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Concept Study of a New Drive System for Industrial Folding Doors

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Master Thesis
Division of Industrial Electrical Engineering and Automation

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This thesis was conducted at ASSA ABLOY Entrance Systems IDDS AB for the Division of Industrial Electrical Engineering and Automation, Faculty of Engineering, Lund University.

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Abstract

As technology advances and population rises, requirements for everyday doors change. In the industrial sector, companies have to keep up with the increase in supply and demand. A well-functioning industrial folding door contributes to a steady flow of goods. Folding doors come in many different sizes and are often installed in narrow spaces. In order for any door supplier to stay competitive, offering a cheap, compact and adjustable drive system for electrically operated industrial folding doors is crucial.

ASSA ABLOY is one of the world's largest suppliers of industrial folding doors. The company currently uses a drive system that takes advantage of a transmission rail to transfer the motor's rotational motion to the movement of the door sections. Although the solution is elegant, it might be hard to fit where space is very limited. The aim of this project is to develop a new drive system that minimizes installation dimensions and focuses on reliability, robustness and cost while staying adaptable for different folding door types and sizes.

To achieve this goal, the report follows a custom-made product development plan based on the Ulrich & Eppinger methodology. This includes concept generation, development, prototyping and testing. The concept generation incorporates a full evaluation of competitors products in order to acknowledge and evaluate existing drive systems to attain an oversight of the market.
The result is an unique new drive system that consists of a rack-pinion inspired solution, where the rack is bent to stay within the door's opening space. The drive system neither requires extra headroom nor side space and has the same installation dimensions as a manually operated folding door. Through testing it was discovered that the drive system's opening speed and current consumption was reduced significantly in comparison to the old drive system. Previous power spikes were eliminated with the new drive system, resulting in smoother operation.
The report concludes that the new drive system might very well be a part of the company's future assortment of products and particularly attractive for customers with very limited installation dimensions.

Acknowledgements

To show our gratitude we would like to thank the Industrial Door Solutions team at ASSA ABLOY Entrance Systems AB for the opportunity to conclude our five year education at the Faculty of Engineering, Lund University with an interesting project. We are especially grateful for our supervisor Sr. Mechanical Engineer Anders Löfgren for all the support, entertainment and laughs at every possible hour of the day. You are an invaluable source of knowledge and expert in your field. A special thanks to Mikael Appeltofft, Linus Bäcklund, Anton Hansson and Bjarne Larsson for assisting with specification lists, model calculations, welding and standard regulations.

Thank you Marcus Daag and Lukas Lundgren for reading and providing feedback on the report.

Finally, we would also like to thank Gunnar Lindstedt for providing assistance, support and clarification with the academic aspects of the project.

I, Patrik Krc Zitny, would like to thank my father Michal Krc for all the support during my years at university and for introducing me to technology, engineering and mathematics. You have made me who I am today and will always be in my thoughts. Vila i frid Pappa. 1971-2020.

Contents

It's a

dangerous business, Frodo,

going out your door.

You step onto the road,

and if you don't

keep your feet,

there's no knowing where

you might be swept off to.

- J.R.R. Tolkien, The Lord of the Rings

1. Introduction

The introduction presents a short background and describes the objectives, aims and delimitations of this project. A description regarding the structure of the report and method used during the project will also be presented.

1.1 Background

1.1.1 ASSA ABLOY Entrance Systems

This master thesis is written at ASSA ABLOY Entrance Systems. ASSA ABLOY is a world leading provider of products and services related to locks, doors, gates and entrance automation. In recent decades, ASSA ABLOY Entrance Systems has acquired several companies in the entrance industry and has become a large supplier of industrial doors.

1.1.2 Industrial Doors in General

Industrial doors are an important part of business flow. The doors help protect, control and speed up access to all kinds of premises. They can also facilitate logistics and provide security to the customer. There are several types of industrial doors available on the market today, serving different purposes. The most common industrial door types are described below:

Folding doors: A folding door usually consists of two halves, and each half consists of at least two door leafs (or sections). The door leaf closest to the hinge mounted on the wall is called the outer leaf, and the other one is called the inner leaf. See figure 1.1. The outer leaf rotates around the hinge mounted on the wall, while the inner leaf rotates around an intermediate hinge connected to the outer leaf. This results in the door sections folding towards each other. For the sections to move properly, the inner door leaf is guided by a rail mounted above. Folding doors are mainly used with small to medium sized door openings.

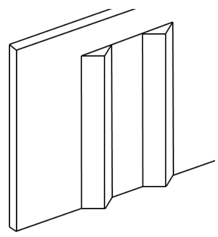

Figure 1.1: *Folding door.*

Overhead sectional doors: The overhead sectional door consists of several horizontal sections that are hinged to each other. See figure 1.2. The sections are pushed upwards and inwards via side-mounted guide rails. This means that the door rests under the ceiling when it is open. In order for the door to be held up, it uses springs and/or wires. This door type can be used in all size categories. Overhead sectional doors are especially attractive when side space and headroom is limited, but depth space is plentiful.

Hinged door: A door that consists of either one or two sections that open horizontally around hinges that are mounted in the door frame. This can be compared to the regular everyday door/double door. See figure 1.3. These types of doors are mainly used with small to medium sized door openings with limited width.

Roller shutter: The roller shutter door consists of horizontal metal slats that are guided up on a roller via vertical rails on each side of the door opening. See figure 1.4. Often used by individual stores in shopping malls. These types of doors can be used with both small and large door openings, if there are no requirements for thermal insulation.

Roller door: Works similar to the roller shutter door. However, the door is instead made up of fabric and can be compared to roller blinds. The roller door is both very fast-acting and insulating. Therefore, this door is mainly used in facilities where doors open and close frequently, such as cold stores.

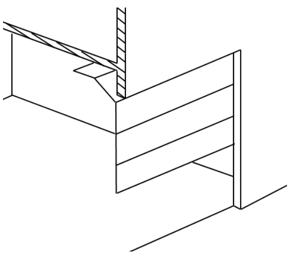

Figure 1.2: *Overhead sectional door.*

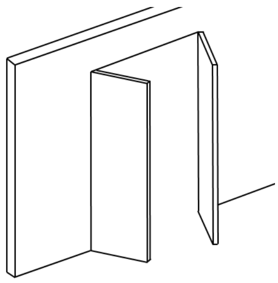

Figure 1.3: *Hinged door.*

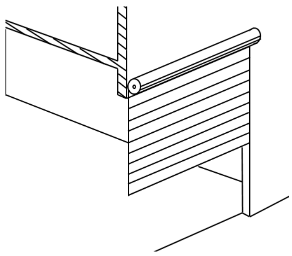

Figure 1.4: *Roller shutter/Roller door.*

All of the above mentioned industrial door types, except the hinge doors, are offered by ASSA ABLOY.

1.1.3 Folding Doors

Industrial folding doors are ideal where space is limited around the opening, minimizing the ceiling space required inside the building. The folding doors are designed to require minimal maintenance. As the doors slide open, lifting hinges raises the door off the floor which reduces the wear. When compared to overhead sectional doors they have several more advantages. Folding doors also have a reduced number of parts that can be damaged. They are also easier to operate manually and are more flexible. The doors can be installed to open both outwards or inwards. Further, overhead sectional doors are often equipped with plastic windows due to their weight, these windows can be bent and reflect light in such a way that it is not possible to see through them from a distance. The plastic can also darken over time. However, folding doors often have glass windows, providing better insulation and visibility. [1]

Since folding doors fold while they move, opening times are cut considerably when compared to other industrial door alternatives. In turn, this keeps the air exchange and temperature loss low when passing through the door. The possibility of partially opening the door also contributes to this. [2]

Folding doors also have a significantly reduced risk of collision. Due to their horizontal movement, the door is always visible as it opens/closes [3]. An image of ASSA ABLOY's electrically operated industrial folding door (Model FD2250P) can be seen in figure 1.5.

Figure 1.5: *One of ASSA ABLOY's electrically operated industrial folding doors. Model: FD2250P. Figure from [2].*

Configurations

The "m+n" configuration is a term describing how many door leafs (or sections) the folding door consists of. The electrically operated offerings from ASSA ABLOY currently use either a 2+0 or 2+2 configuration. As can be seen in figures 1.6 and 1.8, the 2+2 configuration is composed of two mirrored 2+0 configurations. Folding doors also come in an odd number of sections, such as the 2+1 configuration seen in figure 1.7. On electrically operated doors, the single door leaf in the 2+1 configuration is either manual or uses a separate driving system.

Figure 1.6: *2+0 door configuration. Figure from [2].*

Figure 1.7: *2+1 door configuration. Figure from [2].*

Figure 1.8: *2+2 door configuration. Figure from [2].*

Dimensions

Folding doors have standard dimensions and abbreviations in order to declare how much space they require to operate. This section will clarify these abbreviations. The dimensions seen from a top view on a 2+2 folding door configuration are presented in figure 1.9.

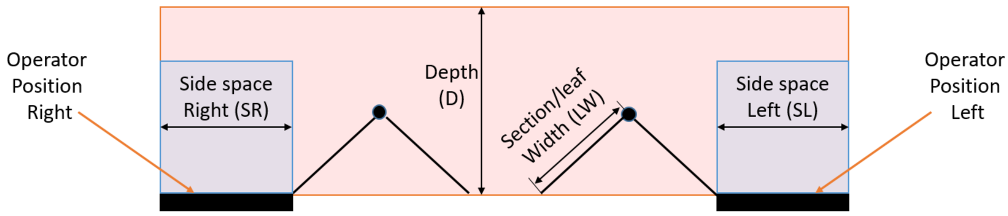

Figure 1.9: *Dimension abbreviations seen from a top view of the door.*

The depth dimension describes how far into the room or building the door will extend. Left and right side space dimensions describe how much side space the door requires in order to fit itself and its components. The operator position indicators denotes possible motor placement locations.

In figure 1.10 the dimensions are shown from a front view. The required headroom and a center mounting location for the motor can be seen. Headroom is the maximum clearance the door requires above the door opening in order to fit itself and its components.

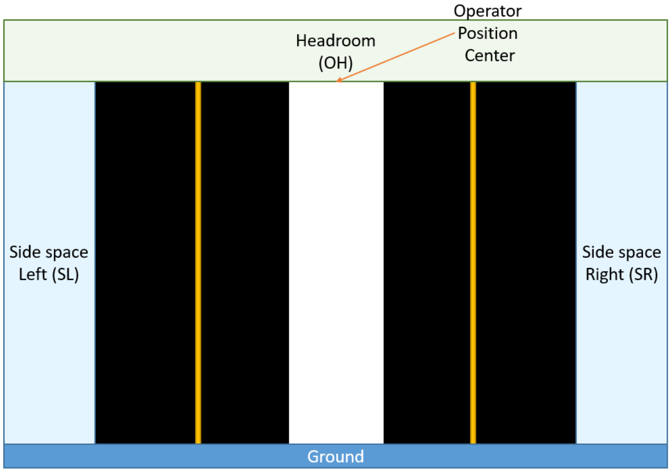

Figure 1.10: *Dimension abbreviations seen from a front view of the door.*

All of the above mentioned dimensions will be used throughout this report.

1.2 Problem Description

The electrically operated folding door available today is opened and closed using either a motor, or in the case of motor malfunction it is possible to operate manually. The drive system consists of a motor and a chain inside a rail that anchors to the door leaves at certain locations. See figure 1.11. This rail extends to the sides and above the door, which gives the design larger side space and headroom dimensions than desired. Another problem with the design is the nonlinear force necessary to close the doors. The company would like to find a technical solution that requires less space than the solution of today while also improving overall robustness and reliability. This project will not reconstruct the folding door or folding door leaves, but rather aims to come up with a new drive system between the door and motor on the electrically operated folding doors available at ASSA ABLOY today.

Figure 1.11: *Today's mechanical transmission rail.*

1.2.1 Delimitations

As mentioned in the previous section 1.2, the folding door itself will not be investigated or altered in this project. Regardless of the concept, the motor must also remain unchanged. The main goal is to develop a new drive system.

1.3 Objectives

The objective of this project is to evaluate the current solutions available and conduct a concept study of a new drive system for industrial folding doors with focus on space, safety and reliability. The methodology used for the product development comes from Ulrich and Eppinger's book "Product design and development". [4]

These are the project goals:

- Find a technical solution that requires less space than the solution of today
- Investigate available solutions on the market
- The solution must follow applicable standards and regulations
- Consider direct cost, installation and reliability
- Create a CAD-model that illustrates the functionality of the drive system concept and provide a calculation model of the chosen drive system.
- Build a full scale prototype for evaluation and presentation

1.4 Method

The first step was to evaluate ASSA ABLOY's current drive system and create a requirement list. The specifications are based on legal regulations, the company's preferences as well as international and internal standards. The list includes targets of both an "obligatory need" and a "nice to have" nature.

The second step was to investigate competitors' solutions, generate concepts and evaluate these towards the generated specification list. The best concept is then selected through two rounds of concept selection.

The third step consisted of creating a detailed design in CAD of the chosen concept to be able to determine specific design parameters.

In the last step, detailed drawings are created and a full-scale prototype is built. Thereafter the prototype is tested in order to evaluate, optimize and refine the concept.

1.5 Report Structure

Introduction

The introduction contains a background of industrial doors, including why the folding door drive system is being investigated. The chapter explains the desired aims, goals and limitations of this project in detail. The method used to achieve the goals is also presented here.

Theory

In this chapter, a summarizing review of the theoretical background and knowledge required in order to conduct the project is presented. This chapter contains the methodology to product design and development, as illustrated by Ulrich and Eppinger. The chapter also contains the different standards and regulations the project follows in order to achieve a product with market incentive.

Concept Development

In this section, an evaluation of the current drive system is made. Operating forces are determined and the current drive system is described in more detail. A summary of the generated specification list and an evaluation of competitor drive systems is presented. Generation of new concepts, concept comparisons and grading is also included. One concept is chosen to continue with.

Design

The functionality of the chosen concept is described in more detail. Different progression steps in the design phase are discussed and the final design is presented. This chapter also contains descriptions of the individual parts and their purpose in the final design.

Testing and Refinement

In this chapter a description of different tests conducted, their results and photos of the prototype are provided. Possible refinement and optimization opportunities are also presented.

Discussion

In this chapter, the project progression and findings is discussed. The chapter also includes possible future work.

Conclusion

A brief summary of the project and what lies ahead.

1.5.1 Division of Labour

Both students involved in this project have, in all parts of the report and practical work, provided an equal amount of time and effort. No specific division of labour can be presented. Both students have written and revised all documents, made calculations, assembled the prototype and performed tests on it to the same extent.

2. Theory

In this chapter a summarizing review of the theoretical background and knowledge required in order to conduct the project will be presented.

2.1 Product Development Methodology

The generic product development method of Ulrich & Eppinger consists of 6 phases as shown in figure 2.1. Traditionally, product planning has no part in a product development process, there are therefore only 5 phases starting from phase 1. This product development process was chosen, with a few alterations, because a well defined development process will simplify decision making and help concretize the work process. This will ensure that the end product is of quality. A well structured development process will also contain milestones corresponding to the end of each phase. This anchors the schedule of the project. Problem areas can be spotted with ease and in time by comparing the actual events to the process. It is also crucial that everything is documented so that the opportunities of improvement can be identified. [4]

Figure 2.1: *The generic product development process in Ulrich & Eppinger.*

Below are the product development phases described, as in the book "Product Design and Development" by Ulrich & Eppinger. [4]

- Phase 0 : Planning - The planning stage is regularly alluded to as "stage zero" since it prepares for the task endorsement and start of the product development process. This stage incorporates appraisal of innovation advancements and market targets. The yield of this stage is the statement of purpose, which indicates the objective market for the product, business objectives and key requirements.
- Phase 1 : Concept Development - In this phase, the requirements of the specific market are distinguished, product ideas are created, assessed and followed by an investigation of competitive products. At least one concept is chosen for additional evaluation and testing. This phase will be further described in chapter 2.1.1.
- Phase 2 : System-Level Design - Incorporates the meaning of the product design and the decomposition of the product into subsystems and components. This phase output often includes specifications of all the subsystems and a geometric layout of the product.
- Phase 3 : Detail Design - Complete specification of the geometry, materials and tolerances of included parts in the product. Standard parts are bought from suppliers in this stage. The yield of this stage is the control documentation which includes drawings or records depicting each part and plans for buying, manufacturing and assembling of the product.
- Phase 4 : Testing and Refinement - Construction and evaluation of preproduction prototypes of the product. Prototypes consists of the same geometry and material properties as expected for the final product and are evaluated to decide if the product will work and fulfill the key client needs.
- Phase 5 : Production Ramp-Up - The motivation behind the ramp-up is to prepare the labor force and to work out any remaining issues in the production processes.

2.1.1 Concept Development

In Ulrich & Eppinger, the generic concept development phase is expanded into what is called the front-end process. The process format, as described by Ulrich & Eppinger, can be seen in figure 2.2 but can vary between different companies. It consists of many different iterative group activities which often overlap each other.[4] The concept development activities are described below:

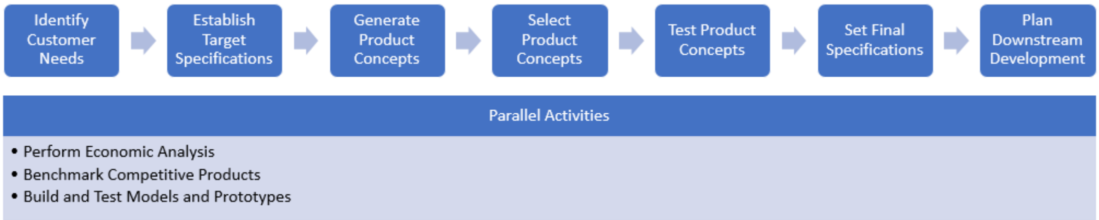

Figure 2.2: *The front-end process in Ulrich & Eppinger.*

- Identifying Customer Needs: The objective of this step is to comprehend clients' needs and forward them to the communication group. The yield of this step are needs statements, with significance weightings for some or the entirety of the needs.
- Establishing Target Specifications: Specifications give an exact depiction of what a product needs to do. They are the interpretation of the client needs into specialized terms. Targets for the specifications are set at the start and represents the expectations of the development team. These specifications are then refined to be compliant with the requirements set by the group's decision of a product idea. The yield of this stage is a rundown of target specifications. Each specification consists of a metric, with a lower, upper and ideal value.
- Concept Generation: The objective of concept generation is to investigate the space of product ideas that may address the client needs. Concept generation incorporates a mix of outside inquiry, imaginative critical thinking inside the group, and investigation of the different solutions the group produces. The result of this activity is normally up to 10 to 20 ideas, each commonly addressed by a sketch and brief descriptive text. This step will be further explained in chapter 2.1.2.
- Concept Selection: In this step, the different product concepts are dissected to distinguish the best concept(s). The cycle typically requires a few iterations and may start extra concepts generations and refinements.
- Concept Testing: At least one concept is tested to evaluate whether the client needs have been met, asses the market capabilities of the product, and identify problems. If the customer reaction is negative, the development project may end or the product is further refined.

- Setting Final Specifications: The target specifications set earlier in the process are returned to after an idea has been chosen and tested. The group will now focus on explicit estimations of the metrics reflecting the constraints of the product concept and compromise between cost and product performance.
- Project Planning: In this last action of concept development, the group makes a point by point advancement plan, devises a methodology to limit advancement time, and distinguishes the assets needed to finish the project.
- Economic Analysis: The group, often with the help of a financial expert, assembles a monetary model for the new product. This model is utilized to legitimize continuation of the development process and is updated throughout the concept development process. Economic analysis is one of the parallel exercises in the concept development phase.
- Benchmarking of Competitive Products: A comprehension of competitive products is essential and can act as a source of thoughts for the product and production process design. Competitive benchmarking is also one of the parallel front-end activities.
- Modeling and Prototyping: Each phase of the concept development process includes different types of models and prototypes. These may incorporate, among others: early "proof-of-concept" models - which can be shown to clients to assess ergonomics and style, spreadsheet models of technical compromises and experimental test models, which can be utilized to set design parameters for reliable and robust performance.

2.1.2 Five-Step Method

Ulrich & Eppinger's five-step concept generation method has been used during the project. In short the method is used to break down an existing complex problem into smaller and simpler subproblems [4]. The generic five-step method can be seen in figure 2.3. The steps involved are described and listed below.

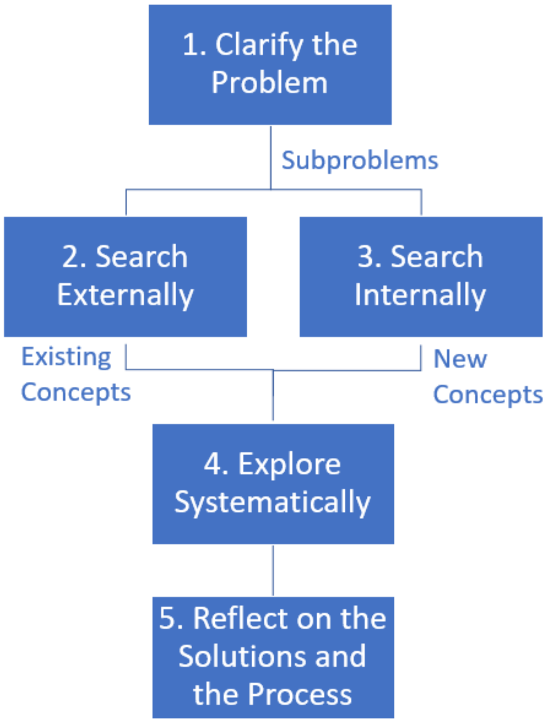

Figure 2.3: *The five-step concept generation method in Ulrich & Eppinger.*

- Clarify the Problem: Clarifying the problem consists of gathering a general understanding of the problem and if necessary breaking it down into subproblems.
- Search Externally: A step where information related to the problem is gathered. This can be patent searches, literature searches, expert consultation, and competitive benchmarking.
- Search Internally: Internal search is the use of personal and team knowledge and creativity to generate solution concepts.
- Explore Systematically: Create a classification tree by combining the different subproblem solutions. This will help the team divide possible solutions into independent categories. The combination table also guides the team in generating concepts.
- Reflect on the Solutions and the Process: Reflections are made throughout each step but a few questions to ask in the end includes: Are there alternative ways to decompose the problem? Have external sources been thoroughly pursued? Have ideas from everyone been accepted and integrated in the process?

2.2 Deviations from Ulrich & Eppinger

As Ulrich & Eppinger state in their book, the product development process in different industries most likely vary from the exact steps mentioned in chapters 2.1, 2.1.1 and 2.1.2. The product development process and the steps taken depends on the company, the staff and the nature of the project.

In this section, the development process used in this project will be described and the reasons why it deviates from the theoretical steps in Ulrich & Eppinger's book are discussed. This will be done by altering figures 2.1, 2.2 and 2.3 respectively. The altered figures with a short explanation follows below.

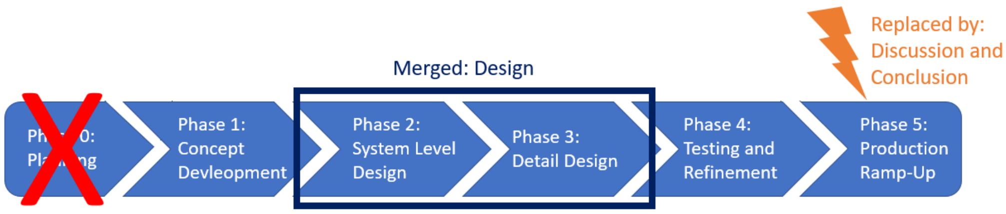

Figure 2.4: *Altered product development steps.*

As seen in figure 2.4, the Planning stage is skipped, since this precedes the project approval. This was already done when the project started.

The Concept Development phase is included and explained more in detail under figure 2.5.

System-Level Design and Detail Design are merged into a single chapter. No distinction is made between these two.

Testing and Refinement is included. Here, the concept prototype is mainly tested and if time allows refined further.

Since the project objective is to create a prototype and generate product concepts, details regarding production and production planning will not be included in the scope of this report. However, this section will instead be replaced with a chapter where the project outcome is discussed, together with thoughts and reflections of the process. Another important aspect that will be included here is future work.

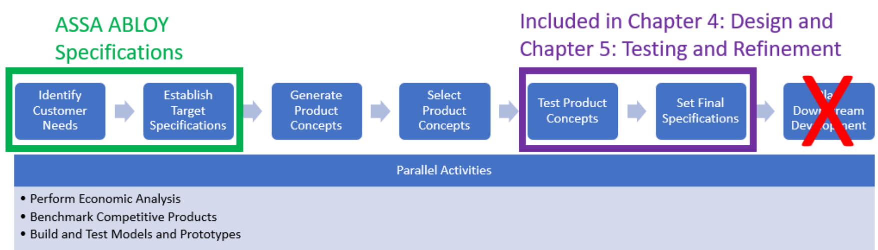

Figure 2.5: *Altered concept development steps.*

Since this project is provided by ASSA ABLOY, the company has already identified customer needs and specified the necessary improvements. Therefore, the two first steps in the concept development method seen in figure 2.5 will be merged together and consist of mainly a specification sheet.

The main part of the project will be performed during the following concept development steps. These steps include concept generation, concept selection, testing and setting final specifications. However, testing and setting the final specifications will be included in the subsequent chapters 4 and 5. Planning downstream development will not be included in the scope of this report.

Economic analysis, benchmarking of competitive products and building and testing prototypes are all included in different parts of the report to some extent.

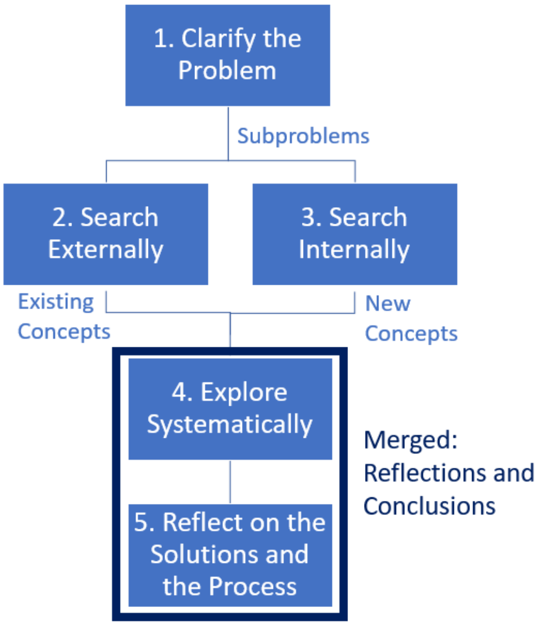

Figure 2.6: *Altered concept generation steps (five-step method).*

The concept generation method or rather, the five-step method presented in chapter 2.1.2 will be used. The two final steps, explore systematically and reflection on solutions will be merged.

2.3 Industry Standards

2.3.1 EN 12424 - Industrial, commercial and garage doors and gates - Resistance to wind load - Classification

The standard used by ASSA ABLOY (and competing companies) relating to wind loads in power-operated doors is EN 12424. This standard defines the classification of the wind loads for closed doors and does not contain any information relating to operating performance of a door during wind load. The wind load is interpreted as a differential pressure from one side of the fully closed door leaf to the other side. The EN 12424 standard distinguishes between 6 wind classes that are used to classify the door's wind resistance. [5]

The 6 wind classes together with the approximate conversion of wind speeds can be seen in table 2.1.

Wind class	Reference wind load [Pa]	Wind speed [m/s]	Remarks
0	-	-	No performance determined
1	300	22	-
2	450	27	-
3	700	34	-
4	1000	41	-
5	>1000	-	Exceptional. Agreement between manufacturer and purchaser.

Table 2.1: *Wind classes in the EN 12424 standard.*

2.3.2 EN 12604 - Industrial, commercial and garage doors and gates - Mechanical aspects - Requirements and test methods

The EN 12605 standard is a broad standard that covers many aspects of industrial doors. Since the standard is 20 pages long, only relevant excerpts will be presented. The standard mentions several structural safety requirements. These will be presented in a bullet list below. Also, the testing procedure to verify these requirements are listed below. [6]

The relevant requirements are:

- The minimum safety factor for materials for calculation purposes is 2.0. However, for components where testing is carried out instead of calculation, the minimum safety factor before yield shall be 1.1.
- The movement of the door leaves shall be limited by end stops. Mechanical stoppers in the terminal positions of the door movement shall withstand the energy developed by the possible impact of the door leaf.
- The door shall incorporate means suitable to prevent movement of the door due to the influence of wind at the terminal positions.
- An industrial door shall be able to open or close manually with a force of 260 N. This force can be exceeded to start the movement. In case of the door being designed for power operation but malfunctions, the physical effort can exceed this value by not more than 50%. [7]
- Sharp edges shall be eliminated or safeguarded to avoid risk of cutting when operating the door. Edges with radius of at least 2mm are considered to be safe.
- Drawing points of steel wire ropes, chains, straps that can be reached during normal operation shall be safeguarded up to a height of 2.5 m above floor level.

The standard also includes how verification of these requirements should take place:

- The door is to be operated in normal use for 10 cycles. The door shall after this continue to operate without any impairment of safety and operability, such as throttling of the movement, increased noise level and increased friction.
- Door leaf shall travel towards an 400x400x400 mm box made by a hard material (e.g. wood, metal, etc.) with a minimum speed of 0.3m/s. The door should be able to operate normally after the impact. In this case, the box is to be put on the floor in the running direction of the main closing edge next to the secondary closing edge.
- Door leaf shall travel towards its terminal positions twice with a minimum speed of 0.3m/s. The door should be able to operate normally after the impact. Damage to the end stops are controlled.

- It is to be verified visually that the means to hold the door leaf in position due to wind loads are effective at the terminal positions of the door.
- The manual operating forces are checked by measurement when the door is in each of the closed, middle and open positions.
- Protection against sharp edges and drawing points of steel wire ropes, chains and straps shall be inspected.

2.3.3 EN 12453 - Industrial, commercial and garage doors and gates - Safety in use of power operated doors - Requirements and test methods

The EN 12453 standard describes the risks related to safety of automatic gates and doors in detail. Again, only relevant excerpts will be presented. The standard includes limits of the crushing force, both in amplitude and duration. [7]

Measuring Equipment

The equipment used to conduct the force measurements shall consist of two contact areas with a diameter of 80 mm, a spring which gives the contact area a spring ratio of 500 N/mm and a load cell with an amplifier and a display unit. It should also be equipped with a plotter. An example of such measuring equipment is displayed in figure 2.7.

Figure 2.7: *Image of a measuring device designed specifically for the EN 12453 test.*

Crushing Force

The EN 12453 standard specifies that measurement of crushing forces shall be carried out on two different locations measuring the duration and value of peak forces for a folding door. The first measurement shall be carried out between a folding leaf and neighbouring stiff parts of the surroundings and the second one between the main closing edge and the opposing closing edge. Three measurements shall be made at each measuring point and the mean value obtained shall fulfil the specified requirements in figure 2.11.

In the first case the force measurement is taken according to figure 2.8, 1000 mm above floor level.

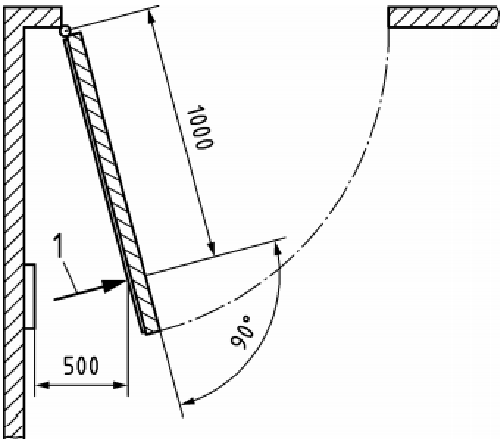

Figure 2.8: *Crushing force test method against neighboring stiff parts. Figure from [7].*

The next force measurements are made in between the folding door leaves' closing edges at three different opening gap widths, according to figure 2.9. Measurements are taken at three different heights, depending on the closing edge length. The heights can be seen in figure 2.10.

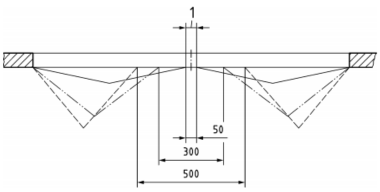

Figure 2.9: *Gap widths where crushing forces shall be measured on a folding door. Figure from [7].*

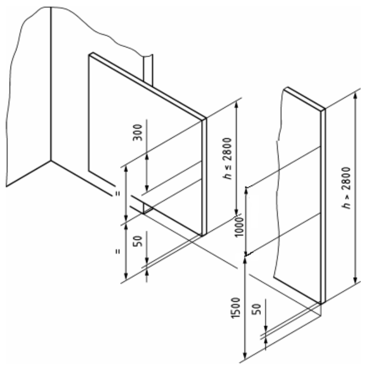

Figure 2.10: *Heights where crushing forces between closing edges shall be measured on a folding door. Figure from [7].*

The mean force value of the three measurements at these points must not exceed the limits seen in figure 2.11. The force diagram consists of 3 different stages. [8]

- IMPACT (Red area): The force during the first instants of contact. The maximum peak value shall be less than 400N and have a duration shorter than 0.75 seconds if the gap is up to 500 mm wide.
- CRUSHING (Yellow area): Represents the crushing force generated by the motor. This force continues to push after the impact. The average value of the static force must be less than 150 N and last for a maximum of 5 seconds.
- END PHASE (Blue area): After 5s from the initial contact, the residual force must be ≤25 N.

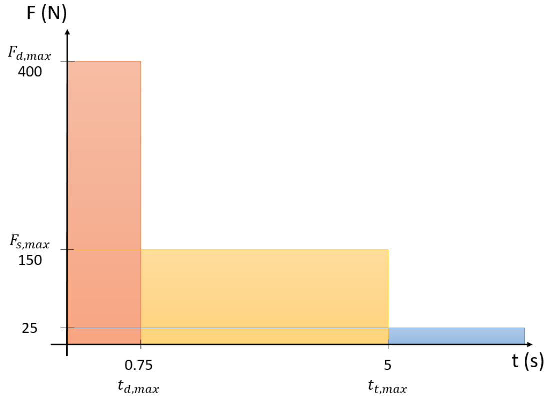

Figure 2.11: *The test graph appearance and its three phases.*

2.4 General about Gears and Transmissions

Gears are often used when there is a need to change speed, torque or direction of a mechanical power source. [9]

Something to keep in mind when using gears is that a downshift in speed (i.e. a gear ratio lower than 1) results in a proportional increase in torque. For example, in theory, a gear ratio of 1:2 results in half speed but double torque [10]. Another useful rule of thumb when manufacturing gears is that the minimal number of teeth should be higher than 17 to avoid undercutting. [11]

There are many different types of gears, but this report will only look at planetary and worm gears. A short explanation to these as well as some mathematical relationships useful when working with gears can be found in the respective sections below.

2.4.1 Mathematical Relationships and Definitions

Some useful definitions when working with gears:

Module m is defined as the relationship between the gear pitch and π.

$$m = \frac{P}{\pi} \tag{2.1}$$

Pitch P is defined as the relationship between the gear pitch diameter, d_0 and the number of teeth, z.

$$P = \frac{d_0 * \pi}{z} \tag{2.2}$$

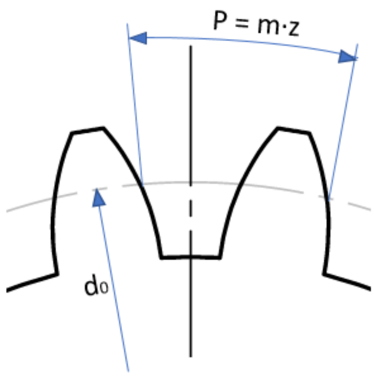

Figure 2.12: *Useful gear definitions.*

Combining equations 2.1 and 2.2 gives an expression that ties the module, pitch diameter and number of teeth together, according to equation 2.3 below.

$$m = \frac{d_0}{z} \tag{2.3}$$

2.4.2 Planetary Gear

A planetary gear consists of four main components, as illustrated in figure 2.13. Planetary gears are popular due to their efficiency and possibility to achieve a high gear ratio on a compact area. [12]

- Sun gear (Green)
- Planet gears (Blue)
- Carrier (Red)
- Ring gear (Grey)

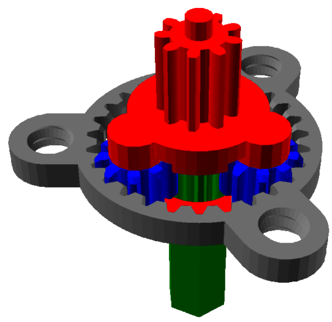

Figure 2.13: *Planetary gear. Figure from [12]. Public domain.*

If we denote some variables:

- T_r = Speed of the ring gear
- T_s = Speed of the sun gear
- T_y = Speed of the planetary gear carrier.
- Z_r = Ring gear teeth
- Z_s = Sun gear teeth
- Z_p = Planet gear teeth

The gear ratio of a planetary gear is [13]:

$$T_y * (Z_r + Z_s) = T_r * Z_r + T_s * Z_s \tag{2.4}$$

Assuming the ring gear is fixed, i.e. the speed of it is zero (T_r=0) and the sun is driving, the gear ratio can be rewritten as:

$$T_y = T_s * \frac{Z_s}{Z_r + Z_s} \tag{2.5}$$

2.4.3 Worm Gear

The worm gear is known for its ability to achieve high torque and low speed. A worm gear set can be seen in figure 2.14, where the worm screw is on the bottom and the gear on top. [9]

Figure 2.14: *Worm gear set. Figure from [9]. Public domain.*

The worm screw can have multiple starts which means that it has multiple helices or threads. For a single start worm screw, the gear reduction ratio is equal to the number of teeth on the gear. For example, if the gear has 32 teeth the gear reduction ratio is 1:32. When the worm screw rotates 360 degrees the gear advances one tooth. On the other hand, for multiple start worm gears the gear reduction equals the number of teeth on the gear divided by the number of starts on the worm screw. [14]
A picture showing the difference of a single start and multiple start worm screw is shown in figure 2.15.

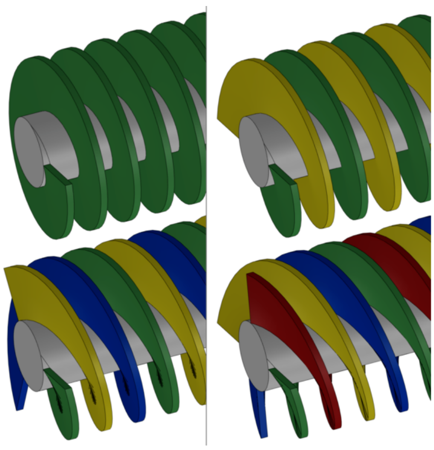

Figure 2.15: *Single and multiple start worm screw. Figure from [15]. Public domain.*

Self-inhibiting Properties of Worm Gears

The worm gear has the ability to self-lock. The worm screw can always drive the gear but if the gear tries to drive the worm and the lead angle is small, the gears teeth will lock against the worms teeth. This happens because the force component circumferential to the worm is not enough to overcome friction. In rare occasions the gear can drive the worm but only if the lead angle is large. [9]

The lead angle is the difference of the worm helix angle and the static friction angle. The static friction angle is the angle where the load begins to slide. [16]
Both angles can be seen in figure 2.16.

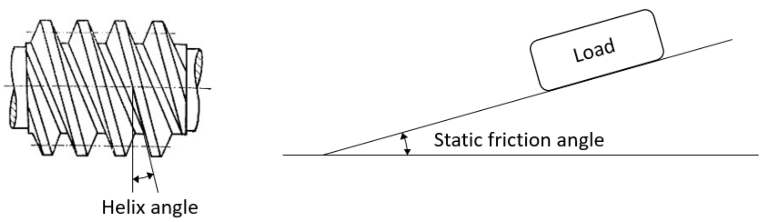

Figure 2.16: *The helix angle and static friction angle. Screw illustration from [17]. Public domain.*

3. Concept Development

In this chapter a comprehensive evaluation of the current drive system at ASSA ABLOY will be made. This will then be followed by a summary of the most critical design points of the product. Both internal and external concepts will then be generated and evaluated. The best candidate is chosen through concept selection.

