brake, being able to brake is of vital importance. In relation to the other main topics, this topic should have the highest weight.

6.2.1.1. B4: Static braking torque
The static braking torque is the maximum braking torque that the brake can exert on the shaft while the shaft has no rotational speed. This is relevant while the winch is not in use but is holding a load. This is a parameter of great importance since it is a safety issue and is strictly regulated to be at least 1.8 times the maximum motor torque as mentioned in Need finding. In relation to the other subtopics this topic should have a weight slightly above medium.

6.2.1.2. B5: Reliable static braking
Even more important is the brake’s reliability to hold a load stationary. Even if the brake was to be stronger in terms of Static braking torque, it would be a worse brake if the reliability of it was deteriorated. Reliable static braking in relation to
6. Specification of demand and decision making

*Static braking torque* should have the weight ratio 10:7, i.e. *Reliable static braking* should be weighted 10 while *Static braking torque* should be weighted 7.

6.2.1.3. B6: Dynamic braking torque
Being able to brake a falling load is the primary task for an emergency brake in a winch application, and the torque available for doing this is of the same importance as the torque available while the load is stationary. This should result in the weight ratio between *Static braking torque* and *Dynamic braking torque* being 1:1, i.e. this parameter should be weighted 7.

6.2.1.4. B7: Predictable dynamic behavior
As a parking brake would need reliability, an emergency brake would need a predictable dynamic behavior and the relation in these two pairs should be the same. But this project is of limited time and the dynamic behavior of a brake is more complex than ever could’ve been coped with during this time, even more complex than deciding the reliability of a brake used in static load cases. A not known parameter shouldn’t have as much influence in decisions as a known one so the weight is slightly decreased. The ratio is thus 9:7 between *Predictable dynamic behavior* and *Dynamic braking torque* instead of 10:7 as between *Reliable static braking* and *Static braking torque*. This means that *Predictable dynamic behavior* should be weighted 9.

6.2.1.5. B8: Small backlash
Load drops when using winch applications are a known problem within Bosch Rexroth as described in need finding. It is a safety problem as well as a comfort problem which desirably would be eliminated. This would mean that the brake would need to have a smaller backlash than currently used brakes experiencing load drops. The importance of this should not be discarded but still it is not near being as important as *Reliable static braking*. The weight ratio between *Reliable static braking* and *Small backlash* should be 10:3, i.e. the weight for *Small backlash* should be 3.

6.2.1.6. B9: Function well with contaminations
Whether a brake functions well with contaminations or not is highly dependent on choice of material which is not known in this project. As previously argued in the case with unknown parameters, these can’t have to much influence in decision making. This speaks for putting a low weight on this parameter, such as 1. Further development of the concept in coming projects where choice of material is known will however sort this out. If deemed reasonable at that stage, the brake’s ability to work even though contaminated might have a bigger influence on decision making as this could be argued as being important.

6.2.1.7. B10: Adjustable braking torque
A brake that brakes to hard in relation to the load lifted and the associated lifting cable dimensions in an emergency brake situation risks damaging the cable and/or the suspending structure. On the other hand it would also be dangerous if the brake wasn’t breaking hard enough, for obvious reasons. It is therefore desirable that the brake torque can be adjusted to suit every application. The adjustability of the brake is deemed to have weight number 5.
6. Specification of demand and decision making

6.2.2. **B11: Engagement mechanism**
The way the brake is engaged is determining how it can be used. It decides whether a brake should be used as a failsafe brake or as an active one. It says something about the reliability of the brake and it could decide its characteristics. To say that the engagement mechanism in relation to the brake’s torque characteristics are in weight ratio 7:10 seems fair. This means that the *Brake torque characteristics* should be given weight number 10 while *Engagement mechanism* should be given a 7.

6.2.2.1. **B12: Self adjusted when worn**
A brake that adjusts itself as the brake pad is worn both saves the customer the effort of adjusting it manually and also provides a safer braking function in case the occasions of maintenance are seldom or random.

6.2.2.2. **B13: Low engagement force**
A low engagement force means that the forces acting upon suspending structure are lower and any auxiliary gear doesn’t have to deliver as much pressure, force or whatnot to engage or disengage the brake as in a case with an engagement mechanism that’s needing a high engagement force. This opens up for the application being lighter, cheaper to produce and easier to install. The company is experiencing the need for having some bearings in the motor dimensioned bigger in order to cope with the forces that some brake engagement mechanisms exert on the motor. This also means lower motor efficiency.

With that said, comparing the importance of the engagement mechanism being *Self adjusted when worn* and demanding a *Low engagement force* it would be fair to set the ratio to 9:7, meaning that *Self adjusted when worn* gets the weight number 9 and *Low engagement force* a 7.

6.2.2.3. **B14: Low disengagement force**
The disengagement force has the same implications on suspending structure and auxiliary gear as the engagement force has. It would also be somewhat inconvenient if the disengagement force were to be higher than the engagement force. If a user one day by some reason were to become hell bent on applying that brake with maximum effort, it might result in a brake that simply won’t disengage. As annoying as that might be, having a low engagement force still is more important as the opposite situation even might be dangerous. The weight ratio between *Low engagement force* and *Low disengagement force* should be 7:5, giving *Low disengagement force* a 5.

6.2.2.4. **B15: Short engagement time**
Keeping the engagement time short means a smoother operation without load drops when using the brake for holding, and in an emergency brake situation it means that a falling load won’t have the time to accelerate and thus causing the brake to have to work harder. Bosch Rexroth even gave a sharp demand concerning this: the brake have to engage within a half second. This is even more important than having a *Low engagement force*. The weight ratio should be 7:10, giving *Short engagement time* a 10.
6. Specification of demand and decision making

6.2.2.5. B16: Short disengagement time
Keeping the disengagement time short is also contributing to a smoother operation but isn’t as important as keeping the engagement time short. The weight number should be half of that of Short engagement time, giving Short disengagement time a 5.

6.2.2.6. B17: Simple engagement
That the engagement mechanism is simple means that it is reliable. And simpler constructions are also cheaper to manufacture and assemble. Compared with Short engagement time, this isn’t as important but still important. The weight ratio should be 10:8, giving Simple engagement an 8.

6.2.2.7. B18: Simple disengagement
Having a simple disengagement is less important than having a simple engagement from a safety point of view. But in the same way as a Simple engagement might reduce cost, Simple disengagement also does. The weight ratio should be 5.

6.2.2.8. B19: Failsafe
That the engagement mechanism is of a failsafe type is crucial for the use in most winch applications as it is regulated by certifying organs such as DNVGL [14][15]. It is important enough to give this the highest weight, thus the weight number should be 10.

6.2.2.9. B20: Gentle emergency release
In order hoist the load in a case of emergency, power out or similar, the brake needs to have a way to be manually disengaged. And in order for the function to be safe and be called emergency hoist and not emergency drop, the engagement mechanism needs to provide a possibility to do this gently. This is as important for the brake as being Failsafe and should be given the weight number 10.

6.2.3. B21: Space intrusion
That the brake doesn’t inflict on the customer working space is of great importance as some applications where winches are used doesn’t leave a lot of it available. It is however less important than, say, the Brake torque characteristics or having a good Engagement mechanism which are weighed 10 and 7 respectively. The weight number should be 3.

