

Faculdade de Engenharia da Universidade do Porto

Redesign of a Retractable Cam

Dinis de Carvalho Tato Alves Miranda

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FEUP Advisor: Prof. Teresa Duarte

FEUP Co-advisor: Prof. Carlos Fernandes



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"Failure is simply the opportunity to begin again, this time more intelligently."

- Henry Ford

Resumo

A presente dissertação foca-se sobre um projeto realizado em parceria com a empresa de elevadores *Schmitt + Sohn*, que surge da necessidade de transformar um componente utilizado em elevadores com portas de patamar de batente: o calço retrátil.

O motivo da proposta de melhoria do calço assenta sobre o facto de este ter sido desenvolvido há mais de 30 anos, de modo que a solução construtiva e certos componentes que o compõe já estão desatualizados.

Deste modo, a primeira etapa do projeto consiste em analisar detalhadamente a solução atualmente comercializada, de forma a notar quais as fraquezas que revela, e assim constituir oportunidades de melhoria. Esta fase engloba assim o estudo cinemático do mecanismo, o estudo estático das forças relevantes para o correto funcionamento do componente, bem como uma avaliação do ponto de vista produtivo das diferentes peças que constituem o calço retrátil.

Numa segunda fase é descrita a solução desenvolvida com base nas oportunidades de melhoria assinaladas na solução atual. É também analisada a cinemática do movimento descrito pelos componentes móveis relevantes, e realizada uma análise de forças com o objetivo de dimensionar os elementos que originam as forças motoras: o solenóide e a mola. O método construtivo da solução, tendo sido totalmente revisto, é também pormenorizadamente caracterizado, referindo todas as peças de produção e normalizadas que foram integradas nesta solução. No que toca às peças desenvolvidas, a seleção dos materiais, bem como a tecnologia de produção são justificados.

Redesign of a Retractable Cam

Abstract

The present dissertation focuses on a project conducted in partnership with the elevators specialized company *Schmitt + Sohn*, which arises from the urge in transforming an already existing component, installed in elevator equipment with manual landing doors: the retractable cam.

This retractable cam improvement proposal lays on the fact that the one used currently was developed more than 30 years ago, and consequently the design is, as well as some components, outdated.

The first project step consisted of analyzing the currently commercialized model in detail, to perceive what weaknesses it reveals and could become improvement opportunities. This part includes the kinematic study of the mechanism, the static study of the forces responsible for the proper functioning of the component, as well as a review of the retractable cam constituents, in a manufacturing point of view.

Then, the developed solution is described, based on the improvement opportunities identified on the current solution. The kinematics of the relevant mechanism components are also analyzed, as well as the forces, so the components producing force can be dimensioned: the solenoid and the spring. The solution design, having been totally reviewed, is also characterized in detail, referring which parts are production parts, and which are standardized. Concerning the production parts, the selection of materials and production technology are justified.

Acknowledgements

This project results from the knowledge acquired during the past years, as well as from people who were involved in it and encouraged me.

To begin with, I would like to thank all the support to all *Schmitt+Sohn* team who somehow contributed to the project, in particular to my mentor, Engineer Cristiano Colciago, who was always there to help with new ideas, and suggest solutions for drawbacks inherent to the development of the project. I appreciate all the ideas and teachings related to the databases and softwares used internally provided to the Research & Development department members.

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1 Introduction

1.1 Project Framework and Motivation

The present dissertation was proposed and conceived in partnership with *Schmitt + Sohn Elevators'* s Research and Development department, which was my first work experience and contact with industrial environment. The project proposed required the application of several theoretical contents acquired during degree and master's curricular units, more specifically in mechanical design and manufacturing areas, and revealed their applicability in industry.

1.2 The Company – Schmitt + Sohn Elevators

Schmitt + Sohn's is a German reputed company specialized in the construction and maintenance of several types of lifts. Its history started in 1861, in Nuremberg, with the foundation of a construction fitting company by Martin Schmitt. By the year of 1900, the company started to develop in-house components, more specifically lift construction drives, controls, cars and doors, and twenty years after, Schmitt + Sohn, at the time named *Aufzugswerke M. Schmitt & Sohn*, already exported their products to eight countries. In 1945, the company suffered a reconstruction driven by two members of Schmitt family, that turned it into a modern lift company. The following period was characterized by innovation in terms of lift constructions technologies, and establishment of new facilities in other countries: Portugal (1963), Austria (1989), and Czechoslovakia (1989).

This notorious evolution resulted in a trustful company with more than 100 000 lift systems built, responsible for the maintenance of 50 000 systems, and capable of producing 2 100 elevators annually, thanks to a team of 1900 employees.

In Portugal, Schmitt + Sohn headquarter is located in São Mamede de Infesta (Porto), where two production facilities are settled as well. In addition, six frontlines taking care of sales, installation and after-sales services are scattered all over the country. The internship has taken place in the headquarter, in an area also including the electrical production line, and Porto front line. A few kilometers away, the metalworking and assembly facility is located.

Nowadays, the company is awarded in terms of design and features innovative and individual technology such machine room-less lifts. Schmitt + Sohn is specialized in production, installation, and maintenance of lifts for new buildings, rehabilitation and modernization in several market segments, such as residential, health care, commerce and industry. The company sells and provides after-sale services in other areas of vertical mobility systems, as well.



Figure 1.1 - *Schmitt + Sohn* logo.

1.3 Project objectives

The aim of this project is the redesign of an already existing and commercialized component, developed and manufactured by the company: a Retractable Cam. This component is extremely important in the matter of user's security given that its function is to control the locking of a lift's landing door.

The urge of developing a different Retractable Cam concept is based on the “voice of the customer” which sometimes complained about the noise emitted during the unlocking of the door. Also, since the current solution is being produced for more than 30 years, its power consumption can be reduced, as well as the production methods can be optimized, reducing manufacturing costs for the company.

Having said that, three S.M.A.R.T. objectives (specific, measurable, achievable, relevant, and time-limited) were established to conduct the project:

1. Reduce power consumption by 10%, or more.
2. Position of spring adjustment should be above the elevator's car roof.
3. Reduce production cost by 20%, or more.

The whole project, since the analysis of the current solution to the complete development of a new design, may not have a duration longer than 19 weeks (from the 2nd march until the 11th of July 2022).

1.4 Methodology

After having a general perception of all company's departments, the beginning of the project took place.

The first step was to deeply analyze the solution currently in use and seek for aspects that could be improved in terms of production, selection of components, and efficiency. In this

part also the “voice of the customer” was considered in order to improve the component’s performance and to identify the weaknesses of the current solution.

Once the factors to be improved were identified, the development of a totally new design will take place. This final solution of the project will be based on requirements and on avoiding the weaknesses found on the currently commercialized model. This final step to be performed combining a CAD software, *Autodesk Inventor 2015*, with knowledge in production matter, in terms of technologies, materials and constructive solutions.

1.5 Dissertation structure

Considering the objectives defined, the dissertation structure is described below:

Chapter 1: On the first chapter, the company is presented, and the project context and objectives are defined.

Chapter 2: An analysis of the lifting systems market is presented as well as a brief description of the types of elevators and elevator doors available. The retractable cam general function and operation is also detailed. The technologies involved to produce the final solution are reviewed, as the materials selected.

Chapter 3: An extensive review of the currently used retractable cam is conducted, including kinematic and static equilibrium studies, power consumption measurements and components production analysis. The improvement opportunities are identified.

Chapter 4: The final solution aiming to solve the weaknesses determined on the previous chapter is presented, and the main components are dimensioned. A comparison of both retractable cam models is performed.

Chapter 5: The project conclusion is realized, and eventual future work is referred.