3.1 Evaluation of the Current Drive System

3.1.1 Motor

The electric motor ASSA ABLOY is using in all of their electrically operated industrial folding doors is the CDM9 model seen in figure 3.2. This motor is a single phase induction motor, also called asynchronous motor, that includes frequency control for soft start and a stop function. Frequency control allows the motor to vary its speed up to 60 rpm. However, the maximum speed at the highest load is 21.5 rpm. Complete motor specifications are listed in table 3.1. This motor is what the concepts will use. Since the motor includes a worm gear, it has self inhibiting properties as described in chapter 2.4.3. The maximum reverse torque the worm gear and motor can withstand in a static situation is 200 Nm.

Parameter	Value
Voltage	1-phase 230 V
Frequency	50 Hz
Rated Current	2 A
Rated Power	0.5 kW
Rated Speed	1440 rpm
Shaft output Speed	60 rpm@140 Hz (Max) 21.5 rpm@50 Hz (τ_{Max})
Peak/rated torque	70/50 Nm

Table 3.1: *Motor specifications. Data from [18] and figure 3.1.*

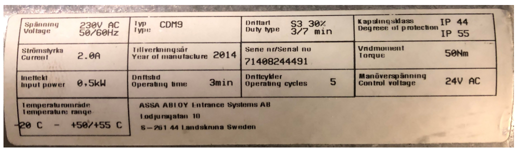

Figure 3.1: *CDM9 Marking.*

Figure 3.2: *Current motor. Figure from [18].*

3.1.2 Current Drive System

The electrically operated folding door available today is opened/closed using the CDM9 motor mentioned in chapter 3.1.1 together with a mechanical transmission rail (see figure 3.3). The transmission rail conceals a chain mechanism with attached carts (see figure 3.4) that slide when the door operates. These carts are anchored to the door leaves via a hinged arm, that rotates as the door moves. As seen in 3.4, the arm ends up at a favorable angle when the door is shut. This almost perpendicular angle means that the motor does not need to work as much when counteracting an outside wind load on the door. The required torque from the motor when a wind load is applied will be calculated in chapter 3.1.4.

Figure 3.3: *The mechanical transmission rail. Figure from [19].*

Figure 3.4: *The current drive system at ASSA ABLOY.*

3.1.3 Tests

Current Consumption

The purpose of evaluating current consumption is to get an estimation of how much torque is needed to operate the door, without any influence from wind load. A current measurement test of the opening and closing cycle of the folding door available in the lab at ASSA ABLOY's premises was conducted. From previous internal tests it has been discovered that the torque and current have a relationship that can be modeled linearly, according to figure 3.5.

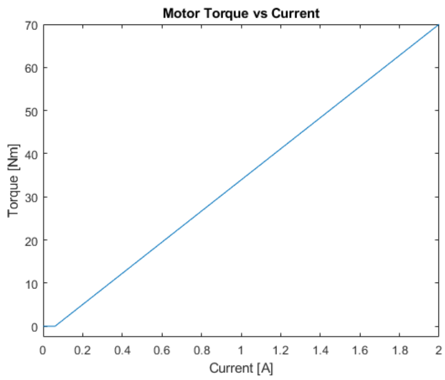

Figure 3.5: *CDM9 torque and current diagram.*

Figure 3.6: *Image of the door available in ASSA ABLOY's lab.*

The test was conducted using a clamp meter on a 2+0 door available in the lab (measuring 2.5x3 m) seen in figure 3.6. Results for the opening and closing cycles are presented in figures 3.7 and 3.10 below. In figures 3.8 and 3.11 the corresponding motor torque is plotted. Motor torque is calculated using the graph in figure 3.5.

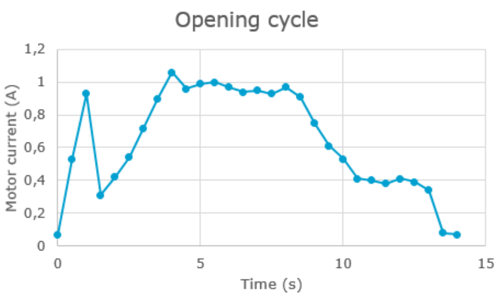

Figure 3.7: *Current consumption of the motor during opening.*

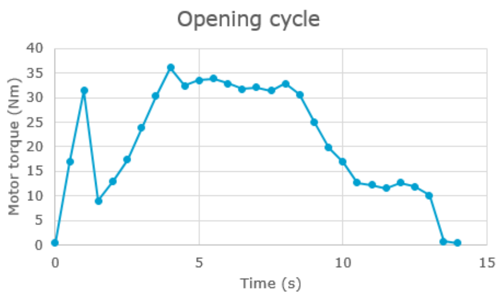

Figure 3.8: *Motor torque during opening.*

The current consumption during opening is reasonable. First an increase in current flow is logged thereafter the consumption levels out and stagnates before it decreases again as the door slows down. The first power spike is due to the fact that the motor has to counteract the friction of the sealing strips. Even if this spike only last for approximately one second this is where the motor outputs a high torque. Next peak is when the arm connecting the driving chain to the door is horizontal, see figure 3.9. After this stage, the current consumption stabilizes and this is because the motor now only has to drag the door open, (i.e. it only needs to counteract the weight of the door and possibly the friction in the rail in which the door pin slides). The test results show that even if the motor is rated for 2 A, the power consumption never reaches this value. This fact provides potential regarding torque increase when designing other drive system solutions.

Figure 3.9: *Arm connecting door to driving chain in horizontal position (4 seconds into the opening cycle).*

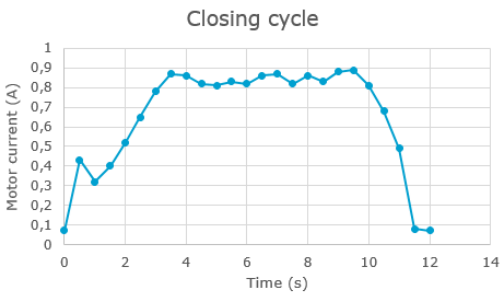

Figure 3.10: *Current consumption of the motor during closing.*

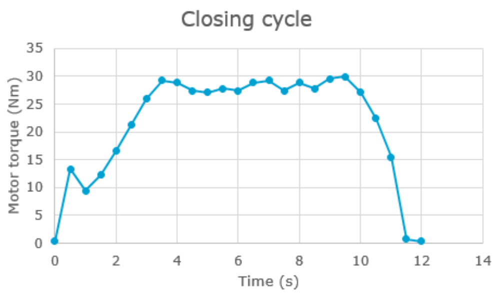

Figure 3.11: *Motor torque during closing.*

Results of the measurements during the closing cycle are also reasonable. The current consumption increases as the door accelerates and then stabilizes before the eventual decline. Peak current value occurs near the end of the cycle due to it using a high amount of torque to compress the sealing strips around the door.

Crushing Forces according to EN 12453

The same door was also tested to determine the crushing forces according to the EN 12453 standard (see chapter 2.3.3). Results from this test can be seen in table 3.2. Peak force values are well below the 400 N and 750 ms limit stated in the EN 12453 standard. It was not possible to measure the crushing forces against neighboring stiff parts due to the absence of walls or nearby rigid objects in the test lab. The crushing forces were only measured between the closing edge and door frame.

- **Door: FD2050P**
- **Control unit: ECS 950**
- **Machinery: CDM9FD**
- **Sensor: Bircher DW 40 Pneumatic**

Crushing force test				
Height from floor [mm]	**Opening gap [mm]**	**Crushing force [N]**	**Time while force >150 N [ms]**	**Unload within 5 s**
50	50	265	157	YES
50	300	264	184	YES
50	500	116	0	YES
1500	50	225	130	YES
1500	300	224	129	YES
1500	500	186	174	YES
2500	50	217	154	YES
2500	300	232	197	YES
2500	500	206	124	YES

Table 3.2: *Results from the EN 12453 test conducted on ASSA ABLOY's current solution with mechanical rail. Crushing force and time while force >150 N are averages from 3 measured values. The maximum allowed force between closing edges is 400 N within a period of maximum 0.75 s and unload (reverse) within 5 s.*

Both current measurement and crushing force tests were also carried out on a 2+2 door, measuring 3.6x3 m. Results of these tests can be seen in Appendix A.

Opening and Closing Speed

The door opening and closing speed was measured using a regular stopwatch. The measurement started when the close/open button was pushed and lasted until the door was completely closed or fully opened.

Results of the opening and closing speed for 5 cycles can be seen below in table 3.3.

Number of test	1	2	3	4	5
Opening cycle	14.16	14.07	13.84	13.98	13.91
Closing cycle	12.29	11.74	12.08	12.01	12.08

Table 3.3: *Results of opening speed test. All values in seconds.*

3.1.4 Wind Load

The electrically operated folding doors at ASSA ABLOY are often tested and rated for up to a class 3 wind load according to the EN 12424 standard. See chapter 2.3.1 for more information on the wind load classification standard.

In order to compare doors where physical tests are not possible, such as competitors doors and generated concepts, it was decided to use a 2+0 door configuration that has the width and height dimensions 2.5x5 meters. This is the largest door available from many companies and will act as a reference point. This door configuration will be used across all calculations in the report if nothing else is stated. A 2+0, 2.5x5 meter door means, one door leaf will be 1.25 meters wide and 5 meters high.

A sketch of the folding door with the applied wind load was drawn. The sketch can be seen in figure 3.12. From the sketch, a free body diagram of the folding door was created to identify the opposing force required to counteract the wind load. The free body diagram can be seen in figure 3.13. In the free body diagram the opposing force on the door (Fu) comes from the motor via the mechanical transmission rail described in chapter 3.1.2.

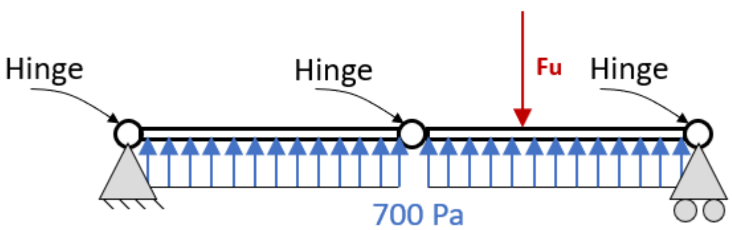

Figure 3.12: *Sketch of folding door with applied wind load.*

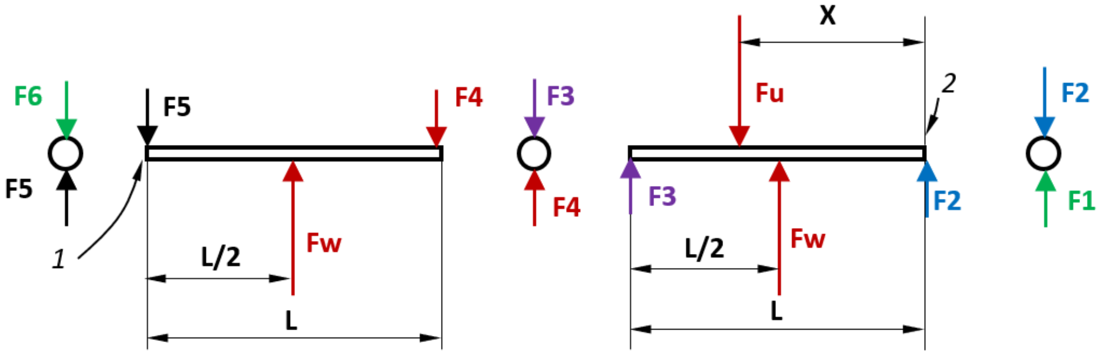

Figure 3.13: *Free body diagram of folding door with applied wind load.*

The free body diagram in figure 3.13 gives the following equilibrium equations:

$F_w = p * A,\ where\ p = 700Pa\ (wind\ class\ 3)\ and\ A = L * 5\ m^2$

$$\overset{\frown}{1} : \frac{L}{2} * F_w - L * F_4 = 0 \tag{3.1}$$

$$\overset{\frown}{2} : x * F_u - L * F_3 - \frac{L}{2} * F_w = 0 \tag{3.2}$$

$$\uparrow : F_4 - F_3 = 0 \tag{3.3}$$

Combining equations 3.1 through 3.3 gives:

$$F_u = \frac{L * F_w}{x} \tag{3.4}$$

Inserting the measurements from the current solution at ASSA ABLOY, $x = 0.59 * L\ and\ L = 1.25$ into equation 3.4 gives:

$$F_u = \frac{1.25 * 700 * 1.25 * 5}{0.59 * 1.25} = 7415\ N \tag{3.5}$$

3.1.5 Operating Forces

When the required opposing force on the door has been determined it is possible to calculate the motor torque required to counteract the wind load. A sketch of the door is drawn where the motor is included. This sketch can be seen in figure 3.14. Also, the link arm connecting the motor to the door via the chain inside the mechanical transmission rail is drawn in a free body diagram. See figure 3.15.

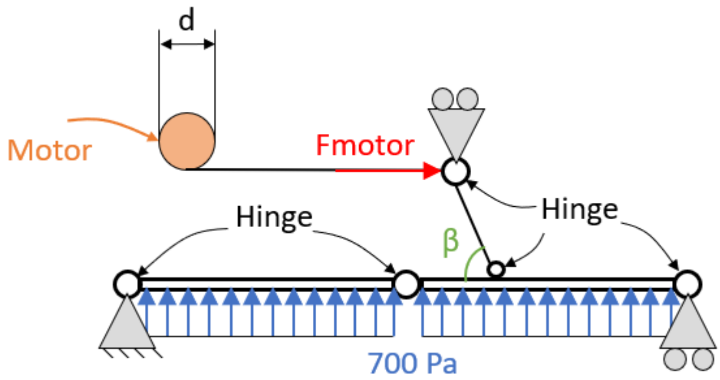

Figure 3.14: *Sketch of ASSA ABLOY's current drive system.*

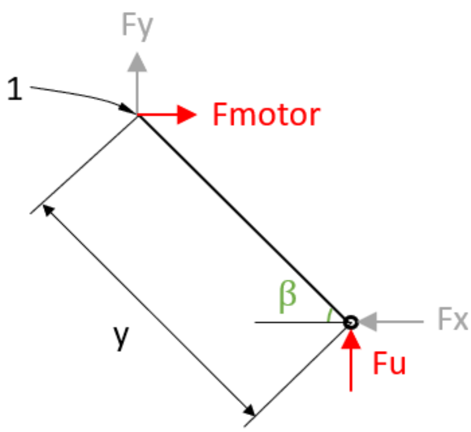

Figure 3.15: *Free body diagram of link arm connecting the door to the chain inside the mechanical transmission rail.*

The free body diagram in figure 3.15 gives the following equilibrium equations:

$$\overset{\frown}{1} : F_u * y * cos(\beta) - F_x * y * sin(\beta) = 0 \tag{3.6}$$

$$\rightarrow: F_{motor} - F_x = 0 \tag{3.7}$$

Combining equations 3.6 and 3.7 gives:

$$F_{motor} = \frac{F_u * y * cos(\beta)}{y * sin(\beta)} = \frac{F_u}{tan(\beta)} \tag{3.8}$$

Inserting values from the current solution at ASSA, β=65.2°, d=0.0524 m and F_u from equation 3.5:

$$F_{motor} = \frac{F_u}{tan(\beta)} = \frac{7415}{tan(65.2°)} = 3426\ N \tag{3.9}$$

$$\tau_{motor} = F_{motor} * \frac{d}{2} = \frac{3426 * 0.0524}{2} = 89.8\ Nm \tag{3.10}$$

For a 2+2 door with the same leaf width and height, the required torque is doubled, resulting in a required motor torque of 179.5 Nm.

When compared to the motor specifications in table 3.1, we can see that for the motor to be able to withstand a wind load of class 3, according to the EN 12424 standard, it has to produce a 179.5 Nm torque. Since the motor torque output peaks at 70 Nm, the conclusion can be drawn that the door would not be able to operate during a 700 Pa wind load. However, in a static situation where the door remains closed, the motor would be able to keep the door shut. This is due to the self-inhibiting properties of worm gears discussed in chapter 2.4.3. The gearbox is able to absorb about 200 Nm of reverse torque in a locked state without moving. This has been tested by Sr. Mechanical Engineer at ASSA ABLOY.

Another conclusion that can be drawn is that the motor is able to operate at about 1/4 of the wind load. Actually, according to the Sr. Mechanical Engineer, this is what the normal operating conditions are for every folding door. This information will be used when comparing ASSA ABLOY's current drive system to competitors and when designing/evaluating concepts.

ASSA ABLOY's 2+2, 5x5 meter folding door requires under normal operating conditions (1/4 wind load): $1/4 * 179.5 = 44.9\ Nm$

The 2+0 counterpart requires half as much torque from the motor, i.e. 22.5 Nm under normal operating conditions.

Summary

A summary of the torque requirements calculated for ASSA ABLOYS's current drive system is presented in table 3.4 below.

Parameter	ASSA ABLOY (2+0)	ASSA ABLOY (2+2)
Torque @ Wind load	89.8 Nm	179.5 Nm
Torque @ Normal conditions	22.5 Nm	45 Nm

Table 3.4: *A summary of the calculated torque requirements for ASSA ABLOY's current drive system.*

3.1.6 Space Requirements

In the table 3.5 below, the space requirements for the current drive system are presented. The minimal required depth is the width of a door leaf + 180 mm. In this case, this means 1430 mm. The SL, SR, OH abbreviations are described in chapter 1.2.

ASSA ABLOY - Current drive system									
No plastic cover on operator									
Configuration	**Op. Pos. Left**			**Op. Pos. Right**			**Op. Pos. Center**		
	SL	SR	OH	SL	SR	OH	SL	SR	OH
2+0	450	50	240	320	200	240	N.A.	N.A.	N.A.
2+2	450	320	240	320	500	240	320	320	380
With plastic cover on operator									
Configuration	**Op. Pos. Left**			**Op. Pos. Right**			**Op. Pos. Center**		
	SL	SR	OH	SL	SR	OH	SL	SR	OH
2+0	470	50	270	320	240	270	N.A.	N.A.	N.A.
2+2	470	320	270	320	540	270	320	320	390

Table 3.5: *Measurements of ASSA ABLOY's current drive system with and without a plastic cover. All values in mm. Data from [19].*

3.2 Customer Needs

3.2.1 Specifications from ASSA ABLOY

Together with ASSA ABLOY's folding door product manager, some important factors when designing concepts and reasonable specifications were discussed in order to create a more compact folding door drive system. The meeting resulted in the following specification table:

Parameter	Demand	Unit	Remarks 1	Remarks 2
Life time	100 000	cycles	Acc. to EN 12605	Same as operator
Life time	10	years	Acc. to EN 12605	
Wind load	Class 3		Acc. to EN 12424	Min. class 3, for 5000x5000 door
Min. width	1400	mm	LW min. 550 mm	
Max width	5000	mm	LW max. 1250 mm	
Min. height	2000	mm		
Max height	6000	mm		
Side space, both sides	200	mm	Max	
Headroom	200	mm	Max	
Installation	Inside and outside			
Opening	Inside and outside			
Opening speed			Twice as today	
Burglar protection SK2/SK3	TBD		To be decided	
Designed for service	YES			
DoC	YES		Declaration of Conformity	
DoP	YES		Declaration of Performance	
Installation time	Reduced by 50%		Compared to FD2250P operated	

Table 3.6: *Requirement specification for the drive system concept.*

The drive system must also be compliant with the maximum crushing forces in the EN 12453 standard.

3.3 Concept Generation

3.3.1 Clarify the Problem

Clarifying the problem is of most importance in order to generate feasible concepts. During this process, the main problem is divided into subproblems, i.e. problem decomposition. Concepts that solve the subproblems are then generated individually. These concepts are later combined to create a single concept. One concept can also solve all subproblems simultaneously.

Folding Motion

The first subproblem of the drive system that needs to be addressed is to implement a mechanical transmission that creates the folding motion. In this section, the principles of folding door motion will be explained and the subproblem will be clarified. Figure 3.16 illustrates the simple principles of motion that the door has to move in order to open or close. The folding motion is created by moving the door leaves either horizontally or vertically, according to figure 3.16. This is due to the guide rail steering the door correctly. Since the doors horizontal movement decreases as the door closes, a vertical force on either door leaf is desired for the final part of the closing movement. The opposite is true when the door is near its open position. Therefore, a combination of these two motions is desired.

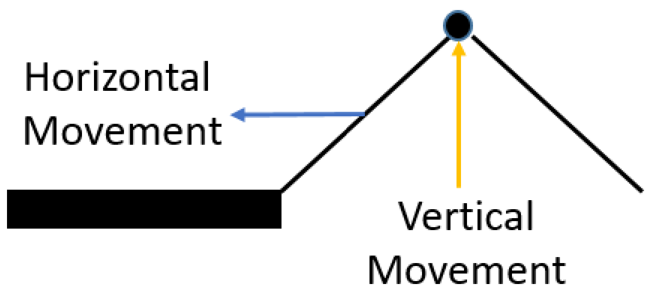

Figure 3.16: *The door's simple principles of motion.*

Wind Load

The second subproblem is the wind load resistance. It is very important for the folding door to be able to withstand a strong wind.

In figure 3.17, it can be seen that the wind load tries to push the door open. If the wind load is so strong that the drive system is not able to resist, the wind will force the door open leading to costly repairs. When closed, the door therefore requires a force that works against the folding motion described in chapter 3.3.1 in order to remain closed.

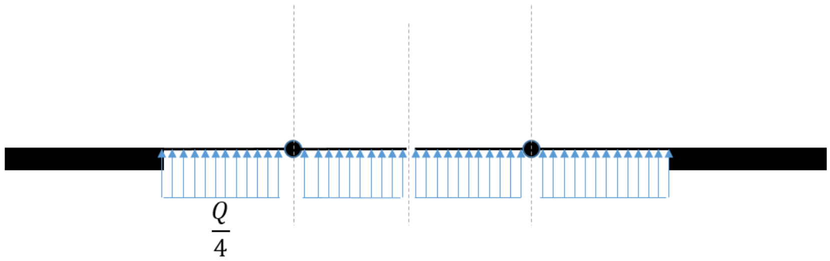

Figure 3.17: *Sketch of wind load applied to a folding door in its closed state.*

3.3.2 Search Internally

Concepts Solving the Folding Motion Subproblem

In figures 3.18-3.27 below, the concepts generated are presented and followed by a brief description. All concepts are based on a 2+0 configuration if nothing else is mentioned. This means that the required number of components is doubled for a 2+2 door configuration.

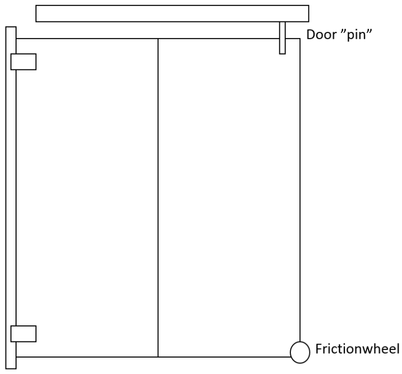

Figure 3.18: *Concept 1: Friction wheels.*

The focus when coming up with the concept in figure 3.18 was minimizing the drive system dimensions. If controlling the friction wheel's direction is achievable, it would be possible to control the door's movement. This concept solves the folding motion subproblem.

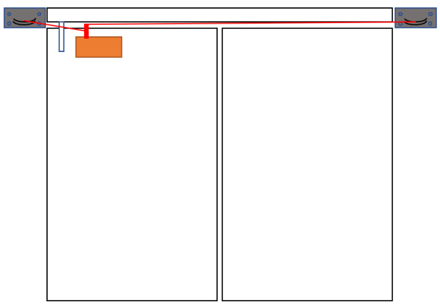

Figure 3.19: *Concept 2: Cable concept. Motor in orange.*

The concept in figure 3.19 shows two red lines resembling a cable. This cable is attached to a cylinder which is connected to the motor shaft. When the motor rotates, the cable pulls the door open. This concept solves the folding motion subproblem.

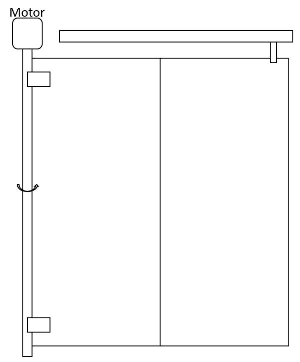

Figure 3.20: *Concept 3: Spinning hinges.*

The spinning hinges concept seen in figure 3.20 works in the following way: The motor applies a torque to the hinges, which makes them rotate opening the door. This concept solves the folding motion subproblem.

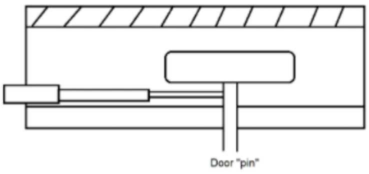

Figure 3.21: *Concept 4: Telescope drive.*

The concept shown in figure 3.21 consists of an arm with similar properties to a telescope that pushes the door pin inside the door rail. This concept will also solve the folding motion subproblem.

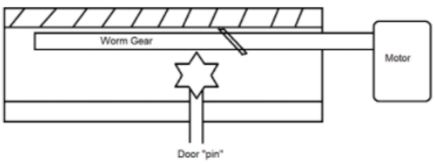

Figure 3.22: *Concept 5: Worm gear.*

The concept shown in figure 3.22 uses a worm gear to achieve the motion of a folding door. The linear motion of the door pin, together with the rail creates the folding motion. The worm screw is directly connected to the motor which, in turn, will force the door pin to move inside the rail. This concept can also be used on a 2+2 folding door with one motor driving both parts. This is possible by having half of the screw left-hand threaded and the other half right-hand threaded. This would allow both halves of the 2+2 door to move in opposite directions.

Concepts Solving the Wind Load Subproblem

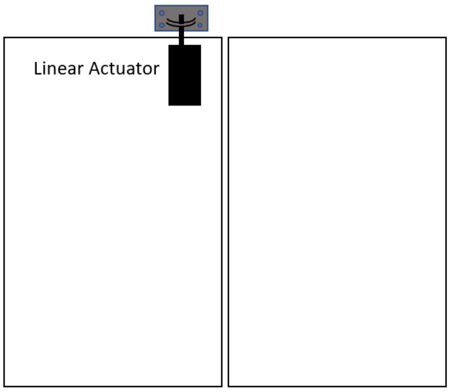

Figure 3.23: *Concept 6: Door lock.*

In figure 3.23 a solution to the wind load subproblem is presented. This concept works similar to a normal door lock. When the door is in its closed state the linear actuator locks the door by extending an arm that goes into a designated loop attached to the wall.

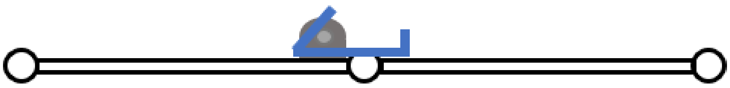

Figure 3.24: *Concept 7: Force diverter.*

Another solution to the wind problem is the force diverter seen in figure 3.24. When the door closes, the diverter grabs onto a fixed arm attached to the wall or door and converts the horizontal force from the movement to a vertical force that resists the wind load. This creates a powerful closing mechanism.

Figure 3.25: *Concept 8: Locking house. Figure from [20].*

The locking house concept shown in figure 3.25 works like it would on a normal door. When the door closes, the locking house connects with the corresponding part on the door frame above the door keeping it shut. This solution only requires input when opening.

Concepts Solving both Subproblems

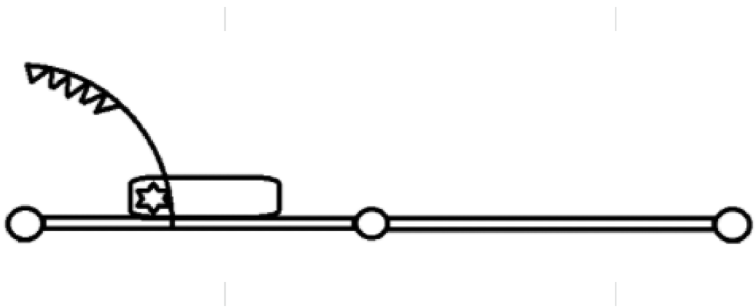

Figure 3.26: *Concept 9: Pinion and bent rack.*

Concept nine, shown in figure 3.26, resembles a straight pinion and rack. The rack is bent and the pinion is mounted on the door leaf. When the motor applies a torque the pinion will start to rotate and climb the rack dragging the door open. When the door is about to open the vertical force will be at its maximum and when the pinion is at the end of the rack the vertical force will be at its minimum. The horizontal force will function inversely to the vertical force. When the horizontal force is at its maximum the vertical force will reach its minimum.

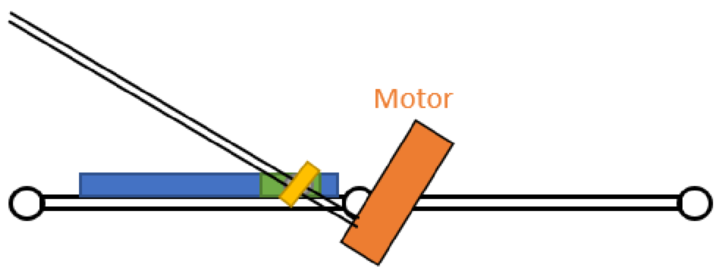

Figure 3.27: *Concept 10: Screw concept.*

Concept ten, shown in figure 3.27, consists of a screw connected to the motor and a nut mounted on a swivel connected to the door via a sliding rail. When the motor applies a torque to the screw, it will start to rotate and subsequently start pushing the nut. This will, in turn, force the door to open. This concept generates a force component perpendicular to the door when closed.

3.3.3 Search Externally

After investigating competitors drive systems, it was possible to divide the findings into four main categories.

Side Mounted

The first category is the side mounted motor. These types of folding doors have the motor mounted on the wall, to the side of the door. There are two motors on each side of the door if it is a 2+2 configuration. As the motor shaft spins, the link arms drag the door open or push the door shut. See figures 3.28 and 3.29.

Figure 3.28: *Side mounted motors on a car wash.*

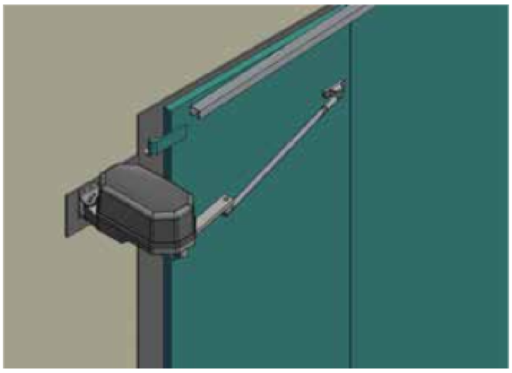

Figure 3.29: *Side mounted FAAC motors. Figure from [21].*

Backstage Mounted

The backstage mounted motor is very similar to the side mounted motor but instead sits above the door. The arm connected to the motor shaft sits in a sliding rail that is attached to the outer door leaf. As the motor rotates, the end of the arm will slide in the rail as it opens the door. See figure 3.30.

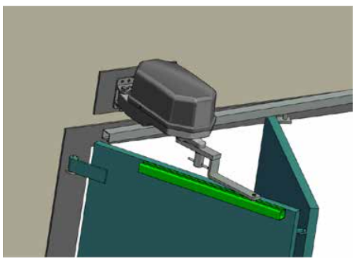

Figure 3.30: *Backstage Mounted FAAC motor. Figure from [21].*

Ditec Dor Solution

The Ditec Dor solution is very compact as the motor sits on one of the door leaves. The arm connected to the motor shaft creates a folding force between the door leaves as the motor rotates. See figures 3.31 and 3.32.

Figure 3.31: *Ditec Dor Motor. Figure from [22].*

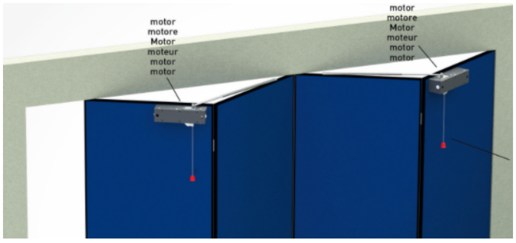

Figure 3.32: *Ditec Dor folding door. Figure from [22].*

Overhead Mounted

The final category has a larger variation over the different competitors. Three variants found can be seen in figures 3.33, 3.34 and 3.35 below. They work by having a centered motor that opens or closes the door via several link arms.

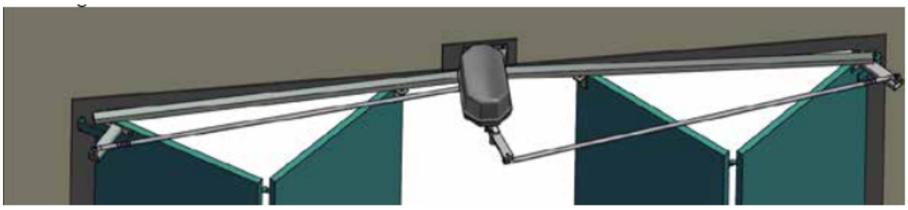

Figure 3.33: *EAB Standard Overhead Mounted. Figure from [21].*

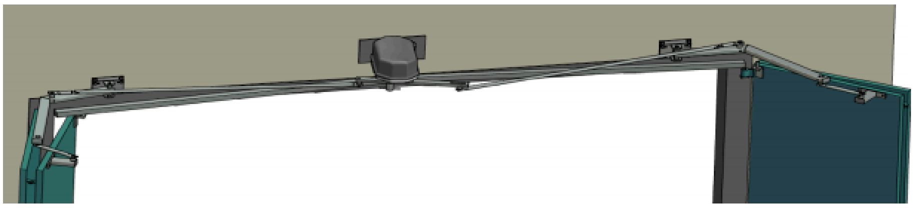

Figure 3.34: *EAB Limited Side Space. Figure from [21].*

Figure 3.35: *Prido Overhead Mounted Boomerang. Figure from [23].*

3.3.4 Reflections and Conclusions

In the first round of evaluation, it was decided to continue and evaluate all competitor concepts to see if any of them are of further interest in pursuing. Since these drive systems already are avaliable today, it would be of interest to not only provide ASSA ABLOY with some data on their competitors, but also see if it would be possible to achieve something similar with the limitations in this project. There might be a chance that a concept similar to these drive systems is very plausible to execute.

However, some of the internally generated concepts were filtered out due to different reasons. These will be discussed below.

Concept 1 - Friction wheels

It was decided that this solution was too hard to implement. This is mainly due to requirement of putting the motor or wheel on a swivel. Another reason for the concept's dismissal is that it only solves the folding motion subproblem. It most likely will not be possible for the wheel alone to withstand a pushing force on the door. This is a problem since there would be no possibility of burglar protection or wind resistance. It would have to be combined with some of the lock concepts (Concept 6 or 8). This would introduce another motor/actuator that operates these locks, further complicating the concept and driving up the cost. This concept was mainly disregarded due to the implementation difficulties with mounting the motor or wheel on a swivel.

Concept 2 - Cable concept

This concept was deemed not reasonable to go further with due to the fact that it would also have to be combined with one of the lock concepts. Another reason it was dismissed is the motor placement. In order for the cable to stay the same length, the motor would have to be placed near the door pin. This would put the motor at a disadvantage mechanically. I.e. the concept solves the folding motion subproblem as described in 3.3.1 but does not meet the requirements to fulfill the wind load subproblem. For the motor to have a chance of withstanding the perpendicular wind loads on the door, it would have to be placed in a center position. This is not possible with a cable since it would at some point during the door's motion require both sides of the cable to extend which is physically impossible. This concept was discussed with the supervisor at ASSA ABLOY, deeming it not feasible.

Concept 3 - Spinning hinges

The spinning hinges is a solution that solves the folding motion subproblem. But similar to the previous concept, it will most likely need locking mechanisms to be able to withstand the wind loads. Mechanically, the motor is at a great disadvantage to be able to counteract the perpendicular forces on the door, such as wind loads. Including locks into the concept will further complicate it, introducing more parts and a higher cost. This concept was discussed with the supervisor at ASSA ABLOY, deeming it not feasible.

Concept 4 - Telescope drive

This concepts solves the folding motion subproblem. The issue with this concept is that there is nothing that retracts the telescopic arm once it reaches its max. It would require another arm, cable, an external force or motion that is able to retract the arm. The service and repair cost would also be high due to the positioning of the arm inside the rail. A total dismantling of the drive system would most likely be necessary. The concept was deemed not feasible due to the high costs and difficulties to implement.

Concept 5 - Worm gear

An interesting and compact concept. The ability of the worm gear to be able to withstand a perpendicular force on the door is problematic due to the mechanical disadvantage. Similar to the telescope drive, service and repair costs would most likely be high since a total dismantling would be necessary if a problem occurs. Also, having the motor mounted on the wall beside the door would eat into valuable side space. Due to large side space requirements and difficulties withstanding a wind load it was decided to not continue with this concept.

Concept 6 - Door lock

The door lock concept works like a hasp where a motor extracts and retracts a pin into the designated loop to achieve a locking mechanism. This mechanism solves the wind load problem but in the end the concept was deemed not feasible due to the requirement of actuators. This would require an update of the controller cards that already exist today and would only add on to the cost of the motors required to move the door in a combined solution. It was decided not to go forward with this concept because of high costs.

Concept 7 - Force diverter

This concept was discussed further with the supervisor at ASSA ABLOY. The concept works as described in chapter 3.3.2. It converts the horizontal force that opens the door to a vertical force, making the door strong when closing. This concept is viable to combine with almost any concept that solves the folding motion subproblem and does not require an external actuator. Therefore it was decided to conduct a deeper investigation. The results of the investigation are presented in chapter 3.4.1.

Concept 8 - Locking house

Similar to the door lock concept (Concept 6), this would also require a motor/actuator to function. As long as an automated locking mechanism with a separate motor/actuator is needed, it should be avoided since it adds cost and complexity to the solution.

Concept 9 - Pinion and bent rack

This solution was deemed feasible by our supervisor at ASSA ABLOY. Since the solution will solve both subproblems and is estimated to be compact and cost efficient, it was decided to continue with this concept.

Concept 10 - Screw concept

Since the solution will solve both subproblems and is estimated to be compact and cost efficient, it was decided to continue with this concept as well.

Summary Table

In table 3.7 below, the internally generated concepts' verdict is shown. The table summarizes the discussion in the sections above.

Concepts	Continue?	Main reasons
1. Friction wheels	NO	Implementation
2. Cable concept	NO	Implementation
3. Spinning hinges	NO	Cost and implementation
4. Telescope drive	NO	Implementation and service cost
5. Worm gear	NO	Implementation, large side space
6. Door lock	NO	Cost, additional components
7. Force diverter	INVESTIGATE	Flexible solution, not a complete concept
8. Locking house	NO	Cost, additional components
9. Pinion and bent rack	YES	Small solution, solving both subproblems
10. Screw concept	YES	Small solution solving both subproblems

Table 3.7: *Table showing all internally generated concept and their verdict.*

3.4 Evaluation of Internal Concepts

3.4.1 Force Diverter Investigation

Initial Changes

Initially, the idea was to mount the force diverter to the rail and drive it with the folding motion of the door leaves. It was realised that an advantageous force conversion from the linear force of the inner door leaf to the perpendicular force on the outer door leaf would not be possible. The initial model can be seen in figure 3.36 below.