6.2.3.1. B22: Small axial dimension
The axial dimension, in the motor’s and winch’s reference system, is where the brake has the most potential to be space intruding and thus the axial dimension should be kept low.

6.2.3.2. B23: Small radial dimension
Less important is the radial dimension as a brake equally big and somewhat conformal compared to the adjacent parts radially, i.e. flush with the motor’s och winch drums outer edges, are likely not to be much more space intruding than the motor and winch already are. Comparing the importance of a Small axial dimension and a Small radial dimension the weight ratio should be 7:4, giving them 7 and 4 as weight numbers respectively.
6. Specification of demand and decision making

6.2.3.3. B24: No need for large external devices
An external device placed in the vicinity of the application would without hesitation be space intruding. What could mitigate this would be if the external device were to be flexible in its positioning but still it is a major concern, equally problematic as a big axial dimension. This means that the weight number should be the same as that of Small axial dimension, namely 7.

6.2.4. B25: User interface and serviceability
A simple use and maintenance of the application adds to customer value as the possibility to use and service the application him-/herself could reduce the need for time consuming production stops and costly services by external companies. It is more important than Space intrusion but not as crucial as a good Engagement mechanism which has 3 and 7 as weight numbers. Being in the middle of these two gives User interface and serviceability a weight number of 5.

6.2.4.1. B26: Not easily tampered with
For the brake to be reliable it needs to be resistant to sabotage and unfortunate mistakes. For example any adjustment screws adjusting the brake torque must be fairly unavailable without some work. This is also stated by DNVGL [14].

6.2.4.2. B27: Easily inspected, maintained and rebuilt
While being of importance that the brake is Not easily tampered with, it’s also important that the brake is easily serviced etc. as argued for in B25: User interface and serviceability. When comparing Not easily tampered with and Easily inspected, maintained and rebuilt it make sense that the weight ratio between these two should be somewhere about 3:7, giving them those very numbers as weight numbers.

6.2.5. B28: Environmental interaction
The line of trade is seeing increasing restrictions on allowed emissions. This topic is about by which means the brake can be operated, hydraulics, pneumatics, electricity and so forth that could result in emissions in a case of a failure. It’s about its emissions that it produces during regular operation, for example any emissions from its brake pads. It is also about how the brake copes with what the environment throws at the break. As many winch applications are of naval nature, sea water is an issue that the brake has to deal with. The brake can of course be isolated from nature by encapsulation for instance and a concept doesn’t necessarily dictate the choice of material. This lessens the influence of this topic as this project isn’t presenting a final product. The weight number should be in between that of Space intrusion and User interface and serviceability, namely 4.

6.2.5.1. B29: Weather resistant
That the brake’s torque isn’t affected by the weather put upon it is a safety concern and therefore of great importance.

6.2.5.2. B30: Connection compatibility
Even if a way of energizing the brake is proposed, the choice of medium isn’t necessarily dictated by the concept. A hydraulic cylinder might in many cases be switched in a later project to a pneumatic one for instance. With that said this should not have a great influence on decision making. Though it would be
6. Specification of demand and decision making

desirable that the brake concept is flexible and could be operated with the same oil as used for driving the motor or any other conveniently available power source or easily adapted as environmental restrictions change.

It seems fair to say that this is half as important as the brake being Weather resistant, thus giving Weather resistant the weight number 2 while Connection compatibility gets a 1.

6.2.5.3. B31: No environmentally hazardous emissions
As argued for Connection compatibility the choice of material isn’t finally set by the concept and shouldn’t affect decision making in a such early stage. The weight number should be 1.

6.2.6. Motor interaction
Any component installed adjacent or coaxial to the motor could affect or be affected by the motor. This could result in shorter lifespan for either one, cause lower application efficiency etc. When comparing the importance of this to the importance of other main topics, the weight number of this should be between that of Engagement mechanism and of User interface and serviceability. This results in the weight number 6.

6.2.6.1. B33: Low motor contamination risk
A brake sharing oil or volume with the motor risks contaminating the motor with wear particles, fluids etc. via leaking flanges, gaskets and so forth. To keep the motor free from contaminations is of big importance as the motor is an expensive product.

6.2.6.2. B34: Low drag
A drag is a constant wear of the brake and also causes the motor to have to work harder which increases to need for service and decreases life span of both motor and brake. This also produces heat which must be kept down to keep the hydraulic fluid at a suitable temperature. It must also be kept low in order to keep the machinery surface temperature at a suitable level as demanded for certain applications to be able to run in explosive environments, see ATEX regulations [8].

This is of as big importance as having a Low motor contamination risk, they should have the weight ratio 1:1.

6.2.6.3. B35: No motor action needed when disengaged
In seen in cases, for instance in the case with the brake developed by the SIRIUS-project in 2007 [2], a brake could need motor action when disengaged. This severely affects the possibility of having a functional emergency release and should thus be avoided. This is of equal importance as the previous subtopics in this group, giving a weight ratio of 1:1:1.

6.2.6.4. B36: Efficient cooling/energy storage
It is important to keep the machinery temperature low, either by having a cooling system powerful enough or having a heat storing capacity big enough within the comprising components, see 5.1.7. Cooling.

Given that the concept sought is a concept that is only producing heat during emergency braking, this makes it reasonable to expect that there is enough time after the heat production for the application to cool down even if relying on energy storing to a higher extent than on a cooling system. This makes this topic
6. Specification of demand and decision making

less important, perhaps half as important as the rest of the topics within this group. Thus the weight ratio should be 2: 2: 2: 1, giving them those very numbers respectively.

6.2.7. B37: Cost
As a company striving for lowering the manufacturing costs in order to make the products a more viable option for more customers, Bosch Rexroth is having Cost as a high priority, second only to safety which in this case is mostly represented by Brake torque characteristics. The ratio could be argued to be 5: 4. With Brake torque characteristics having a 10, Cost should have the weight number 8.

6.2.7.1. B38: Low weight
As a lighter product takes less material to produce than a heavier one, the mass of a product is a good indication on what the cost of a product is.

6.2.7.2. B39: Readily modularized
A good modularization of a product opens up the possibility for larger volumes of each product to be produced as they could be fitted with a larger number of other products. As in this case where a good modularization could make one model of brake swiftly could be fitted to a bigger number of motors. Both in terms of brake-motor-interface and torque.

In comparison with Low weight this has the double importance.

6.2.7.3. B40: Long service intervals
Much customer value is added by designing a brake with long service intervals as this reduces cost for both material and time as it at the same time increases application uptime. This is as important as having a Readily modularized product, but is less demanding during the concept design phase so a slight weight reduction in the decision making is in place. The ratio between Low weight, Readily modularized and Long service intervals should be 5: 10: 9, giving them those numbers as weight numbers.

6.2.7.4. B41: Simple or standard components
The cost for manufacturing a simple and/or standard component compared to the cost for manufacturing a special and/or advanced component is obviously smaller. Standard components produced in large volumes by others on the market are often for sale eliminating the need for having a production facility for these. Having more Simple or standard components in the concept is therefore advantageous. It seems reasonable to say that the weight number of this should be in between those of Long service intervals and Low weight, giving this the weight number 7.