2 Literature Review

2.1 Lifting Systems Market

Increased investment in infrastructure has been responsible for market growth in recent years and is set to continue. According to the United Nations Conference on Trade and Development (UNCTAD) more than forty developing countries intend to invest in infrastructure construction. Further, the increasing interest on smart cities, particularly in Asia, is probably the future driving factor of lifts industry, considering that a notable part of the buildings standing there are skyscrapers, which normally include high-technology and high-speed lifting systems.

On the other hand, the quest for more energy efficient equipment also promotes market growth. In a building, lifts represent between 2% and 7% of the total building energy consumption. This fact motivates the lifting systems manufacturers to develop more efficient components, so they become more competitive in the market, facing other competitors (1).

Another driving factor in lift market is the growing need for repair and maintenance of old and worn equipment. In many cases, the lift is modernized, becoming more aesthetically appealing and more efficient (2).

Even though the COVID-19 pandemic negatively impacted the sector, decreasing the value of the global market by 3.5%, last year (2021) the market was valued at USD 79,70 billion and it is expected to increase from USD 83,86 billion in 2022 to USD 132,08 billion in 2029, representing a compound annual growth rate of 6,7% during this eight-year period (1).

In Europe the estimates are also positive, since in 2027 the Europe Elevator Market Size is estimated to be valued in more than USD 13 billion, meaning a CAGR (compound annual growth rate) of 1,8%. This is due to the proliferation of high-rise buildings in countries such as Germany, as well as a possible growth in maintenance services. As expected, also COVID-19 pandemic lead to a considerable decline in european market value as illustrated on Figure 2.1. The lockdown caused by the pandemic made the construction plunge severely, what directly impacted the market revenue leading to a decline of 11% (3).

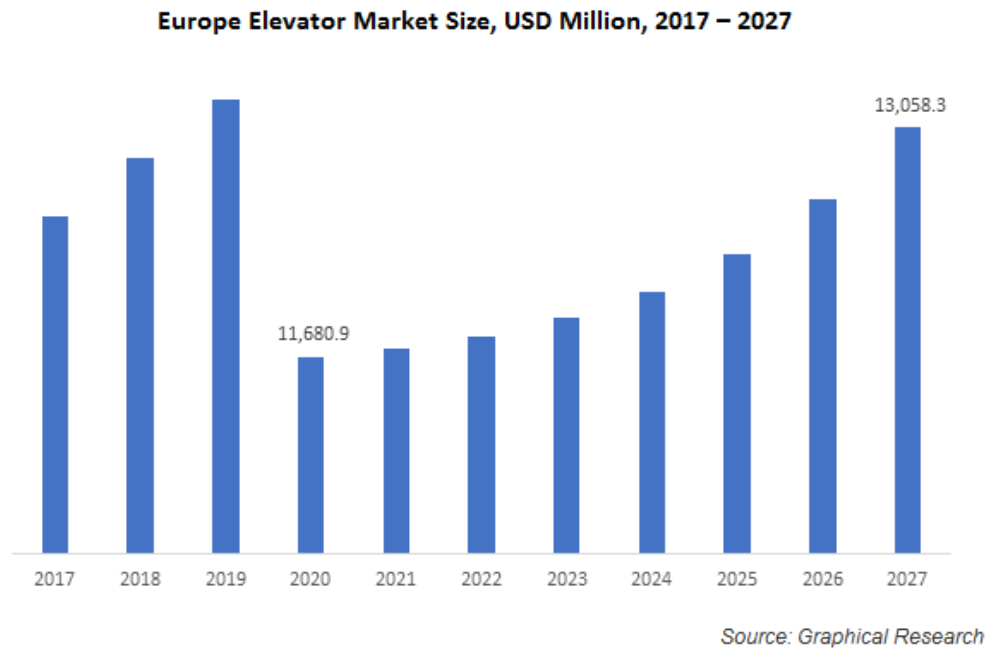


Figure 2.1 - European Elevator Market Size, in million USD, from 2017 to 2027 (3).

2.2 The elevator

According to *Oxford Learner's Dictionary*:

Elevator *noun*

/ˈelɪvətə(r)/

/ˈelɪvətər/

a machine that carries people or goods up and down to different levels in a building or a mine.

2.2.1 History of Elevators

The elevator, mean of vertical transportation widely used on every day's routine, is part of civilization for more than two thousand years already.

By definition, an elevator is simply a moving platform for carrying passengers or freight from one altitude to another. Considering this, tall structures from antiquity, such as the Great Pyramids of Egypt, could not have been built without the use of those mechanisms (4). In fact, the first written report of an elevator dates to the year of 235 BC, approximately, engineered by the Greek mathematician and inventor Archimedes. Apart from many others, a notorious difference from the ancient lifting means to the actual ones is the power source. The first elevators used human or animal force to induce movement to the lifting platform.

The big revolution on elevators technology only happened in the 19th century, due to developments in hydraulics and electricity. In 1853, Elisha Graves Otis, introduced and demonstrated a safety device which prevented the elevator's car from falling in case the lifting chain or rope broke. This turned the passenger elevators possible, given that before the lifts purpose was exclusively to transport goods once the risk of the platform falling was too high. On March 23rd, 1857, Elisha Otis installed, in New York, the first safety elevator for passengers. Later that century, impacting discoveries were made, related to the use of electricity. For instance, in 1875 J. W. Meaker patented an elevator design to open and close elevator's doors safely, and in 1880 Werner von Siemens developed the first electric elevator (5).

2.2.2 Types of Elevators

The lifts installation became more recurrent throughout the years, which lead to constant improvements in vertical transport of people and goods. Nowadays, several types of elevators are commercialized: traction elevators with machine room, machine-room-less elevators, and hydraulic elevators. Each of these types, illustrated on Figure 2.2, have limitations which must be considered while selecting the type of elevator for a certain building.

Traction elevator's cars are lifted by steel ropes or belts and use a counterweight and a pulley system connected to an electric motor, commonly located above the elevators shaft. The hoisting sheave, attached to the motor, turns to both ways, making the car travel up or down. The use of the counterweight increases efficiency reliving the weight that the motor has to move by balancing its weight with the cars and occupants. Those systems are suitable for mid and high-rise applications, are energy efficient and offer a more pleasant and smoother ride for passengers

There are two sub-types of traction elevators with machine room: geared and gear-less traction elevators. The main difference is on the fact that geared traction elevator's feature a gearbox connecting the drive sheave to the engine, while on gear-less solutions the sheave is directly fixed to the motor. Gear-less systems allow greater travelling speeds and are the only possible choice for high-rise applications. On the other hand, gear traction elevators are slower and feature a lower maximum travel distance, however the initial cost is not so significant.

Regarding the Machine-Room-Less (MRL) elevator's operation, it is similar to a traction elevator. In fact, a MRL elevator is a traction elevator without a machine room dedicated, instead the machine is located in the override space or bottom of the elevator's shaft. This is possible by incorporating more compact hoisting sheaves. Lately, these solutions are becoming the most commonly used for mid-rise applications, given their lower necessity of

space, the low energy consumption and the reliability that is comparable to the traction elevators with machine room.

The hydraulic elevators operate following a different principle. The car is supported by a telescopic or direct actuation hydraulic cylinder that extends when an electric motor pressurizes oil (or another fluid) to it, and therefore the car travels upwards. Comparing to the other two types presented, this type of elevators has lower initial and maintenance costs, although the energy consumption is higher, the travelling speed is lower, and are limited to low-rise buildings (6).

In other cases, the actuation is indirect, meaning the hydraulic cylinder is connected to the elevator's car by roping and using a deviation sheave. Consequently, the car travels at twice the speed of the hydraulic cylinder plunger.