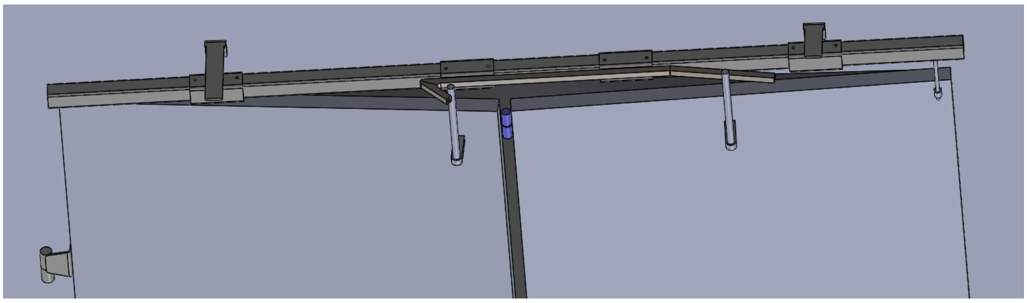

Figure 3.36: *Initial concept of force diverter in CAD.*

To increase the force conversion, a solution where the force diverter is directly driven by the motor was developed. Assuming the motor is connected to the driving pin via either a chain, cable or screw, the driving pin will move inside the rail. The driving pin is then connected to the door pin via a compression spring, according to figure 3.37. The spring is strong enough to not compress when the motor only drives the door, but weak enough so that the motor is able to compress it after the door is fully closed. When the door is fully closed, the motor will start to compress the compression spring. This state is illustrated in figure 3.38. The motor continues to compress the string until the force diverter has locked in the leaf pin sitting on the outer door leaf, completing the closing cycle. This can be seen in figure 3.39.

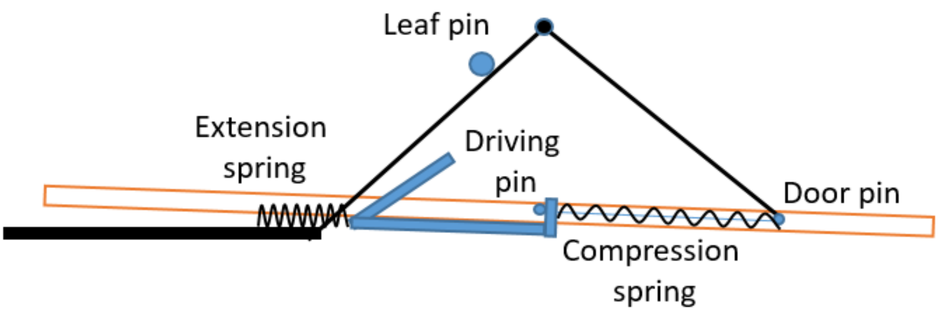

Figure 3.37: *Force diverter in a semi-open state.*

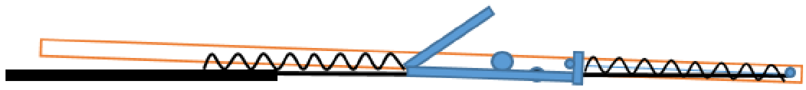

Figure 3.38: Force diverter in a closed state and compression spring half compressed.

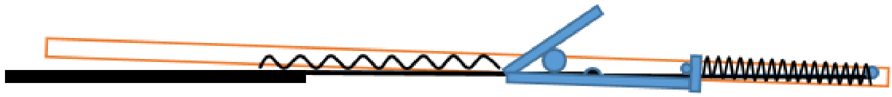

Figure 3.39: Force diverter in a closed state and compression spring fully compressed.

Downsides and Exclusion

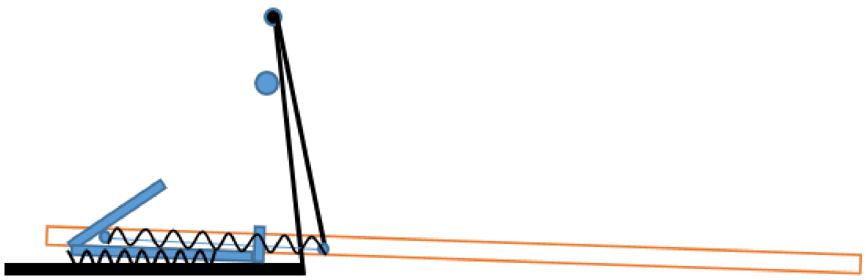

Figure 3.40: Force diverter in a fully opened state.

This concept has some downsides, see figure 3.40. When the door is in its fully opened state the compression spring has to follow the door and extend out into the side space. Not to mention that the motor has to be connected to the driving pin. The only reasonable way to connect the motor to the driving pin located inside the guide rail, is via either a chain, cable or screw. Since the motor now would have to be installed on the side of the guide rail, it would intrude further into the available side space. Therefore, this solution to the wind load subproblem will not be further developed and will not be included in the scoring matrix since it is not a complete concept.

3.4.2 Pinion and Rack

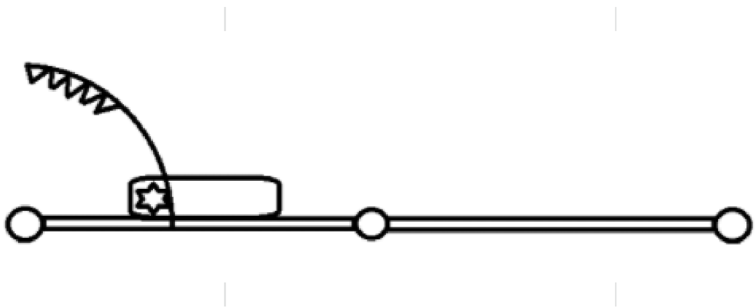

Figure 3.41: *Concept 9: Pinion and bent rack.*

The concept in its original shape can be seen in figure 3.41.

Initial Changes

Since the motor is able to withstand higher wind loads as it is placed closer to the middle hinge (see equation 3.4 in chapter 3.1.4), it was decided that the motor should be placed more towards the middle. However, this introduced another problem. The motor would protrude outside the door dimensions in an open state more when mounted closer to the middle hinge on the outer door leaf. To minimize the side space required for the concept, it was decided to move the motor to the inner door leaf instead, see figures 3.42 and 3.43 for comparison.

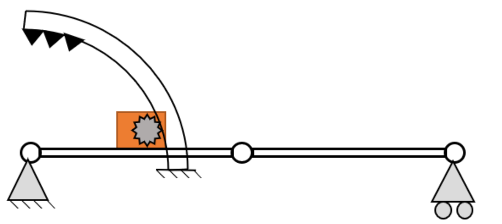

Figure 3.42: *Motor placed on outer door leaf.*

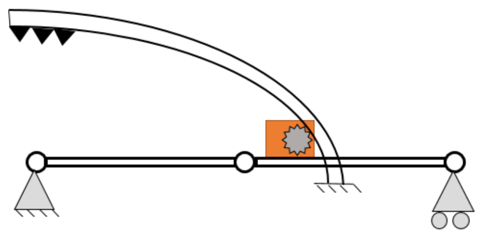

Figure 3.43: *Motor placed on inner door leaf.*

This solution would require a fixed elliptical rack, instead of a fixed circular rack as seen in the figures. The eliptical rack would most likely be hard to manufacture and therefore expensive. Since cost always is a limiting factor in these types of projects, it was decided to replace this with a straight rack mounted on a hinge, according to figure 3.44.

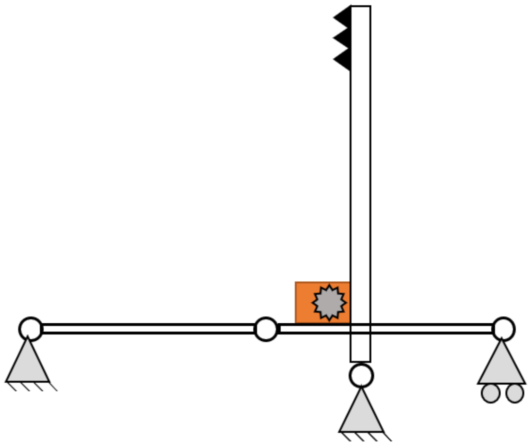

Figure 3.44: *Straight rack on a hinge with motor on inner door leaf.*

Operating Forces

A sketch of the wind load acting on the door is created, see figure 3.45. To be able to calculate a gear pitch diameter, the force necessary to counteract the wind load had to be determined. Inserting values $x = 0.8 * L$ *and* $L = 1.25$ into the formula for $\mathrm{F_u}$ (equation 3.4), derived in chapter 3.1.4, the necessary force in this case is:

$$F_u = \frac{L * F_w}{x} = \frac{1.25 * 700 * 5 * 1.25}{0.8 * 1.25} = 5468\ N \tag{3.11}$$

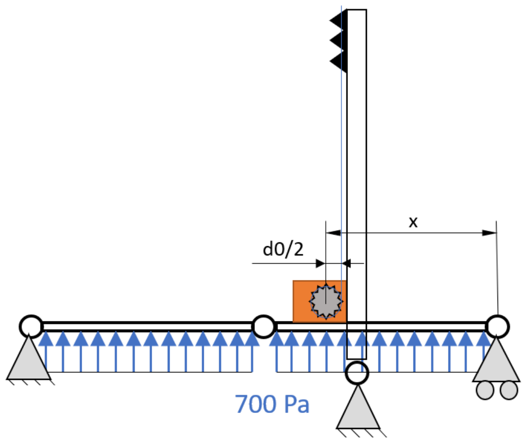

Figure 3.45: *Sketch of the folding door with applied wind load.*

When the force necessary to counteract the wind load has been determined, a free body diagram of the gear is drawn. See figure 3.46. The free body diagram gives an expression for the torque required to withstand the wind load.

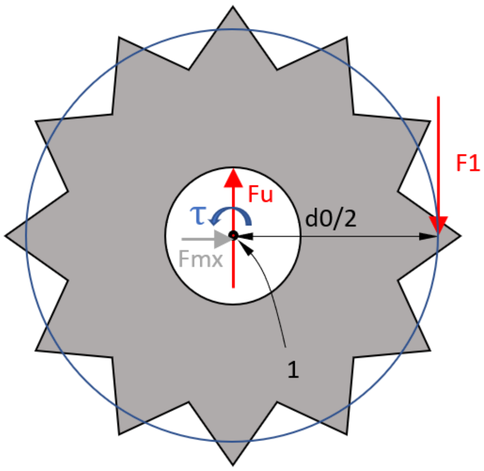

Figure 3.46: *Free body diagram of the gear seen in figure 3.45.*

Equilibrium equations from the free body diagram in figure 3.46:

$$\overset{\curvearrowleft}{1} : \tau - \frac{d_0}{2} * F_1 = 0 \tag{3.12}$$

$$\uparrow \; : F_u - F_1 = 0 \tag{3.13}$$

Combining equations 3.12 and 3.13 and assuming a gear pitch diameter of eg. 0.100 m results in a torque of:

$$\tau = \frac{d_0}{2} * F_u = \frac{0.100}{2} * 5468 = 273\ Nm \tag{3.14}$$

Under normal operating conditions (1/4 wind load) this solution requires $273 * 1/4 = 68\ Nm$ torque from the motor.

Space Requirements

The rack and pinion concept's space requirements are calculated in two positions: opened and closed. See figures 3.47 and 3.48.

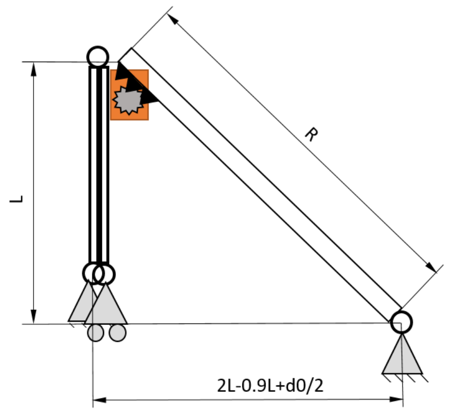

Figure 3.47: *Pinion and rack concept in an opened state.*

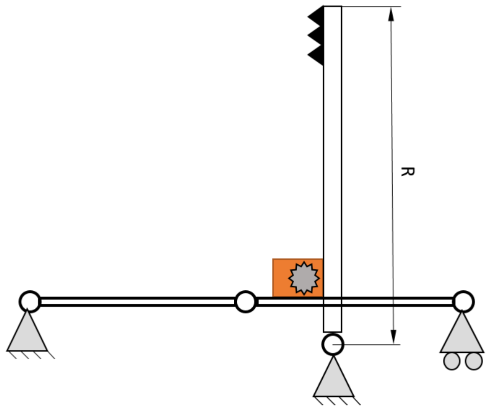

Figure 3.48: *Pinion and rack concept in a closed state.*

Assuming dimensions according to 3.47, with values L=1.25 m and d_0=0.1 m, gives the length R according to Pythagoras's theorem as:

$$R = \sqrt{L^2 + (2L - 0.9L + d_0/2)^2} = \sqrt{1.25^2 + (1.1 * 1.25 + 0.100/2)^2} = 1.90m \tag{3.15}$$

I.e. in order for the door to be able to fully open, a rack with a minimum length of 1.90 m is required. This means that the depth requirement for this solution is at least 1.90 meters. It is important to remember that this dimension is highly dependent on the door leaf width. Regarding side space and headroom, the solution takes up no more than a manual door. A summarizing table of these space requirements is shown in table 3.8.

Straight rack and pinion concept				
Configuration				
	SL	SR	OH	D
2+0	<200	<200	<200	>1900 (depending on LW)
2+2	<200	<200	<200	>1900 (depending on LW)

Table 3.8: *Space requirements of rack and pinion concept. All values in mm.*

Opening Speed and Gearbox

In the chapter describing the CDM9 motor (3.1.1) two speeds were discussed. 21.5 rpm at maximum load and 60 rpm as the highest speed the motor is capable to run at.

Assuming an average motor shaft speed of ω=30 rpm, and dimensions d_0=0.1 m, R=1.9 m, according to the previous space requirements section, gives the following opening speed:

$$t_{open} = \frac{R}{d_0 * \pi * \omega} = \frac{1.9}{0.1 * \pi * 30} = 0.202min = 12.1s \tag{3.16}$$

In this case, both the normal operating torque and the opening time is good. No gearbox is needed.

Summary

A summary of all sections in chapter 3.4.2 and a comparison against ASSA ABLOYS's drive system today is presented in table 3.9 below. The serviceability score was set higher than ASSA ABLOY's drive system due to fewer components.

Parameter	**Value (2+0)**	**ASSA ABLOY (2+0)**
Torque @ Class 3 wind load	273 Nm	89.8 Nm
Torque @ Normal conditions	68 Nm	22.5 Nm
Gearbox necessary	NO	NO
Possible to drive with gearbox	N.A.	N.A.
Required side space	<200 mm	450 mm
Required headroom	<200 mm	240 mm
Required depth	>1900 mm	1430 mm
Serviceability	+	0

Table 3.9: *A comparative summary of all parameters.*

3.4.3 Screw

Operating Forces

A sketch of the wind load acting on the door is created, see figure 3.49. To be able to calculate the required motor torque, the force necessary to counteract the wind load is determined. Inserting values $x = 0.95 * L \; and \; L = 1.25$ into the formula for F_u (3.4), derived in chapter 3.1.4, the necessary force in this case is:

$$F_u = \frac{L * F_w}{x} = \frac{1.25 * 700 * 5 * 1.25}{0.95 * 1.25} = 4605 \; N \tag{3.17}$$

Assuming x equals y, α becomes 45°.

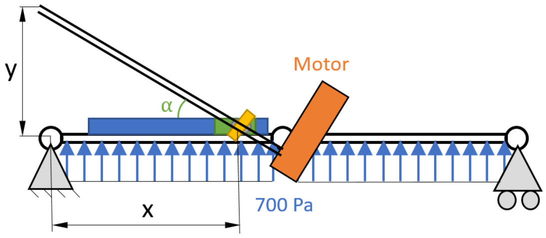

Figure 3.49: *Sketch of the folding door with applied wind load.*

To calculate the required torque to withstand the wind load, the screw axial force that produces a vertical component the same size as F_u is first determined from figure 3.50.

$$F_{axial} = \frac{F_u}{cos(\alpha)} = \frac{4605}{cos(45°)} = 6512 \; N \tag{3.18}$$

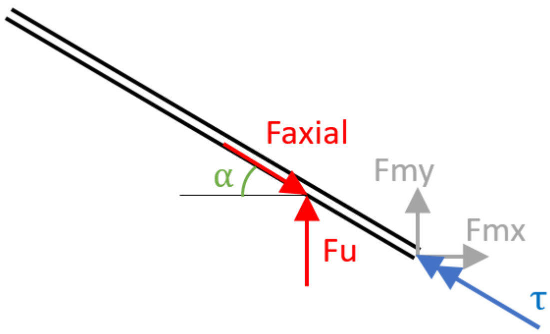

Figure 3.50: *Free body diagram of the power screw seen in figure 3.49.*

Equation to calculate torque for a power screw:

$$\tau = \frac{F_{axial} * P}{2000 * \pi * \eta_s} [24] \tag{3.19}$$

Assuming a screw pitch of P=10mm/revolution and a screw efficiency of η_s=0.2, the required torque can now be calculated through equation 3.19.

$$\tau = \frac{F_{axial} * P}{2 * \pi * \eta_s} = \frac{6512 * 10}{2000 * \pi * 0.2} = 51\ Nm \tag{3.20}$$

Under normal operating conditions (1/4 wind load) this solution requires $51 * 1/4 = 13\ Nm$ torque from the motor.

Space Requirements

One limiting factor is the height of the motor since it is positioned on the wall. The motor is 340 mm high and determines the headroom required. In figure 3.49 it can be seen that the blue sliding rail will be sticking out, taking up side space. This would mean that the side space requirement is defined by the thickness of the rail. The folding door's opening angle is 92°, which would add $L * sin(2°) = 1.25 * sin(2°) = 44\ mm$ to the side space. The combined measurement would still not exceed the 200 mm side space requirement set by ASSA ABLOY (see chapter 3.2.1).

Screw concept				
Configuration				
	SL	SR	OH	D
2+0	<200	<200	340	Door leaf width
2+2	<200	<200	340	Door leaf width

Table 3.10: *Space requirements of screw concept. All values in mm.*

Opening Speed and Gearbox

In the chapter describing the CDM9 motor (3.1.1) two speeds were discussed. 21.5 rpm at maximum load and 60 rpm as the highest speed the motor is capable to run at. Assuming an average motor shaft speed of ω=30 rpm and the dimensions $x = y = 0.9 * L$ from figure 3.49, together with the pitch P=10 mm/revolution from the previous operating forces section, this would give an opening speed of:

$$t_{open} = \frac{sqrt(2 * (0.9 * L)^2)}{P * \omega} = \frac{sqrt(2 * (0.9 * 1.25)^2)}{0.01 * 30} = 5.303\ min = 318.2\ s \tag{3.21}$$

In this case, the normal operating torque would be no problem for the CDM9 motor, but the opening time is too slow. To be able to open the door in 12 s, the optimal opening time for folding doors [3], a gear ratio of 26.5:1 is required.

This also decreases the output torque from 50 Nm to 2 Nm, according to chapter 2.4. This means that after implementing a gearbox, the CDM9 motor is no longer able to drive the folding door under normal operating conditions (required 13 Nm, see section operating forces).

Summary

A summary of all sections in chapter 3.4.3 and a comparison against ASSA ABLOYS's drive system today is presented in table 3.11 below. The serviceability score was set higher than ASSA ABLOY's drive system due to fewer components.

Parameter	**Value (2+0)**	**ASSA ABLOY (2+0)**
Torque @ Class 3 wind load	51 Nm	89.8 Nm
Torque @ Normal conditions	13 Nm	22.5 Nm
Gearbox necessary	26.5:1	NO
Possible to drive with gearbox	NO	N.A.
Required side space	<200 mm	450 mm
Required headroom	340 mm	240 mm
Required depth	1250 mm	1430 mm
Serviceability	+	0

Table 3.11: *A comparative summary of all parameters.*

3.5 Evaluation of External Concepts

3.5.1 General Notes

Since detailed drawings of competitor's drive systems are hard to come by, these concepts were evaluated to the best possible extent. The calculations include assumptions that will be explained in detail in the corresponding text sections.

3.5.2 Side Mounted

The side mounted drive system is a very simple solution often seen where side space is not a limiting factor. This drive system consists of, in addition to the motor, two link arms connecting the motor to the outer door leaf. See figure 3.51.

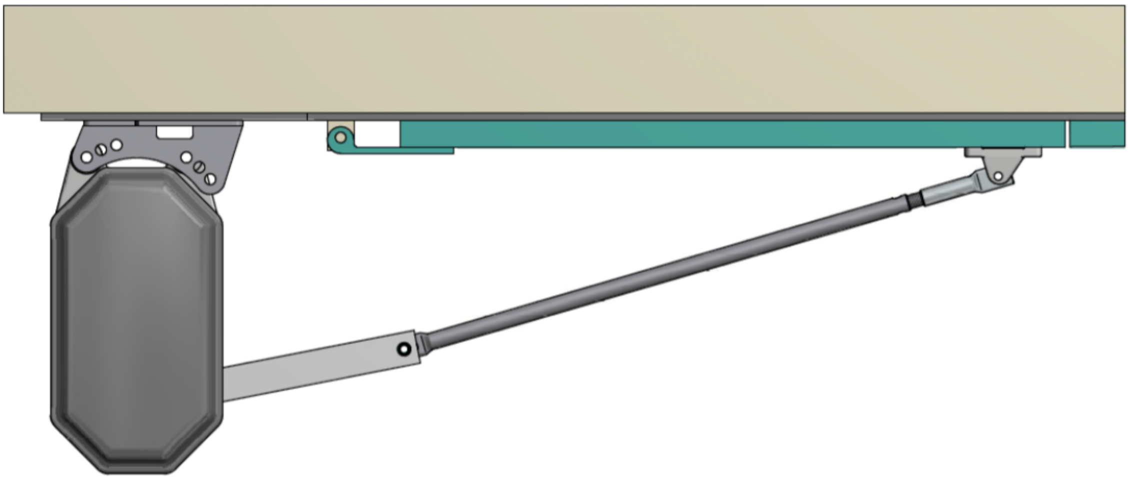

Figure 3.51: *Side-mounted motor. Figure from [21].*

Operating Forces

To determine the required motor torque for operating the folding door using a side mounted drive system some measurements are necessary. It was determined to measure pixels in figure 3.51.

It was known that the motor shaft has its center axis located 421 mm from the wall, this was used as a reference [21]. The pixel counting resulted in the dimensions seen in figure 3.52.

The assumptions made are:

- The angles in figure 3.52 are always 11.03° and 16.30° respectively when the door is closed regardless of the leaf width.
- The motor is always mounted so that the motor shaft's center axis is 421 mm away from the wall, according to figure 3.52.
- The length of the link arm connected to the motor is always 408 mm.
- The fix point on the door leaf is always 989 mm from the hinge.
- The door is rated for wind class 3.

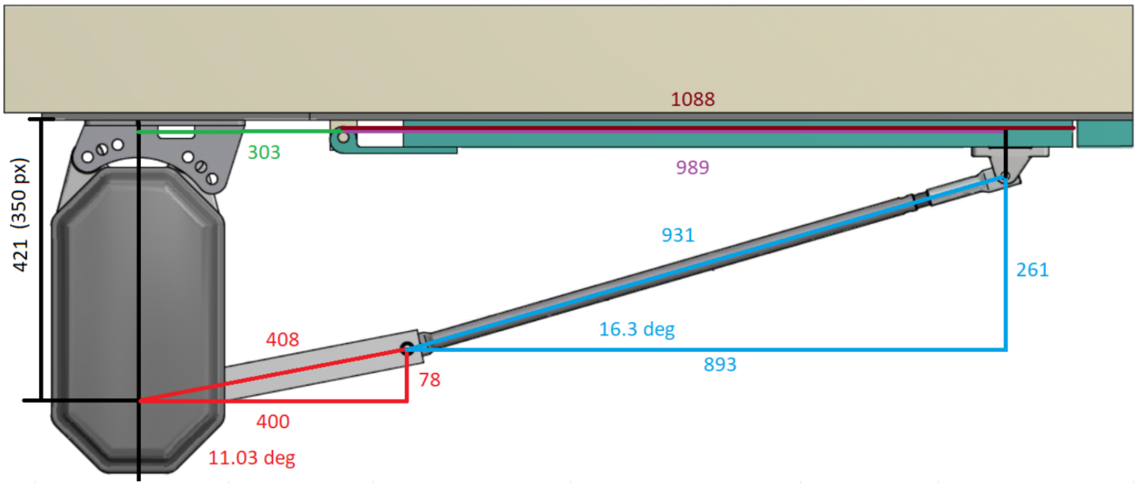

Figure 3.52: *Side-mounted motor with dimensions.*

A sketch of the folding door with a side mounted drive system was created. This sketch can be seen in figure 3.53. A free body diagram of the link arms was also created, see figure 3.54.

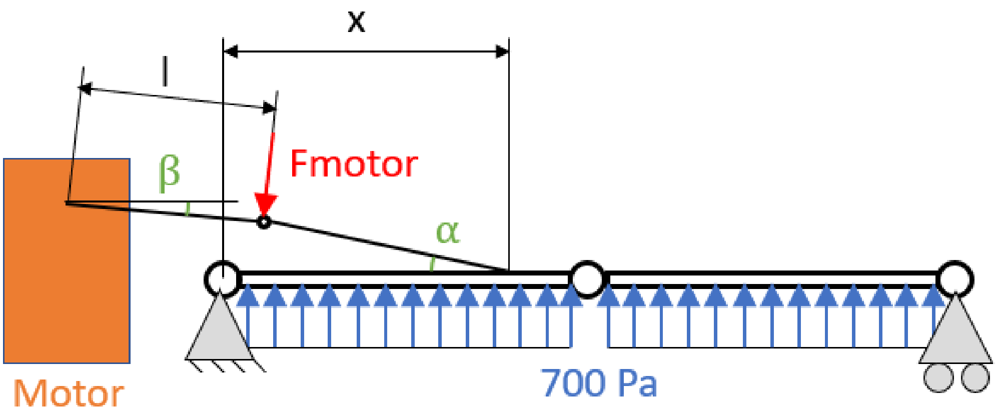

Figure 3.53: *Sketch of the folding door with applied wind load.*

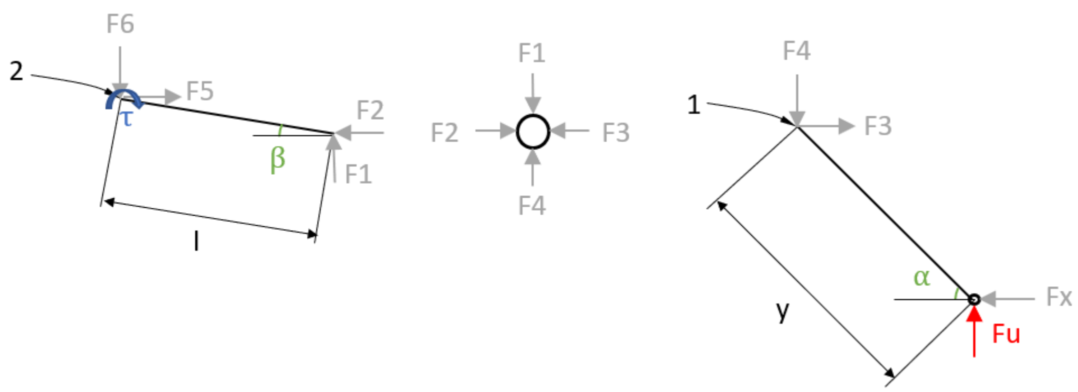

Figure 3.54: *Free body diagram of link arm connecting the motor to the door.*

Equilibrium equations from the free body diagram in figure 3.54:

$$\overset{\curvearrowleft}{1} : y * cos(\alpha) * F_u - y * sin(\alpha) * F_x = 0 \tag{3.22}$$

Equation 3.22 gives:

$$F_x = \frac{y * cos(\alpha) * F_u}{y * sin(\alpha)} = \frac{F_u}{tan(\alpha)} \tag{3.23}$$

$$\overset{\curvearrowleft}{2} : \tau + x * sin(\beta) * F_2 - x * cos(\beta) * F_1 = 0 \tag{3.24}$$

$$\leftarrow: F_x - F_3 = 0 \tag{3.25}$$

$$\leftarrow: F_3 - F_2 = 0 \tag{3.26}$$

Combining equations 3.23 through 3.26 gives:

$$\tau = x * cos(\beta) * F_u - x * sin(\beta) * \frac{F_u}{tan(\alpha)} \tag{3.27}$$

Inserting measurements $x = 0.989$ *and* $L = 1.25$ into the formula for $\mathrm{F_u}$ (3.4), derived in chapter 3.1.4 gives:

$$F_u = \frac{L * F_w}{x} = \frac{1.25 * 700 * 1.25 * 5}{0.989} = 5529\ N \tag{3.28}$$

Inserting values l=0.408 m, α=16.3°, β=11.03° together with F_u=5529 N into equation 3.27 gives the required motor torque of:

$$\tau = 0.408 * cos(11.03°) * 5529 - 0.408 * sin(11.03°) * \frac{5529}{tan(16.3°)} = 738\ Nm \quad (3.29)$$

I.e. the motor has to provide a torque of 738 Nm in order to withstand a wind load of 700 Pa to be classified as wind class 3 resistant.

Under normal operating conditions (1/4 wind load) this solution requires $738*1/4 = 184\ Nm$ torque from the motor.

Space Requirements

There are several different companies offering a side mounted solution. Lists of the competitors respective space requirements is shown below in 3.12 and in 3.13.

EAB - Side mounted				
Configuration				
	SL	SR	OH	D
2+0	800	N.A.	180	Door leaf width
2+2	800	800	180	Door leaf width

Table 3.12: *Space requirements of EAB's side mounted drive system. All values in mm. Data from [25].*

Prido - Side mounted				
Configuration				
	SL	SR	OH	D
2+0	800	N.A.	120	Door leaf width + 100
2+2	800	800	120	Door leaf width + 100

Table 3.13: *Space requirements of Prido's side mounted drive system. All values in mm. Data from [26].*

General Competitor Specifications

The calculated required torque to withstand the wind load is compared to FAAC's motors [27] and is deemed reasonable. The comparison is shown in the table 3.14 below.

Competitor	Opening Speed (s)	Cycles	Wind Class	Calculated Torque (Nm)	FAACs motor torque (Nm)
Prido	Approx. 17	50000	2-5	738	275-2600
EAB	Approx. 17	N.A.	N.A.	738	450-1600

Table 3.14: *General specifications gathered from respective competitors webpages Prido and EAB. Data from [27], [28] and [29].*

Opening Speed and Gearbox

If the CDM9 motor was used in this drive system, a gear reducing its speed is needed. For the door to fully open using the side mounted solution, the motor shaft has to spin 200 degrees [21]. To open the door in 12 s, the optimal opening time for folding doors [3], an average motor speed of

$$\frac{\frac{200°}{360°}}{\frac{12}{60}} = 2.7rpm \tag{3.30}$$

is required. This would mean a reduction ratio of 2.7/21.5 = 1:8.

This also increases the output torque from 50 Nm to 400 Nm, according to chapter 2.4. The CDM9 motor is now able to drive the folding door under normal operating conditions (required 184 Nm, see section operating forces).

Constructing a planetary gear with 18 teeth (minimum number of teeth, see chapter 2.4) on the sun gear, gives the following number of teeth on the ring gear (see equation 2.5 in chapter 2.4.2):

$$\frac{18}{Z_r + 18} = \frac{1}{8} \Rightarrow Z_r = 126 \tag{3.31}$$

Assuming gear module 4, this would give the ring gear a pitch diameter of $4 * 126 = 504\ mm$. This is too large. It was not possible to find a planetary gear with more gear stages that is able to achieve this in a smaller space, since the torque required to withstand a full wind load (738 Nm) is too high.

The possibility of using a worm gear (see chapter 2.4.3) with 24 teeth on the gear and three starts on the worm screw exists. Assuming module 8 (due to the high torques) results in a gear pitch diameter of $8 * 24 = 192\ mm$.

Summary

A summary of all sections in chapter 3.5.2 and a comparison against ASSA ABLOYS's drive system today is presented in table 3.15 below. The serviceability score was set higher than ASSA ABLOY's drive system due to fewer components.

Parameter	Value (2+0)	ASSA ABLOY (2+0)
Torque @ Class 3 wind load	738 Nm	89.8 Nm
Torque @ Normal conditions	184 Nm	22.5 Nm
Gearbox necessary	1:8	NO
Possible to drive with gearbox	YES	N.A.
Required side space	800 mm	450 mm
Required headroom	<200 mm	240 mm
Required depth	1250 mm	1430 mm
Serviceability	+	0

Table 3.15: *A comparative summary of all parameters.*

3.5.3 Backstage Mounted

The backstage mounted drive system is another solution often used where headroom is not limited. This solution consists of one motor mounted on the wall above the folding door. The motor is connected to the outer door leaf via an arm combined with a sliding rail that sits on the door leaf. See figure 3.55.

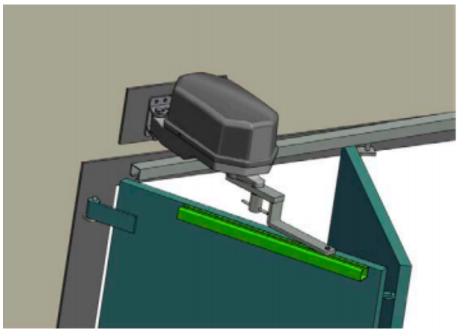

Figure 3.55: *Backstage mounted motor solution. Figure from [21].*

Operating Forces

To determine the required motor torque for a backstage mounted drive system, a sketch and free body diagram was created. These illustrations can be seen in figure 3.56 and 3.57 respectively. The assumptions made are listed below. Reasonable values were chosen, from images, to be able to compare the different transmission solutions. The assumptions made are:

- The attachment point x is in the middle of the door, i.e. $x = 0.5 * L$.
- The motor is always mounted so that the motor shaft's center axis is 421 mm away from the wall, according to figure 3.52.
- The angle between the arm and the door (labeled α) in figures 3.56 and 3.57 is 30°.
- The door is rated for wind class 3.

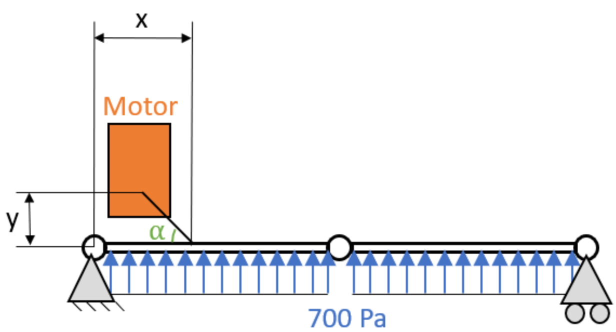

Figure 3.56: *Sketch of the folding door with applied wind load.*

Inserting measurements $x = 0.5 * L$ and $L = 1.25$ into the formula for F_u (3.4), derived in chapter 3.1.4 gives:

$$F_u = \frac{L * F_w}{x} = \frac{1.25 * 700 * 1.25 * 5}{0.5 * 1.25} = 8750\ N \tag{3.32}$$

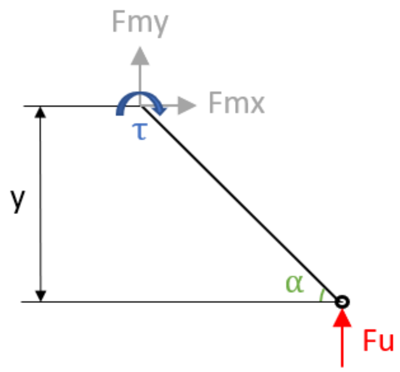

Figure 3.57: *Free body diagram of link arm connecting the motor to the door.*

The motor torque τ can now be calculated from the free body diagram, using the angle α=30° and distance y=0.421 m.

$$\tau = \frac{y}{tan(\alpha)} * F_u = \frac{0.421}{tan(30)} * 8750 = 6380\ Nm \tag{3.33}$$

Under normal operating conditions (1/4 wind load) this solution requires $6380 * 1/4 = 1595\ Nm$ torque from the motor.

Space Requirements

The space requirements from EAB and Bator can be seen in tables 3.16 and 3.17 below.

EAB - Backstage mounted				
Configuration				
	SL	SR	OH	D
2+0	<200	<200	450	Door leaf width
2+2	<200	<200	450	Door leaf width

Table 3.16: *Space requirements of EAB's backstage mounted drive system. All values in mm. Data from [30].*

Bator - Backstage mounted				
Configuration				
	SL	SR	OH	D
2+0	285	N.A.	310	Door leaf width + 100
2+2	295	295	345	Door leaf width + 100

Table 3.17: *Space requirements of Bator's backstage mounted drive system. All values in mm. Data from [31], [32] and [33].*

General Competitor Specifications

The calculated required torque to withstand the wind load is compared to FAAC's motors [27]. The conclusion can be drawn that either the companies' doors use a stronger motor or lack a class 3 wind load specification. The comparison is shown in table 3.18 below.

Competitor	Opening Speed (s)	Cycles (million)	Wind Class	Calc. Torque (Nm)	FAACs motor torque (Nm)
EAB	Approx. 56	N.A	N.A.	6380	450-1600
Bator	N.A.	1	5	6380	N.A.

Table 3.18: *General specifications gathered from respective competitors webpages EAB and Bator. Data from [34] and [35].*

Opening Speed and Gearbox

If the CDM9 motor was used on this type of solution, a gear described in chapter 2.4 reducing its speed is needed. For the door to fully open using the backstage mounted solution, the motor shaft has to spin 200 degrees [21]. To open the door in 12 s, the optimal opening time for folding doors [3], an average motor speed of

$$\frac{\frac{200^\circ}{360^\circ}}{\frac{12}{60}} = 2.7rpm \tag{3.34}$$

is required. This would mean a reduction ratio of 2.7/21.5 = 1:8.

This also increases the output torque from 50 Nm to 400 Nm, according to chapter 2.4. The CDM9 motor is still not able to drive the folding door under normal operating conditions (required 1595 Nm, see section operating forces).

Since the motor will not be able to drive the door during normal operating conditions, no gearbox is discussed.

Summary

A summary of all sections in chapter 3.5.3 and a comparison against ASSA ABLOYS's drive system today is presented in table 3.19 below. The serviceability score was set higher than ASSA ABLOY's drive system due to fewer components.

Parameter	**Value (2+0)**	**ASSA ABLOY (2+0)**
Torque @ Class 3 wind load	6380 Nm	89.8 Nm
Torque @ Normal conditions	1595 Nm	22.5 Nm
Gearbox necessary	1:8	NO
Possible to drive with gearbox	NO	N.A.
Required side space	<200 mm	450 mm
Required headroom	310 mm	240 mm
Required depth	1250 mm	1430 mm
Serviceability	+	0

Table 3.19: *A comparative summary of all parameters.*

3.5.4 Ditec Dor Solution

The Ditec Dor solution is a compact solution and an interesting candidate. A figure of the Ditec Dor's drive system can be seen in figure 3.58.

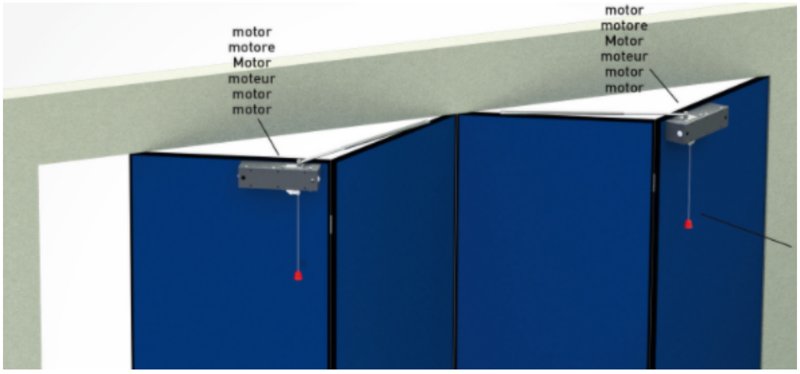

Figure 3.58: *Ditec Dor motor solution. Figure from [36].*

Operating Forces

To determine the required motor torque for a Ditec Dor solution, a sketch and free body diagram was created. These drawings can be seen in figure 3.59 and 3.60 respectively. The measurements in figure 3.59 are retrieved from the Ditec Dor technical manual [22]. Since the Ditec Dor solution puts two points of force on the folding door, the mathematical model will be more complicated. The assumptions made for this drive system are listed in bullet points below:

- The thickness of the door leaves is ignored. I.e. the contribution to the moment equations from the force F_{armx} in figure 3.60 is zero.
- The door is rated for wind class 3.