6.2.7.5. B42: Easy assembly and installation
The time it takes and what tools are needed in the assembly and installation of the product also drives cost and decreases the rate in which the products can be made. It’s not necessarily forcing the company to extend the factory floor and inventories substantially so this should have a slightly lower weight number than Simple and standard components. 6 might suit.
7. Concept generation

By looking at the problems and needs present and the available solution space, multiple brainstorming sessions are held. The concept ideas and ideas of specific parts of such concepts generated during these sessions are then organized in such ways that the needs are satisfied. These concepts are then refined by drawing or CAD-modeling until they’re comprehensible enough to be either accepted or dropped in a rough screening considering their properties soundness and a rough estimation on whether they’re achievable or not.

It is written [10] that the primary preparation phases of any engineering project is of considerable importance, and that they should be done carefully before jumping into concept generation to avoid proceeding with misinterpreted project goals and purposes. But to avoid only finding the already found ideas and miss the opportunity to come up with something new the concept generation at this occasion was divided into two time periods. The first one being at a quite early stage in the project after only having done some light investigation on customer needs, motor functions, currently available brakes and seeing the methods of production. After having done the primary phases more thoroughly, now with more insight, more ideas were generated to cover the solution space to a wider extent. This method was tried in order to make this project as rewarding as possible in terms of generated concepts since this is one of the higher ranked goals of the work.

During the first sessions of brainstorming the following ideas were found.

7.1. Hydraulically regenerative brake

The hydraulically regenerative brake idea as depicted in Figure 10 is a method of braking that is used. In conversation with the sales personnel at Bosch Rexroth it emerged that on docks where multiple cranes are used, the energy regenerated as one crane lowers its load is used by the other one to lift. At such occasions the energy is transferred via their respective electrical motors driving their hydraulic pumps and the electrical grid used on site. That differs from the idea at hand since the thought here is to store the pressurized hydraulic fluid in a hydraulic accumulator for the hydraulic motor to use later as it is about to lift.

Figure 10: Hydraulically regenerative braking with the hydraulic motor.
7. Concept generation

7.2. Multi mode drum brake

This idea is about a multi mode drum brake using one mode as a self energizing parking brake (upper left in Figure 11) and the other mode as a non self energizing emergency brake (upper right in Figure 11). To switch between the modes an additional rod holding the rod with the brake pad on the other end (left ends in Figure 11) is turned about a shaft at its lower end. The rod holding the brake pad is pushed upwards against the brake drum by a roll to engage the brake as either a parking brake or as an emergency brake depending on mode.

Figure 11: Multi mode drum brake. The blue position is for directional parking brake, the red is for emergency braking.
7. Concept generation

7.3. Symmetric conical brake

This conical drum brake idea comprises two conical drums working on the same axis mirrored on each side of a mid plane as seen in Figure 12. The two bodies located inside the drums are equipped with friction material and kept from rotating. A lever pushing these bodies apart axially and into the drums causes the braking torque. The axial force acting between the drum and the friction material body on one side is matched and counteracted by the arrangement on the other side. Due to the symmetry of this concept, no axial forces would be exerted on any bearing. The conical brake also has the advantage of amplifying the force applied by the lever to a higher normal force between the cones.

Figure 12: Symmetric conical brake.
7.4. Self-energizing hydraulic disc brake

The idea seen in Figure 13 is a hydraulic brake caliper with one piston for emergency braking situations and another two for self energized braking in either direction. (MER)

This idea is about a self energizing brake caliper for use with a brake disc. The caliper is forming a converging slot between the brake pad seat and the disc which the pad is forced into by the brake disc as torque is applied. The more torque that is applied, the more brake torque the brake delivers. This reduces the need for external force and thus cost and also adds to brake reliability. The brake can be made to function in only one direction or in both. It can be made as a floating caliper engaged by the application of force from one side or, if deemed better, it can be made as a fixed caliper engaged by the application of force from both sides.

Figure 13: Self energizing, hydraulic disc brake caliper.
7. Concept generation

7.5. Integrated conical brake

*Figure 14* shows a conical brake integrated in the cylinder block of the motor. The cone (green) is pressed into the rotating cylinder block to exert a braking torque. It is splined to the motor housing (red) to keep it from rotating. The purple chamber either contains a mechanical spring or is pressurized with oil or air.

*Figure 15* is the same brake but with an additional chamber (yellow) separated from the first chamber (purple) by a sliding wall or membrane (blue). During actuation the yellow chamber is filled gas generated by an explosive charge much like in an airbag to provide a quick engagement. At the same time the purple chamber is filled with hydraulic/pneumatic pressure to compensate for the cooling and thus contracting gas volume loss, and keep the brake applied.

*Figure 14: Conical brake integrated in the cylinder block.*

*Figure 15: Integrated conical brake with a secondary chamber for explosive actuation.*
7. Concept generation

7.6. Linearly acting drum brake
This is a linearly acting drum brake with some modification in order to be self energizing to a higher extent than the conventional version. Several of these brakes can be installed coaxially if advantageously. Instead of making one axially long brake, several small ones installed coaxially would increase the number of these produced and thus decrease unit cost and possibly overall cost. It could be fitted inside the hollow motor axle or inside the winch drum for a compact winch application.

7.7. Band brake
The concept is a band brake with a hub that’s speeding up the engagement of the brake. The concept is much like a conventional band brake.
8. Screening

After the first phase of the concept generation a screening is done to eliminate some of the least interesting ideas. Some of the ideas are to complicated, not robust enough, to space demanding etc.

8.1. Hydraulically regenerative brake
The idea about a hydraulically regenerative brake as described in section 7.1. Hydraulically regenerative brake and depicted in Figure 10 is excluded as such a brake is out of the project scope. It isn’t going to fulfill the functions described in the need analysis. It is not going to work as a parking brake as hydraulics tend to leak, which would mean that a load meant to be kept stationary over a long period of time would creep. In an emergency situation which could mean that hydraulic hoses or other equipment has suffered a failure putting the hydraulic motor out of order, relying on hydraulic braking isn’t sufficient.

8.2. Multi mode drum brake
The multi mode drum brake described in 7.2. Multi mode drum brake and depicted in Figure 11 is deemed too rickety, and if more of the drum peripheral was to be paired with friction material the brake would be crammed with rods. The shifting between the modes also would cause movement of the brake drum if the winch were to be loaded during this switch. All this excludes the brake from further development in this project.

8.3. Symmetric conical brake
The dual cone brake described in 7.3. Symmetric conical brake and depicted in Figure 12 is excluded since it has a few problems concerning how to keep the drums rotating together while there’s a lever and matching bodies between them that are to be kept from rotating. That problems might be solved but they aren’t in this project. The size also makes the idea less attractive.

8.4. Self-energizing hydraulic disc brake
In order to have a self-energizing function capable of delivering any usable torque, the parking brake pistons in the caliper described in 7.4. Self-energizing hydraulic disc brake and depicted in Figure 13 would need to be substantially inclined. For this inclination to be realizable the caliper would have to become longer to still house the piston in a good way. Having a piston inclined in this way would also mean that the brake application force would be applied on the cylinder wall of the caliper. In an environment like that of a brake caliper, this would mean considerable wear of the cylinder and piston and soon a seizing and or leaking cylinder. This makes this concept not interesting for further development.