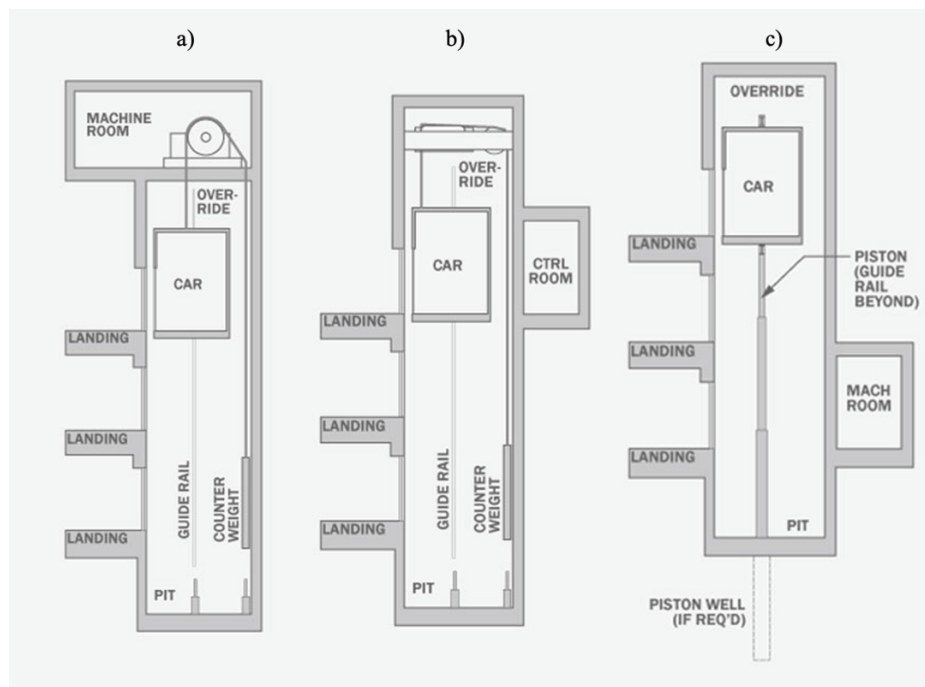


Figure 2.2 - Types of elevators: a) traction elevators with machine room; b) Machine-Room-Less elevator; c) Hydraulic elevator (6).

2.2.3 Elevator doors

One of the most important safety elements of a lift are the doors, that prevent the occupants from injuries while the car is moving and falling into the elevator shaft from the landing floors. Understanding how an elevator doors system is composed and how it operates is fundamental to realize the importance of the component in question.

Typically, an elevator has a car door, which moves jointly with the car, and a landing door (or hall door) statically located on each floor, as represented on Figure 2.3.

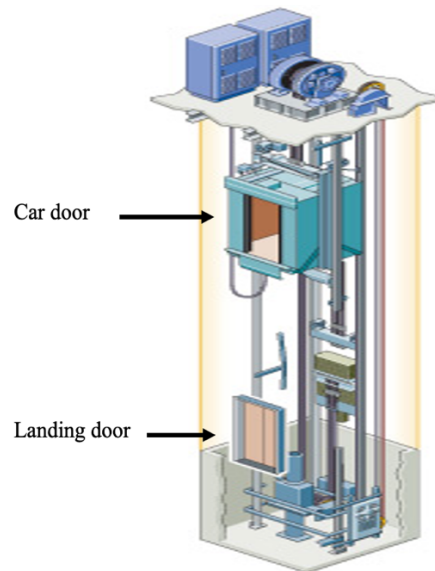


Figure 2.3 - Elevator diagram indicating the car door and landing door (7) (adapted).

According to the standard EN 81-20 5.3.1.1, “The openings in the well giving normal access to the car shall be provided with landing doors and the access to the car shall be through a car door”. This means that, the elevators must have a landing door in each floor and a car door.

The landing doors that are mounted on new lifts are automatic doors that are characterized by the number of panels that form the door and how it opens. Normally, the landing doors are composed by 2, 3, 4 or 6 panels. Regarding the opening typology, the elevator can feature center opening landing doors or sliding telescopic doors. A center opening door always opens symmetrically, this forces to have a pair number of panels. Further, this opening type requires to be telescopic when the door has more than two panels.

Other types of automatic doors do exist, nevertheless they are rare to find in the market, or are not used anymore.

On older buildings, sometimes manual landing doors can be found, as the one shown on Figure 2.4. Presently, recent elevator models do not feature manual landing doors, however they can still be installed in case of modernizations of already existing lifts for architecture or space reasons. This type of doors differs from the automatic ones once, instead of automatically when the car stops on the floor, it is user’s job to manually pull the door to access the car.

Afterwards, the door closes automatically by mean of a spring, and a buffer absorbs the shock when touching the frame, to reduce noise. Although the opening is manual, closing and locking of the door must be automatic for both manual and automatic landing doors (8).



Figure 2.4 - Manual landing door (9).

Regarding the car doors, when the landing doors are automatic, they will also be automatic, and both will have the same number of panels and opening type. For instance, if the landing door is center opening with four panels, the car door will also have four doors and center opening.

When the landing doors are manual, several options are possible for car doors. Before standard EN 81-20 came into effect, the use of car doors was not mandatory. In those cases, if the landing door is manual, the car door could be a scissor gate or even inexistant. However, it is also possible to have automatic car doors paired with a manual landing door.

It is important to refer that if a lift without a car door is intended to be modernized, the modernization project must include a car door.

To conclude, Table 1 summarizes the possible combinations of car and landing doors.

Table 1 - Possible combinations of landing and car doors.

Landing Door	Car door
Automatic	Automatic (Same number of panels and opening)
Manual	Automatic
	Scissor gate (before EM 81-20)
	Inexistant (before EM 81-20)

The lift doors play an important role in terms of safety. The car doors prevent the occupants from having contact with the landing door, and the landing doors restrict the access to non-authorized people to the elevator's shaft. When both doors are automatic, they open simultaneously, when the car stops on the desired floor. On the other hand, manual landing doors operate differently: as the car stops, the car doors open (if they exist) and only then the landing doors are unlocked by the landing door locking system, so the user can push it to open.

2.2.4 Manual landing door locking system

A manual landing door locking system is installed in every manual landing door frame of a lift, and has three purposes: lock the landing door, detect if the door is closed, and if it is locked.

The main components that integrate this type of locking system are pointed on Figure 2.5 and explained below:

- Arm with roller: when the roller is fully pushed (to the right, considering Figure 2.5) the locking pin and locking contacts pin retract, and the locking contacts open.
- Locking pin: it locks the landing door, when extended.
- Locking pin spring: forces the locking pin to extend when the roller is not being pushed.
- Door contacts: gives information to elevator's command if the landing door is closed (if contacts are closed). If the doors are opened, the contacts are opened.
- Locking contacts: gives information to elevator's command if the landing door is locked (if contacts are closed).
- Locking contact pin: when retracted, and the roller is not pushed, the locking contacts are closed.