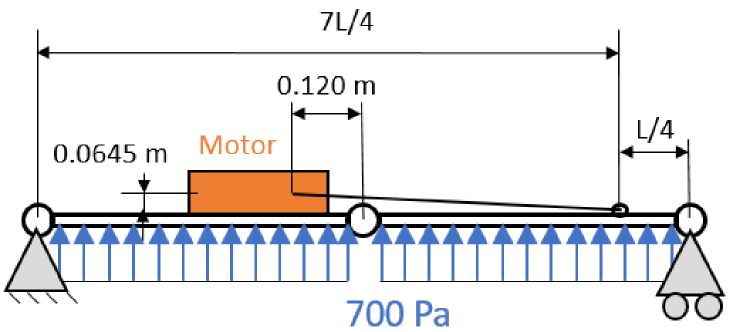

Figure 3.59: *Sketch of the folding door with applied wind load.*

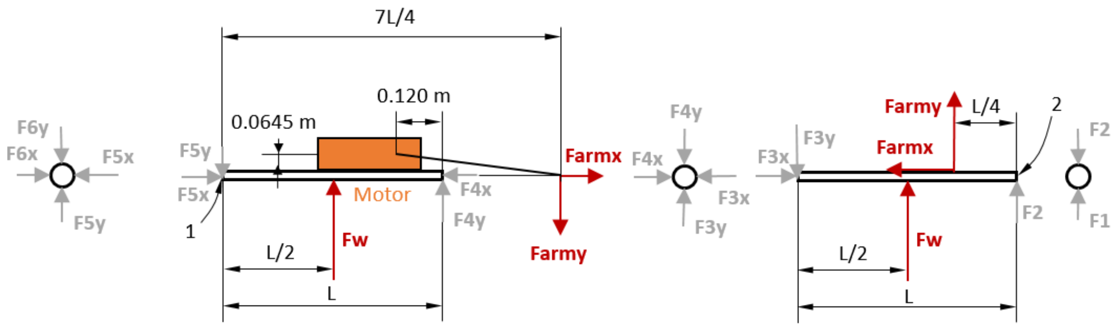

Figure 3.60: *Free body diagram of the Ditec Dor solution.*

From chapter 3.1.4: the wind load Fw can be written as
$F_w = p * A,\ where\ p = 700\ Pa\ (wind\ class\ 3)\ and\ A = L * 5\ m^2$.

Equilibrium equations from the free body diagram in figure 3.60:

$$\overset{\frown}{1} : \frac{L}{2} * F_w + L * F_{4y} - \frac{7 * L}{4} * F_{army} = 0 \tag{3.35}$$

$$\overset{\frown}{2} : L * F_{3y} - \frac{L}{2} * F_w - \frac{L}{4} * F_{army} = 0 \tag{3.36}$$

$$\uparrow : F_{3y} - F_{4y} = 0 \tag{3.37}$$

Combining equations 3.35 through 3.37 gives:

$$F_{army} = \frac{2 * F_w}{3} = \frac{2 * 700 * 1.25 * 5}{3} = 2916\ N \tag{3.38}$$

$$\tau = \frac{3 * L}{4} * F_{army} = \frac{3 * 1.25}{4} * 2916 = 2733\ Nm \tag{3.39}$$

Under normal operating conditions (1/4 wind load) this solution requires $2733 * 1/4 = 683\ Nm$ torque from the motor.

Space Requirements

The space requirement of Ditec Dor's drive system can be seen in table 3.20 below. Dt stands for the thickness of the door leaves.

Ditec Dor							
Configuration	**Op. Pos. Outer Leaf**			**Op. Pos. Inner Leaf**			
	SL	SR	OH	SL	SR	OH	D
2+0	120+Dt	N.A.	35	60 + Dt	N.A.	35	Door leaf width
2+2	120+Dt	120+Dt	35	60 + Dt	60 + Dt	35	Door leaf width

Table 3.20: *Space requirements of Ditec Dor's drive system. All values in mm. Data from [22].*

General Competitor Specifications

The calculated required torque to withstand the wind load is compared to Ditec's motor. The comparison is shown in table 3.21 below. Through this comparison it can be seen that the Ditec Dor drive system is not applicable where wind load resistance is required.

Competitor	Opening Speed (s)	Cycles	Wind Class	Calc. Torque (Nm)	Ditecs motor torque (Nm)
Ditec	N.A.	365000	N.A.	2733	300

Table 3.21: *General specifications gathered from Ditec's webpage. Data from [22].*

Opening Speed and Gearbox

If the CDM9 motor was used on this type of solution, a gear described in chapter 2.4 reducing its speed is needed. For the door to fully open using the Ditec Dor solution, the motor shaft has to spin approximately 180 degrees. To open the door in 12 s, the optimal opening time for folding doors [3], an average motor speed of

$$\frac{\frac{180°}{360°}}{\frac{12}{60}} = 2.5rpm \tag{3.40}$$

is required. This would mean a reduction ratio of 2.5/21.5 = 1:9.

This also increases the output torque from 50 Nm to 450 Nm, according to chapter 2.4. The CDM9 motor is still not able to drive the folding door under normal operating conditions (required 683 Nm, see section operating forces).

Since the motor will not be able to drive the door during normal operating conditions, no gearbox is discussed.

Summary

A summary of all sections in chapter 3.5.4 and a comparison against ASSA ABLOYS's drive system today is presented in table 3.22 below. The serviceability score was set higher than ASSA ABLOY's drive system due to fewer components.

Parameter	**Value (2+0)**	**ASSA ABLOY (2+0)**
Torque @ Class 3 wind load	2733 Nm	89.8 Nm
Torque @ Normal conditions	683 Nm	22.5 Nm
Gearbox necessary	1:9	NO
Possible to drive with gearbox	NO	N.A.
Required side space	<200 mm	450 mm
Required headroom	<200 mm	240 mm
Required depth	1250 mm	1430 mm
Serviceability	+	0

Table 3.22: *A comparative summary of all parameters.*

3.5.5 Overhead Mounted

The overhead drive system is desired in places where side space is limited. As seen in the external search chapter 3.3.3, there are three main categories used by competitors. However, since they are so similar in function, only one of them will be evaluated and discussed in this chapter. In order to simplify calculations of the operating forces, they are made on a 2+2 folding door and then divided by 2.

Operating Forces

In order to solve for the required motor torque, given a pressure difference between the inside and outside of the door, some dimensions had to be determined. The assumptions made are:

- The motor is always mounted so that the motor shaft's center axis is 421 mm away from the wall, according to figure 3.63.
- The link arm attached to the motor is 570 mm long. I.e. it has a rotation radius of 285 mm.
- The door is rated for wind class 3.

Further, some estimations regarding the size of the "boomerang" were made. Assuming that the person standing behind the door in figure 3.61 has a shoulder height of 1400 mm, the boomerang width can be estimated to 750 mm by counting pixels. This assumption is then used for determining the other dimensions of the boomerang using the same method. The boomerang dimensions determined can be seen in figure 3.62. From these assumptions, two sketches of the overhead mounted solution was created. See figures 3.63 and 3.64.

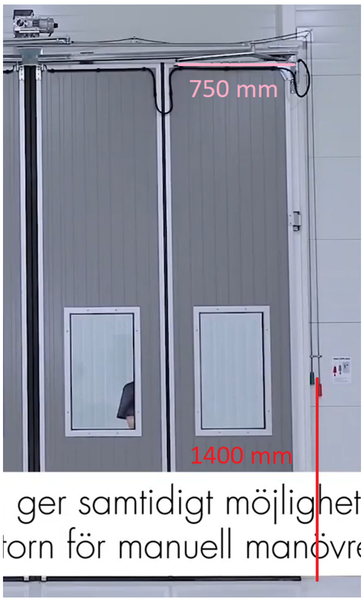

Figure 3.61: *Boomerang length estimation from YouTube video. Figure from [37].*

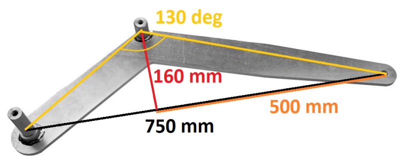

Figure 3.62: *Boomerang dimension estimation. Figure from [38].*

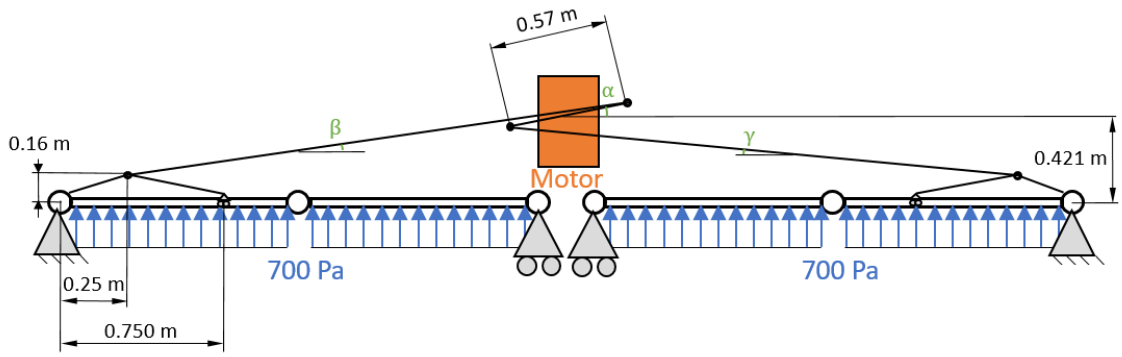

Figure 3.63: *Sketch of the folding door with applied wind load.*

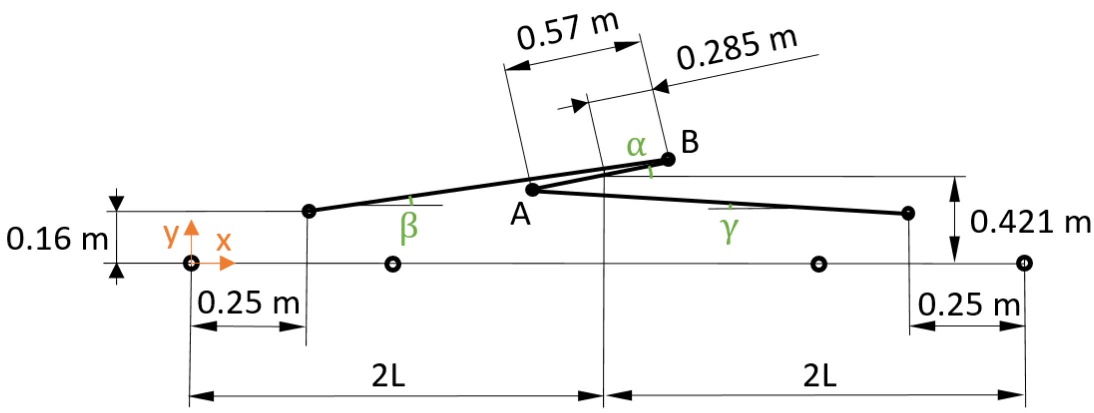

Figure 3.64: *Figure of link arms in the overhead mounted solution.*

From figure 3.64 it is possible to calculate the angles β and γ if the angle α is given. The angle α is approximately 10° when the folding door is closed [39]. To make this task a bit easier, the coordinates for the points A and B are calculated.

$$A_x = 2 * L - 0.285 * cos(\alpha) = 2 * 1.25 - 0.285 * cos(10°) = 2.219\ m \tag{3.41}$$

$$A_y = 0.421 - 0.285 * sin(\alpha) = 0.421 - 0.285 * sin(10°) = 0.371\ m \tag{3.42}$$

$$B_x = 2 * L + 0.285 * cos(\alpha) = 2 * 1.25 + 0.285 * cos(10°) = 2.780\ m \tag{3.43}$$

$$B_y = 0.421 + 0.285 * sin(\alpha) = 0.421 + 0.285 * sin(10°) = 0.470\ m \tag{3.44}$$

Using trigonometry, the angles β and γ are then determined:

$$\beta = arctan(\frac{B_y - 0.16}{B_x - 0.25}) = arctan(\frac{0.470 - 0.16}{2.780 - 0.25}) = 6.99° \tag{3.45}$$

$$\gamma = arctan(\frac{A_y - 0.16}{4 * L - A_x - 0.25}) = arctan(\frac{0.371 - 0.16}{4 * 1.25 - 2.219 - 0.25}) = 4.77° \tag{3.46}$$

The motor torque required for the door to withstand a class 3 wind load is calculated using two free body diagrams. See figures 3.65 and 3.66.

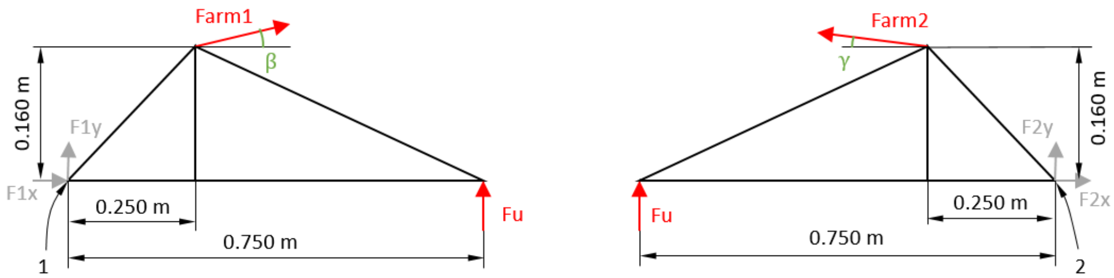

Figure 3.65: *Free body diagram of the boomerang.*

Unlike the other free body diagrams such as 3.15 and 3.54, where each hinge results in two force components, these two components are in this case replaced by a resultant force in the link arms direction. Since the angles β and γ are known, the problem is possible to solve. The following equilibrium equations are created:

$$\overset{\frown}{1} : 0.75 * F_u - 0.16 * F_{arm1} * cos(\beta) = 0 \tag{3.47}$$

$$\overset{\frown}{2} : 0.16 * F_{arm2} * cos(\gamma) - 0.75 * F_u = 0 \tag{3.48}$$

Inserting measurements $x = 0.750$ and $L = 1.25$ into the formula for F_u (3.4), derived in chapter 3.1.4 gives:

$$F_u = \frac{L * F_w}{x} = \frac{1.25 * 700 * 1.25 * 5}{0.750} = 7291\ N \tag{3.49}$$

With these three equations, the magnitude of the forces in each one of the link arms can be determined:

$$\overset{\frown}{1} \Rightarrow F_{arm1} = \frac{0.75 * F_u}{0.16 * cos(\beta)} = \frac{0.75 * 7291}{0.16 * cos(6.99°)} = 34432\ N \tag{3.50}$$

$$\overset{\frown}{2} \Rightarrow F_{arm2} = \frac{0.75 * F_u}{0.16 * cos(\gamma)} = \frac{0.75 * 7291}{0.16 * cos(4.77°)} = 34295\ N \tag{3.51}$$

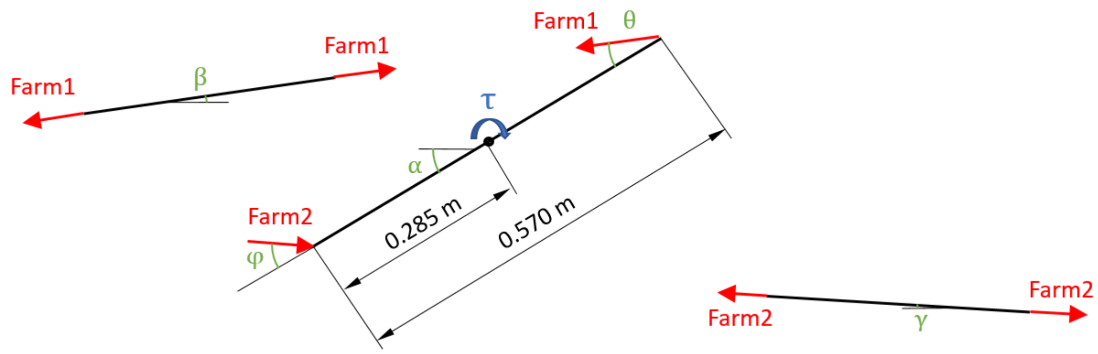

Figure 3.66: *Free body diagram of the link arms.*

From figure 3.66, the angles Θ and φ can be calculated:

$$\Theta = \alpha - \beta = 10^\circ - 6.99^\circ = 3.01^\circ \tag{3.52}$$

$$\phi = \alpha + \gamma = 10^\circ + 4.77^\circ = 14.77^\circ \tag{3.53}$$

Finally, the required motor torque can be calculated:

$$\tau = 0.285 * F_{arm1} * sin(\Theta) + 0.285 * F_{arm2} * sin(\phi) \tag{3.54}$$

$$\tau = 0.285 * 34432 * sin(3.01) + 0.285 * 34295 * sin(14.77) = 3007\ Nm \tag{3.55}$$

Under normal operating conditions (1/4 wind load) this solution requires $3007 * 1/4 = 751\ Nm$ torque from the motor.

For a 2+0 door, the corresponding values would be 1504 and 376 Nm respectively. This is under the assumption that the motor torque is split equally between both halves of the 2+2 door.

Space Requirements

Prido - Overhead Mounted							
Configuration	**Op. pos. Center (PriDrive)**			**Op. Pos. Center (FAAC)**			
	SL	SR	OH	SL	SR	OH	D
2+2 Standard	380	380	420	380	380	380	Door leaf width + 100
2+2 High Speed	380	380	420	N.A.	N.A.	N.A.	Door leaf width + 100
2+2 Boomerang	250	250	420	N.A.	N.A.	N.A.	Door leaf width + 100

Table 3.23: *Space requirements of Prido's overhead mounted drive system. All values in mm. Data from [40].*

EAB - Overhead Mounted				
Configuration	**Operator Position Center**			
	SL	SR	OH	D
2+2	460	460	455	Door leaf width
2+2 + Limited Side	250	250	620	Door leaf width

Table 3.24: *Space requirements of EAB's overhead mounted drive system. All values in mm. Data from [25].*

General Competitor Specifications

The calculated required torque to withstand the wind load is compared to FAAC's motors [27] and is deemed reasonable. Since the FAAC motors have a maximum torque of 1600 Nm for a wind class 1 door, the 3007 Nm result is fair for wind class 3 door. The comparison is shown in table 3.25 below.

Competitor	Open. Speed (s)	Cycles (million)	Wind Class	Calc. Torque (Nm)	FAACs motor torque (Nm)
EAB		N.A.	1	N.A.	450-1600
Prido (Standard)	11	1	2-5	N.A.	N.A.
Prido (High Speed)	7.5	1	2-5	N.A.	N.A.
Prido (Boomerang)	9.5	1	2-5	3007	N.A.

Table 3.25: *General specifications gathered from respective competitors webpages EAB and Prido. Data from [41] and [42].*

Opening Speed and Gearbox

If the CDM9 motor was used on this type of solution, a gear described in chapter 2.4 reducing its speed is needed. For the door to fully open using the overhead mounted solution, the motor shaft has to spin approximately 135 degrees. To open the door in 12 s, the optimal opening time for folding doors [3], an average motor speed of

$$\frac{\frac{135°}{360°}}{\frac{12}{60}} = 1.9rpm \tag{3.56}$$

is required. This would mean a reduction ratio of 1.9/21.5 = 1:11.

This increases the output torque from 50 Nm to 550 Nm, according to chapter 2.4. The CDM9 motor is now able to drive a 2+0 folding door under normal operating conditions (required 376 Nm, see section operating forces).

Constructing a planetary gear with 18 teeth (minimum number of teeth, see chapter 2.4) on the sun gear, gives the following number of teeth on the ring gear (see equation 2.5 in chapter 2.4.2):

$$\frac{18}{Z_r + 18} = \frac{1}{11} \Rightarrow Z_r = 180 \tag{3.57}$$

Assuming gear module 4, this would give the ring gear a pitch diameter of $4 * 180 = 720\ mm$. This is too large. It was not possible to find a planetary gear with more gear stages that is able to achieve this in a smaller space, since the torque required to withstand a full wind load (1504 Nm) is too high.

The possibility of using a worm gear (see chapter 2.4.3) with 22 teeth on the gear and two starts on the worm screw exists. Assuming module 8 (due to the high torques) results in a gear pitch diameter of $8 * 22 = 176\ mm$.

Summary

A summary of all sections in chapter 3.5.5 and a comparison against ASSA ABLOYS's drive system today is presented in table 3.26 below. The serviceability score is equal to ASSA ABLOY's drive system due to the number of components included.

Parameter	**Value (2+0)**	**ASSA ABLOY (2+0)**
Torque @ Class 3 wind load	1504 Nm	89.8 Nm
Torque @ Normal conditions	376 Nm	22.5 Nm
Gearbox necessary	1:11	NO
Possible to drive with gearbox	YES	N.A.
Required side space	250 mm	450 mm
Required headroom	420 mm	240 mm
Required depth	1250 mm	1430 mm
Serviceability	0	0

Table 3.26: *A comparative summary of all parameters.*

3.6 Concept Selection

3.6.1 Concept Scoring Criteria

The concept selection is based on 6 different scoring criteria explained in the sections below.

Wind Resistance

The wind resistance criterion is based on the torque required to withstand a full wind load as described in 3.1.4. It has a 0.15 weight score since it is of less importance than the door being able to operate under normal conditions. ASSA ABLOY's current drive system is used as a base reference (3 out of 5 points). The pinion and rack concept's rating is worse relative to ASSA ABLOY's drive system and therefore scores 2 out of 5 points. The same rating is applied to the side mounted solution. Both concepts require a large torque in order to withstand a full wind load of 700 Pa. The worst concepts regarding wind resistance are the backstage mounted, Ditec Dor and overhead mounted drive systems because they require a very large torque to withstand the wind load. Since switching motors is a limitation to our project these are given a score around 0-1. The best concept is the screw concept. It scores a 5 due to the small amount of torque required from the motor.

Operating Force

The second criterion is the torque required for the door to operate during normal conditions (1/4th of the wind load). This is considered important and therefore receives a 0.2 weight score. The scoring follows the same pattern as the wind resistance section above, since both use the same required torque calculations. Worth mentioning is that the side mounted and overhead mounted drive systems are able to operate if an additional gearbox is used. Therefore they receives a high score in this criterion. The screw concept wins in this criterion.

Gearbox

The third criterion denotes the need of a gearbox, this criterion was deemed not as important since it is considered an evaluation and exploration of possibilities. The criterion was given a weight score of 0.15. If the door is able to open at a reasonable speed ($\backsim$ 12 s) using the CDM9 motor, without the need of a gearbox, the score will be 5. However, if the motor speed needs to be adjusted using a gearbox and the CDM9 is able to perform during normal operating conditions (1/4th of the wind load) with said gearbox, a score of 3 is given. Finally, if the speed needs to be adjusted with a gearbox and the CDM9 motor remains/becomes incapable of driving the door during normal operating conditions, a score of 0 is given. All concepts except the pinion and rack, side mounted and overhead mounted drive systems are given a score of 0 since it was not possible for the door to be open under normal operating conditions. The side mounted and overhead mounted concepts were the only drive systems given a score of 3. The pinion and rack was given a 5.

Side Space

One of the most important project goals ASSA ABLOY was interested in, was minimizing the side space and headroom requirements. Therefore, these two criteria were given a 0.2 weight score. These space requirements were, for each concept compared to ASSA ABLOY's current drive system. If the required space was less/much less than ASSA ABLOY's current drive system the concepts received a 4/5. If the space requirement was larger/much larger than ASSA ABLOY's current drive system the concepts were given scores of 2/1 respectively. If the space requirements were similar to ASSA ABLOY's current drive system, the concepts were given a score of 3. The best concept in this criterion is the pinion and rack and the Ditec Dor drive system. This is mainly due to the motor being mounted on the door instead of the wall in both concepts.

Headroom

This criterion follows the same scoring principle as the side space criterion. The current drive system at ASSA ABLOY has a large headroom requirement, compared to competitor solutions. The two best solutions regarding headroom are the pinion and rack concept and Ditec Dor drive system again due to their motor placement.

Serviceability

It was chosen to include a more qualitative criterion as well. The serviceability is mainly based on the number of parts, overall complexity of the concept and how easy it is to service or repair. If the complexity or number of parts of the concept are close to ASSA ABLOY's current drive system a score of 3 is given. If the concept has a greater number of parts and/or is more complex, a score of 1 or 2 is given, depending on the severity. The opposite applies to concepts with less parts or simpler constructions, where a 4 or 5 is assigned. This criterion received a weight score of 0.1 since it was considered a side goal.

3.6.2 Concept Choice

In table 3.27 the pinion and rack concept scored a 4.15 out of 5 possible points. This makes it the best candidate and will, throughout this project and report, be the concept of choice. This decision was made, even though the concept scores low in some criteria such as wind resistance and operating force, because it was deemed manageable for smaller doors. However, superior serviceability, space requirements and no need for a gearbox became the deciding factors. On top of this, it was a unique solution that has never been seen before. This made it a very interesting concept to explore.

		Concepts													
		ASSA ABLOY's current drive system (ref)		**Pinion and Rack**		**Screw**		**Side Mounted**		**Backstage Mounted**		**Ditec Dor Solution**		**Overhead Mounted**	
Selection Criteria	Weight	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Wind Resistance	0.15	3	0.45	2	0.3	5	0.75	2	0.3	0	0	1	0.15	1	0.15
Operating Forces	0.2	3	0.6	3	0.6	5	1	4	0.8	1	0.2	2	0.4	2	0.4
Gearbox	0.15	5	0.75	5	0.75	0	0	3	0.45	0	0	0	0	3	0.45
Side Space	0.2	3	0.6	5	1	4	0.8	0	0	5	1	5	1	3	0.6
Headroom	0.2	3	0.6	5	1	2	0.4	5	1	2	0.4	5	1	1	0.2
Serviceability	0.1	3	0.3	5	0.5	4	0.4	5	0.5	4	0.4	5	0.5	3	0.3
Total Score		3.3		4.15		3.35		3.05		2		3.05		2.1	
Rank		3		1		2		4		6		4		5	

Table 3.27: *Concept scoring matrix.*

4. Design

In this chapter the final design of each component is discussed. The complete assembly is presented in the last section.

4.1 CAD-model

In order to visualise the pinion and rack drive system concept and aid in component design, a CAD-model of the folding door was created. ASSA ABLOY currently migrates from Autodesk's Inventor software to Dassault Systems' Solidworks counterpart. As the project progressed it was found that the old files for the folding door were corrupt. The software migration and corrupt files meant recreating the door from scratch.

The CAD-assembly is supposed to be a model and only includes the most important parts of the door. I.e. details such as weather strips and insulation inside the door leaves were not included in the 3D model. Only the parts required to illustrate the folding motion were included. The model can be seen in figures 4.1 and 4.2.

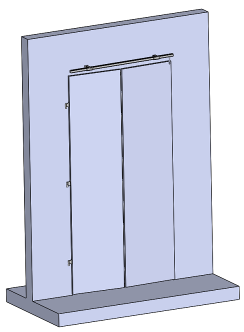

Figure 4.1: *Basic door in a closed state.*

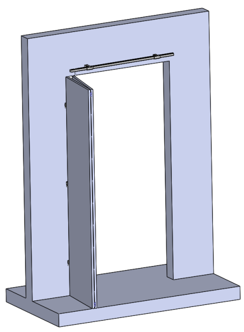

Figure 4.2: *Basic door in an open state.*

4.2 Initial Changes

The winning concept in an early stage of design can be seen in figures 4.3 and 4.4 below.

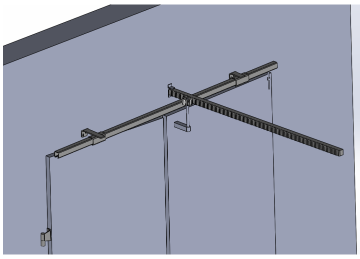

Figure 4.3: *Door with straight rack in a closed state.*

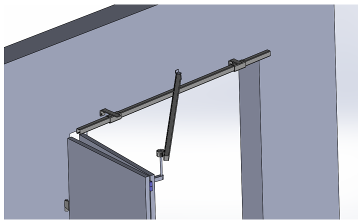

Figure 4.4: *Door with straight rack in an open state.*

When designing the CAD model for the initial concept it was quickly realized that a bent rack would be more space efficient. As discussed in chapter 3.4.2, a elliptical rack would be needed if the motor was mounted on the inner door leaf as in this case. The elliptical rack would be difficult to manufacture and more expensive. However, combining the hinge from the straight rack with the circular rack from chapter 3.4.2, would improve the depth required simultaneously as being simpler to manufacture. See figures 4.5 and 4.6 for an illustrative comparison.

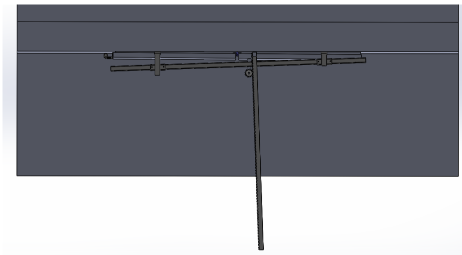

Figure 4.5: *Door with straight rack in a closed state, seen from above.*

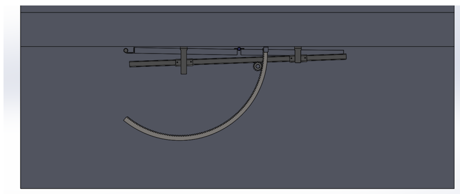

Figure 4.6: *Door with circular rack in a closed state, seen from above.*

4.3 Components

In this chapter, all component's design processes are described and their final design is presented. The motor shaft, shaft keys and standard bolts provided by ASSA ABLOY will not be included in this chapter. Detail drawings of all components can be found in Appendix D.

4.3.1 Motor Attachment

The first obstacle to overcome was attaching the motor to the inner door leaf. A simple plate that hangs over the door was initially designed according to figures 4.7 and 4.8.

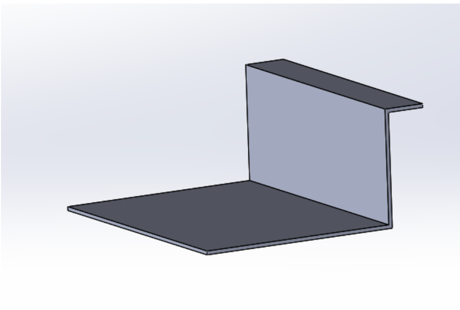

Figure 4.7: *The initial motor attachment design.*

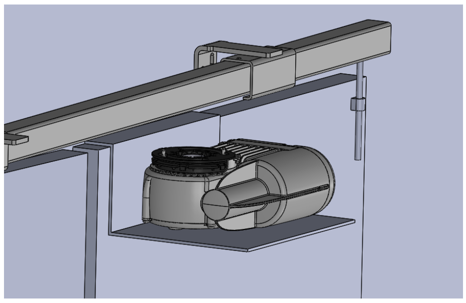

Figure 4.8: *The initial motor attachment attached to the door in CAD.*

During a consultation with the Sr. Mechanical Engineer at ASSA ABLOY, it was brought up that a motor attachment had already been developed in search for new ways of attaching the motor to other industrial doors. Their method used a 3 mm thick aluminum bracket (see figure 4.9) that was welded to the motor hull. ASSA ABLOY's way of attaching the motor to the door was elegant and used less material. Therefore it was decided to implement this in the CAD-model instead. The implementation can be seen in figure 4.10. However, the newly introduced bracket made the motor collide with the guide rail of the folding door. A new design introducing two more bends to move the motor downwards was therefore created. The new design also introduces two additional holes to be screwed into the metal frame inside the door for stability reasons. The final design can be seen in figure 4.11 and 4.12.

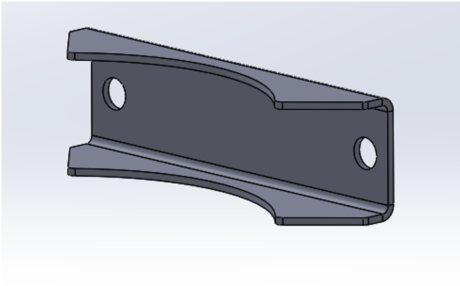

Figure 4.9: *ASSA ABLOY's design of the motor attachment.*

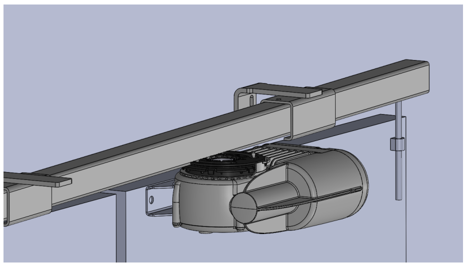

Figure 4.10: *ASSA ABLOY's design attached to the door in the CAD-model.*

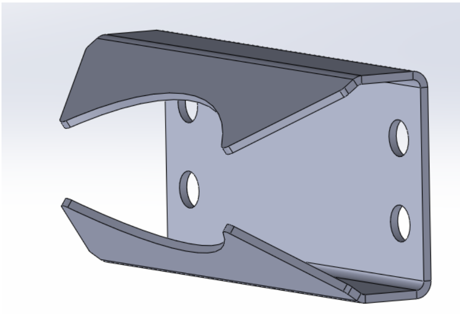

Figure 4.11: *Final design of the motor attachment.*

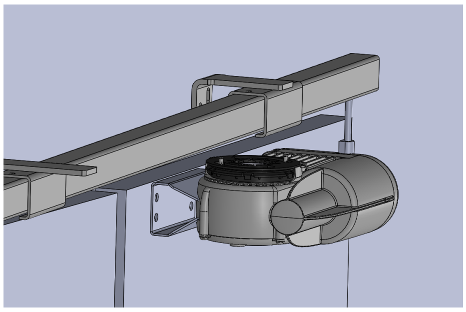

Figure 4.12: *The final motor attachment design attached to the door in the CAD-model.*

4.3.2 Rack Hinge

For the rack to be able to rotate, some sort of hinge had to be constructed. Since sheet metal would be used for creating the rack, a fitting screw mounted on an additional, separate guide rail holder seen in figure 4.12. This was evaluated using FEM to see if the screw would endure a class 3 wind load. Most fitting screws are CLASS 12.9, meaning they have an ultimate strength of 1200 MPa and 90% (1080 MPa) yield strength [43]. The guide rail holder is modeled with a fixed hinge support. A simplified rack is included in the analysis as a steel sheet rectangle. The boundary conditions used in the analysis are shown in figure 4.13 and listed below.

- Wind load (purple arrows in figure 4.13). 5468 N (700 Pa), according to chapter 3.4.2.
- Fixed hinge on the upper cylindrical surface of the fitting screw (green arrows in figure 4.13).
- Bonded connection between the guide rail holder and the fitting screw. (See highlighted surfaces in figure 4.13).

The stress results of this simulation can be seen in figure 4.15.

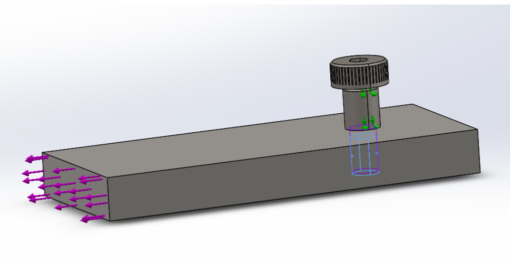

Figure 4.13: *Boundary conditions used in the first FEM analysis.*

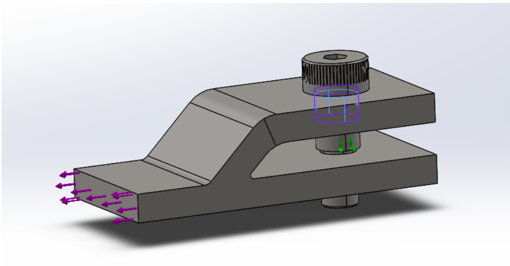

Figure 4.14: *Boundary conditions used in the second FEM analysis.*

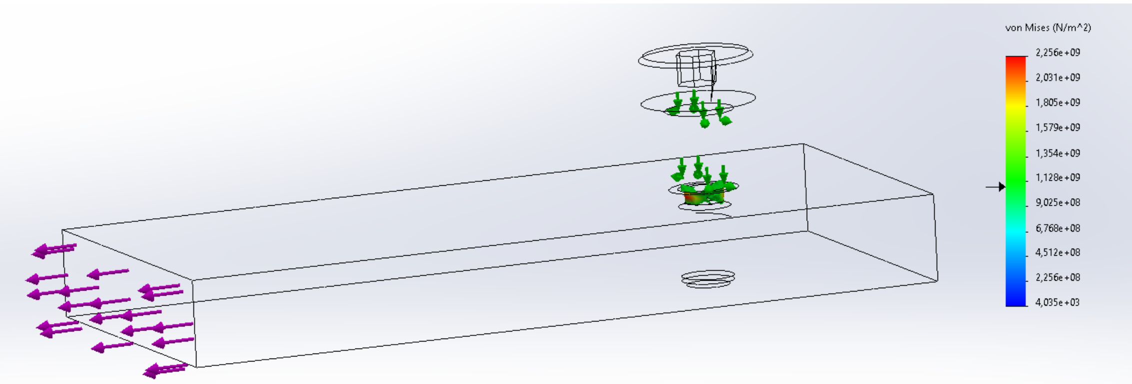

Figure 4.15: *ISO-surface of von-Mises equivalent stress results in the first FEM analysis.*

The finite element analysis showed that the screw would plasticize if mounted in this configuration. Another configuration using a "fork" attachment was tried instead, according to figure 4.14. A second FEM analysis was set up using this design. The boundary conditions are the same as before, except the "no penetration" boundary condition that was added between the two highlighted surfaces in figure 4.14. The complete set up and stress results of this analysis can be seen in figures 4.14 and 4.16 respectively.

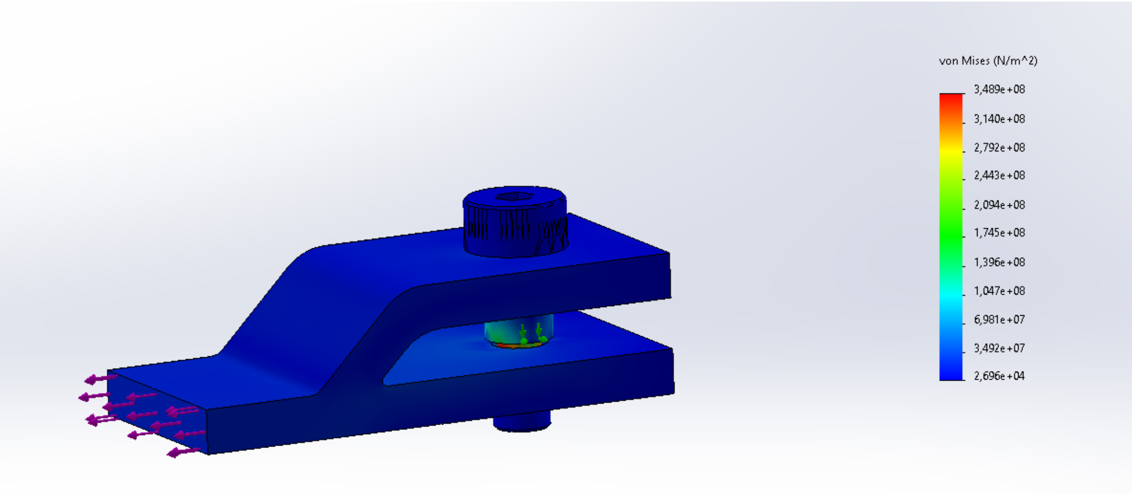

Figure 4.16: *Von-Mises equivalent stress results of the second FEM analysis.*

Since the fork method decreased the bending force on the screw, it was deemed as being more reliable and is therefore the final design. See figure 4.17. Only the upper part of the "fork" will be included, since this will be welded to the rack, resulting in the "fork" seen in figure 4.14. This will then be mounted on the guide rail holder provided by ASSA ABLOY, according to figure 4.32.