8.5. Integrated conical brake
This concept is an interesting one. The angle of the cone as described in 7.5. Integrated conical brake and depicted in Figure 14 and Figure 15, makes the brake not needing any large forces to be engaged, but still is pressed against the drum with a considerable force. It is easily integrated into the cylinder block of the motor without the need for a separate brake drum. It is compact in its design and easily understood. This brake passes the screening.

8.6. Linearly acting drum brake
The concept as described in 7.6. Linearly acting drum brake is a simple brake with some features that makes its response time very short. The conventional linearly acting drum brake
is a well known brake concept that works. This slightly altered brake takes advantage of those benefits while being slightly quicker. The concept passes the screening.

8.7. Band brake
A conventional band brake is a brake concept presently used by the company. It is simple and known to work. The hub of the brake makes it faster and thus interesting. This concept described in 7.7. Band brake also passes the screening.
9. Concept refinement
When the concept generation phase and subsequent screening is over the work with refining the remaining concepts begins. The work is done iteratively as new ideas of how the concepts are to fulfill the demands are getting more refined and are ending up better, simpler, lighter, cheaper etc. This phase leaves the concepts developed up to the point where a rough quantitative analysis can be done. This means that a concept ranking with help of the earlier mentioned decision matrix is possible.

9.1. Integrated conical brake
The first and most critical thing to work on with the conical brake is the cylinder block of the motor which also acts as brake drum for this concept. In order to do this an CAb 40 cylinder block was picked and modified in order to house a cone with friction material.

When modifying the cylinder block to house the brake it becomes apparent that the space is quite narrow. In order to make the brake perform as needed the cone has to be wider at both ends and longer than the cylinder block permits. Figure 17 shows a cylinder block roughly extended lengthwise and also in radius. The radius of the cone is this illustration enlarged so much that if further enlarged it would need alteration of other components. On the right hand side of the cylinder block holes are seen, these holes provide hydraulic pressure to the motor.

Figure 15: CAb40 modified to house a conical brake.
pistons through channels seen running axially through the motor. The flat faces radially inside and outside the right ends of these channels are matching the face of the ring formed oil distributor in the complete assembly. This ring would have to become enlarged in order for the cone to be further enlarged. This is a severe operation that’s followed by much work and cost on redesigning the motor, not necessarily to the better.

At this size the brake contact area is approximately $37 \text{ cm}$ long, has a $5.3 \text{ cm}$ radius on the wider end and $2.7 \text{ cm}$ on the thinner, the angle between the cone and the plane orthogonal to the axial direction of the cone is $86^\circ$.

The CAb 40 produces $40 \text{ Nm}$ for every $\text{ bar}$ of hydraulic pressure applied. At maximum operating pressure, $250 \text{ bar}$, the motor thus produces $10 \text{ kNm}$. This means that the brake needs to exert a braking torque of $18 \text{ kNm}$ to pass the certification demands for lifting appliances [14][15].

In order to roughly estimate the braking torque that the brake can exert some simple formulas are used:

$$T_{\text{brake}} = F_{\text{normal}} \cdot \mu \cdot r_{\text{mean}} = p_{\text{brake}} \cdot A_{\text{brake}} \cdot \mu \cdot r_{\text{mean}} \quad (\text{Eq. 9.1})$$

Rearranging Eq. 9.1 gives

$$p_{\text{brake}} = \frac{T_{\text{brake}}}{A_{\text{brake}} \cdot \mu \cdot r_{\text{mean}}} \quad (\text{Eq. 9.2})$$

$$A_{\text{brake}} = \sum_{l=0}^{L} 2 \cdot \pi \cdot r(l) \cdot dl$$

$$r(l) = r_0 + \frac{dr}{dl} \cdot l$$

(Eq. 9.3)

(Eq. 9.4)

Equating Eq. 9.3 with Eq. 9.4 gives

$$A_{\text{brake}} = \sum_{l=0}^{L} 2 \cdot \pi \cdot r_0 + \frac{dr}{dl} \cdot l \cdot dl$$

(Eq. 9.5)

Which equates to

$$= 2 \cdot \pi \left( r_{\text{min}} \int_{0}^{L} dl + \frac{dr}{dl} \cdot \int_{0}^{L} l \cdot dl \right) = 2 \cdot \pi \left( r_{\text{min}} \cdot [L]_0^L + \frac{dr}{dl} \cdot \left[ \frac{l^2}{2} \right]_0^L \right)$$

$$= 2 \cdot \pi \left( r_{\text{min}} \cdot (L) + \frac{dr}{dl} \cdot \frac{L^2}{2} \right) = 2 \cdot \pi \left( 0.027 \cdot 0.369 + \frac{0.053 - 0.027}{0.369} \cdot \frac{0.369^2}{2} \right)$$

$$= 0.0927 \text{ m}^2$$

Assuming that the mean radius is

$$r_{\text{mean}} = \frac{r_{\text{min}} + r_{\text{max}}}{2} = \frac{0.027 + 0.053}{2} = 0.04 \text{ m}$$

and choosing $\mu = 0.3$, Eq. 9.2 says that the surface pressure would have to be

$$p_{\text{brake}} = \frac{18 \text{ 000}}{0.0927 \cdot 0.3 \cdot 0.04} \approx 16 \text{ MPa}$$

This brake pad pressure is not feasible and the brake concept is excluded from further development.
9. Concept refinement

9.2. Linearly acting drum brake

The linearly acting drum brake is during this stage developed so that a rough estimation on what size the brake needs to be of to provide the torque necessary can be done. Two cases are stated that the brake needs to be able to handle; the first case is a least case scenario where the brake is acting directly on the winch drum and the winch is supposed to lift a load of two tons with the safety factor three, the other case is a worst case scenario with a lift of 200 tons with the security factor 5.

9.2.1. Lift two tons with the security factor three

To determine the dimensions of the winch and what motor to use for this, the procedure used by the company (see Appendix II) is used here. That safety factor and that load states that the lifting cable needs to be of 4.95 mm radius and thus needs a drum core radius, $DCR$, of

$$DCR = r_{cable} \cdot 20 = 99 \text{ mm}$$  \hspace{1cm} (Eq. 9.6)

The brake drum is assumed to be placed on the outer edge of the winch drum wall, and the winch drum wall is designed to be five cable radiuses high to prevent the cable from leaving the winch drum.

$$r_{wall} = DCR + 5 \cdot r_{cable} = 124 \text{ mm}$$

The pitch core radius, the radius where the force is applied is 104 mm. The torque needed to lift the load with the cable center acting at that distance from the winch drum center is thus

$$T_{required\ motor} = m_{load} \cdot g \cdot PCR = 2000 \cdot 9.81 \cdot 0.104 = 2040 \text{ Nm}$$  \hspace{1cm} (Eq. 9.7)

The smallest motor alternative to satisfy this is the Cab 10 which delivers 3000 Nm. This means that the brake needs to brake with

$$T_{brake} = T_{Cab\ 10} \cdot 1.8 = 5400 \text{ Nm}$$  \hspace{1cm} (Eq. 9.8)

The braking torque is at this stage calculated via the simplified formula Eq. 9.1.

$$T_{brake} = P_{brake} \cdot A_{brake} \cdot \mu \cdot r_{mean}$$

where

$$A_{brake} = \left(\frac{2}{3}\right) 2 \cdot \pi \cdot r_{brake} \cdot w_{brake}$$  \hspace{1cm} (Eq. 9.9)