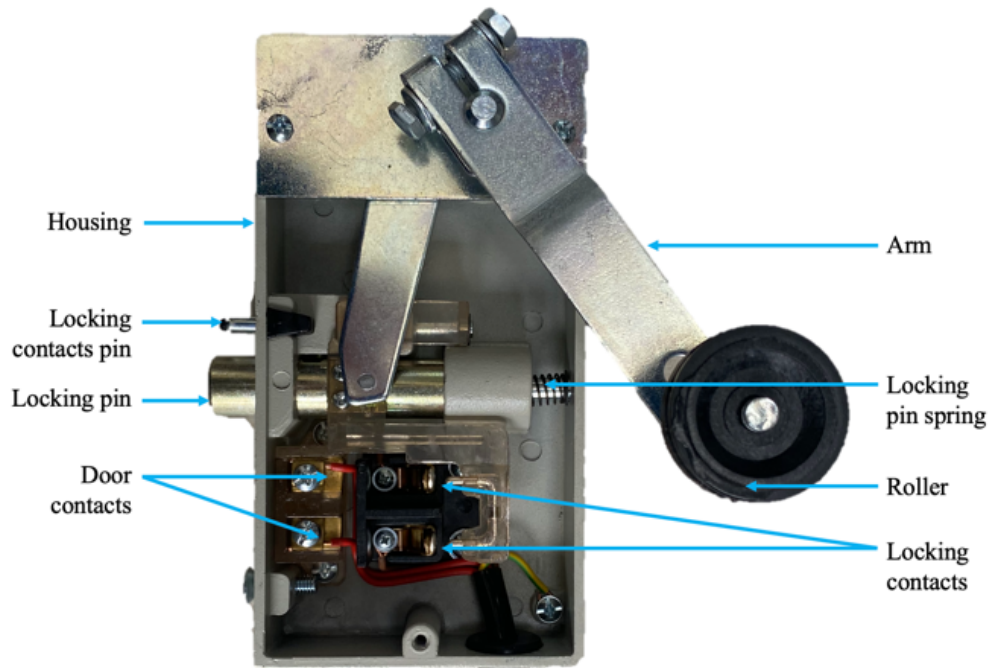


Figure 2.5 - Landing door locking system's main components.

There are two positions where the locking system can be installed: on top and on the side of the landing door. On both positions, the locking contacts pin as well as the locking pin shall be aligned with the door. If the device is on top of the door, the roller arm can be on position (1) (linked to the locking mechanism indirectly) or (2) directly attached to the mechanism.

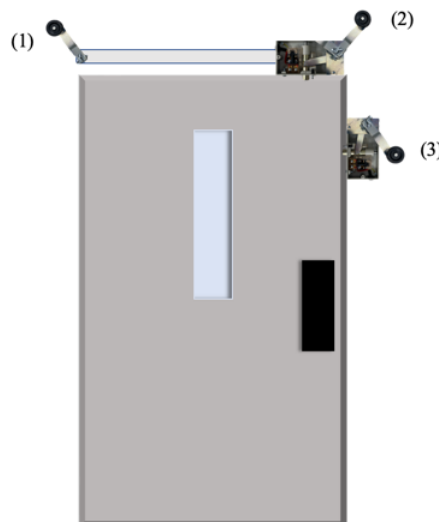


Figure 2.6 - Possible positions for locking system and its roller arm.

2.3 Function of a General Retractable Cam

To begin with, a retractable cam is a component located on the top or side of the elevator's car, in case of a system of manual landing doors is used. This assembly mainly consists of a solenoid, a spring, a structure, a lever mechanism, and a ramp. The solenoid, when actuated, is responsible for retracting the ramp, by transferring its pushing force along the lever's mechanism. When that force is no longer exerted, the elastic force of the spring must lead to the ramp extension, which is a mechanical equilibrium position. The structure's main function is to hold the remaining parts together.

To comprehend the role of the retractable cam, it is essential to understand the locking operation of the landing door. A landing door must be locked in case the car is not on that floor but unlocked if it is. The component that locks the door has been described on section 2.2.4. The application of a force on the lever's roller results in the unlocking of the landing door. That force is to be applied by the ramp when extended. In summary, the retractable cam's function is to press or release the locking mechanism's roller lever, by extending or retracting its ramp, resulting in unlocking, or locking the landing door.

The detailed opening door cycle order is the following:

1. The car travels towards the floor destination. While the car is in movement, the ramp is retracted (solenoid actuated), so it does not unlock all landing doors of floors where the car passes, by pushing their rollers.
2. The car stops on a given floor. The ramp is still retracted and the landing door of the floor locked.
3. In case of having car doors, these open. The ramp is still retracted and the landing door of the floor locked.
4. Once the car doors are fully open, the solenoid relaxes, and the retractable cam's spring forces the ramp to extend, pushing the landing door locking mechanism lever. This will unlock the landing door and open the locking contacts.
5. When the landing door opens, the doors contact opens too.
6. When the landing door closes its door contacts close as well, and the retractable cam ramp can be retracted.
7. Then, the retractable cam solenoid is actuated, causing its ramp to retract. The spring of the locking pin forces it to extend and lock the landing door. The locking pin is also extended, and if it confirms that the door is locked, the locking contacts close.

8. Only when the landing door is closed and locked (both doors contact and locking contacts are closed) the car can initiate another ride.
9. The car starts the ride towards another floor.

There are retractable cams for different applications and with distinct characteristics as analyzed on section 2.4, nevertheless, there are points in common that must be respected, so it can be commercialized. For instance, in case of electrical failure in the building, the landing doors must be unlocked. This is accomplished associating the ramp extending force to the spring. This way, if there is no electric current on the solenoid, the ramp is extended, unlocking the door. Also, the solenoid and electric components associated to must feature an ingress protection rating of at least IP20, according to the standard IEC 60529.

As illustrated on Figure 2.6 the locking system can be located in different positions, so the retractable cam has to be adjustable to align in height its ramp with the locking system arm. There are two types of retractable cams in terms of adjustment: the component can be fixed on different heights on the cabin wall, or it is fixed on the top of the cabin and internal adjustments allow to rise or descend the ramp as represented on Figure 2.7.

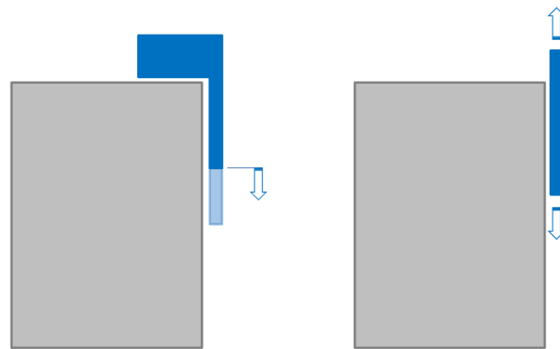


Figure 2.7 - Retractable cam adjustment methods.

2.4 Existing models on the Market

Given the indispensable use of the retractable cams on manual landing doors elevators with no car door, all elevator manufacturers that produce this type of lifts must develop or acquire externally this component. Also, some solenoid manufacturers produce their own retractable cams to sell to elevator companies.

2.4.1 Model 1: *Kuhse* Type V16 (GL)

This solution is developed and commercialized by *Kuhse Industrial Components*, a German supplier for industrial equipment, including solenoids. The returning force comes from a compression spring that is not visible on Figure 2.8. The solenoid actuates directly on the ramp. This solution is to be mounted on the side of elevator's car. Technical data on Table 2, page 26.



Figure 2.8 - *Kuhse* Type V16 (10)

2.4.2 Model 2: *Kuhse* Type GR 70 (GL)

This retractable cam is also a *Kuhse Industrial Components* series of products, however, is to be mounted on the top of the lift's car. It consists of two parts: the cam with a compression spring and a pulling solenoid. The two parts are linked by a metallic chain, which is subjected to traction when the solenoid is actuated, causing the ramp retraction. Technical data on Table 2.

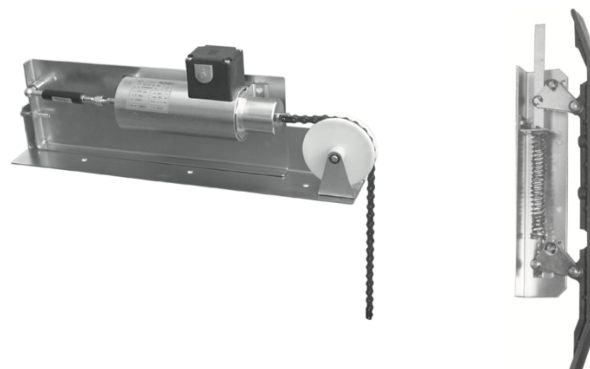


Figure 2.9 - *Kuhse* Type GR 70 (GL) (10).