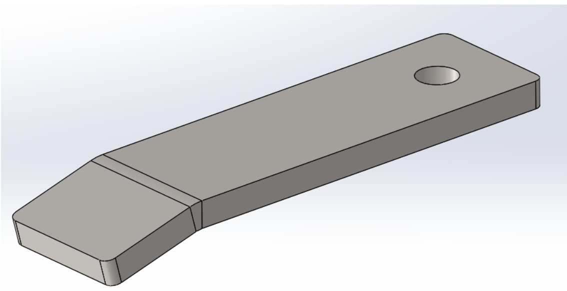

Figure 4.17: *Final design of the rack hinge.*

4.3.3 Gear

To determine the gear size, the required gear pitch diameter was calculated using the maximum torque of the CDM9 motor (70 Nm) and by assuming normal operating conditions (1/4 of the class 3 wind load). According to equation 3.11 in chapter 3.4.2 the force during full wind load on the gear is 5468 N.

Equilibrium equations 3.12 and 3.13 from chapter 3.4.2 gives the maximum required gear pitch diameter according to equation 4.1 below when combined.

$$d_0 = \frac{2 * \tau}{\frac{1}{4} * F_u} = \frac{2 * 70}{\frac{1}{4} * 5468} = 102.4\ mm \tag{4.1}$$

Henceforth the maximum gear diameter, for the motor to be able to operate under normal operating conditions, is 102.4 mm. The next step was to choose the teeth module. A table of relationships between gear pitch diameter, teeth modules, number of teeth and torque capacity was created. See table 4.1.

Module	**4**		**5**		**6**		**8**	
Diameter	**No. of teeth**	**Torque cap.**	**No. of teeth**	**Torque cap.**	**No. of teeth**	**Torque cap.**	**No. of teeth**	**Torque cap.**
75	18	40	15	NaN	12	NaN	9	NaN
80	20	46	16	NaN	13	NaN	10	NaN
85	21	48	17	NaN	14	NaN	10	NaN
90	22	50	18	65	15	NaN	11	NaN
95	23	57	19	70	15	NaN	11	NaN
100	25	70	20	80	16	NaN	12	NaN
105	26	75	21	85	17	NaN	13	NaN
110	27	80	22	90	18	110	13	NaN
115	28	85	23	100	19	140	14	NaN
120	30	97	24	110	20	160	15	NaN
125	31	100	25	125	20	160	15	NaN
130	32	117	26	140	21	170	16	NaN
135	33	128	27	150	22	180	16	NaN
140	35	143	28	160	23	190	17	NaN
145	36	150	29	175	24	200	18	220
150	37	157	30	190	25	230	18	250
155	38	165	31	195	25	230	19	275
160	40	175	32	205	26	275	20	300

Table 4.1: *Table showing the relationship between gear pitch diameter [mm], teeth module, number of teeth and torque capacity [Nm] at 200 rpm. Data used comes from Appendix B.1 and chapter 2.4.*

Studying table 4.1 on the 100 mm diameter row, it is possible to identify two candidates. Module 4 with 25 teeth and module 5 with 20 teeth. Module 5 with 20 teeth was chosen for the design due to it having a higher torque capacity than the module 4 option. Note that the torque capacity values are calculated for gears in motion.

For the prototype, it was determined to create a laser-cut version of the gear. The maximum sheet thickness of 10 mm was chosen due to laser limitations cutting through thicker steel sheets.

The final design of the gear is illustrated in figure 4.18 below.

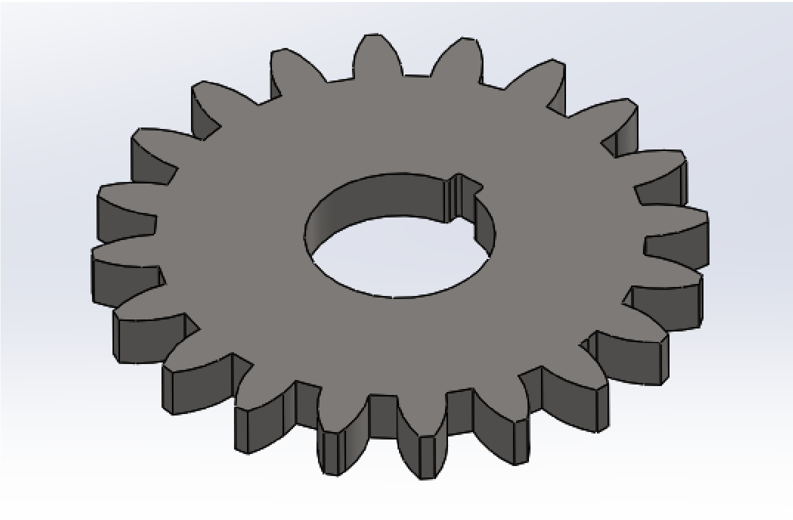

Figure 4.18: *Final design of the gear.*

4.3.4 Rack

Optimizing side space requirements is part of the main goal of this thesis. Henceforth an elaborate model of the folding door movement and how this would affect the rack's position was created in MATLAB. This model is presented in Appendix C. This model was used to iterate to the optimal rack radius with the objective of minimizing side and depth space requirements. For a door with a 1.25 m door leaf width, the optimal rack pitch radius proved to be 1.05 m with an angle of 135 degrees.

In chapter 4.3.3, teeth module 5 was selected for the gear. The same module has to be used on the rack. Module 5 combined with the rack radius, gives the number of teeth on the 360° circular rack, according to equation 2.3 in chapter 2.4. The 135° rack seen in figure 4.19 is then cut from the 360° rack.

For the rack prototype, a 10 mm thick (maximum thickness possible to laser-cut) laser-cut steel version was used. In order for the guide rail holders to not interfere with the rack support rollers described in the next chapter, the rack had to be 70 mm wide.

It was also decided to introduce slots in order to reduce the weight of the rack. The final design of the rack can be seen in figure 4.19.

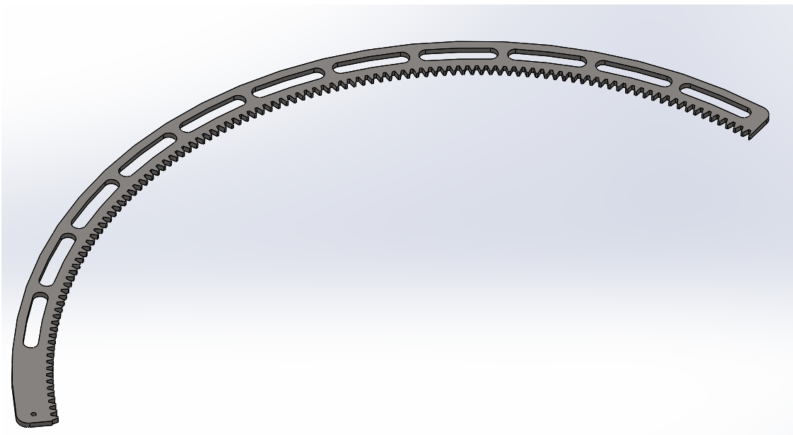

Figure 4.19: *Final design of the circular rack.*

4.3.5 Rack Support

To hold the rack in place and keep it in contact with the gear, a supporting structure is needed. Two sketches of how this could be achieved were created. See figures 4.20 and 4.21.

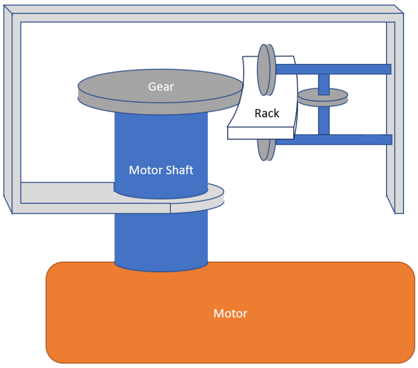

Figure 4.20: *Rack support concept 1.*

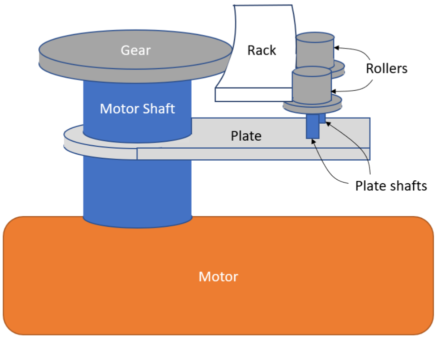

Figure 4.21: *Rack support concept 2.*

The design chosen for the rack support is concept 2 illustrated in figure 4.21. This choice was based on the complexity of the solution with regards to bearing locations and the fact that it requires less headroom.

Plate Shafts

Between the rollers and plate shaft seen in figure 4.21, bearings are needed. The 6202RS bearings ASSA ABLOY already had in stock determined the main dimensions of the plate shafts. The 6202RS bearings can be seen in figure 4.22. The bearings have an inner diameter of 15 mm and a height of 11 mm. In order for the bearings to sit tight, a greater diameter was used beneath them and a SGA15 snap ring, see figure 4.23, above them. An assembly illustrating the 6202RS mounted in place is shown in the figure 4.25.

Figure 4.22: *Bearing with rubber seal, 6202RS.*

Figure 4.23: *Snap ring SGA15.*

The height of the plate shafts was adjusted to allow the other components to assemble well. The final design is presented in figure 4.24.

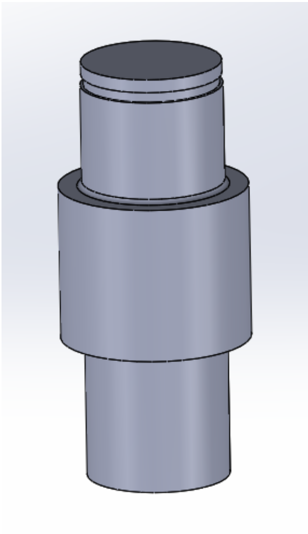

Figure 4.24: *Final design of the plate shaft.*

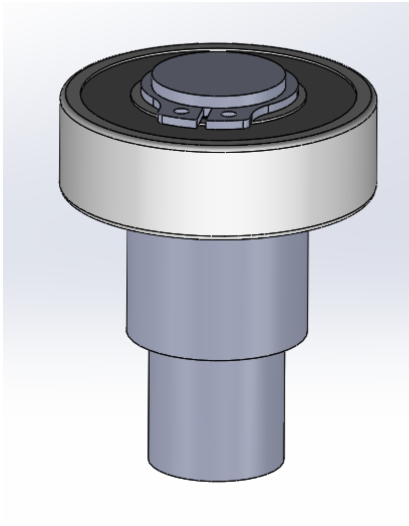

Figure 4.25: *Plate shaft assembly.*

Rollers

The rollers support the rack and keep the teeth of the rack and gear in contact. The rollers must be large enough to house the 6202RS bearings. The bearings allow the rollers to rotate and the rollers will act as protective housings for the bearings. In order for the bearing to sit tight inside the roller, a slot for a SGH35 snap ring and an elevation at the bottom were created. This can be seen in figures 4.26 and 4.27. An assembly showing the positions for the SGH35 and bearing is presented in figure 4.28.

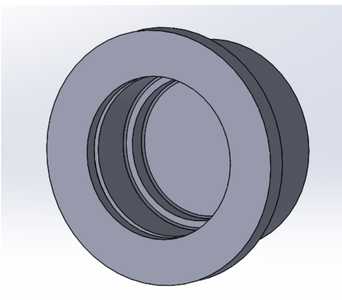

Figure 4.26: *The roller with an elevation at the bottom and a slot for the snap ring.*

Figure 4.27: *SGH35 snap ring.*

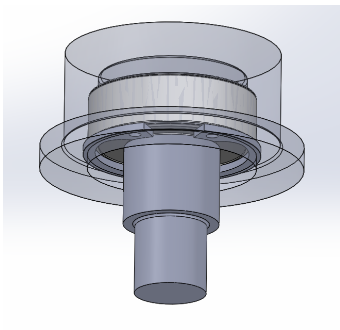

Figure 4.28: *An assembly showing the roller, bearing and SGH35.*

Plate

As illustrated in 4.21 some sort of bearing is needed to mount the plate to the motor shaft. The motor shaft available at ASSA ABLOY has a diameter of 34.9 mm. A bearing called SB207 (see figure 4.29) has an inner diameter of 35 mm and includes grub screws. The grub screws remove the need of press fitting the bearing onto the shaft. The bearing also includes holes to easily mount the plate, simplifying the installation and is therefore chosen.

Figure 4.29: *SB207 bearing and included holder.*

The plate was designed with a slot, allowing the mechanic installing the drive system to engage/disengage the teeth of the gear and rack. The design also includes holes for the two plate shafts, see in figure 4.21. Fixing the plate shafts to the plate is done by fitting them through the holes and welding them to the opposite side. A hole for a magnetic sensor was also made. The magnetic sensor will help the motor controller recognize when the door has reached a fully open position. To enable the gear and rack to be cut from the same sheet of metal the thickness of the plate was set to 10 mm. The final design can be seen in figure 4.30.

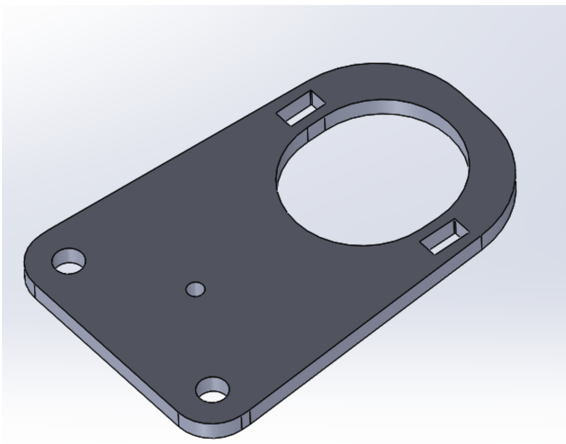

Figure 4.30: *Final design of the plate.*

4.4 Final CAD-model and Design

In this section, a final CAD-model and an assembly including all parts will be presented.

4.4.1 Rack Assembly

The final rack assembly consists of four parts. The "fork", a guide rail holder, a fitting screw and the rack. The hole in the rack is threaded and requires no nut. An exploded view and collapsed view, including the previously mentioned weld, is shown in figures 4.31 and 4.32 respectively.

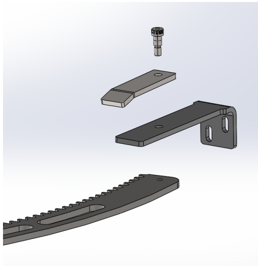

Figure 4.31: *Exploded view of the rack assembly and its components.*

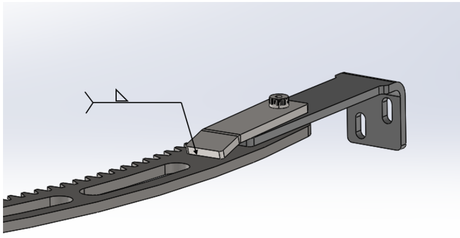

Figure 4.32: *The corresponding collapsed view of the rack assembly with a weld designation.*

4.4.2 Motor Assembly

All in all, the final motor assembly consists of nine individual parts. A CDM9 motor, motor shaft, two stop rings used on either side, two keys, a SB207 bearing, the gear and the motor attachment discussed earlier. An exploded view of the assembly and the corresponding collapsed view can be seen in figures 4.33 and 4.34 respectively. A figure with weld designations for the motor attachment can be seen in figure 4.35.

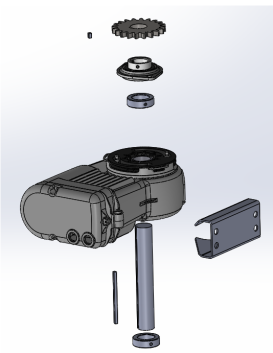

Figure 4.33: *Exploded view of the motor assembly.*

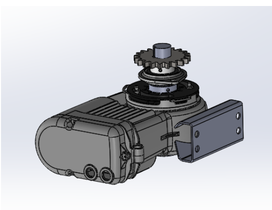

Figure 4.34: *The corresponding collapsed view of the motor assembly.*

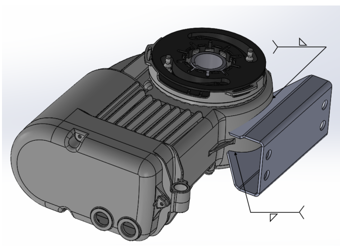

Figure 4.35: *Weld designations for the motor attachment.*

4.4.3 Rack Support

The final assembly consists of the plate, plate shafts and rollers as main components. The snap rings and the 6202RS bearings are also part of this assembly. An exploded and collapsed view of the assembly can be seen in figures 4.36 and 4.37 respectively. The plate shafts will be welded to the underside of the plate, as seen in figure 4.38.

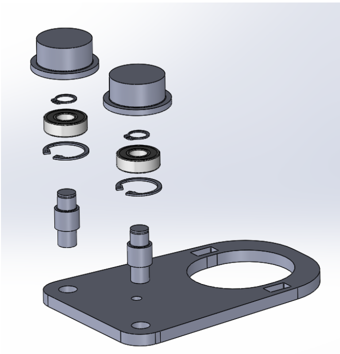

Figure 4.36: *An exploded view of the rack support assembly.*

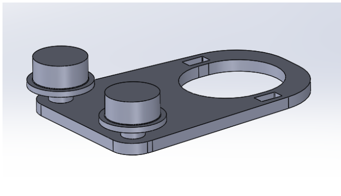

Figure 4.37: *The corresponding collapsed view of the rack support assembly.*

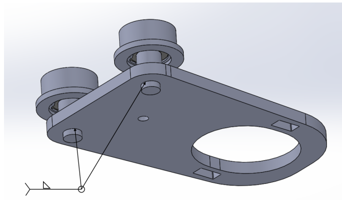

Figure 4.38: *A bottom view of the plate, including weld designations.*

4.4.4 Final Design

Figures 4.39-4.44 illustrate how the final combined design looks and how the different assemblies are mounted.

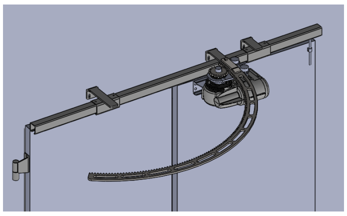

Figure 4.39: *The final concept in a closed state.*

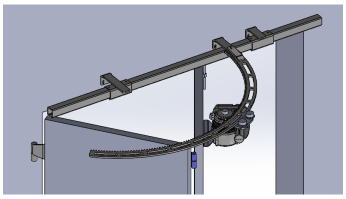

Figure 4.40: *The final concept in a half open state.*

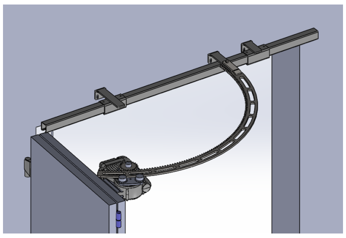

Figure 4.41: *The final concept in an open state.*

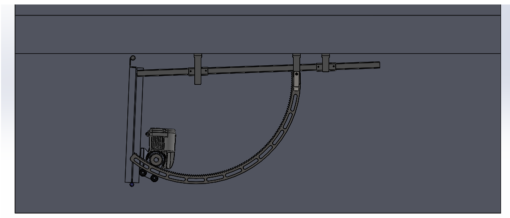

Figure 4.42: *View from above in a fully opened state.*

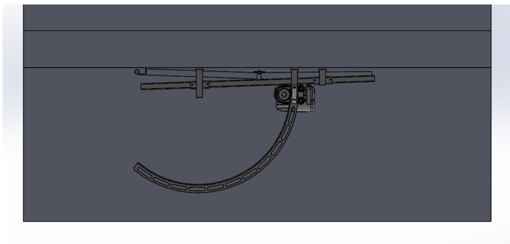

Figure 4.43: *View from above in a closed state.*

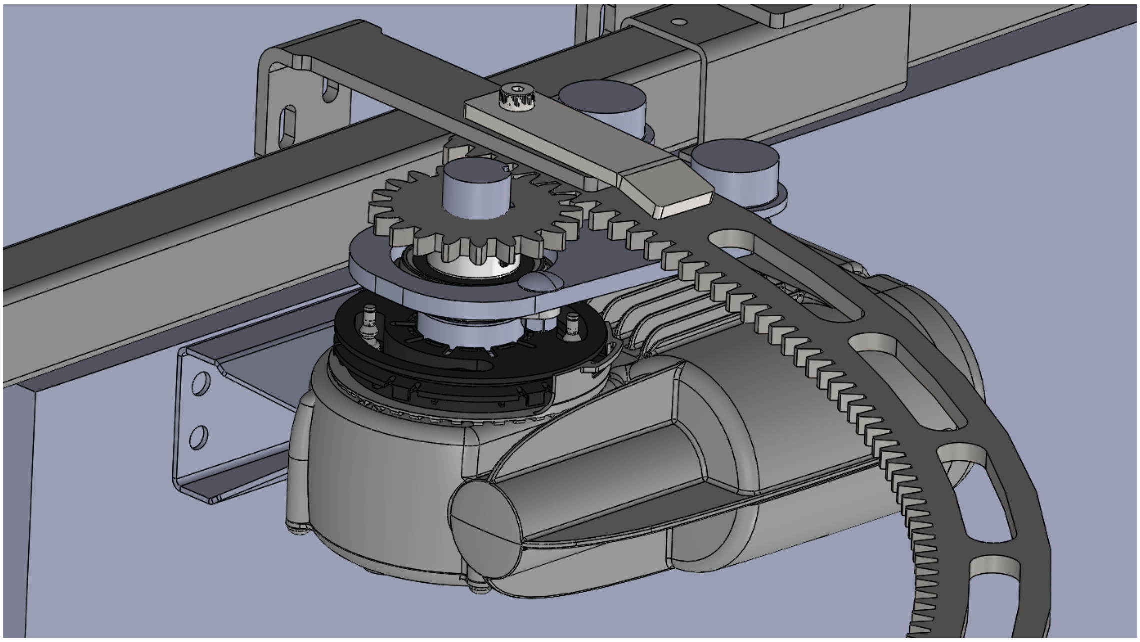

Figure 4.44: *A detailed view of the final design.*

Space Requirements

The new drive system's space requirements are measured in the final CAD model and are presented in table 4.2. These values confirm that the design meets the specifications in chapter 3.2.1 regarding side space and headroom. The depth required is also less than the door leaf width. This means that the drive system requires no extra space depth-wise.

Side space	Headroom	Depth
130	145	1230

Table 4.2: *Space requirements for the new drive system. All values in mm. Values measured in the final CAD-model.*

Bill of Materials

In table 4.3 below all the included main components in the final design are listed. It is also specified for which sub assembly the components belong to. (1) Rack sub assembly, (2) Motor sub assembly, (3) Rack support sub assembly.

Name	Notes	Quantity	Material
Motor attachment	(2)	1	EN AW-1050 A H14/H24 SS4007 (AL99.5)
Rack fork	(1)	1	S355J2 + N
Fitting screw	(1),12X16/M10	1	Tempered steel grade 12.9 according to DIN ISO 7379
Gear	(2),Module 5, 20 teeth	1	S355J2 + N
Rack	(1),Module 5, 410 teeth	1	S355J2 + N
Plate shaft	(3)	2	S355J2C + C
Bearing	(3),6202RS	2	
Snap ring	(3),SGA15	2	Phosphated spring steel according to DIN471
Rollers	(3)	2	S355J2C + C
Snap ring	(3),SGH35	2	Phosphated spring steel according to DIN472
Plate	(3)	1	S355J2 + N
Bearing	(2),SB207	1	
Rail holder	(1),Provided by ASSA ABLOY	1	S355J2 + N
Motor shaft	(2),Provided by ASSA ABLOY	1	S235JGR2 AC + C
Motor shaft keys	(2),Provided by ASSA ABLOY	2	S235JGR2
Stop rings	(2),Provided by ASSA ABLOY	2	

Table 4.3: *Bill of materials for the final design.*

5. Testing and Refinement

This chapter will describe the different tests conducted on the new drive system. The results of these tests are summarized and compared to the current solution. Possible refinements will also be presented.

5.1 General Notes

The door tested for opening speed, current consumption and crushing forces in chapter 3.1.3 is also tested with the new drive system. The original drive system can be seen in figure 5.1 and the new design in figure 5.2.

Figure 5.1: *Original drive system.*

Figure 5.2: *Chosen drive system.*

5.2 Initial Changes

Before testing could start, a few design changes had to be made. The rack turned out heavier than expected. Precautionary measures were therefore taken before mounting the rack on the wall. The wall mount was flipped and a triangle support was added to increase its rigidity and the opposite end of the rack was tied up to the roof with a lace. The flipped wall mount increased the required headroom to 190 mm, still less than the desired 200 mm. See figures 5.3 and 5.4.

Figure 5.3: *Triangle added to the wall mount.*

Figure 5.4: *The lace tying up the rack.*

The motor attachment designed in chapter 4.3 could not be manufactured due to the two bends being too close to each other. This was changed to a design with only one bend that has a larger bending angle. Compare figures 5.5 and 5.6

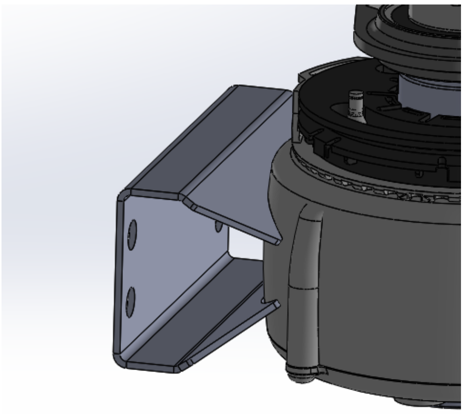

Figure 5.5: *The final design of the motor attachment.*

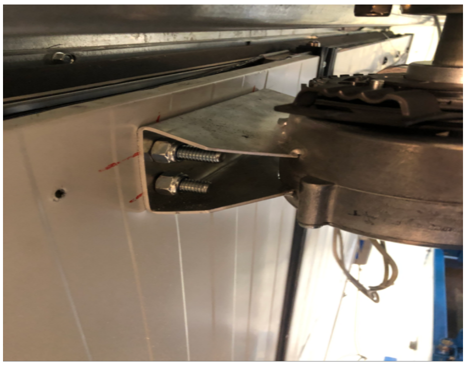

Figure 5.6: *The redesigned motor attachment.*

When assembling and mounting the solution on the door it was found that the motor shaft was too short to be fastened on the bottom of the motor using a stop ring as intended according to chapter 4.4.2. Due to this, the motor shaft slid out resulting in the gear teeth losing contact when operating the door. To fasten the shaft to the bottom of the motor, preventing it to slide out, a screw was welded to the end of the shaft, according to figure 5.7. A nut together with a washer was then used to fasten the motor shaft in place, according to figure 5.8.

Figure 5.7: *The shaft extension.*

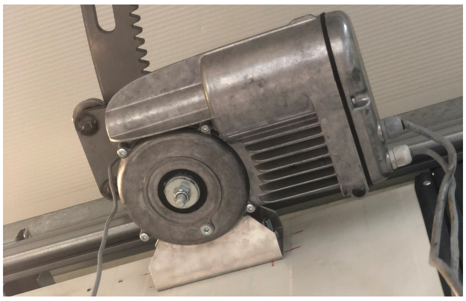

Figure 5.8: *The shaft washer and nut in place.*

A detailed view of the prototype assembly including all changes can be seen in figure 5.9.

Figure 5.9: *Detailed view of the prototype assembly.*

5.3 Opening Cycle Test

The first test conducted on the prototype is the opening cycle test where the door is run through 10 opening/closing cycles, according to the EN 12604 standard (see chapter 2.3.2) with the purpose of ensuring an operational door. Results of this test showed that the drive system passed.

5.4 Current Consumption

The current consumption test is performed as described in chapter 3.1.3.

5.4.1 Results

Results of the current consumption test during opening can be seen in figure 5.10 with corresponding calculated motor torque presented in figure 5.11.

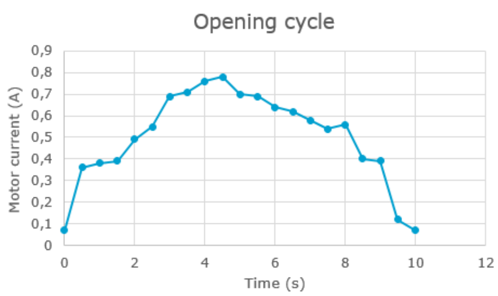

Figure 5.10: *Current consumption of the motor during opening.*

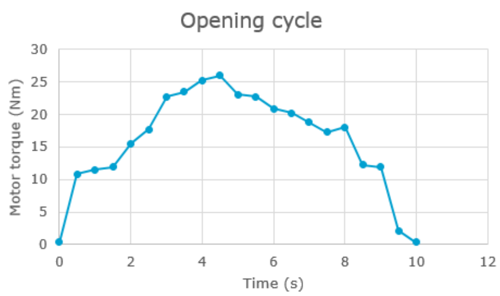

Figure 5.11: *Motor torque during opening.*

Corresponding results for the closing cycle can be seen in figures 5.12 and 5.13.

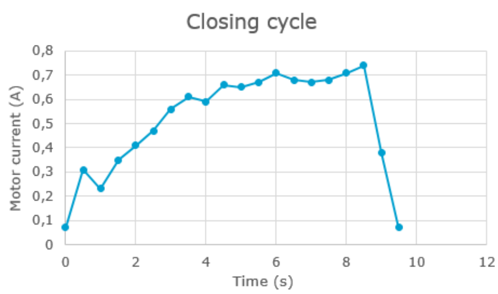

Figure 5.12: *Current consumption of the motor during closing.*

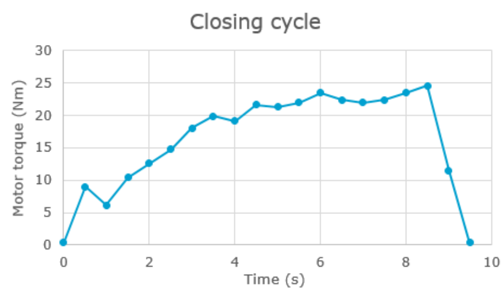

Figure 5.13: *Motor torque during closing.*

5.4.2 Comparison

Compared to the current drive system at ASSA ABLOY, evaluated in chapter 3.1.3, it is seen that the new drive system's current consumption is substantially lower and smoother during both opening and closing cycles. The lower current consumption can be explained by a reduction of losses due to the use of gears in the new drive system, instead of using a chain with carts sliding inside a rail. Similarly, the reduction of power spikes can be attributed to the force always acting in the direction of motion in the new drive system. The smoother operation and lower current consumption opens up opportunities for an increase in speed if required. Current and torque consumption for both the current and the new drive system can be seen in figures 5.14 through 5.17.

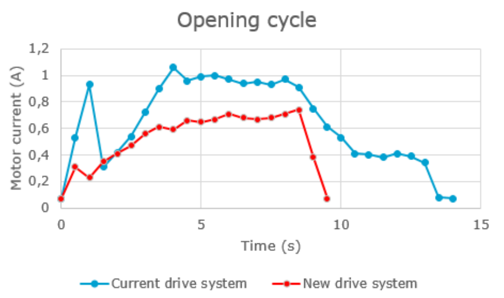

Figure 5.14: *Motor current consumption during opening for the current and new drive system.*

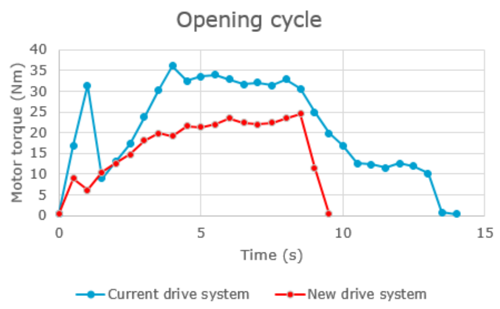

Figure 5.15: *Motor torque during opening for the current and new drive system.*

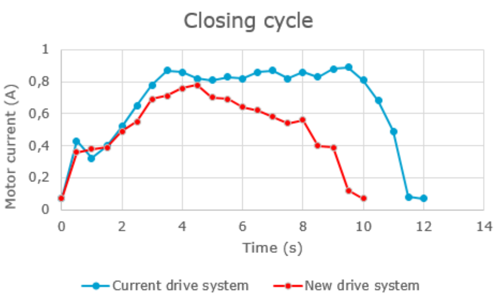

Figure 5.16: *Motor current consumption during closing for the current and new drive system.*

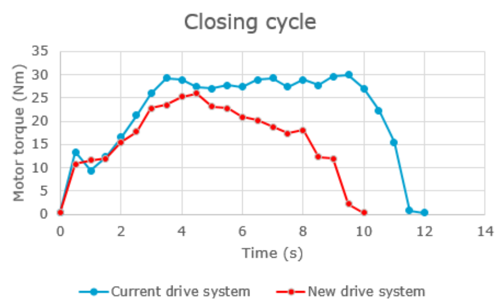

Figure 5.17: *Motor torque during closing for the current and new drive system.*

5.5 Crushing Force

The crushing force test is performed as described in chapter 2.3.3. It was not possible to measure the crushing forces against neighboring stiff parts because there were no walls or other rigid objects nearby in the test lab.

5.5.1 Results

Results of the crushing force test can be seen in table 5.1 and they verify that the new drive system is compliant with the EN 12453 standard.

- **Door: FD2050P**
- **Control unit: ECS 950**
- **Machinery: CDM9FD**
- **Sensor: Bircher DW 40 Pneumatic**

Crushing force test				
Height from floor [mm]	**Opening gap [mm]**	**Crushing force [N]**	**Time while force >150 N [ms]**	**Unload within 5 s**
50	50	228	177	YES
50	300	280	139	YES
50	500	264	137	YES
1500	50	111	0	YES
1500	300	214	130	YES
1500	500	226	129	YES
2500	50	129	0	YES
2500	300	252	142	YES
2500	500	263	134	YES

Table 5.1: *Results from the EN 12453 test conducted on the new drive system. Crushing force and time while force >150 N are averages from 3 measured values. The maximum allowed force between closing edges is 400 N within a period of maximum 0.75 s and unload (reverse) within 5 s.*

5.6 Opening and Closing Speed

The opening and closing speed test is carried out as described in chapter 3.1.3. Opening and closing times for the new drive system are presented in table 5.2.

Number of test	1	2	3	4	5
Opening cycle	10.12	10.03	9.86	9.96	9.91
Closing cycle	10.36	9.66	10.05	10.09	10.00

Table 5.2: *Results of opening and closing speed test. All values in seconds.*

Results show that the opening speed is improved by four seconds and closing speed by two seconds. According to ASSA ABLOY's specifications a speed of twice as today, 6-7 seconds, is desirable. This goal is unfortunately not achieved, but according to the current test in chapter 5.4 there is room for speed improvements.

5.7 Refinements

Detail drawings of all proposed refinements in this chapter can be found in appendix D.2.

5.7.1 Rack

The rack designed in chapter 4.3.4 weighs approximately 10 kilograms. This results in a significant deflection of the rack when the door is in a closed position. In turn, the deflection poses a risk of teeth disengagement between the rack and gear. The refinements necessary to counteract this problem is to reduce the width of the rack. Reducing the width from 70 mm to 15 mm results in a weight reduction of 65%. The refined rack weighs 3.5 kg and can be seen in figure 5.18.

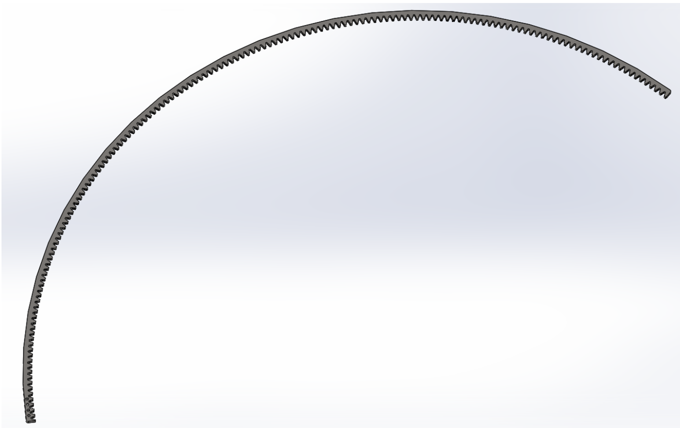

Figure 5.18: *The refined rack.*

”Hockey” Rack

To achieve the same results but further decrease the depth protrusion into the facility when the door is closed, at the expense of side space, the design of the rack can be modified. Our proposed modification uses a combination of a straight and circular rack and can be seen in figure 5.19.

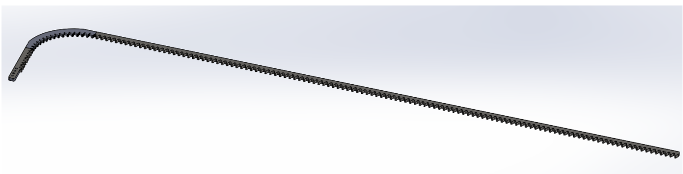

Figure 5.19: *The proposed ”hockey” rack design.*

In table 5.3 below different combinations of these racks and how they affect the space required for a folding door with a section width of 1.25 meters are presented. A comparison of the 310 mm diameter "hockey" rack and the refined circular rack can be seen in figures 5.20 through 5.22.

Diameter of the circular rack	Length of the straight rack	Side space required	Depth space required
310	1900	470	2100
500	1750	420	1900
625	1700	420	1800
1000	1450	350	1600
1500	1150	330	1350
1750	950	260	1290
2000	750	<200	<1250

Table 5.3: *The different "hockey" rack combinations for folding doors with a section width of 1.25 metres. All values in mm.*

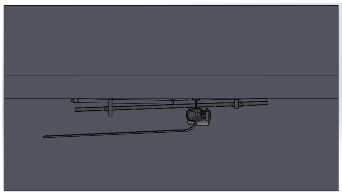

Figure 5.20: *"Hockey" rack with a small circular rack and large straight racks.*

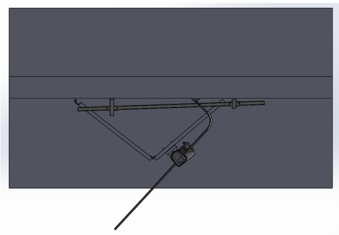

Figure 5.21: *"Hockey" rack with half-open door.*

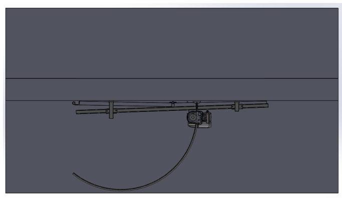

Figure 5.22: *The refined circular rack.*

Conclusions regarding cross-traffic when the door is closed can be drawn. The "hockey" rack design works better when there is a need for cross-traffic close to the door as it does not take up a large amount of depth into the facility. While the refined circular rack is better regarding side space since it never takes up more than the door itself.

5.7.2 Rack Assembly

The refinements done to the rack implies a change in the design of the rack wall mount. Since the new rack is very narrow, a new design with screws and slots was created. Bushings with flanges were used to prevent rubbing wear and noise in the refined design. See figures 5.23 and 5.24.