At this stage the brake radius and the brake torque is known. $\mu$ is assumed to be 0.3. The friction material is assumed to cover 2/3 of the brake drum circumference and the brake pad pressure is assumed to be 6 MPa. This means that the width can be determined by inserting Eq. 9.9 and rearranging Eq. 9.1.

$$w_{brake} = \frac{T_{brake}}{P_{brake} \left(\frac{2}{3}\right) 2 \cdot \pi \cdot r_{brake} \cdot \mu \cdot r_{mean}} = \frac{T_{brake}}{P_{brake} \left(\frac{2}{3}\right) 2 \cdot \pi \cdot r_{brake}^2 \cdot \mu}$$  \hspace{1cm} (Eq. 9.10)

which equates to
9. Concept refinement

\[ w_{\text{brake}} = \frac{5400}{6_{10^6} \cdot \left(\frac{2}{3}\right) 2 \cdot \pi \cdot 0.124^2 \cdot 0.3} = 0.0465 \approx 47 \, \text{mm} \]

9.2.2. Lift 200 tons with the security factor five

This case is done in the exactly same way but with other numbers. The safety factor and that load states that the lifting cable needs to be of 64 mm radius and thus needs a drum core radius, DCR, of

\[ DCR = r_{\text{cable}} \cdot 20 = 1278 \, \text{mm} \]

The radius of the winch drum wall edge and also the brake drum is

\[ r_{\text{wall}} = DCR + 5 \cdot r_{\text{cable}} = 1598 \, \text{mm} \]

The pitch core radius, the radius where the force is applied is 1342 mm. The torque needed to lift the load with the cable center acting at that distance from the winch drum center is thus

\[ T_{\text{required motor}} = m_{\text{load}} \cdot g \cdot PCR = 200 \, 000 \cdot 9.81 \cdot 1.342 = 2632 \, 848 \, \text{Nm} \]

The smallest motor alternative to satisfy this is two CBm 5000 which delivers 3 284 000 Nm. This means that the brake needs to brake with

\[ T_{\text{brake}} = T_{2\text{-CBm5000}} \cdot 1.8 = 5911 \, 200 \, \text{Nm} \]

The braking torque is at this stage calculated via the simplified formula Eq. 9.1.

\[ T_{\text{brake}} = p_{\text{brake}} \cdot A_{\text{brake}} \cdot \mu \cdot r_{\text{mean}} \]

where

\[ A_{\text{brake}} = \left(\frac{2}{3}\right) 2 \cdot \pi \cdot r_{\text{brake}} \cdot w_{\text{brake}} \]  
(Eq. 9.9)

At this stage the brake radius and the brake torque is known. \( \mu \) is assumed to be 0.3. The friction material is assumed to cover 2/3 of the brake drum circumference and the brake pad pressure is assumed to be 6 MPa. This means that the width can be determined by inserting Eq. 9.9. and rearranging Eq. 9.1.

\[ w_{\text{brake}} = \frac{T_{\text{brake}}}{p_{\text{brake}} \left(\frac{2}{3}\right) 2 \cdot \pi \cdot r_{\text{brake}} \cdot \mu \cdot r_{\text{mean}}} = \frac{T_{\text{brake}}}{p_{\text{brake}} \left(\frac{2}{3}\right) 2 \cdot \pi \cdot r_{\text{brake}}^2 \cdot \mu \cdot r_{\text{mean}}} \]  
(Eq. 9.10)

which equates to

\[ w_{\text{brake}} = \frac{5911 \, 200}{6_{10^6} \cdot \left(\frac{2}{3}\right) 2 \cdot \pi \cdot 1.598^2 \cdot 0.3} = 0.3070 \approx 307 \, \text{mm} \]
9. Concept refinement

The brakes corresponding to these two cases are modeled and could be seen in Figure 18.

9.3. Band brake
This concept is also designed to fit the two cases that the linearly acting drum brake is dealing with. The difference in the equations may arise from the placing of the friction material. The bands do not completely encircle the brake drum but it’s still a better coverage peripherally
9. Concept refinement

than that of the drum brake. Instead of having gaps between the pads, the band brake has a
gap between the two bands. The radius of the brake drum is the same for both brakes. One
could roughly assume that the braking torque is the same for the two concepts as long as only
Eq. 9.1 is used. This substantially reduces calculation work during this stage. The outer
dimensions and the braking torque for these two brakes are identical to the ones of drum
brake type.

9.3.1. Lift two tons with the security factor three
The band brake on the application dealing with the least case scenario, lifting two tons
with a safety factor three has a brake surface radius of 124 mm, a brake drum width of
47 mm and brakes with 5.4 kN m.

9.3.2. Lift 200 tons with the security factor five
The band brake on the application dealing with the worst case scenario, lifting 200 tons
with a safety factor five has a brake surface radius of 1598 mm, a brake drum width of
307 mm and brakes with 5.9 MN m.

The brakes corresponding to these two cases are modeled and could be seen in Figure 19 and
Figure 20.

Figure 17: The winch drum and band brake for lifting 200 tons with the safety factor five.

Figure 19: The winch drum and band brake for lifting two tons with the safety factor three.
## 10. Concept evaluation

<table>
<thead>
<tr>
<th>Property</th>
<th>Demand</th>
<th>Weight to distribute (Wd)</th>
<th>Internal weight (Wij)</th>
<th>Weight share (Wd/Wij)</th>
<th>Score Concept 1 (1-5)</th>
<th>Result Concept 1</th>
<th>Score Concept 3 (1-5)</th>
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Figure 18: The decision matrix filled in for the C1 Band brake and the C3 Drum brake.
As the properties of the concepts are better known a concept ranking can be done using the decision matrix (see Specification of demands and decision making). This evaluation is done in order to decide where it is most worth to invest in continued work. The concepts that are previously deemed without prospect of performance or to complex to manufacture aren’t evaluated in this step as they’re already excluded in the screening process described in Screening. That leaves C1 and C3 for evaluation.

The first main topic in the decision matrix is how the brakes are performing in terms of braking and there’s a maximum of 10 points to distribute within this subject.

The first subtopic up is Static braking torque. As the concepts are designed to withstand a worst case scenario load it is also known that both of them are capable of performing a sufficient static braking torque and both concept thus should be given the score five on this topic.

Second is

10.1. B3: Brake torque characteristics

10.1.1. B4: Static braking torque

10.1.1.1. C1 Freehub
Both the concepts are designed to deal with a worst case scenario which says that they’ll be able to deal with heavy static loads. The score is 5.

10.1.1.2. C3 Drum brake
See C1 Freehub. The score is 5.

10.1.2. B5: Reliable static braking

10.1.2.1. C1 Freehub
This brake uses an ordinary band brake which would yield a full score but since many other parts of the brake is untested as heavy duty brake components the reliability has to be scored lower.

The score is 2.

10.1.2.2. C3 Drum brake
This brake is a slightly modified linearly acting drum brake which in its conventional version would get a full score, but the modification makes it something else. As this engagement mechanism has a measure of redundancy and not being as novel as C1 Freehub, it still gets a higher score.

The score is 3.