2.4.3 Model 3: *Kuhse* Type Group V10

The Type Group V10 is a *Kuhse Industrial Components* series of products that consist of a hinged ramp that is retracted by actuation of a squared solenoid on the hinges. The solenoid is vertically positioned above the ramp, which makes this solution ideal for applications with narrow space between the car and the door's locking device.

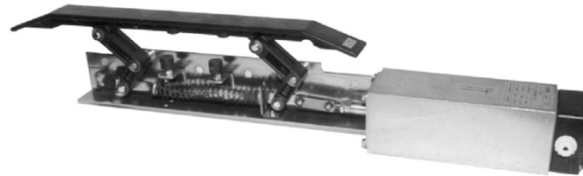


Figure 2.10 - *Kuhse* Type Group V10 (10).

2.4.4 Model 4: *Kronenberg* EMT 15 K

Kronenberg GmbH is a German firm that became one of leader suppliers of switches in elevators industry. This retractable cam model is ideal to install where installation depth is short. This solution is to be mounted on the side of the car (11).



Figure 2.11 - *Kronenberg* EMT 15 K (11).

2.4.5 Model 5: *Kronenberg* EMT 13

This compact retractable cam was designed to be installed on the top of the car. The returning force is provided by a compressions spring concentric with the solenoid responsible for the ramp retraction. Both elements actuate directly on a ramp lever (12).



Figure 2.12 - *Kronenberg* EMT 13 (12).

2.4.6 Technical data comparison between models

To compare all the models presented on this section, Table 2 was elaborated.

Table 2 - Technical data of other solutions existing in the market (10) (11) (12).

	Type V16 (GL)	Type GR70 (GL)	Type Group V10	EMT 15 K	EMT13
Stroke [mm]	30	30	33	30	30
Ramp pushing Force [N]	56	36	35	40	40
Relative On-time [%]	100	60	100	100	100
Solenoid power [W]	280	95	70	46	48,3
Ingress Protection Ratio	IP20/IP43	IP42	IP42	IP20	IP20
Mounting	Cabin wall	Cabin Roof	Cabin Wall	Cabin Wall	Cabin Roof

2.5 Solenoid

A solenoid is a type of electromagnet formed by tightly wound electrical conductor wire (typically made of copper) forming a helical coil around a magnetic core, named armature. When an electric current runs along the wire, an intense and controlled magnetic field is produced, corresponding to the one produced by the ensemble of the number of turns. As shown on Figure 2.13, in the centre of the solenoid, far from the ends, the magnetic lines are parallel to the solenoid's axis and form an intense and uniform magnetic field, while at the extremities, the lines split. Outside the coil, the magnetic field is way less dense (13).

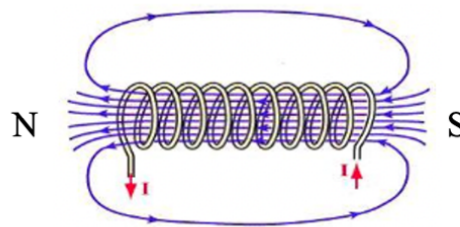


Figure 2.13 - The magnetic field produced by the coil of a solenoid.

As on a permanent magnet, the magnetic field lines are oriented from the south pole to the north pole of the magnet.

In the case of an ideal solenoid, by other words, having an infinite length and the coils as tightly wound as possible, the magnetic field produced at its core is parallel to the central axis and can be calculated using Ampère's law.

$$B = \mu_0 n I \quad (1)$$

Where:

B , is the magnetic field

μ_0 , is the permeability of free space

n , is the number of turns per length unit

I , is the current through the coils of the solenoid

Nevertheless, commonly used solenoids have two extremities meaning that Ampère's law cannot be applied, and the magnetic field is not constant during its entire length. In fact, the magnetic field in the end of a relatively long solenoid, is approximately half than at its centre, as illustrated on Figure 2.14 (14).

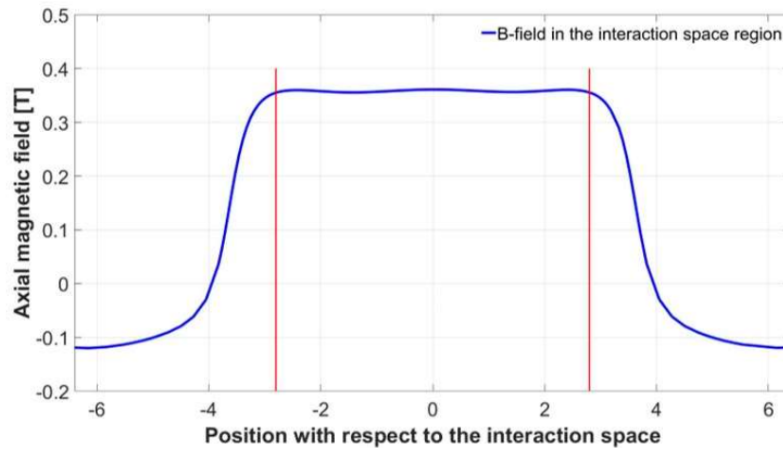


Figure 2.14 - Relative magnetic field strength relative to position in solenoid's axis (15).

In practice, this phenomenon results in a non-constant force exerted by the armature, throughout the core stroke. Consider the armature coordinate system of

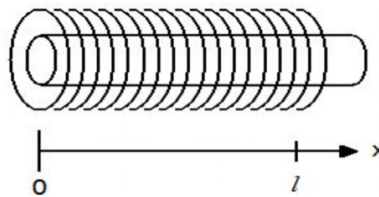


Figure 2.15 - Armature coordinate system considered (16).

When the armature is totally inside the solenoid's coil, $x=0$, and when $x=l$ it is at the entry extremity of the coil, as represented on Figure 2.15. For those two positions, the inductance (L) can be calculated using equations (2) and (3):

$$L(0) = \frac{\mu_r \mu_0 N^2 A}{l} \quad (2)$$

$$L(l) = \frac{L(0)}{\mu_r} = \frac{\mu_0 N^2 A}{l} \quad (3)$$

Where:

L is inductance [H]

μ_0 is the relative permeability of the iron armature [Wb/A·m]

μ_r is the relative permeability of free space [Wb/A·m]

N is the number of turns of the coil [-]

A is the cross-sectional area of winding [m²]

l is the length of coil [m]

Given that the permeability of iron is significantly larger than free space's, $L(0)$ is much larger than $L(l)$. This means that the more the armature is inside the coil, larger is the inductance. The variation shape of inductance between these two values depends on factors inherent to the solenoid's construction.

The force produced by the solenoid can be determined by the deriving the energy in order to x :

$$F(x) = \frac{dE_L}{dx} = \frac{d}{dx} \left[\frac{V^2}{2R^2} L(x) \right] \quad (4)$$

Where:

E_L is the energy stored in the coil [J]

V is the electric potential difference [V]

R is the resistance of the entire length of the wire [Ω]

$L(x)$ is the inductance for a given position [H]

Therefore, the greater the inductance, the greater the force produced by the solenoid, by other words, the more the armature is inside the coil, the larger is the force exerted. This phenomenon can be observed for a given pull-effect solenoid on Figure 2.16 (16).