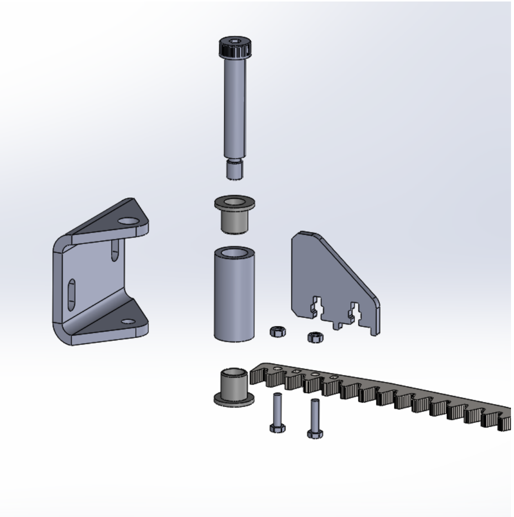

Figure 5.23: *Exploded view of the refined rack assembly.*

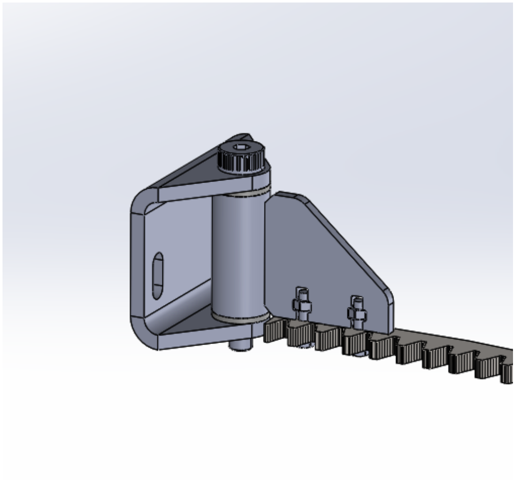

Figure 5.24: *Collapsed view of the refined rack assembly.*

5.7.3 Safety Device

To comply with the EN 12604 standard, all moving parts under a 2.5 m height must be encapsulated or in other means made inaccessible to avoid personal injuries. In installations where the door has a height less than 2.5 meters, a safety device must therefore be incorporated. The safety device would, with the help of springs and a magnetic sensor, attached to the plate, detect if it was moved. The motor would disengage and stop the door from opening or closing. A sheet metal and a plastic version of the safety device is proposed. See figures 5.25-5.27 below.

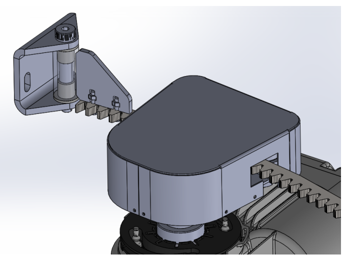

Figure 5.25: *Safety device in sheet metal.*

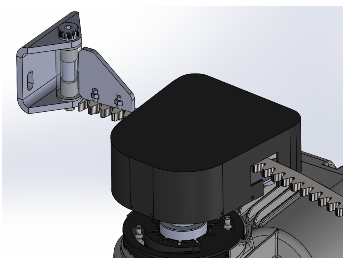

Figure 5.26: *Safety device in plastic.*

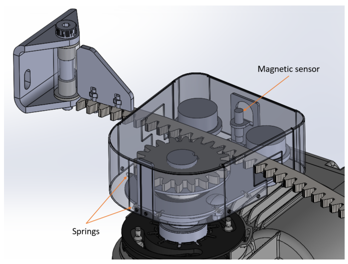

Figure 5.27: *A transparent view of the safety device showing the springs attached and the magnetic sensor.*

5.7.4 SB207 Bearing

The SB207 bearing used caused problems since it was spherical and had the ability to roll if the screws attaching it to the plate were not tightened enough. See figures 5.28 and 5.29. However, these are the only bearings available with grub screws. The grub screws are needed for fixing the bearing to the motor shaft. The solution proposed is to either replace the spherical bearing through a special order or replace the holder which enables the rolling ability. A new holder design attached to the plate can be seen in figure 5.30.

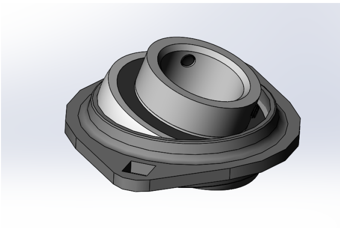

Figure 5.28: *The misalignment possibility of the SB207 bearing in CAD.*

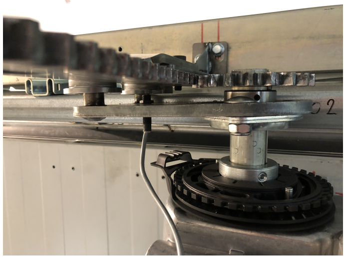

Figure 5.29: *Image of the misalignment possibility of the SB207 bearing in the prototype.*

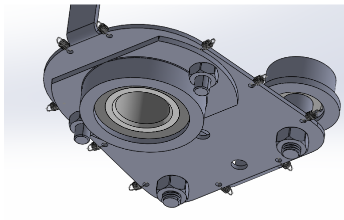

Figure 5.30: *A bottom view of the refined holder for the SB207 bearing attached to the plate.*

5.7.5 Plate

The previous plate design, developed in chapter 4.3.5, had a 10 mm thickness. This thickness was chosen because it would make it possible to cut the rack, gear and plate from the same sheet of metal. This turned out to be unnecessarily thick, adding material cost and weight to the drive system. The thickness was therefore decreased to 3 mm. Also, the implementation of the safety device described in chapter 5.7.3 implies a change in design of the plate. An edge or point, on the plate, where the safety device can rest and holes where the springs and magnetic sensor can be mounted are needed. The most simple solution is to include two bent flanges according to 5.31. The new plate design in the rack support with all refinements can be seen in figure 5.32.

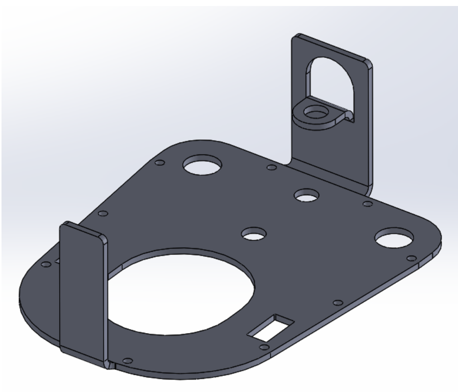

Figure 5.31: *The refined plate.*

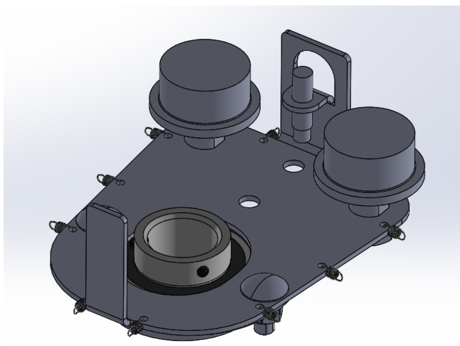

Figure 5.32: *The refined rack support assembly.*

5.7.6 Plate Shafts

The refinements regarding the plate shaft are minimal. A thread at the end of each shaft is introduced in order to remove the necessity to weld them to the plate. As they will be screwed in place, slots will be milled to make it easier to mount them. The changes done also includes a length adjustment. The thick parts of the shaft now needs to be longer since the plate's thickness was decreased. The refined plate shaft can be seen in figure 5.33.

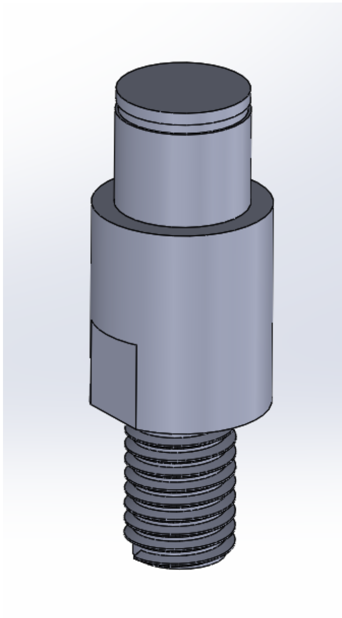

Figure 5.33: *The refined plate shaft.*

5.7.7 Motor Attachment

Since the motor attachment designed in chapter 4.3.1 was not possible to manufacture, and to avoid the need for an overbend as it would further complicate the manufacturing, the attachment needs to be redesigned. The new design must be simple to manufacture. It was therefore decided to split the attachment into two pieces, possible to put together similar to puzzle pieces. See figures 5.34 and 5.35. This way, the main design is kept and made easier to manufacture.

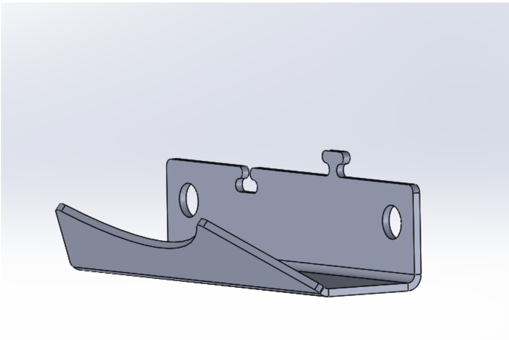

Figure 5.34: *One half of the refined motor attachment.*

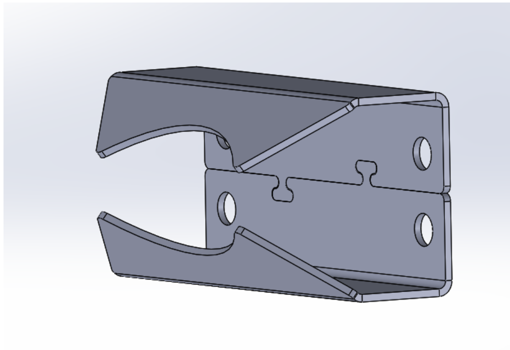

Figure 5.35: *The complete refined motor attachment.*

Also, for the safety device discussed in section 5.7.3 to fit, the motor needs to be moved further from the door. The motor attachment is therefore made longer to compensate for the additional dimensions this added. The old and new design can be seen and compared in figures 5.37 and 5.36.

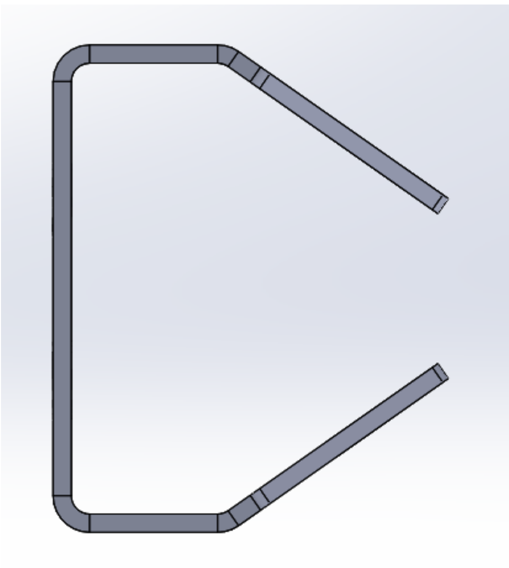

Figure 5.36: *The initial motor attachment's design. View from the side.*

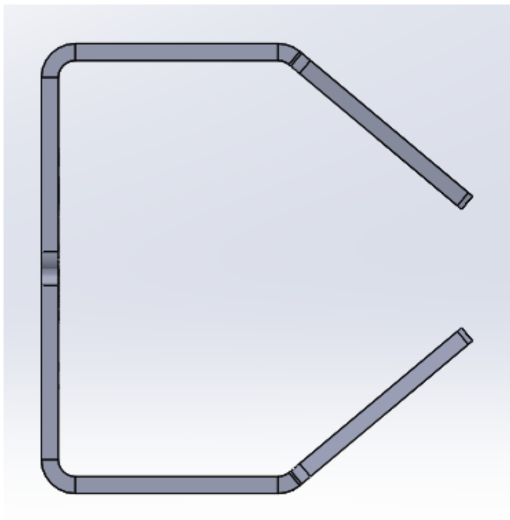

Figure 5.37: *The refined motor attachment's design. View from the side.*

5.7.8 Gear

The gear used in the prototype does not fulfill the desired properties. The gear should in theory be made of different material than the rack or have a special coating to minimize the strain and tear on the teeth. An important factor is the requirement of having good teeth quality and using a gear that is laser-cut does not fulfill these requirements. The gear has to be manufactured from a proper gear supplier. In order to help stabilize the rack a circular plate is added beneath the gear preventing the teeth of the rack to tilt. See figures 5.38 and 5.39.

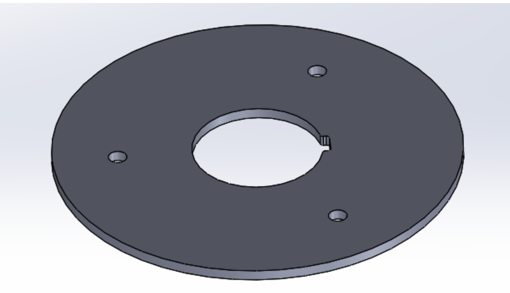

Figure 5.38: *The gear plate that is added beneath the gear with spot weld holes.*

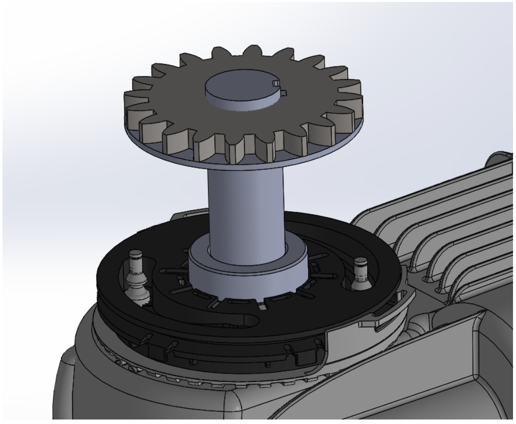

Figure 5.39: *The assembled gear and plate.*

5.7.9 Refined Final Design

In figures 5.40-5.45 the refined final design can be seen. The refined design includes all changes introduced in this chapter. This design will not be tested due to time limitations.

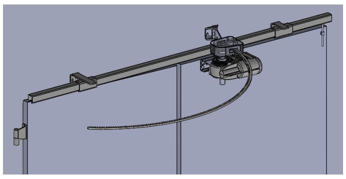

Figure 5.40: *The refined final concept in a closed state.*

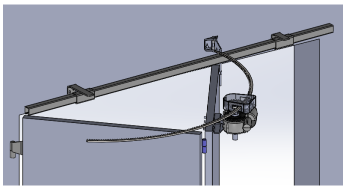

Figure 5.41: *The refined final concept in a half open state.*

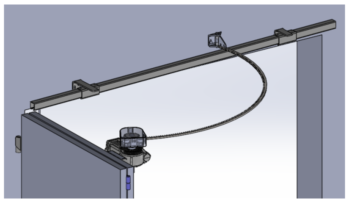

Figure 5.42: *The refined final concept in an open state.*

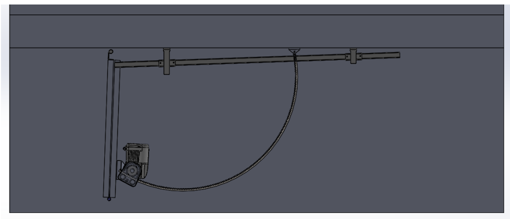

Figure 5.43: *View from above in a fully opened state.*

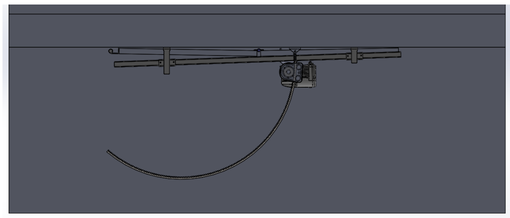

Figure 5.44: *View from above in a closed state.*

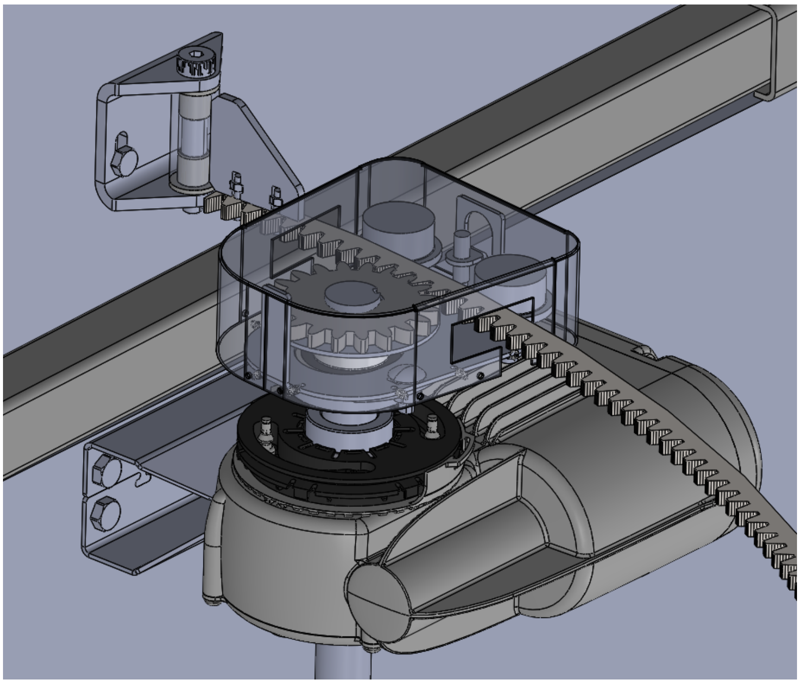

Figure 5.45: *A detailed view of the refined final design.*

Space Requirements

The refined drive system's final space requirements are measured in the refined CAD-model and are presented in table 5.4. These values confirm that the design meets the specifications in chapter 3.2.1 regarding headroom and side space.

Side space	Headroom	Depth
110	200	1200

Table 5.4: *Space requirements for the refined drive system. All values in mm. Values measured in the refined CAD-model.*

Bill of Materials

In table 5.5 below all the included components in the refined final design are listed.

Name	Notes	Quantity	Material
Circular rack		1	S355J2 + N
Hockey rack	z400, z64, replaces the circular rack if chosen	1	S355J2 + N
Rack support		2	S355J2C + C
Wall support		1	S355J2 + N
Wall mount		1	S355J2 + N
Wall hinge		1	S355J2 + N
Fitting screw	ISO7379 16X80/M12	1	
Bushing with flanges	LB30201623S	2	
Nut	DIN934 A4 M6	2	Stainless, acid-proof steel according to DIN 934/ISO 4032
Screw	DIN933A4 M6X25	2	Stainless, acid-proof steel according to DIN 933/ISO 4017
Safety device	Could be made both in sheet metal and plastic	1	EN AW-1050 A H14/H24 SS4007 (Al99.5)
Extension spring	9462	10	Spring steel EN 10270-1-SM Stainless steel EN 10270-3-1.4310
Bearing holder	SB207 bearing	1	S355J2C + C
Bearing	SB207	1	
Plate		1	DX51 2275
Screw	DIN933A4 M12X70	4	Stainless, acid-proof steel according to DIN 933/ISO 4017
Magnetic sensor	K0046623, M10X0.75, Provided by ASSA	2	
Plate shaft		2	S355J2C + C
Snap ring	SGA15	2	Phosphated spring steel according to DIN471
Snap ring	SGH35	2	Phosphated spring steel according to DIN472
Carriage bolt	DIN603 A4 M10X25	2	Stainless steel, acid-proof according to DIN603
Nut	DIN985 A4 M10	2	Stainless steel, acid-proof according to DIN985
Motor attachment		2	EN AW-1050 A H14/H24 SS4007 (Al99.5)
Screw	DIN933A4 M10X70	2	Stainless, acid-proof steel according to DIN 933/ISO 4017
Gear		1	S355J2 + N
Gear support		1	S355J2 + N
Motor shaft	Provided by ASSA ABLOY	1	S235JGR2 AC + C
Motor shaft keys	Provided by ASSA ABLOY	2	S235JGR2
Stop rings	Provided by ASSA ABLOY	2	
CDM9	Provided by ASSA ABLOY	1	

Table 5.5: *Bill of materials for the refined drive system.*

6. Discussion

In this chapter the project progression and findings are discussed. Suggestions on how this project can be continued are also brought up.

6.1 Project Progression

The project held the intentional time frame both during the concept development and design phase. It was not until the testing and refinement phase that delays disrupting the initial time plan became a problem. The delays were a consequence of longer than planned delivery times and manufacturing difficulties. During this "downtime" the majority of the report was written.

6.2 Findings

6.2.1 General Discussion

The tests that were conducted had many indications that this new drive system has potential. With smaller installation dimensions, lower current consumption, smoother operation and faster opening speed it will be seen as a working concept and probably have better market incentive than the current drive system. The new drive system can be configured in such a way that the side space and overhead dimensions completely vanishes which makes this solution, in this aspect, a very compact and robust solution. The downside with the new drive system is that it takes up quite much depth space when closed, but this can be easily solved by choosing the proposed "hockey" rack design. Nevertheless, there will always be a trade-off in dimensions between required side space and depth. If one is low the other one is bound to be large. As the new design system consists of simpler and fewer components in a 2+0 installation, from a economical point of view, the designed drive system is deemed more cost efficient than the current drive system. However, the new design has to be mirrored to be able to open a 2+2 folding door. This doubles the number of components required and a complete cost analysis has to be conducted in order to determine if the new drive system is economically beneficial in 2+2 installations.

6.2.2 Unique Rack Variants

Circular Rack

In order to examine how many rack variants would be required to cover ASSA ABLOY's available folding door sizes, a table was constructed. The complete table (that can be seen in Appendix C) shows the relationship between leaf width, rack radius and depth/side space requirements. A summary of the recommended rack radii where the leaf widths range from 550 to 1250 mm is presented in table 6.1. If these radii are chosen, the drive system's space requirements will be the same as for a manually operated door. If a leaf width falls in between two of the intervals in table 6.1, choose the larger rack. This will however compromise on depth space. This can be seen in Appendix C.

Leaf Width [m]	0.55-0.6	0.65-0.7	0.75-0.85	0.9-1.05	1.1-1.25
Rack radius [m]	0.4	0.5	0.6	0.75	0.95

Table 6.1: *Five different rack radii cover all ASSA ABLOY's available folding door configurations/sizes. All units in m.*

The table shows that five different rack variants are enough to cover ASSA ABLOY's entire folding door size range.

"Hockey" Rack

The "hockey" rack discussed in chapter 5.7 (and that can be seen in figure 5.19) has potential to be the only rack variant needed for all ASSA ABLOY folding door sizes. The straight part rack can be cut to fit each door size and is the perfect choice if cross-traffic exists, due to its minimal depth protrusion when the door is closed. Having only one rack that can be adapted to all door sizes is attractive to the company because it will reduce warehousing costs and allow the company to buy the racks in larger quantities.

6.2.3 Wind Class

The limiting factor of the new drive system design is the gear, since the torque the teeth are able to withstand is greatly affected by the diameter and teeth module. One goal for the project, according to ASSA ABLOY's specifications, was to achieve a drive system that, when mounted on a 5x5 m large folding door, is able to withstand a wind load of class 3. In figure 6.4 it is possible to see that this is achieved only if the door leaf width is less than 0.7 meters due to the gear's torque capacity. This gives a maximum door width of $0.7 * 4 = 2.8\ meters$. The only way to increase the gear's torque capacity is to increase its diameter and/or module. It may be possible for the CDM9 to drive the door with larger gears. From the current consumption tests in chapter 5.4 it is apparent that the full potential of the motor is not used. If, for example, a 120 mm diameter gear with module 6 was used instead, the gear's torque capacity would increase from 80 Nm to 160 Nm (see table 4.1 in chapter

4.3.3). Worth mentioning here is that the theoretical values in table 4.1 are for gears in motion. The gears are believed to have higher torque capacities in static situations. Since the motor's maximum reverse torque is 200 Nm due to the self inhibiting properties of it's worm gear (see chapter 2.4.3), the gear with 160 Nm torque capacity would act as a safety device that prevents damage to the motor by failing first. According to the wind class 3 graph in figure 6.4, a gear with 160 Nm torque capacity can be mounted on a door with 1 m wide leaves. For other door heights, the correlation between leaf width and torque requirement for different wind loads can be seen in figures 6.1-6.3. The MATLAB code used to create the figures can be found in Appendix C.

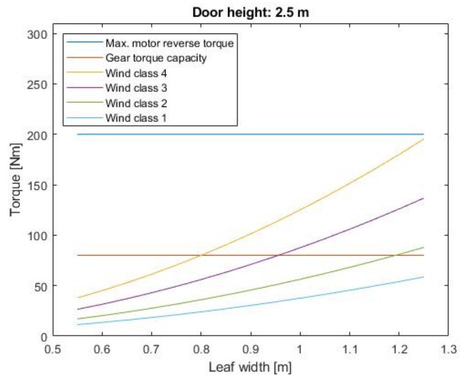

Figure 6.1: *Correlation between leaf width and torque required to resist different wind loads for a 2.5 m high door.*

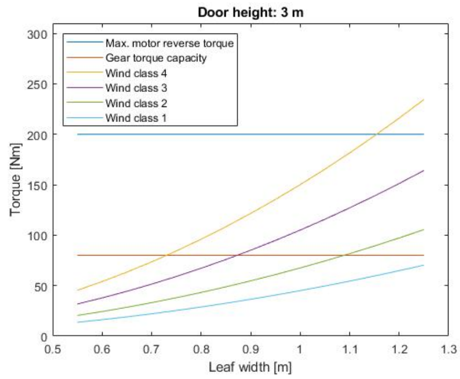

Figure 6.2: *Correlation between leaf width and torque required to resist different wind loads for a 3 m high door.*

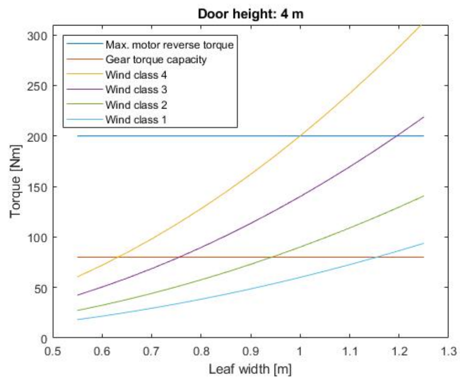

Figure 6.3: *Correlation between leaf width and torque required to resist different wind loads for a 4 m high door.*

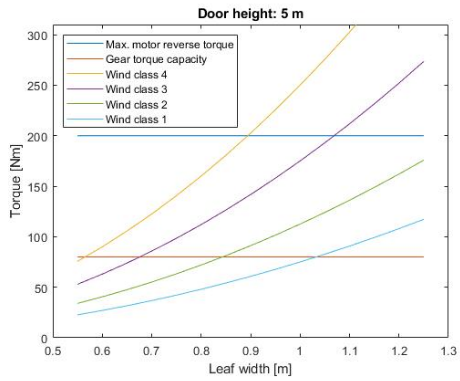

Figure 6.4: *Correlation between leaf width and torque required to resist different wind loads for a 5 m high door.*

6.2.4 Evaluation of Specification List

Specifications comparison					
	ASSA ABLOY		Refined concept		
Parameter	**Demand**	**Remarks**	**Results**	**Remarks**	**Future work**
Life time (years)	10	Acc. to EN12605, same as operator	Not tested		YES
Life time (cycles)	100000	Acc. to EN12605	Not tested		YES
Wind load	Class 3	Acc. to EN12424 Min. class 3, for 5x5 door	Not fulfilled	See discussion 6.2.3	YES
Min. width (mm)	1400	LW min. 550 mm	1400		
Max. width (mm)	5000	LW max. 1250 mm	5000		
Min. height (mm)	2000		2000		
Max. height (mm)	6000		6000		
Side space, both sides (mm)	200	Max	<200		
Headroom (mm)	200	Max	200		
Installation	Inside and outside		YES		
Opening	Inside and outside		YES		
Opening speed		Twice as today	30% faster	This was done @21.5 rpm	Could be improved further
Burglar protection SK2/SK3	TBD		NO		YES
Designed for service	YES		YES	Few rack variations	
DoC	YES	Declaration of Conformity	Not done		YES
DoP	YES	Declaration of Performance	Not done		YES
Installation time	Reduced by 50%	Compared to FD2250P operated	Not tested		YES

Table 6.2: *Specification comparison between ASSA ABLOY's specification demands and the achieved result from our concept.*

As can be seen in table 6.2 the life time regarding both years and cycles have not been tested and is considered future work. It looks more promising after the refinements done since the weight has been reduced.
ASSA ABLOY's demand for a class 3 wind load resistance was not fulfilled. With the calculations done on the new drive system it is verified that this is not possible for a 5x5 m door due to the gear not being able to handle the torque required. As discussed in chapter 6.2.3, there are room for improvements such as using a manufactured gear, changing teeth modules and perform torque capacity testing since the conclusions drawn in the calculation model are purely theoretical and during motion.
In chapter 6.2.2 it is shown that five circular rack variants are enough to cover all available ASSA ABLOY folding door sizes. If a "hockey" rack design is used, side space will increase but only one rack variant will be required to cover all of the same folding door sizes. This design is desired when there is enough side space and cross traffic is present. All dimensions regarding width, height, side space and headroom are fulfilled and even in some cases zero. The side space is set to <200 due to the drive systems adjustable rack design.

There has not been any problem with the outside and inside installation and opening. It is assumed that if opening and installation works on one of the alternatives the other one works as well. Outside installations will require stainless or galvanized steel to avoid corrosion.
The opening speed measured in chapter 5.6 was 30% faster than the current drive system. The opening speed was measured at a motor speed of 21.5 rpm. It is possible to run the motor at higher speeds, as described in chapter 3.1.1. Since the current consumption tested in chapter 5.4 is lower than ASSA ABLOY's current solution, there is room to increase motor speed (and opening speed) further.
It has not been thought of a potential burglar protection at this stage of the project and is at this point considered future work. The refined drive system has been designed with serviceability in mind which refers to having as few required welding

points as possible, minimum number of components and few rack variants. Declaration of Performance and Declaration of Conformity was not done due to time restrictions. Since the refined design will not be built during this project, no conclusions regarding installation time can be made.

7. Conclusion

The main focus and goal of this project was to conduct a concept study of a new drive system for industrial folding doors with the aim of improving, compared to the current drive system, reliability, robustness and cost. This was done following the product development process in Ulrich and Eppinger's book "Product design and development". The project resulted in a new drive system design and a thorough competitor evaluation. The drive system developed has potential in becoming a future product and deserves to be further pursued. With a complete cost analysis and further testing and refinements it is believed to fulfill all project objectives and desired product specifications. In future endeavours this project will be of great value to ASSA ABLOY.

7.1 Future Work

For future work the refinements mentioned in chapter 5.7 should be evaluated and tested. Also, the proposed "hockey" rack and the larger module 6 gear discussed in chapter 5.7 and 6.2.3 respectively should be looked into further. The refined design should be tested to see if it complies with the EN 12604 and EN 12453 standards. In order to be able to introduce the drive system on the market a DoP (Declaration of Performance), DoC (Declaration of Conformity), instruction manual and a product sheet has to be made. It is also recommended to conduct a complete cost analysis regarding installations on 2+2 folding doors. Currently, it is not sure that there will be economical benefits switching to the new drive system in these cases. Burglar protection is another aspect of the drive system solution that has not been looked into in this project and will have to be investigated if the project continues. Staying up to date with the competitors solutions and products on the market will be beneficial for ASSA ABLOY. As the concept is developing and improving, there is a possibility that a solution solving many of the problems mentioned in this report will emerge.

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Appendices

A. Additional Testing

In this Appendix, additional tests on a four fold are presented. The tests include current consumption, calculated torque and crushing forces. This door has a 2+2 configuration and measures 3.6x3 m. The door can be seen in figure A.1.

Figure A.1: *The 2+2 door additional tests were performed on.*

A.1 Current Consumption

The motor current is shown in figure A.2 and torque is shown in figure A.3 during the entire opening cycle. The corresponding figure for the closing cycle are A.4 and A.5.

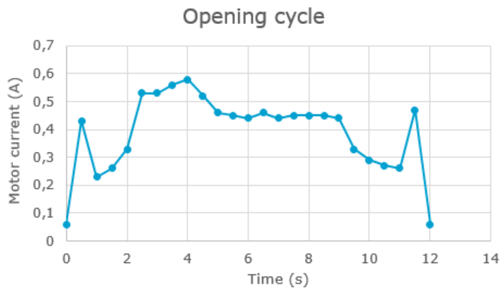

Figure A.2: The power consumption of the motor during opening.

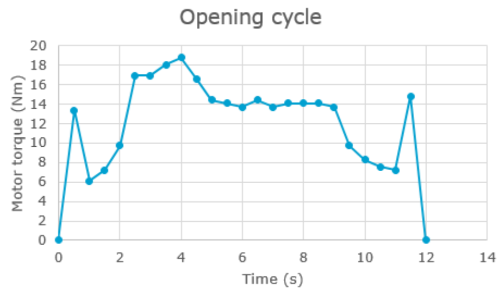

Figure A.3: The motor torque during opening.

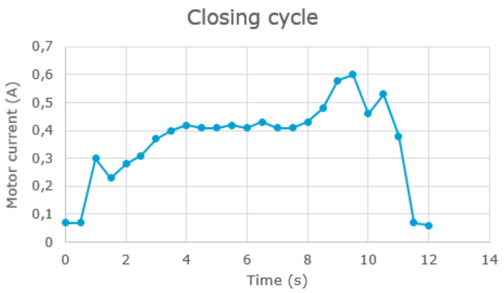

Figure A.4: The power consumption of the motor during closing.

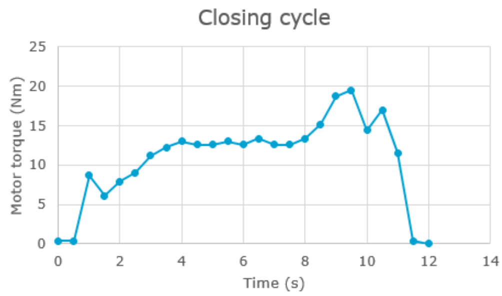

Figure A.5: The motor torque during closing.

A.2 Crushing Force

The measured crushing forces according to the EN 12453 standard are presented in table A.1 below. It was not possible to measure the crushing forces against neighboring stiff parts because there were no walls or other rigid objects nearby in the test lab.

- **Door: FD2250P**
- **Control unit: ECS 950**
- **Machinery: CDM9FD**
- **Sensor: Bircher DW 40 Pneumatic**

Crushing force test				
Height from floor [mm]	**Opening gap [mm]**	**Crushing force [N]**	**Time while force >150 N [ms]**	**Unload within 5 s**
50	50	124	0	YES
50	300	190	122	YES
50	500	206	117	YES
1500	50	134	0	YES
1500	300	215	144	YES
1500	500	177	124	YES
2500	50	155	77	YES
2500	300	193	134	YES
2500	500	170	137	YES

Table A.1: *Results from the EN 12453 test conducted on ASSA ABLOY's current solution with mechanical rail. Crushing force and time while force >150 N are averages from 3 measured values. The maximum allowed force between closing edges is 400 N within a period of maximum 0.75 s and unload (reverse) within 5 s.*

B. Modules

In this chapter the table used when calculating the required gear teeth module is presented.

B.1 Charts

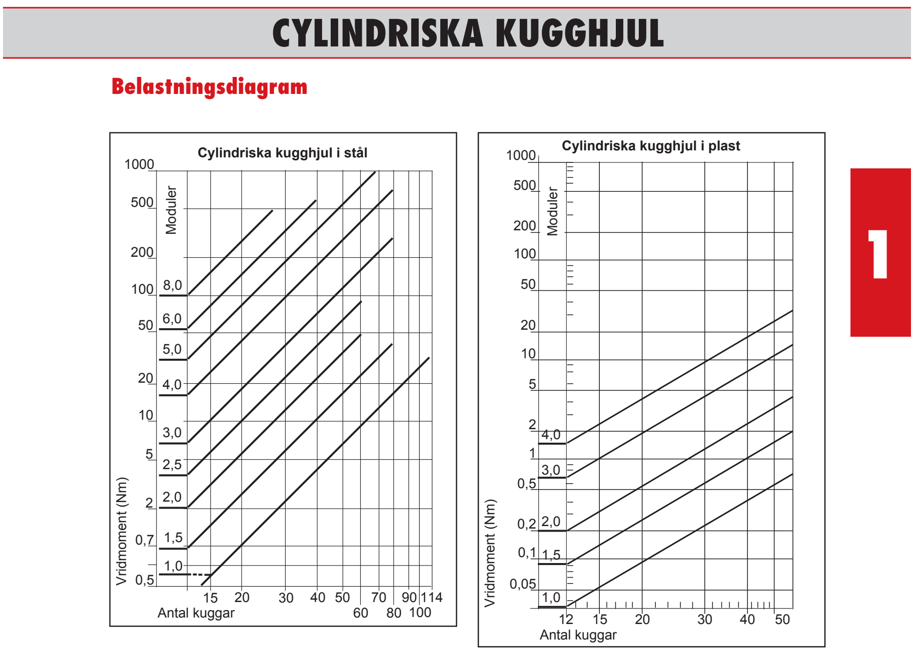

Figure B.1: Charts describing the correlation between no. teeth, module and torque capacity of gears. The torque capacity values are calculated for gears moving at 200 rpm. Figure from [44].

C. Calculation Models

C.1 Calculation Model for Circular Rack

In this chapter the approximative calculation model created in MATLAB and used for designing the circular rack is presented. The folding door approximation model can be seen in figure C.1.