10.1.3. B6: Dynamic braking torque

10.1.3.1. C1 Freehub
As the brake probably will use a friction material with a difference between static and dynamic frictional coefficient, the same difference will be seen between the static and dynamic braking torque.

The score is 4.

10.1.3.2. C3 Drum brake
See C1 Freehub.

The score is 4.
10. Concept evaluation

10.1.4. B7: Predictable dynamic behavior
10.1.4.1. C1 Freehub
As this brake uses a band brake and it is known that this brake type is a self energizing one, questions needs to be answered concerning the dynamical behavior.

The score is 2.

10.1.4.2. C3 Drum brake
A linearly acting drum brake can be made to not have any self energizing tendencies which would speak for a score 5. This is a modified version which relies on being self energizing, but it also has ways to limit that. Without that limitation the brake would get the same score as C1 Freehub, but these limitations are mitigating this.

The score is 3.

10.1.5. B8: Small backlash
10.1.5.1. C1 Freehub
Band brakes aren’t having any big problems with backlash and the other parts of this brake should neither have any problems; with emphasis on should.

The score is 4.

10.1.5.2. C3 Drum brake
This modified version does have some backlash, it can be limited but the fact remains. To say that the concept works poorly because of this would be an exaggeration, but it isn’t getting a full score.

The score is 3.

10.1.6. B9: Function well with contaminations
10.1.6.1. C1 Freehub
Dry band brakes are a dry brake which has tendencies to have a deteriorated performance when subjected to contaminations such as sea water as known by the company.

The score is 2.

10.1.6.2. C3 Drum brake
If the drum brake doesn’t have another set of materials in the friction contact than that of the band brake, this concept shouldn’t have any higher score.

The score is 2.

10.1.7. B10: Adjustable braking torque
10.1.7.1. C1 Freehub
The braking torque of this concept can be adjusted by simply applying the brake harder/lighter.

The score is 5.
10. Concept evaluation

10.1.7.2. C3 Drum brake
The braking torque can be adjusted by either applying the brake harder or adjusting the level of self-energizing.

The score is 5.

10.2. B11: Engagement mechanism

10.2.1. B12: Self adjusted when worn

10.2.1.1. C1 Freehub
As the brake is worn, the engagement mechanism enables for the controlling of it to find different idle positions.

The score is 5.

10.2.1.2. C3 Drum brake
As the brake is worn, the engagement mechanism enables for the controlling of it to find different idle positions.

The score is 5.

10.2.2. B13: Low engagement force

10.2.2.1. C1 Freehub
The engagement force of this concept should be that of other band brakes in comparable size.

The score is 3.

10.2.2.2. C3 Drum brake
The engagement force of the concept is low as the brake is self energizing, but more force can be applied if deemed necessary.

The score is 5.

10.2.3. B14: Low disengagement force

10.2.3.1. C1 Freehub
There is nothing substantial restricting the tension relieving of the bands during disengagement.

The score is 5.

10.2.3.2. C3 Drum brake
There is nothing substantial restricting the disengagement.

The score is 5.

10.2.4. B15: Short engagement time

10.2.4.1. C1 Freehub
The brake is engaged very shortly after if not immediately when the controlling unit decides to apply the brakes, faster than other brakes on the market.

The score is 5.
10. Concept evaluation

10.2.4.2. C3 Drum brake
The brake is engaged faster than conventional linearly acting drum brakes, but not as fast as C1 Freehub.

The score is 4.

10.2.5. B16: Short disengagement time
10.2.5.1. C1 Freehub
The brake is disengaged as fast as any other band brake of comparable size.

The score is 3.

10.2.5.2. C3 Drum brake
The brake is disengaged as fast as any other linearly acting drum brake of comparable size.

The score is 3.

10.2.6. B17: Simple engagement
10.2.6.1. C1 Freehub
The engagement mechanism is complex compared to other brakes on the market.

The score is 2.

10.2.6.2. C3 Drum brake
This brake has a very simple engagement mechanism.

The score is 5.

10.2.7. B18: Simple disengagement
10.2.7.1. C1 Freehub
The disengagement mechanism is complex compared to other brakes on the market.

The score is 2.

10.2.7.2. C3 Drum brake
This brake has a very simple disengagement mechanism.

The score is 5.

10.2.8. B19: Failsafe
10.2.8.1. C1 Freehub
The brake has nothing stopping it from being operated as a failsafe brake.

The score is 5.

10.2.8.2. C3 Drum brake
The brake has nothing stopping it from being operated as a failsafe brake.

The score is 5.
10. Concept evaluation

10.2.9. B20: Gentle emergency release

10.2.9.1. C1 Freehub
This brake can be gently released just as easy any band brake.
The score is 5.

10.2.9.2. C3 Drum brake
This brake may be released gently, but questions needs to be answered first.
The score is 3.

10.3. B21: Space intrusion

10.3.1. B22: Small axial dimension

10.3.1.1. C1 Freehub
In order to deliver the torque demanded, the brake needs to have either a sufficient radial dimension or a sufficient axial dimension. Focusing on keeping the axial dimension low and instead widening it radially also gives the advantage of not having to use as high engagement forces. The axial dimension is thereby small, but not significantly smaller than other brakes of comparable torque.
The score is 4.

10.3.1.2. C3 Drum brake
The same can be said about this brake.
The score is 4.

10.3.2. B23: Small radial dimension

10.3.2.1. C1 Freehub
The brake is designed to be as radially big as adjacent parts of the application, and that might very well be bigger than other brakes of comparable torque.
The score is 2.

10.3.2.2. C3 Drum brake
The same can be said about this brake.
The score is 2.

10.3.3. B24: No need for large external devices

10.3.3.1. C1 Freehub
The brake needs an external force acting on the engagement mechanism, a pneumatic/hydraulic cylinder for instance.
The score is 3.

10.3.3.2. C3 Drum brake
The same can be said about this brake.
The score is 3.
10. Concept evaluation

10.4. **B25: User interface and serviceability**

10.4.1. **B26: Not easily tampered with**

10.4.1.1. **C1 Freehub**

The internal parts of this brake is situated in a way so that they cannot be easily accessed. The brake bands on the other hand is placed visibly, they’re also very crucial for the function of the brake. Any force supplying part such as an external cylinder might also end up easily accessed.

The score is 2.

10.4.1.2. **C3 Drum brake**

The frictional contact surface as well as the engagement mechanism is placed centrally in the brake that could be enclosed. External components for supplying force for the engagement mechanism may be placed in an easily accessed way.

The score is 4.

10.4.2. **B27: Easily inspected, maintained and rebuilt**

10.4.2.1. **C1 Freehub**

The brake bands are placed in a way so that they can be easily accessed, the same probably goes for any external cylinders or similar acting upon the engagement mechanisms. Parts of the engagement mechanism may be situated more inaccessibly, and the engagement mechanism has many parts to deal with which makes the process more time consuming.

The score is 3.

10.4.2.2. **C3 Drum brake**

This brake is simpler, it may be inspected and probably adjusted without much disassembly. Once the center of the brake with brake pads etc. is removed all components can be swiftly replaced.

The score is 5.

10.5. **B28: Environmental interaction**

10.5.1. **B29: Weather resistant**

10.5.1.1. **C1 Freehub**

The design is not weather resistant, it would need to be encapsulated to be able to deal with the weather.