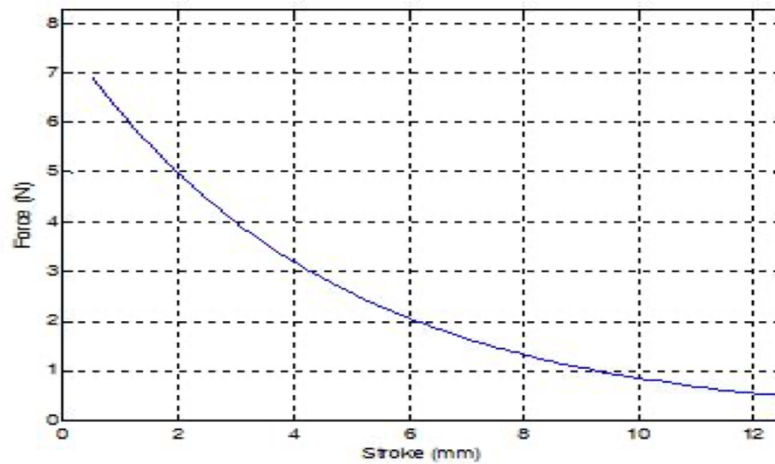


Figure 2.16 - Force exerted by the solenoid's armature function of stroke (16).

Another factor that must be considered when analyzing solenoids is the duty cycle, normally designated by “ED%” by the manufacturers. It represents the relative time-on that the solenoid can support without overheating.

$$ED\% = \frac{T_{ON}}{(T_{ON} + T_{OFF})} \quad (5)$$

Where:

T_{ON} is the time the solenoid is energized

T_{OFF} is the time the solenoid is relaxed

2.6 Production Technologies

2.6.1 Laser Cut

A laser can be described as a powerful and narrow beam of light as well as the device that produces it (17). Nowadays, utility of lasers covers different areas such as entertainment (light shows), medical and industrial, more specifically in terms of material cutting. Actually, it has become the most used thermal power process-based material cutting technology, being able to cut a panoply of different materials including wood, glass, plastics, and metals (18).

LASER is an acronym for Light Amplification by Stimulated Emission of Radiation, so allowing to understand the laser's principle. The emission of radiation is caused by an atom or molecule that, by absorbing energy, transited from its ground state to a higher energy level. Naturally, the particle returns to its ground state, releasing part of its energy as a photon. That photon can cause another excited atom or particle to return to its ground state, also emitting a photon equivalent to the first one. This second photon emission caused by another photon is a light amplification phenomenon (19).

A laser source mainly consists of an optical resonator, which is a cavity where a gain medium (or lasing medium) is located in between two mirrors, and an energy source. The lasing medium contains the atoms, and it can be in every state of matter (solid, liquid or gas), and on its extremities, two mirrors are located. One is totally reflecting and is named the high reflector (HR), the second one is partially reflecting and referred as the output coupler (OC). The energy source itself can be of various types such as a radiation source (flash lamp or diode laser) for solid state lasers, electrical discharge for gas lasers or even a chemical reaction (19) (20).

The laser operation can be described in five sequent steps. Initially, the atoms/molecules in the gain medium are in their ground state (step 1), then the energy source is responsible to bring the medium particles to an excited state (step 2). Once the electrons tend to decay to the ground level, in this phase, the spontaneous emission is greater than the stimulated one. Given the existence of the two mirrors, the photons bounce back and forth in the optical resonator, stimulating the emission (steps 3 and 4). The phenomenon of the increasing ratio between atoms in excited state and ground state is called Population Inversion. At a certain point, a full stimulated emission is attained, resulting on a coherent and highly monochromatic laser beam (step 5). The fact that one of the mirrors is not fully reflective, allows some photons to escape the resonator cavity, and those photons are the ones that form de laser.

The complete laser operation is illustrated, step-by-step on Figure 2.17.

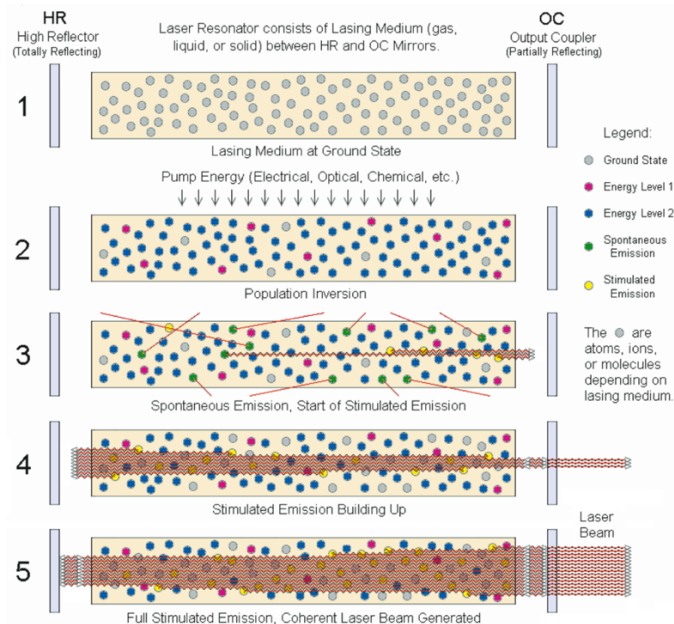


Figure 2.17 - Description of the five steps composing a basic laser operation (19).

For such a vast panoply of applications, several types of lasers exist. Given the different wavelength of the different varieties, each one is ideal for a specific purpose. For instance, in

the matter of metal laser cutting, the best laser type for a given metal may not be adequate for another metal. The diagram of Figure 2.18 categorizes the different laser types and graphic illustrates the absorption of the laser for a given wavelength for several metals.

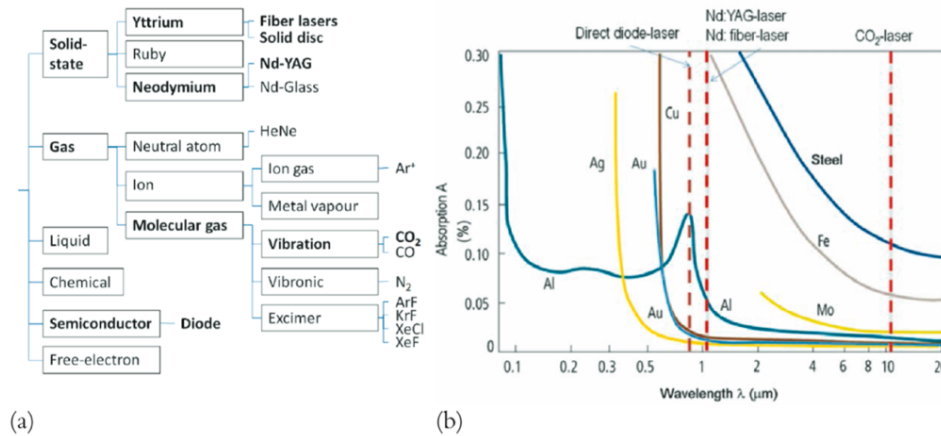


Figure 2.18 - a) Diagram specifying and categorizing different types of lasers. b) Absorption of the laser as function of the wavelength for different metals (19).

The higher absorption means that the material is absorbing a greater percentage of the energy provided by the laser.

For this project's purpose, only metal laser cut is relevant, given it is the only type of material that will be cut during the production of the components.

While laser cutting metals, the laser melts the metal and expels it from the kerf, assisted by an inert gas, such as nitrogen (N₂) once it does not react exothermically with the molten metal (18). This type of cut is named fusion cut and the sequence of phases to cut a part from a metal sheet is the following:

1. The laser beam focuses on a specific spot on the metal sheet with high intensity.
2. On that spot, the material starts to melt, creating a melt pool. At the same time, a pressurized inert gas jet, coaxial with the laser, is responsible for removing the molten material.
3. When the metal sheet is fully pierced, piercing phase has ended, and the laser beam starts to move, melting the material where it localizes, generating the cut.