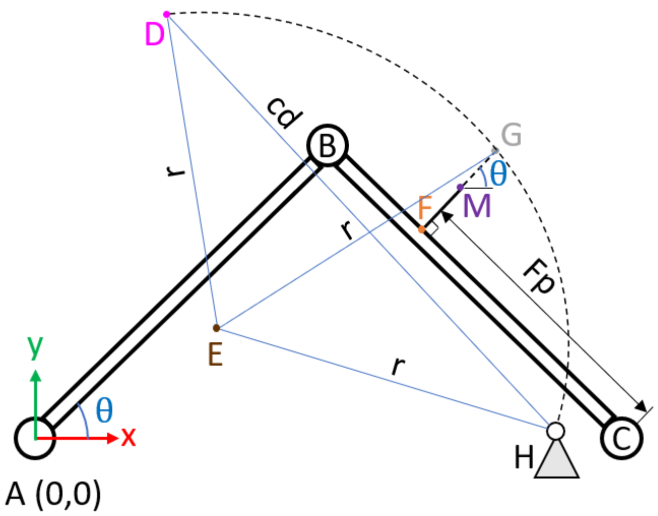

Figure C.1: *Illustration of the variables used in the model.*

```
clc
clear all
close all


%INPUTS
motor_depth=0.150; %Measured in CAD
gear_radius=0.1/2; %from excel
Fp=1;  %Fixpoint motor. DLW=0.9-->Fp=0.7, DLW=1.25-->Fp=1
radius_rack=1.05; %r in figure
L=1.25; %DLW (DOOR LEAF WIDTH)
```

```
theta=1:1:95; %Opening angle door

Hx=1.55; Hy=0.126;  %DLW=0.9-->: Hx=1.15; Hy=0.126; DLW=1.25-->: Hx=1.55; Hy=0.126;

%Door hinge coordinates during opening/closing (Simplified).
Ax=0; Ay=0;
Bx=L*cosd(theta); By=L*sind(theta);
Cx=2*L*cosd(theta); Cy=0;

%Motor fixture point on inner door leaf during opening/closing.
Fx=Cx-Fp*cosd(theta);
Fy=Fp*sind(theta);

%Motor shaft coordinates during opening/closing.
Mx=Fx+motor_depth*cosd(90-theta);
My=Fy+motor_depth*sind(90-theta);

%Gear engagement point (Simplified).
Gx=Mx+gear_radius*cosd(theta);
Gy=My+gear_radius*sind(theta);

%E denotes center of circular rack and D denotes endpoint of rack.
Ex=zeros(1,95);
Ey=zeros(1,95);
Dx=zeros(1,95);
Dy=zeros(1,95);

%Calculation of some rack parameters.
rack_closest_distance=sqrt((Hx-Gx(95))^2+(Hy-Gy(95))^2); %cd in figure
rack_angle=acosd((2*radius_rack^2-rack_closest_distance^2)/(2*radius_rack^2));
rack_angle_secure=rack_angle*1.1;
rack_length=deg2rad(rack_angle)*radius_rack;
rack_length_secure=rack_length*1.1;

for i=1:1:95
[Eintersect_x,Eintersect_y]=circcirc(Hx,Hy,radius_rack,Gx(i),Gy(i),radius_rack);
 %selection of real intersection point.
if Eintersect_x(2)>Eintersect_x(1)
    Ex(i)=Eintersect_x(1);
    Ey(i)=Eintersect_y(1);
```

```
else
    Ex(i)=Eintersect_x(2);
    Ey(i)=Eintersect_y(2);
end

end

for j=1:1:95
[Dintersect_x,Dintersect_y]=circcirc(Hx,Hy,rack_closest_distance,Ex(j),Ey(j),radius_rack);
 %selection of real intersection point.
 if Dintersect_y(2)>Dintersect_y(1)
     Dx(j)=Dintersect_x(2);
     Dy(j)=Dintersect_y(2);
 else
     Dx(j)=Dintersect_x(1);
     Dy(j)=Dintersect_y(1);
 end

end

 plot(theta,Ey);
rack_max_side=-min(Dx); %Maximum side space requirement during opening/closing
rack_max_depth=max(Dy); %Maximum depth space requirement during opening/closing
```

C.2 Calculation Model for Wind Load

```
clear all
close all
clc

L=0.55:0.05:1.25;   %Leaf width
H=5;     %Door height
P=[300,450,700,1000];    %Wind class 1: 300Pa, Wind class 2: 450Pa,
                         %Wind class 3: 700Pa, Wind Class 4: 1000Pa
x=1;     %Motor placement from closing edge
d=0.1;   %Gear pitch diameter

Tmax_gear=80*ones(1,length(L)); %Gear torque capacity (80Nm)
Tmax_motor=200*ones(1,length(L));    %Max. motor reverse torque (200Nm)
```

```

%——————————————OUTPUT————————————————
for i=1:length(P)
Fw(i,:)=L*H*P(i);
Fu=(Fw.*L)./x;
T=d/2*Fu;
end
plot(L,Tmax_motor,L,Tmax_gear,L,T(4,:),L,T(3,:),L,T(2,:),L,T(1,:))
title("Door height: "+H+" m");
legend('Max. motor reverse torque','Gear torque capacity','Wind class 4','Wind class 3','Wind class 2','Wind class 1')
legend('Location','northwest')
xlabel('Leaf width [m]')
ylabel('Torque [Nm]')
ylim([0,310])
```

C.3 Circular Rack Variants

Internal

	<0,2	<LW	<180			
Rack radius	Side space	Depth space	Angle	Fp=0,8*SW		Best variant
LW 1,25				Hx :	1,55	
1,1	0,04	1,33	120 deg			
1,05	0,07	1,24	130 deg			
1	0,11	1,12	140 deg			
0,95	0,15	1,04	155 deg			0,95
0,9	0,22	1,04	181 deg			
LW 1,2				Hx:	1,5	
1,05	0,04	1,28	125 deg			
1	0,07	1,19	130 deg			
0,95	0,1	1,06	145 deg			0,95
0,9	0,15	0,99	160 deg			
LW 1,15				Hx :	1,45	
1	0,04	1,23	125 deg			
0,95	0,06	1,13	135 deg			0,95
0,9	0,1	1	145 deg			
0,85	0,15	0,94	165 deg			
LW 1,1				Hx	1,35	
0,95	0,01	1,19	120 deg			0,95
0,9	0,04	1,1	135 deg			(High depth)
0,85	0,08	0,97	145 deg			
0,8	0,14	0,94	170 deg			
LW 1,05				Hx	1,3	
0,9	0,01	1,14	125 deg			
0,85	0,04	1,04	135 deg			0,75
0,8	0,08	0,9	150 deg			
0,75	0,15	0,89	175 deg			
LW 1				Hx	1,25	
0,85	0,01	1,08	125 deg			
0,8	0,04	0,98	135 deg			0,75
0,75	0,08	0,84	155 deg			
LW 0,95				Hx	1,2	
0,8	0	1,03	125 deg			
0,75	0,03	0,92	140 deg			0,75
0,7	0,08	0,79	160 deg			
LW 0,9				Hx	1,15	
0,75	0	0,98	130 deg			
0,7	0,03	0,86	145 deg			0,75
0,65	0,08	0,74	170 deg			(High depth)
LW 0,85				Hx	1,05	
0,7	0	0,94	125 deg			
0,65	0,01	0,83	140 deg			0,6
0,6	0,07	0,74	170 deg			
LW 0,8				Hx	1	
0,65	0	0,88	130 deg			
0,6	0,01	0,77	145 deg			0,6
0,55	0,08	0,69	183 deg			
LW 0,75				Hx	0,95	
0,6	0	0,83	130 deg			
0,55	0,01	0,7	150 deg			0,6
0,5	Not possible					(High depth)
LW 0,7				Hx	0,9	
0,55	0	0,77	135 deg			
0,5	0,01	0,63	160 deg			0,5
0,45	Not possible					
LW 0,65				Hx	0,85	
0,5	0	0,71	135 deg			
0,45	0,01	0,54	170 deg			0,5
0,4	Not possible					(High depth)
LW 0,6				Hx	0,75	
0,5	0	0,75	115 deg			
0,45	0	0,68	135 deg			0,4
0,4	0	0,55	175 deg			
LW 0,55				Hx	0,7	
0,45	0	0,7	115 deg			0,4
0,4	0	0,65	140 deg			(High depth)
0,35	Not possible					

Figure C.2: *Correlation between leaf width, rack radius and space requirements.*

D. Drawings

D.1 Detail Drawings of Components

In this section the detailed drawings of all the included components in the final concepts design are presented.

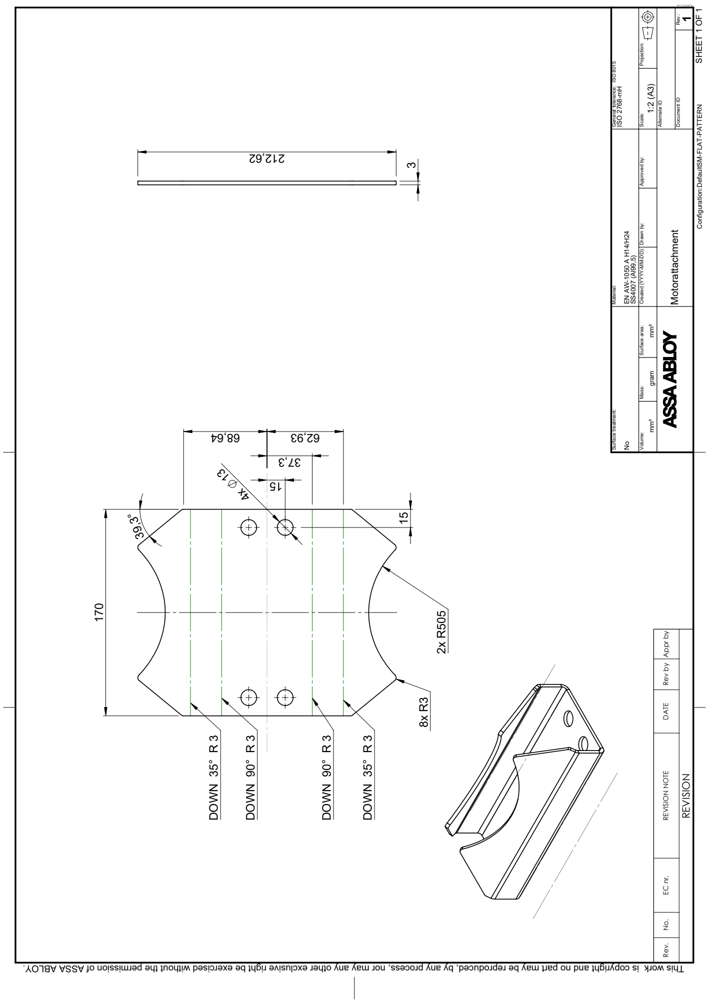

Figure D.1: *The final design of the motor attachment, featuring 2 bends on each side.*

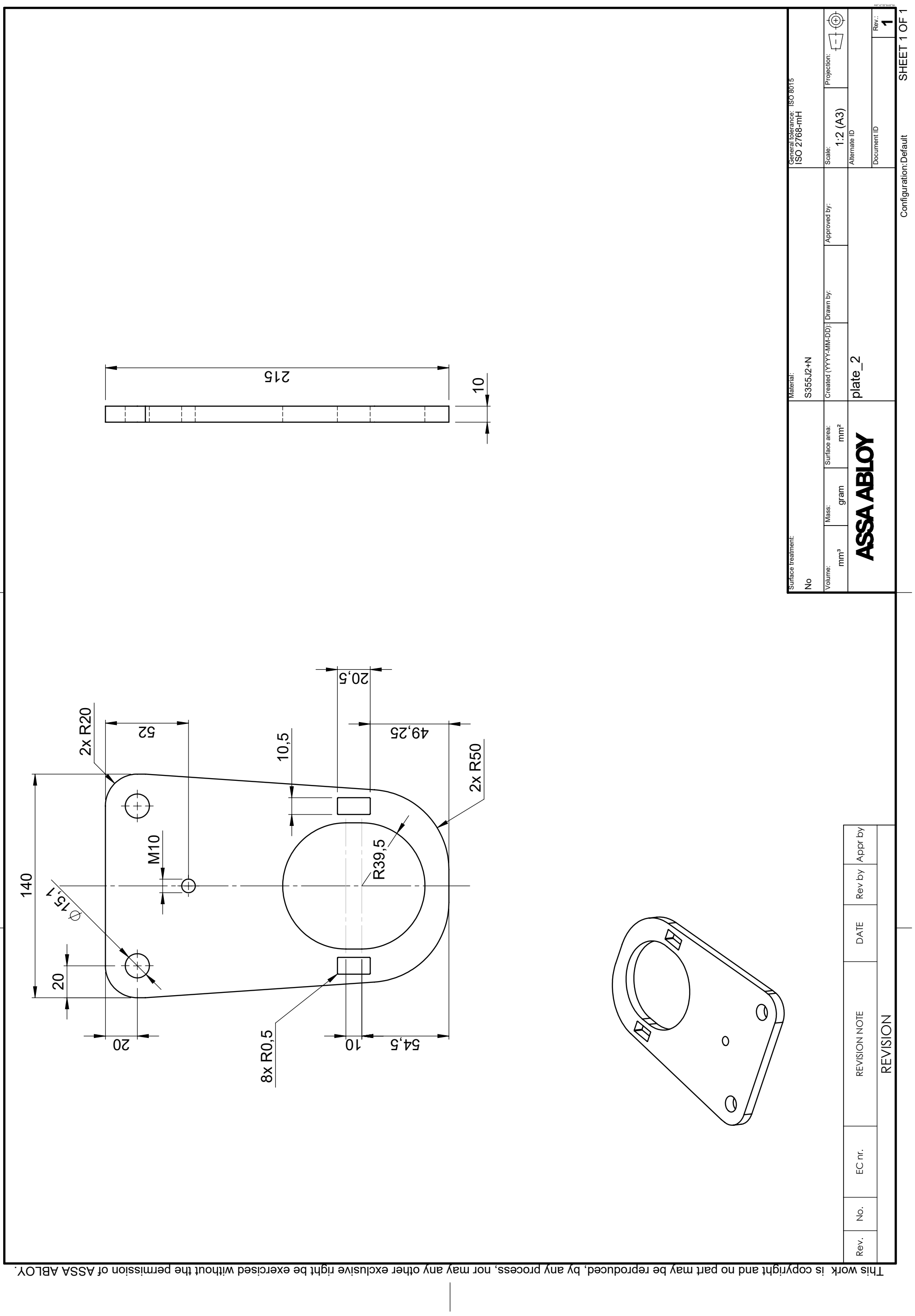

Figure D.2: *The final design of the plate.*

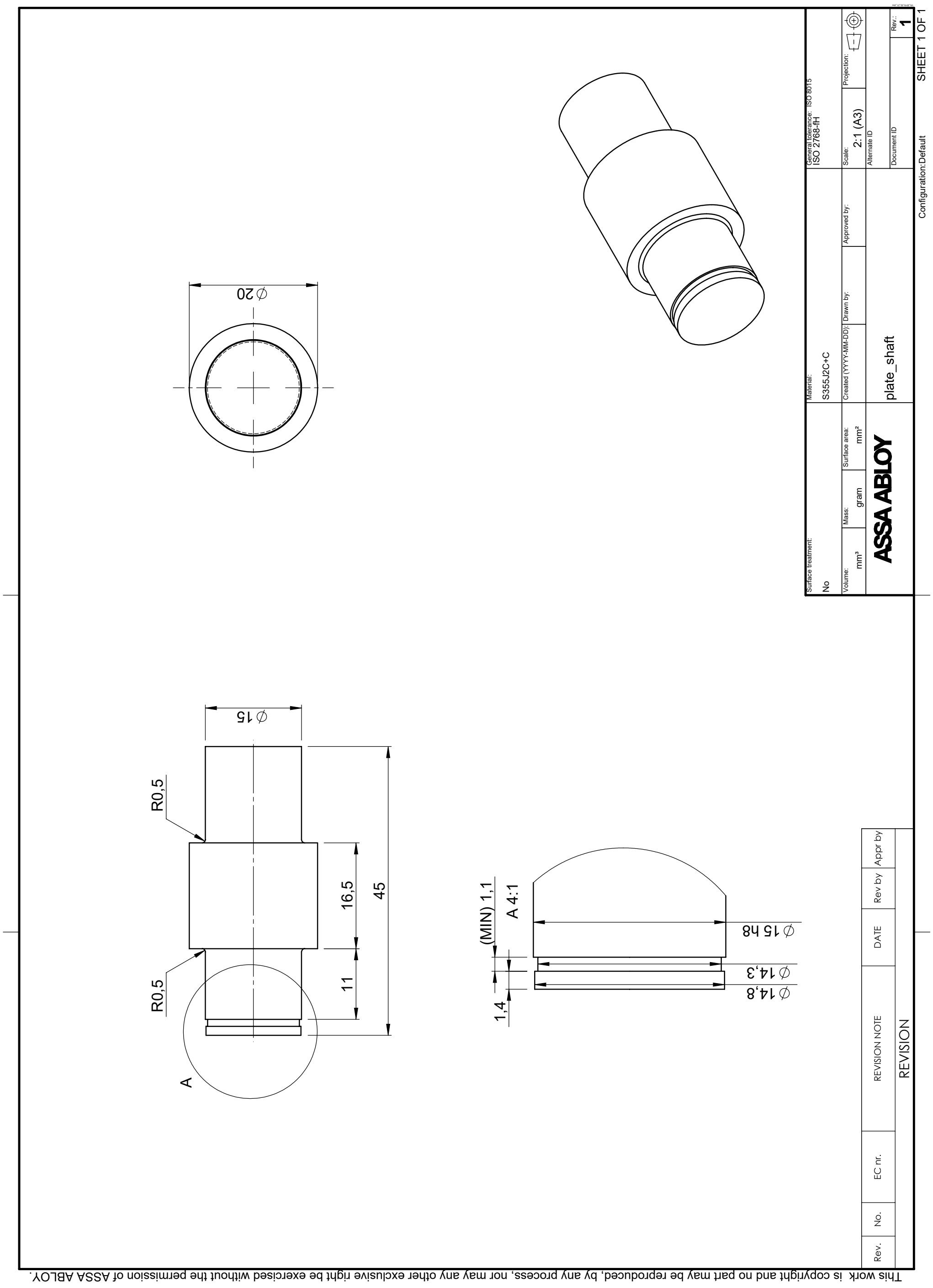

Figure D.3: *The final design of the plate shafts.*

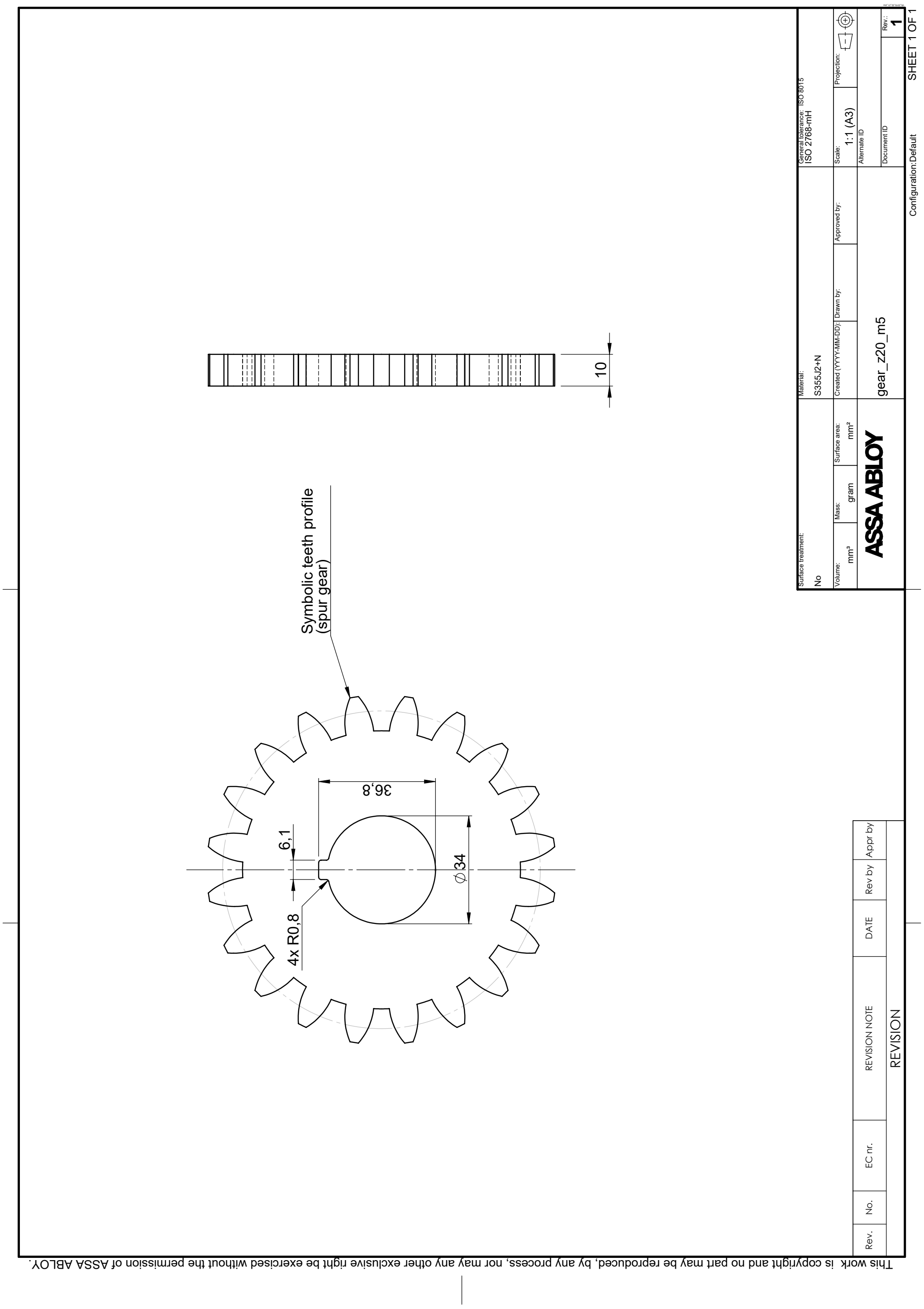

Figure D.4: *The final design of the gear.*

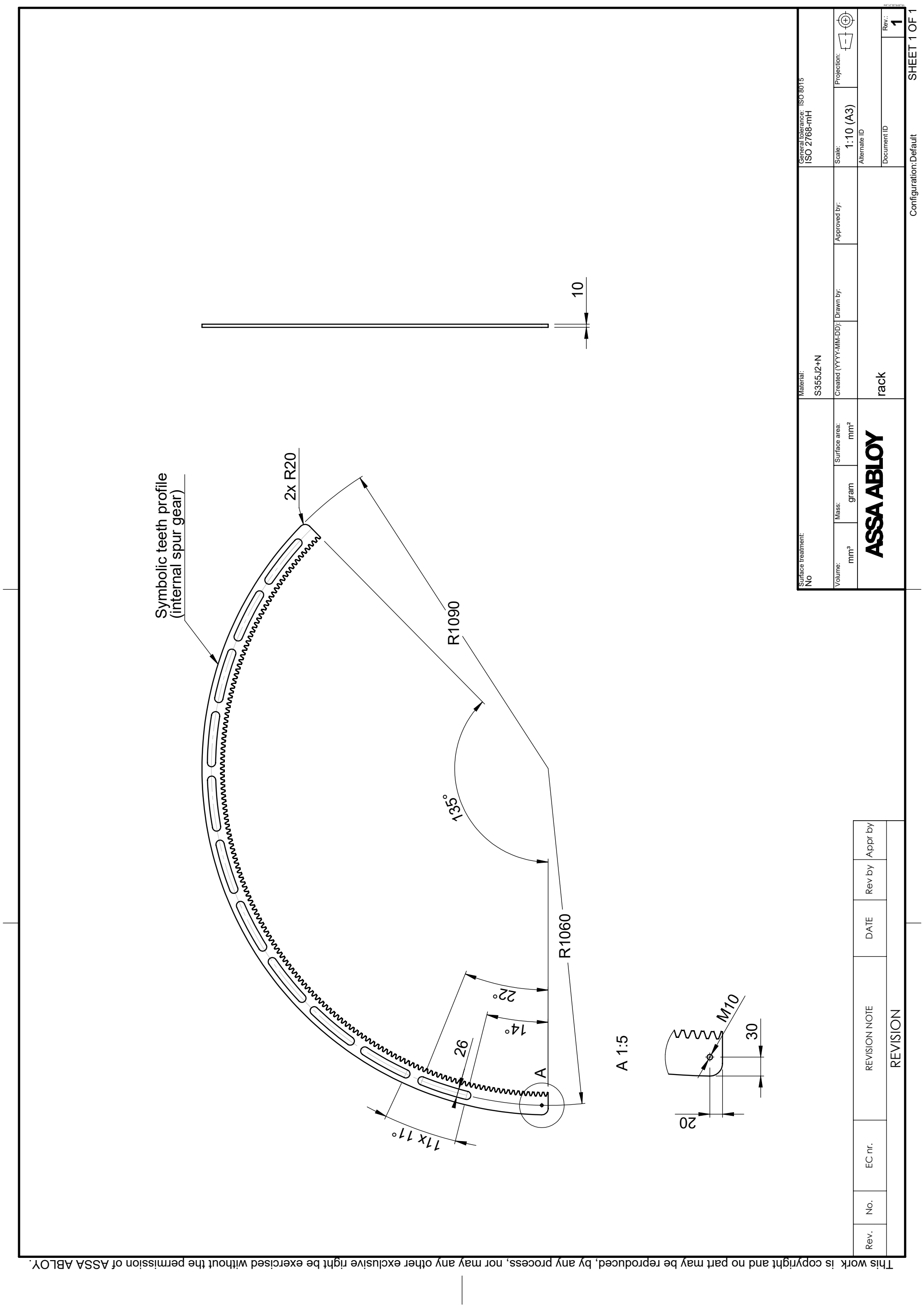

Figure D.5: *The final design of the circular rack.*

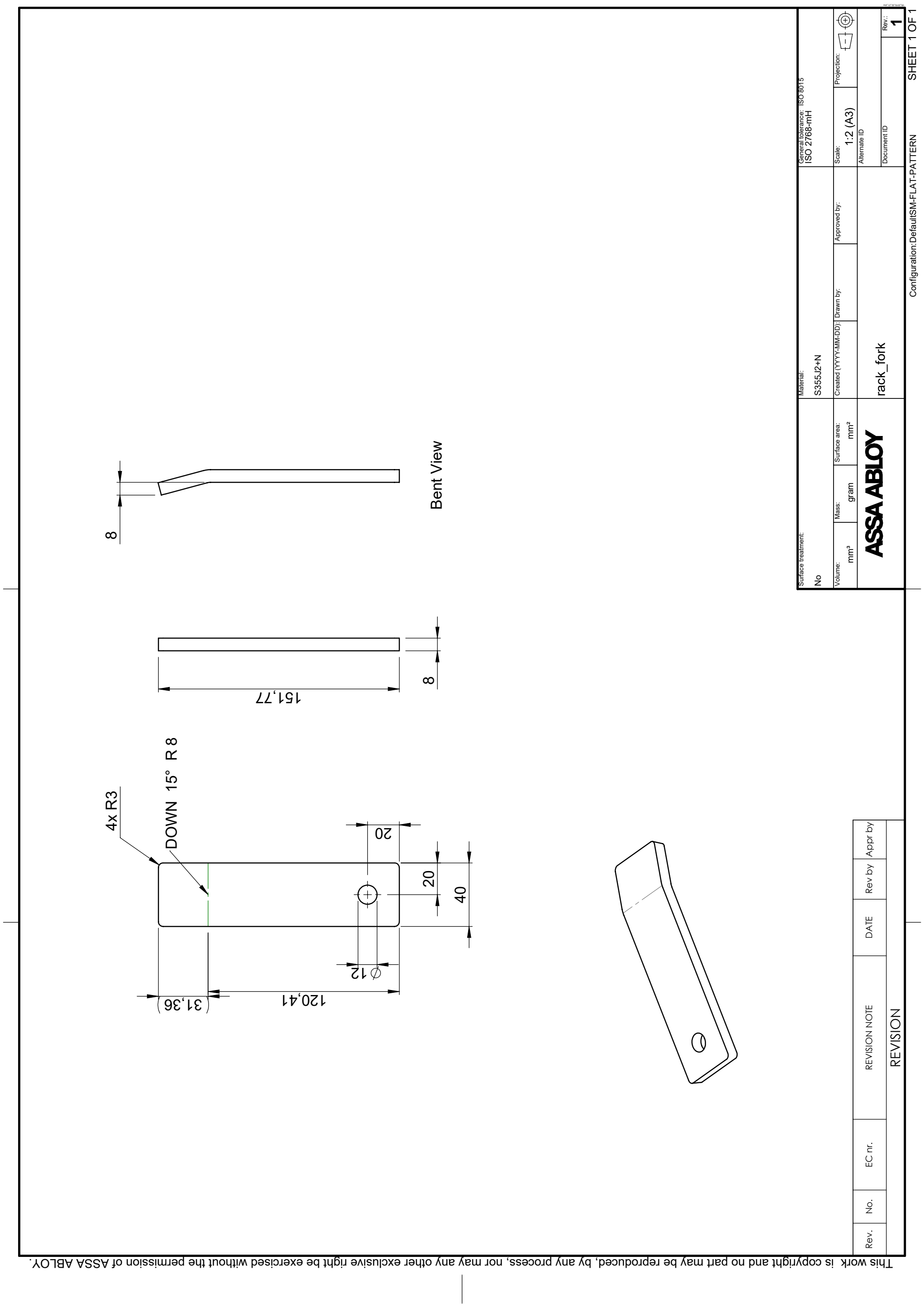

Figure D.6: *The final design of the fork.*

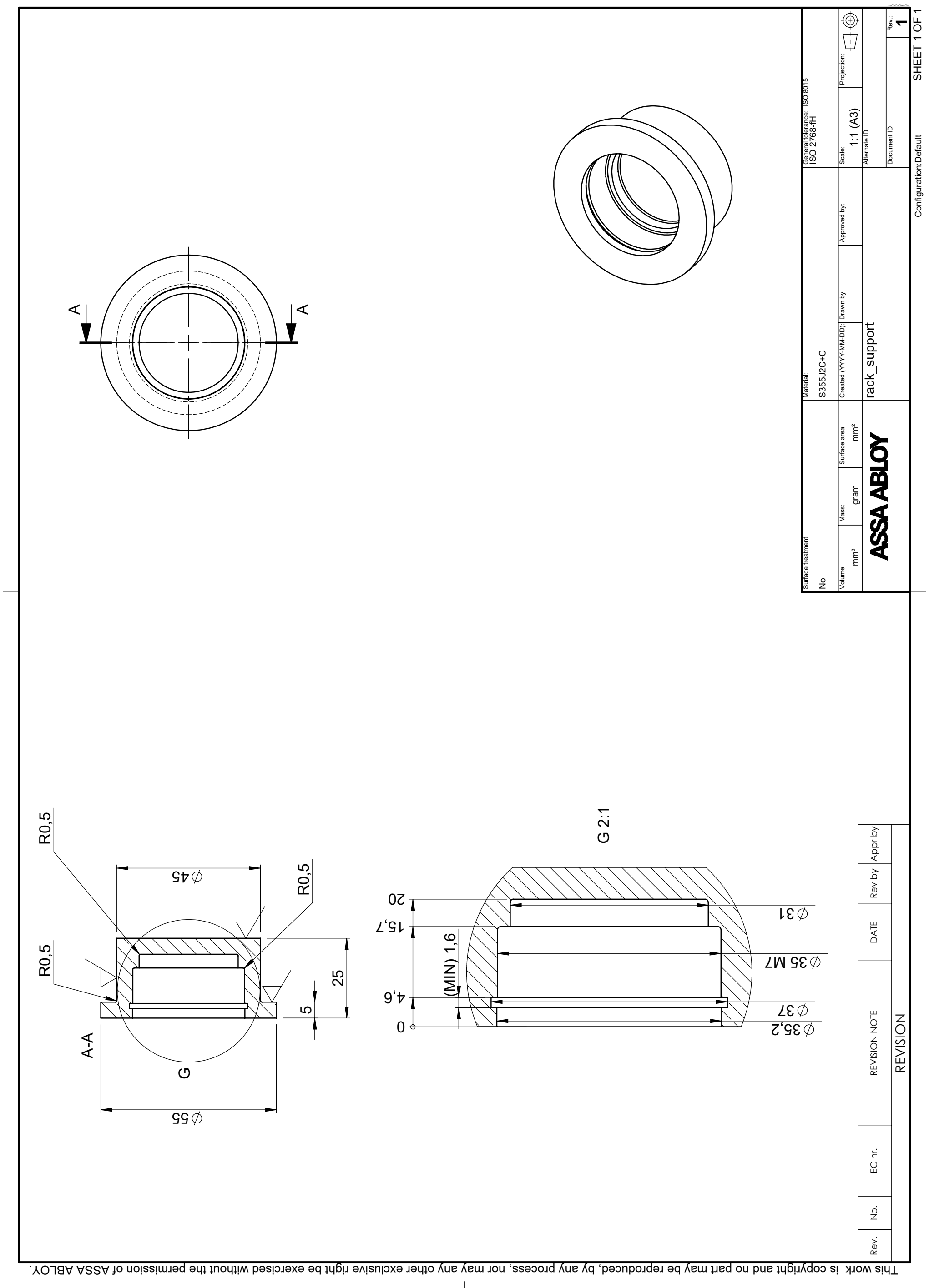

Figure D.7: *The final design of the rack supports.*

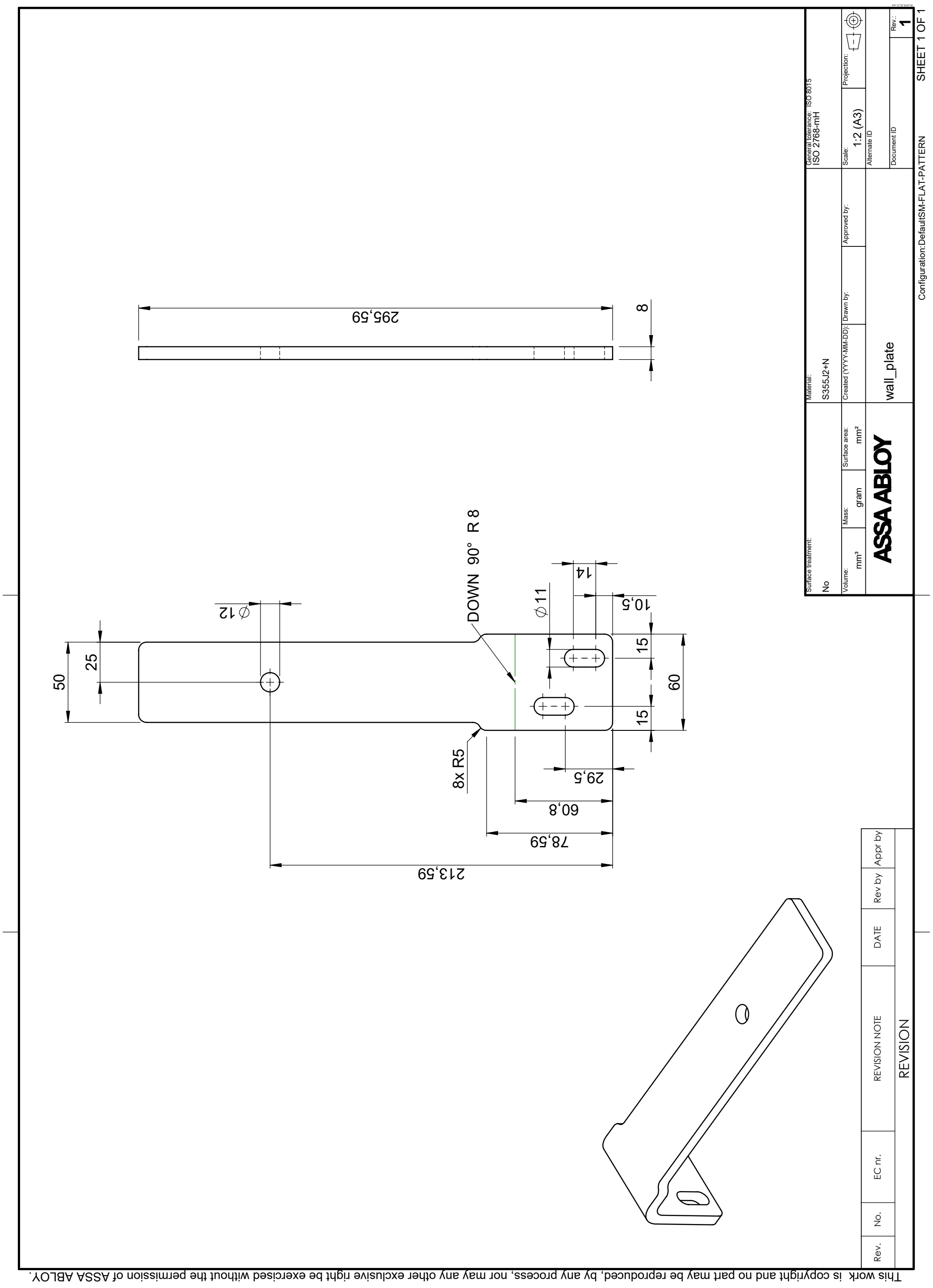

Figure D.8: *ASSA ABLOY's guide rail holder.*

D.2 Detail Drawings of Refined Components

In this section the detailed drawings of all the included components in the refined concepts design are presented.