The score is 1.

10.5.1.2. **C3 Drum brake**

The design is not weather resistant, it would need to be encapsulated to be able to deal with the weather.

The score is 1.
10. Concept evaluation

10.5.2. B30: Connection compatibility

10.5.2.1. C1 Freehub
This concept may be designed to be operated with hydraulics, pneumatics, electrics etc.
The score is 5.

10.5.2.2. C3 Drum brake
Neither this concept has any restrictions in ways to operate it.
The score is 5.

10.5.3. B31: No environmentally hazardous emissions

10.5.3.1. C1 Freehub
Neither the brake material nor the way of applying force on the engagement mechanism has been set and thus the score can’t be either good or bad.
The score is 3.

10.5.3.2. C3 Drum brake
Neither the brake material nor the way of applying force on the engagement mechanism has been set and thus the score can’t be either good or bad.
The score is 3.

10.6. B32: Motor interaction

10.6.1. B33: Low motor contamination risk

10.6.1.1. C1 Freehub
The motor and the brake doesn’t share any volume. The brake isn’t necessarily using the hydraulic fluid of the motor, and if so, it wouldn’t have it in close contact with the friction material wear debris.
The score is 5.

10.6.1.2. C3 Drum brake
The same goes for this concept.
The score is 5.

10.6.2. B34: Low drag

10.6.2.1. C1 Freehub
The drag between brake band and drum wouldn’t exist, and the rest of the brake would cause no or very little drag.
The score is 5.

10.6.2.2. C3 Drum brake
Drag in this brake would have to be completely eliminated in order to function. Fortunately drag also is easily eliminated.
The score is 5.
10. Concept evaluation

10.6.3. B35: No motor action needed when disengaged

10.6.3.1. C1 Freehub
As with any band brake, motor action for disengagement is unnecessary.

The score is 5.

10.6.3.2. C3 Drum brake
As with any drum brake, motor action for disengagement is unnecessary.

The score is 5.

10.6.4. B36: Efficient cooling/energy storage

10.6.4.1. C1 Freehub
As being a band brake, the contact surface and thus the surface producing heat is placed in a way that makes it exposed to the surrounding air. The surface is also spread over a large volume and in contact with the brake drum which could store, conduct and dissipate the energy. The brake bands that also could fulfill this function is separated from the contact surface only by the friction material. With a quantified set of brake energy, energy storage capacity and subsequent energy dissipation/conduction rate and optimization where applicable, this concept has the prospect of scoring 5. Should the temperatures be found to rise above what’s tolerable or the rate of dissipation/conduction to low, the concept can be complemented with an active cooling system such as a forced convection system or; if in a maritime application, a desiccant assisted system. Since these calculations still hasn’t been carried out the score is left at average.

The score is 3.

10.6.4.2. C3 Drum brake
The contact surface in this concept is also spread over a large volume spread in the peripheral of the brake. Even better for this brake is the fact that the brake drum which is a component of much mass and thus considerable heat storage capacity and a probably good conductivity is the component closest to the surrounding air. The friction material separates the surface producing heat from the rest of the structure which also is of considerable volume and mass. This system could also be equipped with an active cooling system if deemed necessary. The same quantification and subsequent optimization needs to be carried out for this concept in order for it to score 5, but it appears as if it is better at dissipating the heat than C1 Freehub.

The score is 4.

10.7. B37: Cost

10.7.1. B38: Low weight

10.7.1.1. C1 Freehub
Comparing the large size (200t-SF5) brake with currently used brakes, the torque to mass ratio of the C1 Freehub outperforms the other brakes except the MDA10. It is worth saying that the concept does have material that could be trimmed off. Corresponding figures for concept designed for the lightest case (2t-SF3) isn’t evaluated.
10. Concept evaluation

The score is 3.

10.7.1.2. C3 Drum brake
Comparing the large size (200t-SF5) brake with currently used brakes, the torque:mass ratio of the C3 Drum brake outperforms the others. This concept probably also has material that could be trimmed off. Corresponding figures for concept designed for the lightest case (2t-SF3) isn’t evaluated.

The score is 4.

10.7.2. B39: Readily modularized
This isn’t evaluated for any of the concepts. Both of them get the same average score.

The score is 3.

10.7.3. B40: Long service intervals
10.7.3.1. C1 Freehub
The service intervals would highly depend friction material thickness which is not set in this project.

The score is 3.

10.7.3.2. C3 Drum brake
The service intervals would highly depend friction material thickness which is not set in this project.

The score is 3.

10.7.4. B41: Simple or standard components
10.7.4.1. C1 Freehub
This concept has many parts that couldn’t be called simple.

The score is 2.

10.7.4.2. C3 Drum brake
This concept uses very few different parts and all of them are easily produced. Few of them are standard components.

The score is 4.

10.7.5. B42: Easy assembly and installation
10.7.5.1. C1 Freehub
The number of parts in this concept makes it time consuming to assemble. Installation on motor should be simple.

The score is 2.

10.7.5.2. C3 Drum brake
This is a very simple brake to assemble and installation shouldn’t be any harder.

The score is 5.
11. Finishing work

As the Concept evaluation concludes, the linearly acting drum brake is the concept that is most worth spending more time working on. This is done by further refining the brake torque calculations to see how it can be optimized.

The linearly acting drum brake is much like a shoe brake. The brake shoe in such a brake is fastened in one end and pushed towards the drum on the other end and thus causing a rotating motion towards the drum. This means that the brake shoe is pushed against the drum as the drum rotates in one direction, and pushed from the drum when rotating in the other direction. Being a linearly acting drum brake means that both ends (A & C in Figure 22) of the brake shoe or brake pad is pushed towards the brake drum. That both point A and C are pushed towards the drum is reducing the leverage that causes a moment around either one of them dependent on the direction of rotation. Designing a brake where A and C are further away from the drum increases this leverage and vice versa. This makes the brake being less prone to self-energizing, but does not eliminate it.

Figure 19: Linearly acting external and internal drum brakes as described by William C. Orthwein in ‘Clutches and Brakes - Design and Selection, 2nd ed.’ [3]. The same nomenclature is used in this section.
Orthwein argues that the pressure distribution is an effect of the deforming condition of the friction material lining so that

\[ p = k \cdot \Delta \cdot \cos(\theta) \]  
(Eq. 11.1)

Where \( k \) is the stiffness of the lining, \( \Delta \) is the deformation and \( \theta \) is the angle as shown in Figure 2.2. Except for the lining, all the pad is assumed to be completely rigid.