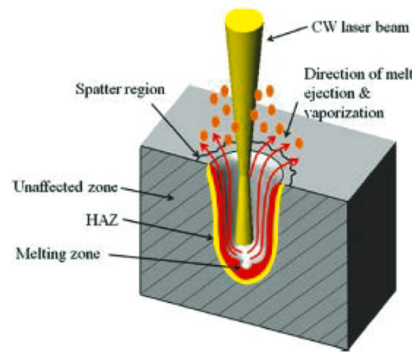


Figure 2.19 - Piercing phase of fusion cut (19).

Piercing is the first operation in laser fusion cut, except when the cut starts at a free edge of the metal sheet. In this case, no piercing is needed. The aim of this operation is to pierce the metal sheet to establish a cut front, allowing the energy to melt the material from top to bottom, as the laser moves, producing the cut. During the cut, the assistance gas is directed to the kerf, in order to remove the molten material from it as shown on Figure 2.20.

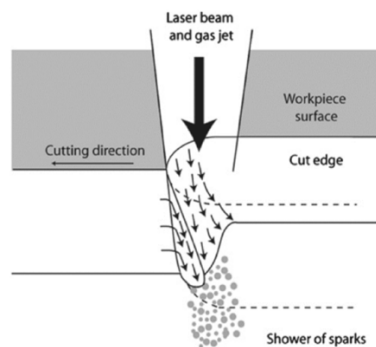


Figure 2.20 - Laser cut process.

There are some other cutting technologies that can be compared to laser cutting. Regarding mechanical cutting techniques, sawing, water jet are two technologies that can be compared to laser cut. On the other hand, the thermal cutting techniques that can be used in alternative to laser cut are, for instance Oxy-cut and wire electric discharge machining (Wire EDM). Despite the high investment cost, laser cutting presents several advantages such as flexibility in terms of geometry, precision, and edge quality (higher for thinner sheets), small heat affected zone (HAZ) and the process can be easily automated (CNC).

Cut quality can be classified according to ISO 9013:2002 (Thermal cutting – Classification of thermal cuts – Geometrical product specification and quality tolerances), by defining the quality levels of the cutting surface (21). Other relevant quality parameters related to laser cut are the formation of dross, which is undesirable, and kerf width that should as small as possible.

The quality parameters can be optimized for certain values of the following cutting parameters:

- Laser power
- Cutting speed
- Focal distance
- Assistance gas pressure
- Stand-off distance
- Type and diameter of nozzle

The assistance gas itself has a major impact on the process and can be either an inert gas, such as nitrogen (N_2), or oxygen. When an inert gas is used, its only function is remove the molten material from the cut zone, resulting in a lean cut edge, on the other hand, when oxygen serves as assistance gas, it reacts with the material increasing cutting speed but an oxidized cut edge is obtained.

2.6.2 Bending Operation

Metal sheet forming process has proven to be widely used and revealed to be essential in many industries, such as transportation industry.

Among this type of technologies, bending process is the most common, and consists of plastic deformation of a sheet made of a ductile material (normally metal) along a linear axis, named bending axis, and interfering the less possible with the surface area. The focus will be metal sheet bending once in this project, this is the only type of raw material that will be bent (22).

The process is performed in a bending machine, which is similar to a press, where the tools are mounted: a punch and a die. The operation starts by positioning the sheet in the correct position to obtain the desired shape, then the punch presses the sheet towards the die, deforming it plastically. While the bending is being produced, the force applied and the punch displacement generate a stress gradient, where the face of the sheet that is in contact with the punch is being compressed while the opposite is being stretched.

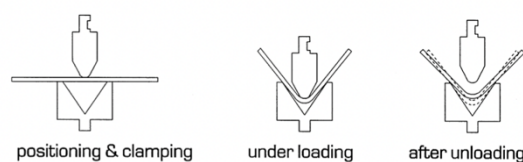


Figure 2.21 - Main stages of sheet bending (23).

Even though the principles are the same, the bending operation can be realized in different ways and can be characterized according to the interaction between the sheet, the punch and the die, and the tool's geometry. There are essentially three conventional types of bending performed with a V-shaped die: air bending, coining, and bottoming.

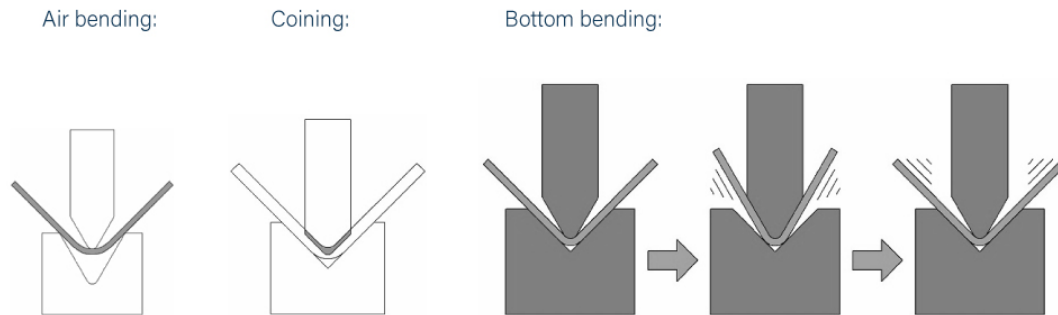


Figure 2.22 - Illustration of the 3 types of bending operation (24).

Air bending is more relevant for this purpose, so it will be analyzed more in detail.

On air bending, the metal sheet is plastically deformed by the punch movement towards the die until the desired angle is attained. The angle between flanges is established by the displacement of the punch and die opening (distance “W” on Figure 2.23). This means that the same set of tools is suitable to bend different angles.

A simple bending model can relate the punch penetration with the bend angle, tools geometry, and sheet thickness, considering that the cross section of the sheet is formed by a circular part and the two flanges.

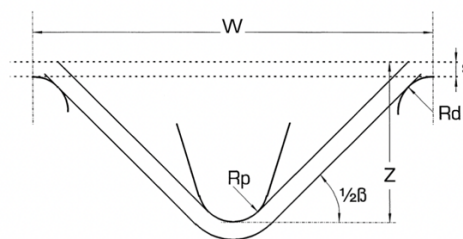


Figure 2.23 - Air bending dimensions for bend angle calculation (25).

Using the dimensions annotated on Figure 2.23, the model results on the equation (6).

$$Z = \frac{W \tan\left(\frac{\beta}{2}\right)}{2} + \left[1 - \cos\left(\frac{\beta}{2}\right) - \tan\left(\frac{\beta}{2}\right) \sin\left(\frac{\beta}{2}\right)\right] \cdot (R_p + R_d + s) \quad (6)$$

Where:

Z , is the punch penetration

β , is the bend angle

R_p , is the punch radius

R_d , is the die radius

s , is the sheet thickness

This model is based on a large number of assumptions, so the results obtained from it are quite distinct from the ones obtained experimentally (23).

One of the main advantages of this bending operation is the low force required, resulting in a low tool wear, and machine effort. On the other hand, the angle precision associated to this technique is compromised, due to a considerable spring back effect, but this defect can be solved increasing the punch penetration and adjusting the value experimentally (25).

Coining is also a quite common type of bending operation for thin sheets, typically around 1 to 2 mm. It consists of pressing the sheet against the die and acquiring the contour's shape of the tools and preventing the spring back effect. Comparing to air bending, this operation presents a better precision along successive operations, smaller bend radius, and the material strength in the bent zone is increased by the compression applied. In terms of required force, coining requires a lot more compressive force exerted by the bending machine, reducing tools operating life. Also, once it is the tools that define the bend angle, a set is required for each specific desired angle.