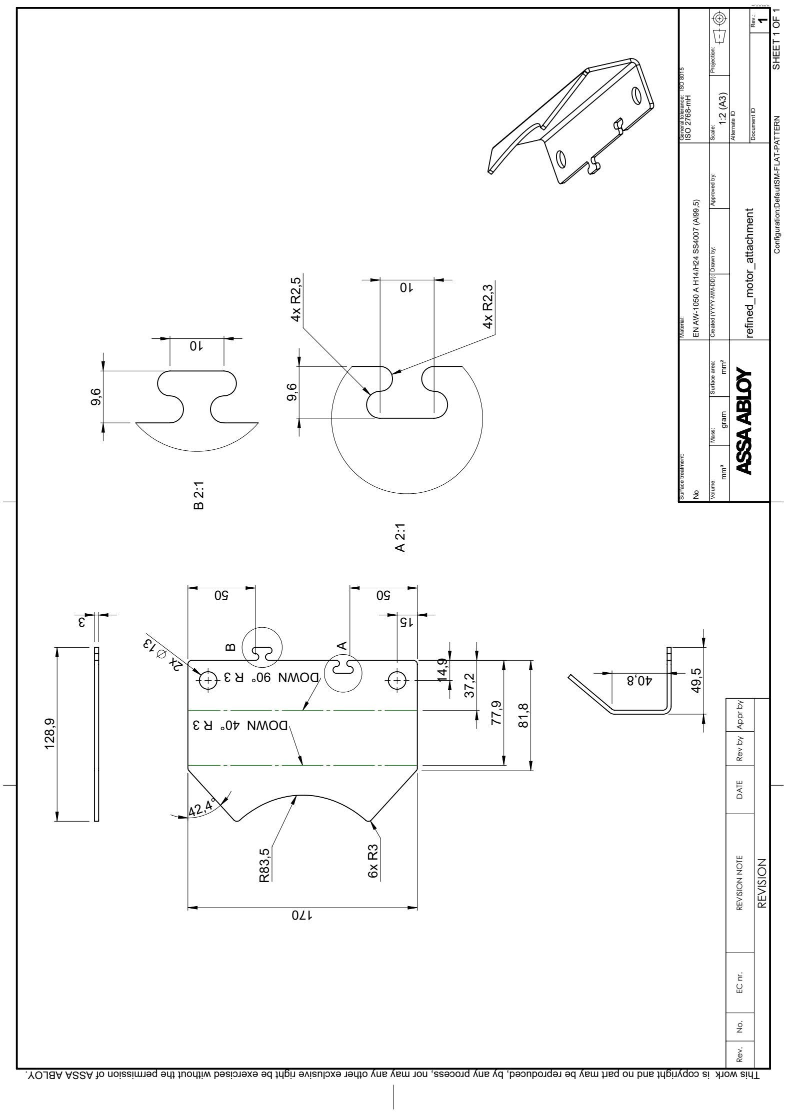

Figure D.9: *The final design of the refined motor attachment*

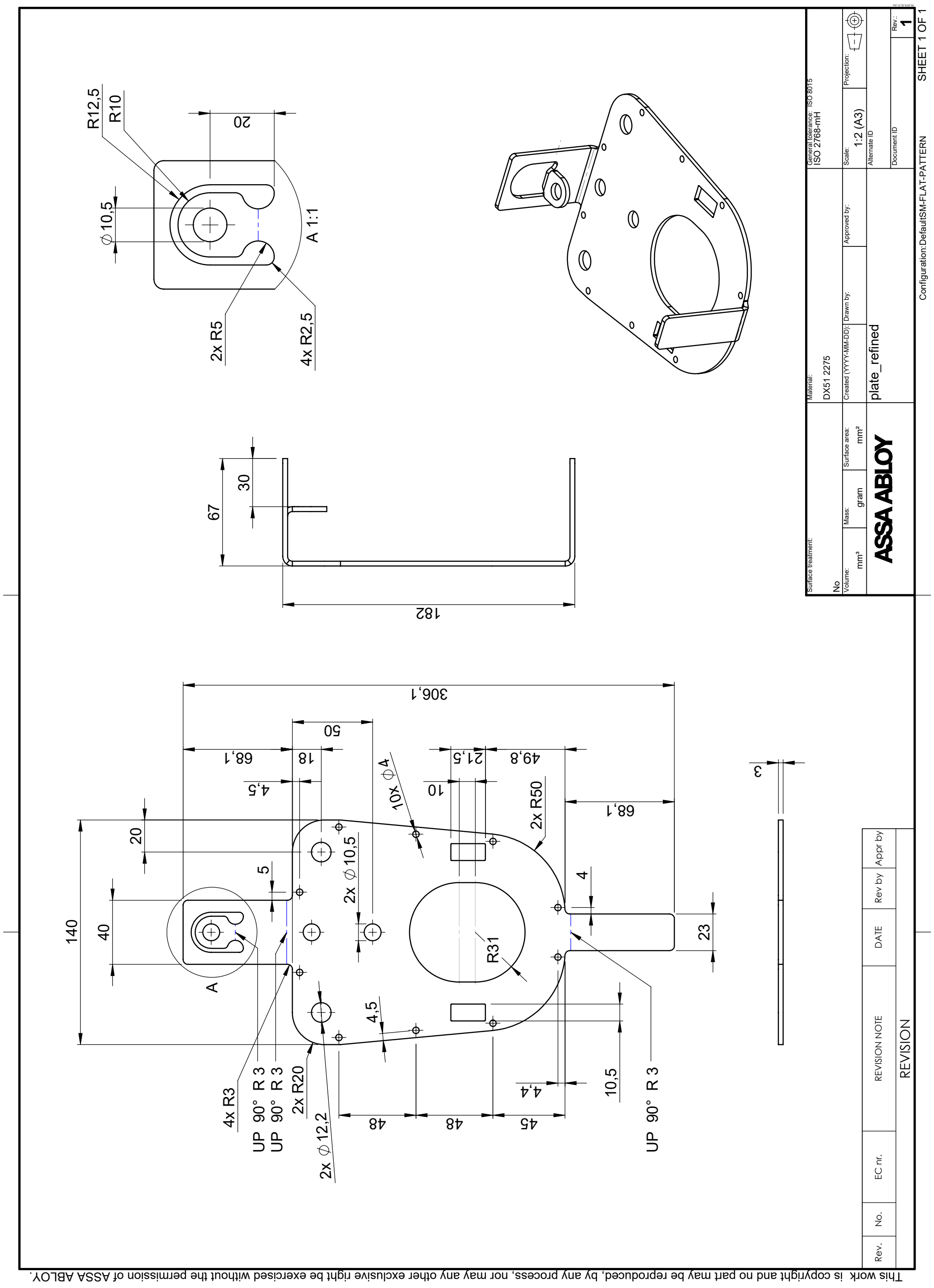

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Figure D.10: *The final design of the refined plate.*

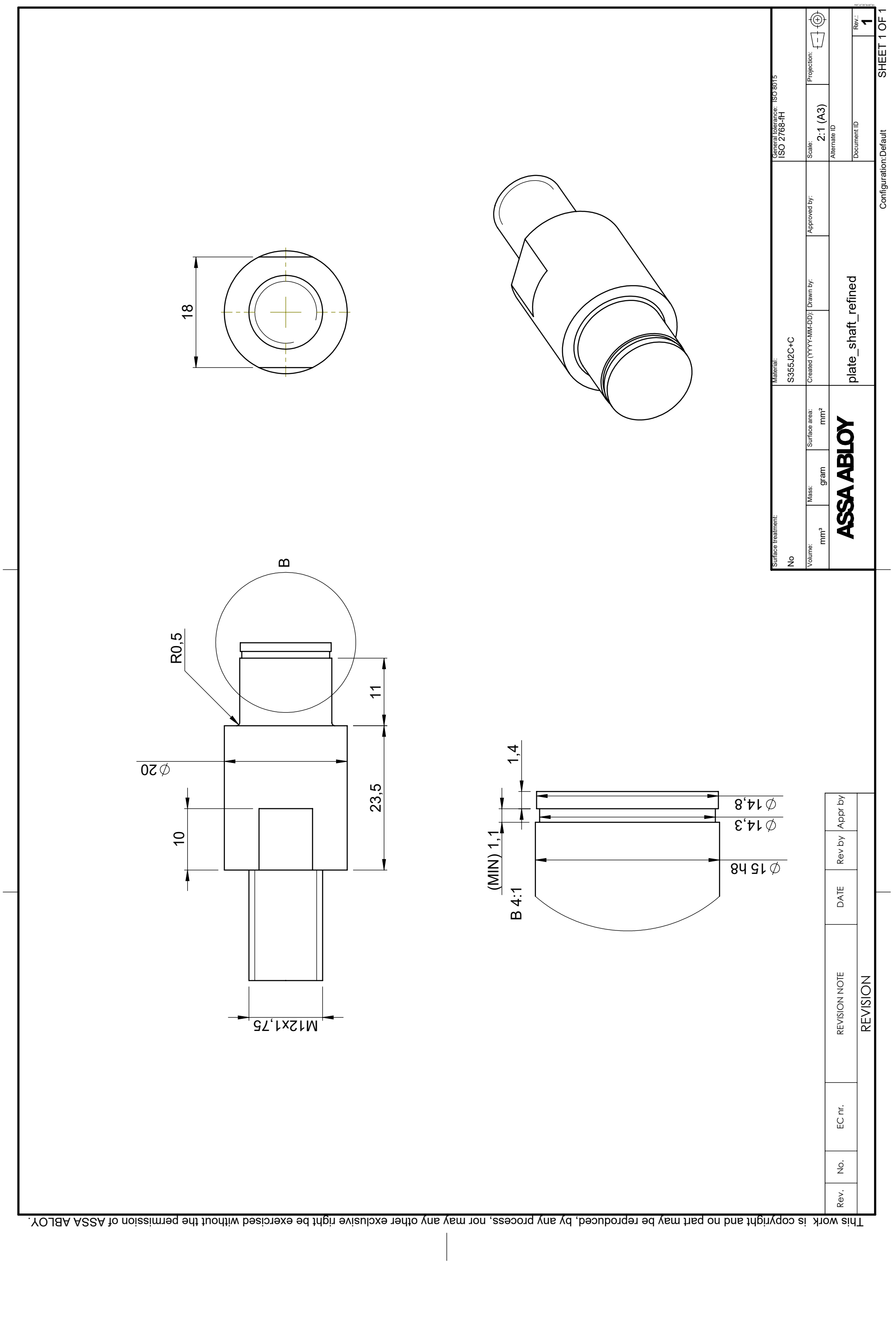

Figure D.11: *The final design of the refined plate shafts.*

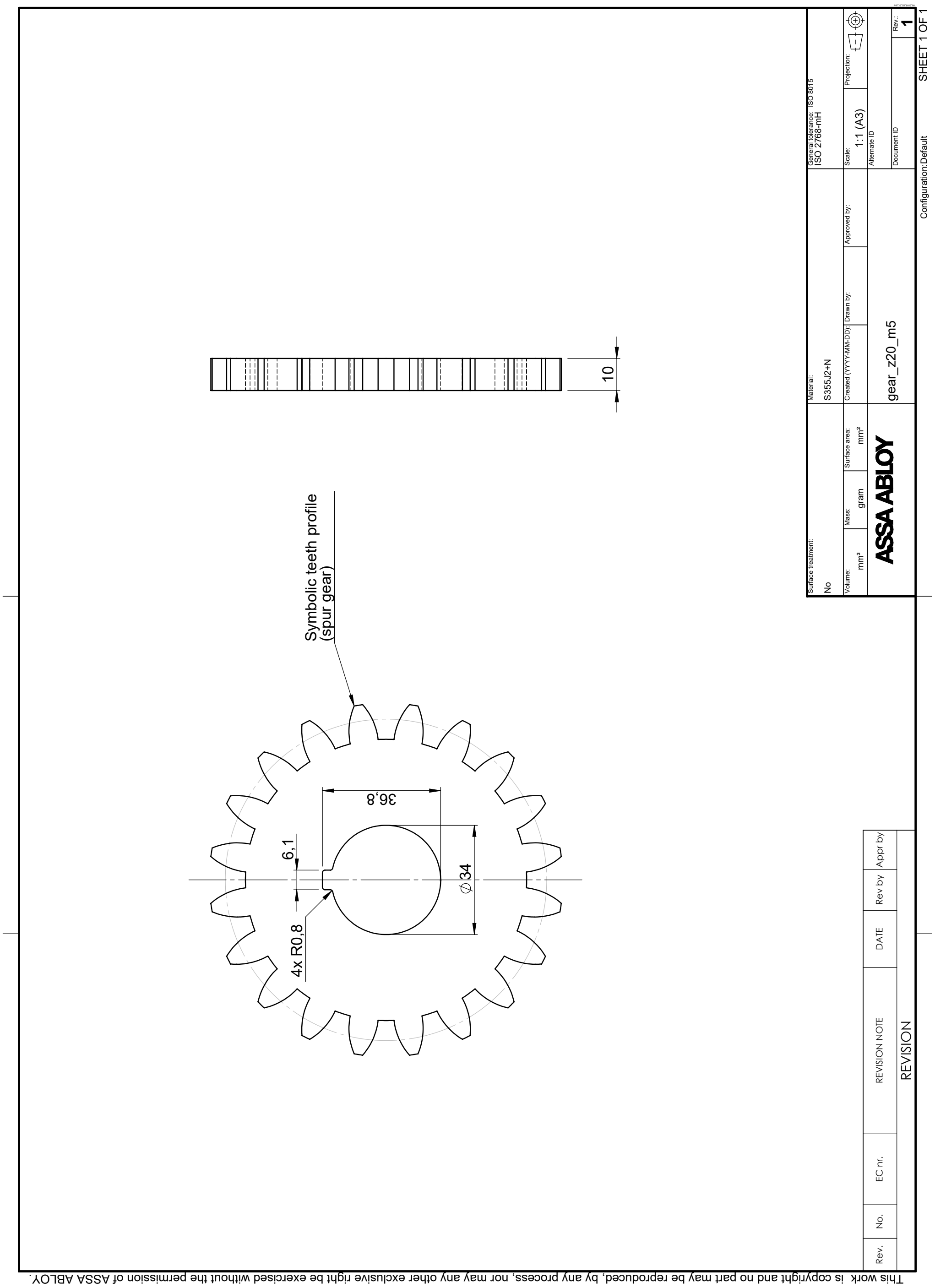

Figure D.12: *The final design of the refined gear.*

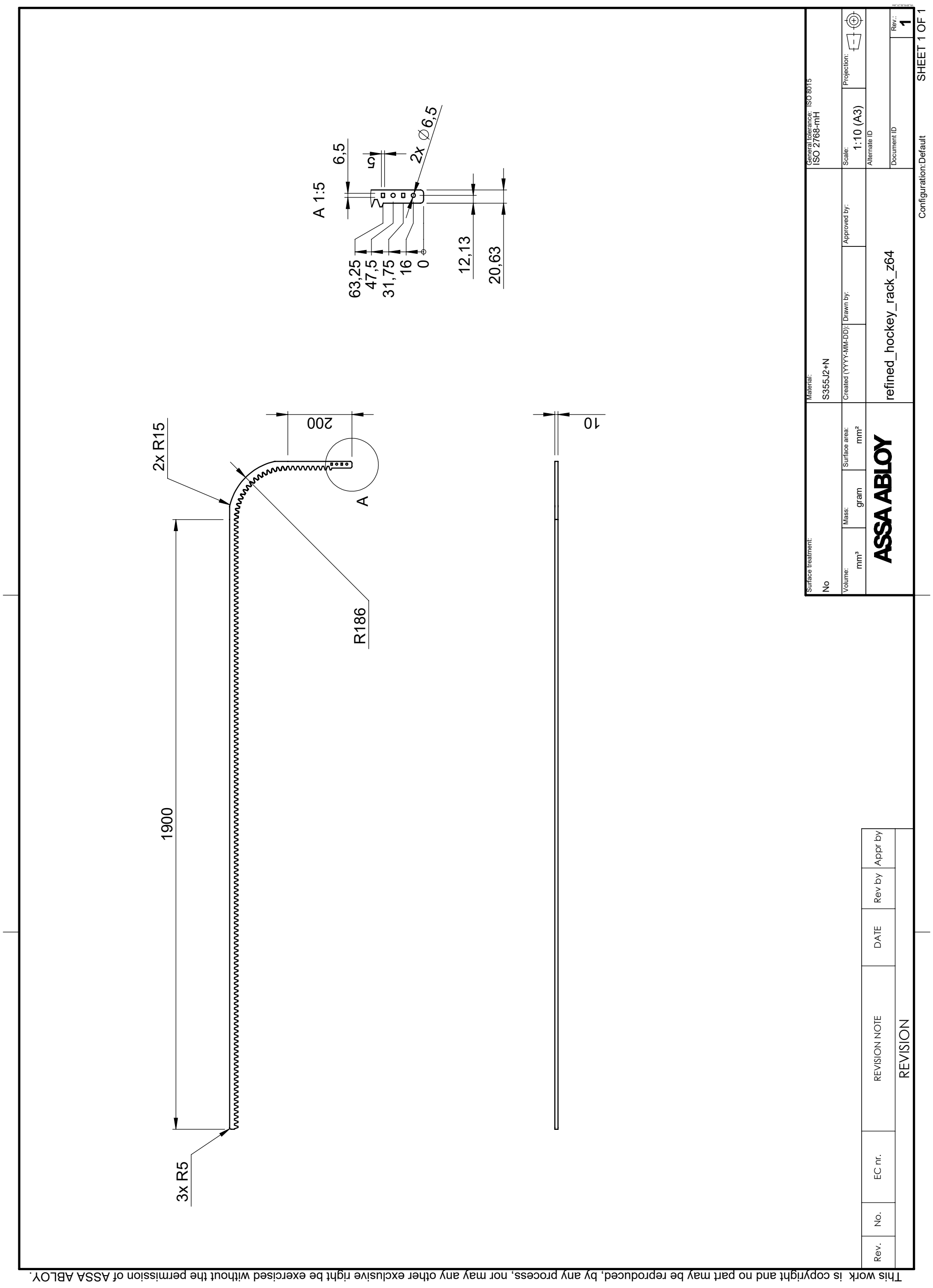

Figure D.13: *The final refined design of the proposed "hockey" rack.*

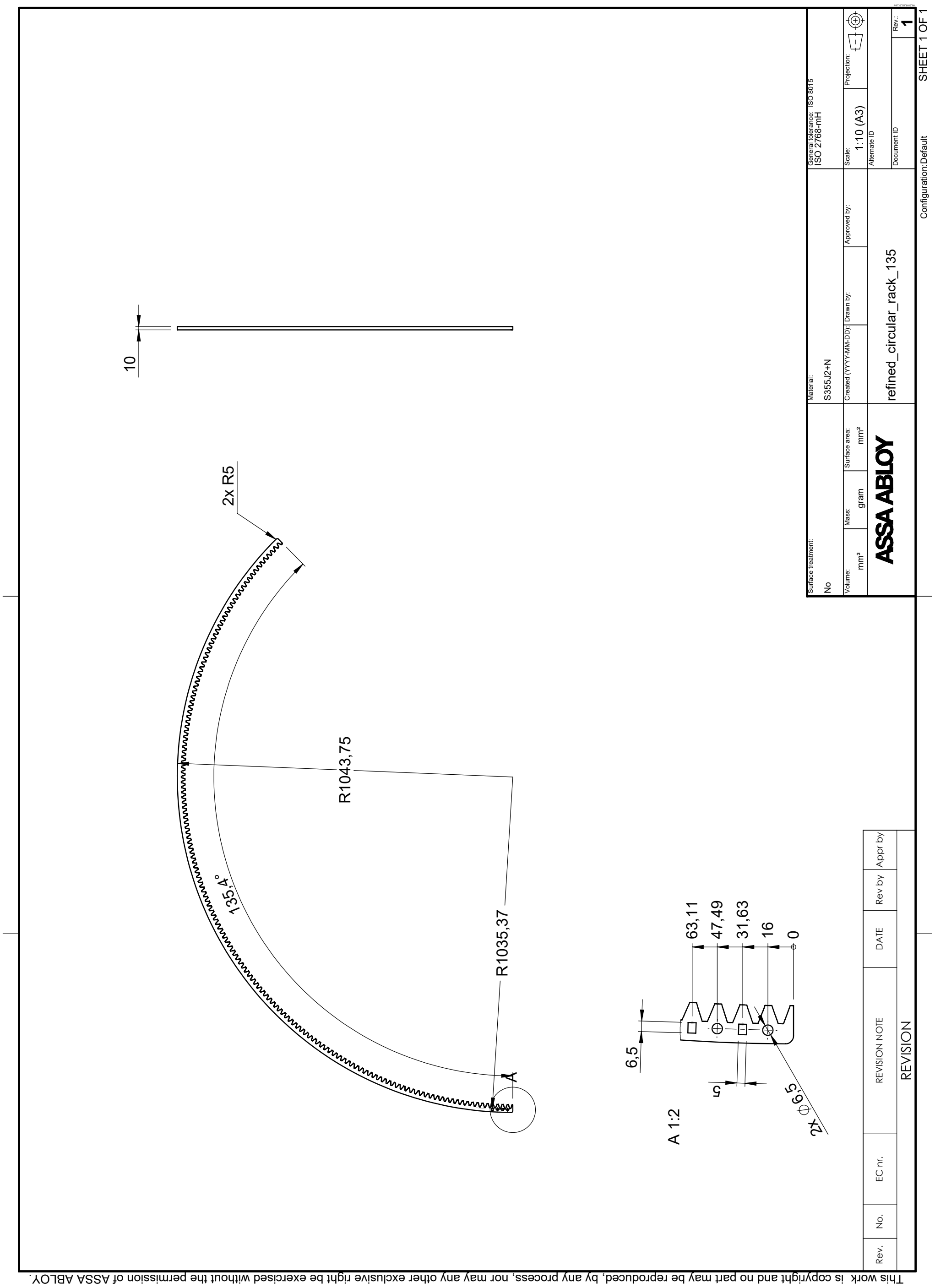

Figure D.14: *The final design of the refined circular rack.*

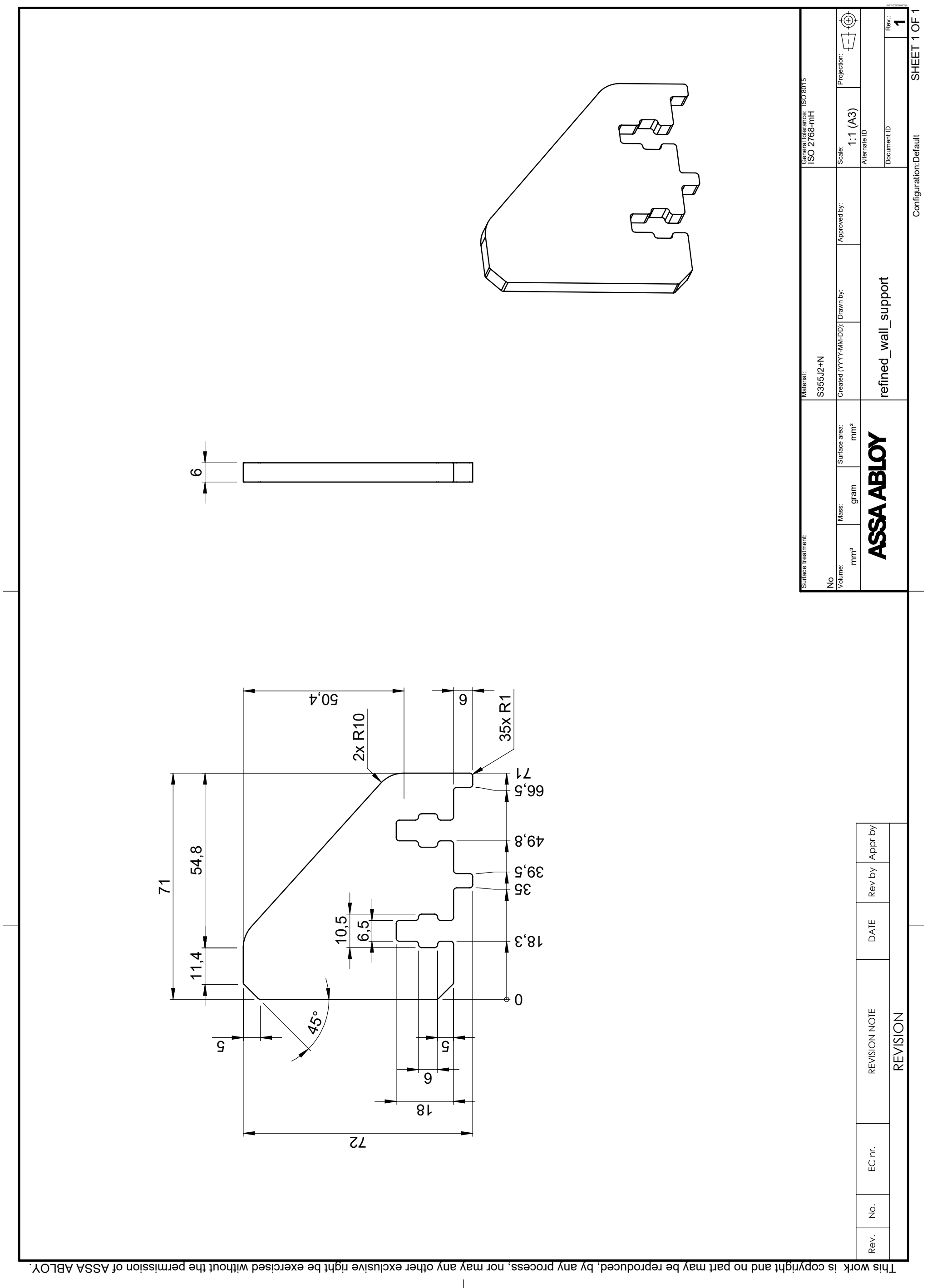

Figure D.15: The final design of the refined wall support.

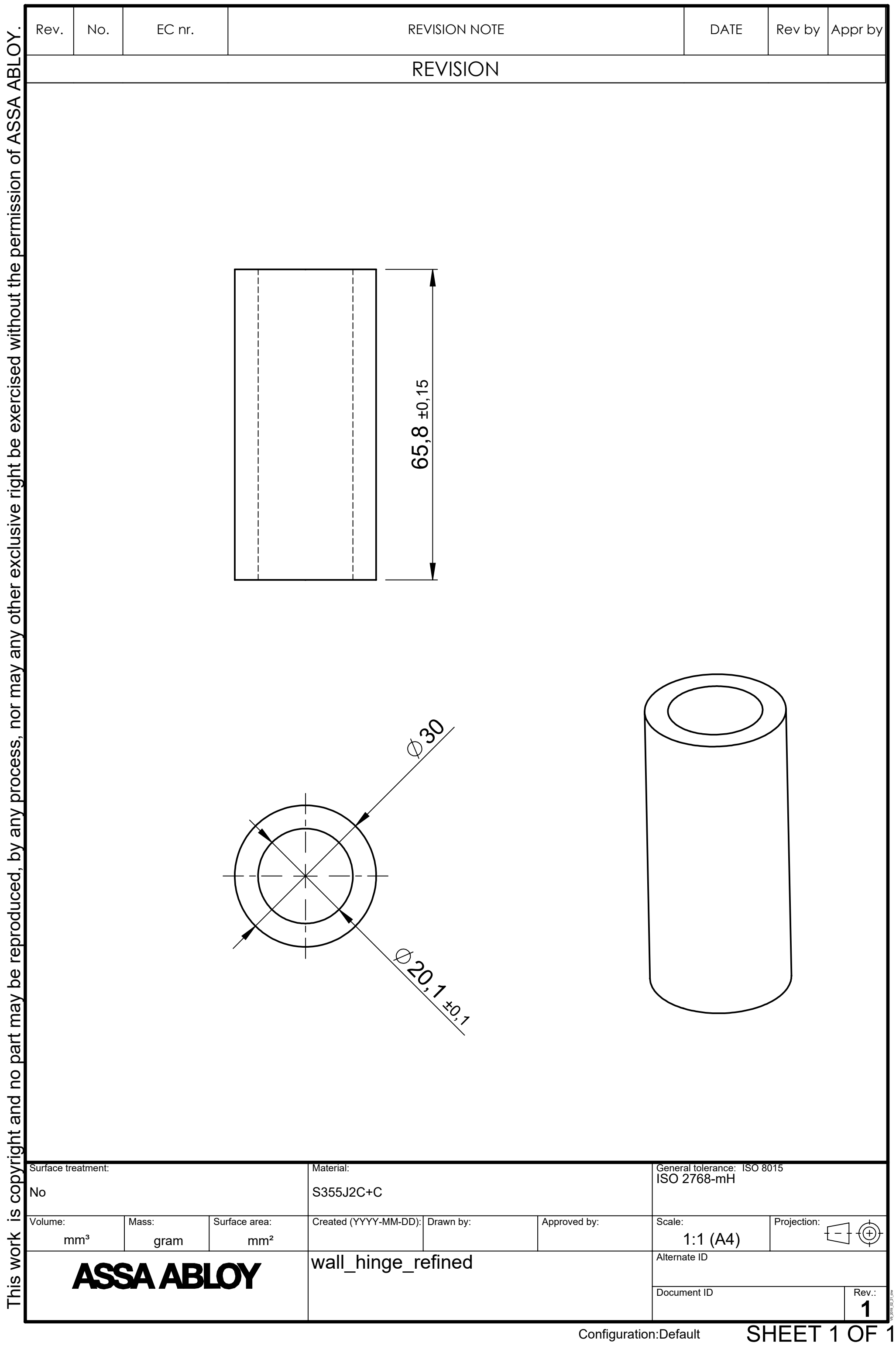

Figure D.16: *The final design of the refined wall hinge.*

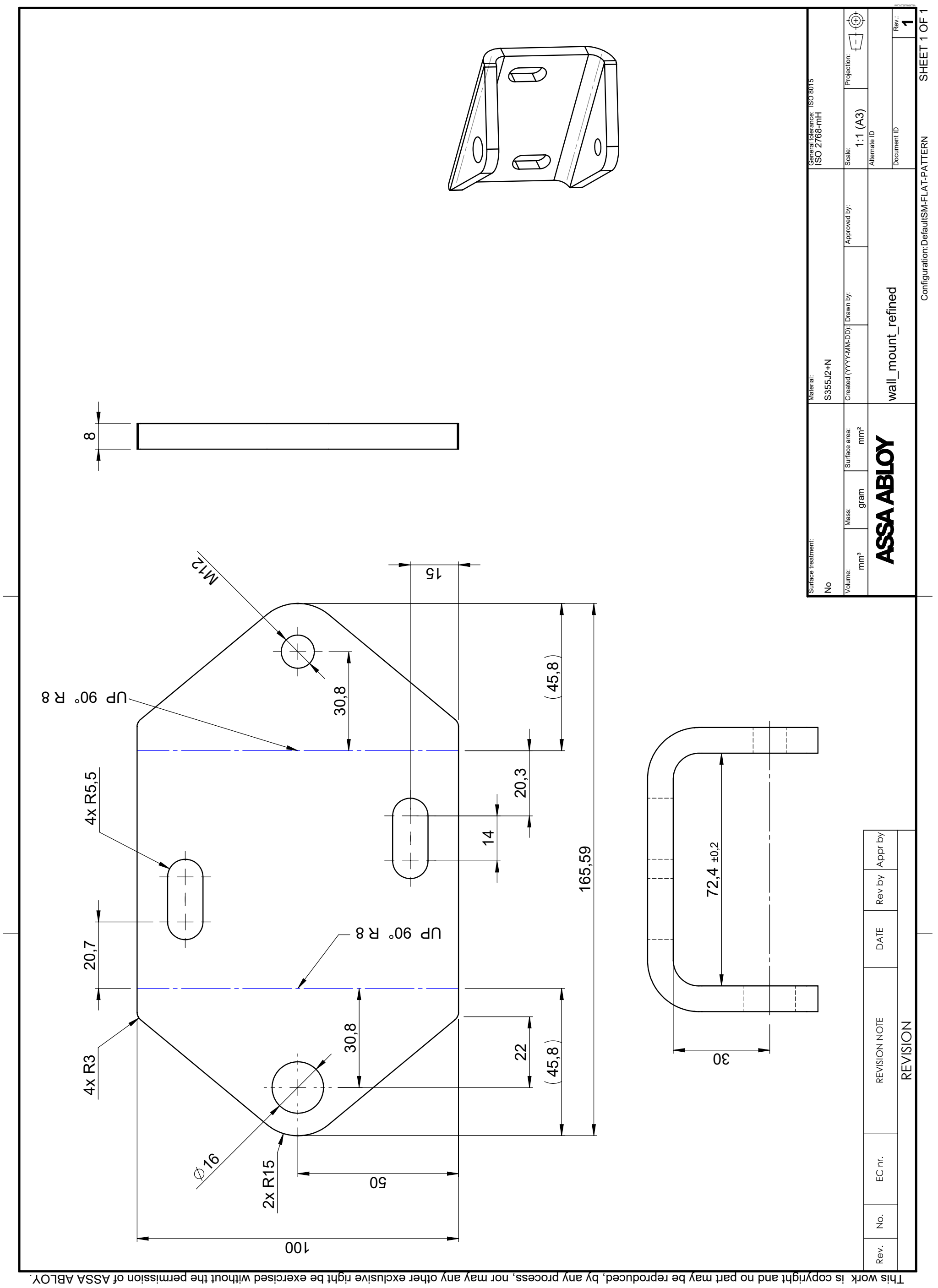

Figure D.17: *The final design of the refined wall mount.*

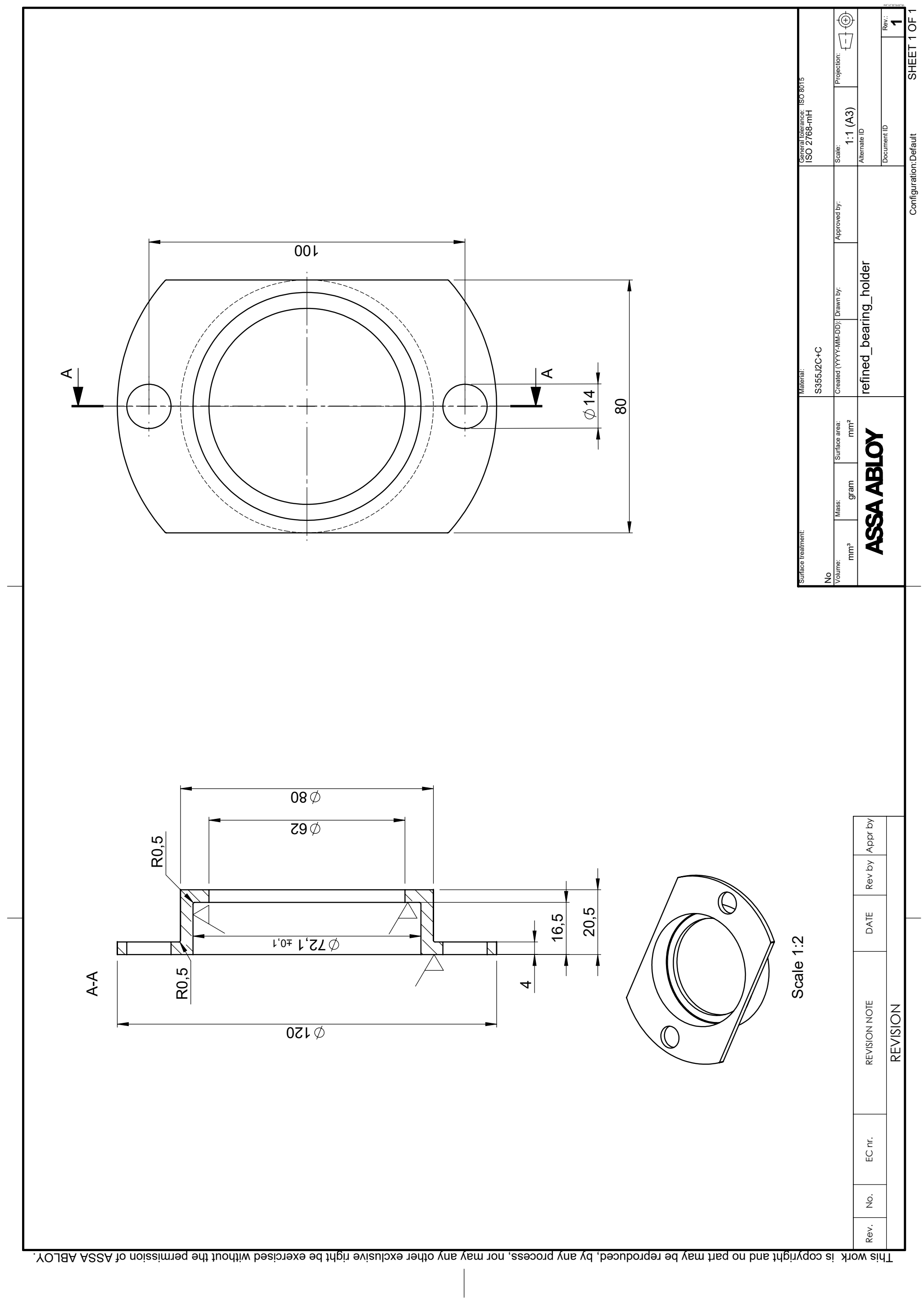

Figure D.18: *The final design of the refined bearing holder.*

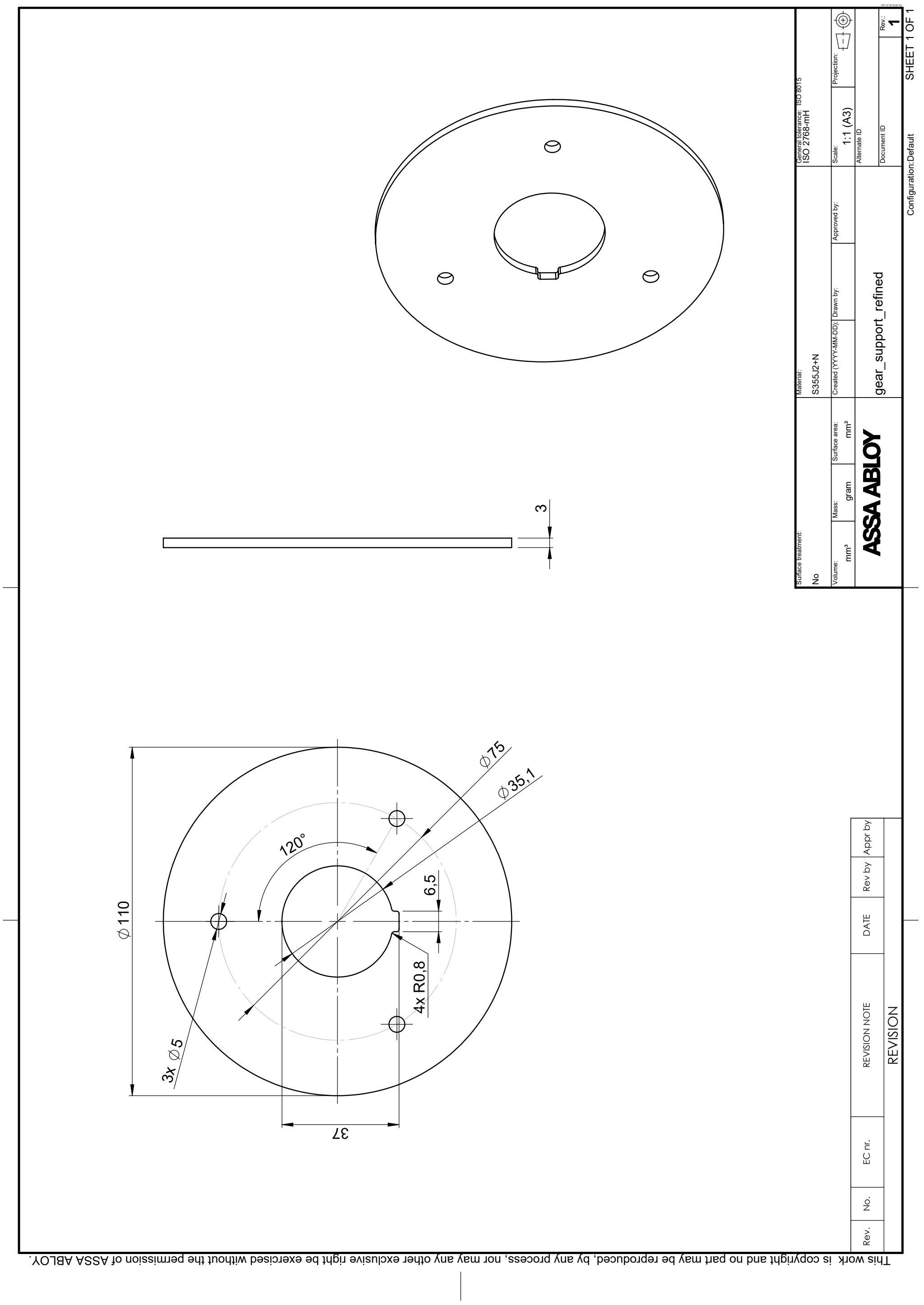

Figure D.19: *The final design of the refined gear support.*

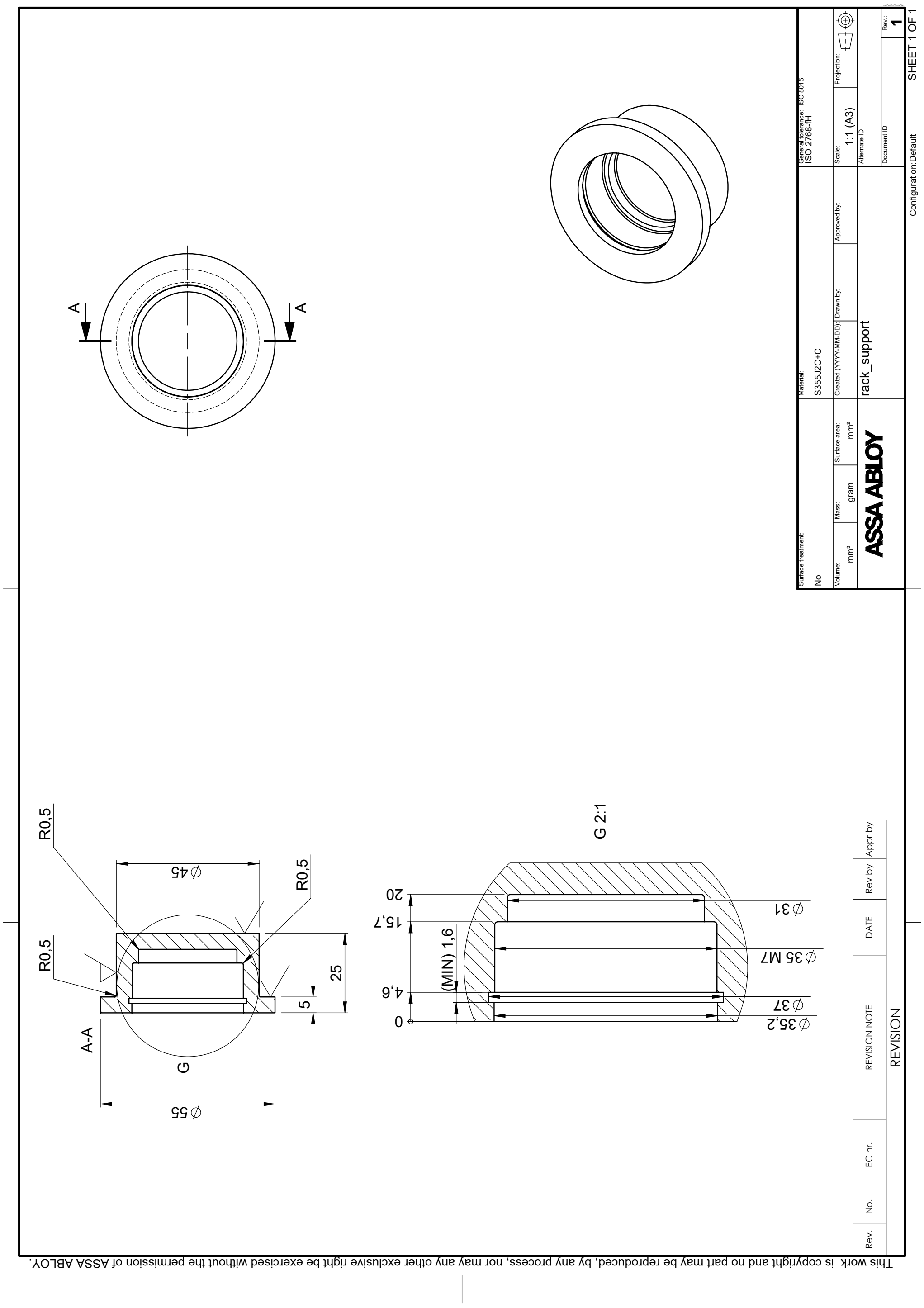

Figure D.20: *The final design of the refined rack support.*