This makes it possible to rewrite Eq. 11.1 as a function of the maximum pressure, \( p_{\text{max}} \).

\[ p = p_{\text{max}} \cdot \cos(\theta) \]  
(Eq. 11.2)

If this is multiplied by radius and area of contact an expression for torque emerges.

\[ dF = \mu \cdot p_{\text{max}} \cdot \cos(\theta) \cdot r \cdot w \cdot d\theta \]  
(Eq. 11.3)

Now, if the pad is assumed to only move in the radial direction and not in the peripheral one, \( p_{\text{max}} \) is obtained at \( \theta = 0 \). That means that integrating Eq. 11.3 is simplified by doubling half the integral and thus expressing the torque as

\[ T = 2 \cdot \mu \cdot p_{\text{max}} \cdot r^2 \cdot w \cdot \sin\left(\frac{\phi_0}{2}\right) \]  
(Eq. 11.4)

where \( T \) is the torque and \( \phi \) is the angle measured from the end of the lining rather than the middle. \( \frac{\phi_0}{2} \) is half the angle covered by the lining. From Eq. 11.4 the following expression for width can be derived.

\[ \frac{T}{2 \cdot \mu \cdot p_{\text{max}} \cdot r^2 \cdot \sin\left(\frac{\phi_0}{2}\right)} = w \]  
(Eq. 11.5)

This is quite close to the equation that describes the torque for the drum brake concept of this project. This equation shows that the brake drum has to be widened slightly to mitigate the fact that the maximum pressure doesn’t act over the whole brake lining. Instead of having a brake drum that is 47 mm wide, 58 mm is needed for the least case scenario. Instead of the worst case scenario brake having to be 307 mm, it is 379 mm wide.
12. Discussion and subsequent work

The work as described in the work description (see *Project introduction*) can be divided into seven operations that are listed below.

<table>
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<tr>
<th>Operation</th>
<th>Status</th>
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<tr>
<td>Need analysis.</td>
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</tr>
<tr>
<td>Competitor study.</td>
<td>Done.</td>
</tr>
<tr>
<td>Literature study.</td>
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<tr>
<td>Benchmarking.</td>
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<td>Done.</td>
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<td>Concept selection.</td>
<td>Done.</td>
</tr>
<tr>
<td>Concept refinement.</td>
<td>Done.</td>
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</tbody>
</table>

The project has ended up with a few concepts in different stages of development. The *C2 Conical brake* concept was discarded early since it isn’t delivering the required torque due to the limited volume available for it in the motor shaft. Propositions have been left to Bosch Rexroth on how to use this space for future brake concept generation. Together with this concept idea, a way to eliminate the axial engagement/disengagement force currently exerted on bearings when using some brakes is proposed. Sadly, these ideas were found to be too late in order for them to be further developed in this project.

The *C1 Freehub* is a concept that with a further developed engagement mechanism, could reduce the brake engagement time in an emergency brake situation. This would mean that the forces exerted on the structure would decrease at the same time, which means a safer product. Shorter response time also reduces the need for a high braking torque; could this mean that smaller and cheaper brakes can be used without compromising safety in the future? This engagement mechanism can also be combined with other brake types than a band brake. The heat production during emergency braking also needs to be investigated to ensure suitable temperatures.

The concept deemed most interesting in the concept evaluation this time, the *C3 Drum brake*, also has refining to be done. The brake internals and the suspending structure needs to be redesigned to make sure that the material is in its right places. This can prove to be useful in order to reduce weight as well as cost. The heat production during emergency braking also needs to be investigated to ensure suitable temperatures. Further investigation on it’s dynamical behaviors would also have to be carried out and ways to restrict these might improve the reliability of the brake.
13. References


13. References


Appendix

Appendix I – Interview questions
Questions for Hans Sahlin and Roger Granström during the interview 27/2-2018:

1. Where’s the winches normally situated? Cranes? Ships? Else where?
2. How heavy loads are relevant?
3. How is a winch operated? Could you describe the different sequences?
4. How is the brake used? Only as an emergency brake? Only for holding? What about braking during normal operation?
5. How are the winches driven? Are other hydraulic pumps used than those delivered by Bosch Rexroth?
6. How much space is available in the vicinity of the pump?
7. What power outputs are typically available in the vicinity of the application? (Electricity? Hydraulic pressure? Compressed air? Anything else?)
8. How much space is available in the vicinity of the motor?
9. What demands are there concerning brake torque?
10. What demands are there concerning winches and brakes stated by certifying organs?
11. What certifying organs are there to consider?
12. Is there anything about the Viking motor that makes it better suited for winch applications than other motors?
13. Is there anything about the CA motor that makes it better suited for winch applications than other motors?
14. What work does the customer typically do to the motor? (Service? Troubleshooting? Maintenance?)
15. What brake properties are valued by the customer? (Size? Accessibility?)
16. Why is there dual acting band brakes delivered by Bosch Rexroth?
Hägglunds solutions for winches. Dimensioning of winch motor.

**Winch motor sizing**

Pitch core diameter = PCD

PCD = D₀ + (2(N−1)+d

Drum diameter D₀ mm
Rope diameter d mm
Number of rope layers N

**Example:**
For a winch drum of diameter
D=600 mm with 4 layers of rope.
Rope diameter d=20 mm
The PCD in the outer layer will be:

PCD = 600 + (2×4−1)+20 = 740 mm

- **Line pull**
  \[ S = \frac{T}{g_e \cdot [0.5 \cdot D + (N−0.5) \cdot d]} \]  
  [ton]

- **Drum speed**
  \[ n = \frac{\sqrt{1000}}{\pi \cdot \left( \frac{D + d \cdot (2N - 1)}{2N - 1} \right)} \]  
  [rpm]

- **No of layers**
  \[ N = \sqrt{\frac{4D + d^2 - W}{2 \cdot d}} - D \]

- **Drum length**
  \[ L = \frac{W \cdot 1050 \cdot d}{N \cdot (D + N \cdot d) \cdot \pi} \]  
  [mm]

- **Drum diameter**
  \[ D = \frac{T}{0.5 \cdot g_e \cdot S - \frac{d \cdot (N−0.5)}{0.5}} \]  
  [mm]

- **Rope speed**
  \[ v = \frac{n \cdot D + d \cdot (2N - 1) \cdot \pi}{1000} \]  
  [m/min]

- **Torque**
  \[ T = S \cdot [g_e \cdot [0.5 \cdot D + (N−0.5) \cdot d]] \]  
  [Nm]

- **Wire rope capacity**
  \[ W = \frac{L \cdot N \cdot \pi \cdot (D + N \cdot d)}{1.06 \cdot d \cdot 1000} \]  
  [m]
Hägglunds solutions for winches. Dimensioning of winch motor.

Winch motor sizing

- Winches are normally specified by the pull available on the first layer. This is the layer of wire sitting on the drum core. Have to be confirmed/agreed with the customer.

- For a 12 tonnes pull the table below suggests an 24mm wire diameter.

<table>
<thead>
<tr>
<th>Rope Diameter</th>
<th>18</th>
<th>16</th>
<th>14</th>
<th>12</th>
<th>10</th>
<th>8</th>
<th>6</th>
<th>4</th>
<th>3</th>
<th>2</th>
<th>1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pull</td>
<td>6</td>
<td>5</td>
<td>7</td>
<td>8</td>
<td>9</td>
<td>10</td>
<td>11</td>
<td>12.5</td>
<td>14</td>
<td>16</td>
<td>18</td>
</tr>
<tr>
<td>Min Breaking Load</td>
<td>12</td>
<td>16</td>
<td>20</td>
<td>25</td>
<td>31</td>
<td>37</td>
<td>43</td>
<td>50</td>
<td>65</td>
<td>80</td>
<td>100</td>
</tr>
</tbody>
</table>

- The core diameter of a winch is normally sized between 14 and 20 times the diameter of the wire being used. 16 times the wire diameter is most commonly used for simple winches.

- 16-22 times for winches with light duty.

- 22-25 times for ropes with anti rotation function.