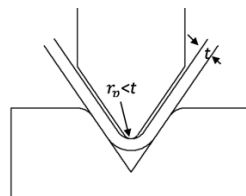


Figure 2.24 - Coining bending operation (25).

Bottoming is the method which offers the least spring back effect but is mostly adopted to obtain bend angles of 90° or slightly superior. The punch bottoms the sheet into the die, forcing the bend radius to be equal to the punch radius. When bending, the gap between punch and die is the thickness of the sheet (24) (25).

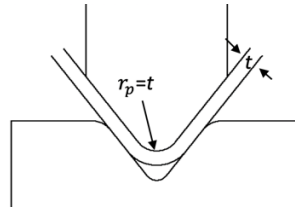


Figure 2.25 - Bottom bending operation (25).

After the bending process, the final result can be analyzed through some specific dimensions illustrated on Figure 2.26: bending angle (θ), inner bending radius (R_i), sheet thickness (t) and the bending width (w).

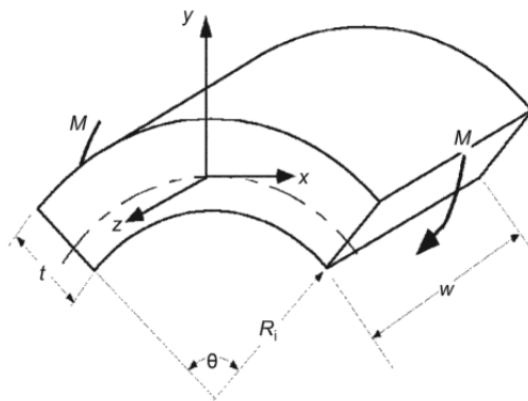


Figure 2.26 - Relevant dimensions after sheet bending (26).

Some of those dimensions, such as the inner radius and bending angle, highly depend on the bending operation, parameters, and tools and on the material as well.

Some other dimensions, such as the neutral radius and mean bending radius can be evaluated to perform a more complex analysis, nevertheless, they are not relevant for this project, will not be described.

2.6.3 Orbital Riveting

Riveting is a joining technology used in several industrial sectors such as aeronautical and elevators and comprises several different procedures, for instance: pop riveting, press riveting and radial frictional riveting. The main problem of riveting has to do with the resistance of the revit, that directly depends on many factors related to the tension of the joint, as the riveting load, tolerances, and friction ratio.

The demand of a measurable controlled process capable of producing a joint with lower residual stress led to the use of an innovative technology: orbital riveting or orbital forming (27).

This technology is of a cold forming process and serves as alternative to conventional joining processes, such as welding, riveting, and crimping. It consists of exerting on the rivet a compressive axial and radial load with an angled peen (forming tool) which spins at a constant angle (typically from 3° to 6°), until it deforms to a certain shape, taking approximately 1,3 to 3 seconds per joint. This joining technology can be used in different materials with hardness up to 54 Rockwell C, such as steel, aluminum, copper and even some plastics.

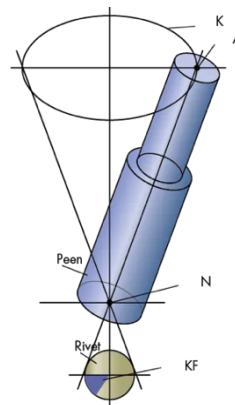


Figure 2.27 – Peen movement in orbital riveting and contact surface between rivet and peen (28).

The forming tool typically have the same shape of the final part (revit), so they can be flat, conical, eyelet or crown headed. The material of which they are made is normally M2 drill-rod steel, with a heat treatment to harden it to 60 to 64 Rockwell C (28).

As illustrated on Figure 2.27, the forming tool is only in contact with a part of the revit, not its whole surface, reducing the compressive axial forces exerted are 80% lower than in impact riveting. The result is a joint with lower stress transmitted to the fixed parts, a smooth and regular surface finish without cracks on rivet's head, the fastener shank bending is avoided. Also, the presses required are more economical, smaller, and less powerful, leading to longer lasting tools (27).

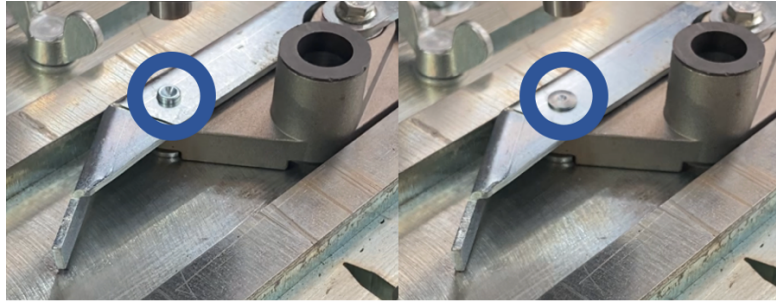


Figure 2.28 - Orbital rivet before and after being applied (left and right respectively).

2.7 Materials

A large selection of materials composes the currently used retractable cam, from polymers to metals. As expected, structural parts requiring greater mechanical strength are made of metal, commonly cold rolled or hot rolled steel, and the secondary parts such as washers or bushings are made of polymers.

2.7.1 Steel Sheet – EN10130 DC01

The steel sheet characterized by the standard EN 10130:2006 (Cold rolled low carbon steel flat products for cold forming), is made of a low carbon steel, indicated for cold forming. This European standard applies to flat products featuring widths greater than 600 mm and thickness between 0,35 and 3 mm at the time of acquisition of raw material. There are six grades included in EN10130:2006, featuring different chemical compositions and therefore different mechanical properties.

Table 3 - Mechanical properties and chemical composition of steels included on EN10130:2006.

Grade	Max. yield strength (MPa)	Tensile strength (MPa)	Ladle analysis chemical composition				
			C max. (%)	P max. (%)	S max. (%)	Mn max. (%)	Ti max. (%)
DC01	280	270 - 410	0,12	0,045	0,045	0,6	-
DC03	240	270 - 370	0,1	0,035	0,035	0,45	-
DC04	210	270 - 350	0,08	0,03	0,03	0,4	-
DC05	180	270 - 330	0,06	0,025	0,025	0,35	-
DC06	170	270 - 330	0,02	0,02	0,02	0,25	0,3
DC07	150	259 - 310	0,01	0,02	0,02	0,2	0,2

2.7.2 Steel Sheet – EN10111 DD11

The standard EN 10111:1998 (Continuously hot-rolled low carbon steel sheet and strip for cold forming – Technical delivery conditions) specifies 4 grades (DD11, DD12, DD13 and DD14) of hot-rolled low carbon sheets and strips for cold forming, with a thickness between 1,5mm to 8mm. For each grade, the mechanical properties as well as chemical composition and forming characteristics are detailed.

Table 4 - Mechanical properties and chemical composition of steels included on EN10111:1998

Grade	Yield strength (MPa)		Max. tensile strength (MPa)	Ladle analysis chemical composition			
	$1,5 \leq e \leq 2 \text{ mm}$	$2 \leq e \leq 8 \text{ mm}$		C max. (%)	P max. (%)	S max. (%)	Mn max. (%)
DD11	170 - 360	170 - 340	440	0,12	0,045	0,045	0,6
DD12	170 - 340	170 - 320	420	0,1	0,035	0,035	0,45
DD13	170 - 330	170 - 310	400	0,08	0,03	0,03	0,4
DD14	170 - 310	170 - 290	380	0,08	0,025	0,025	0,35

2.7.3 Steel 9S20 – DIN 1651

The 9S20 steel is a free cutting steel with high machinability and is not intended for heat treatment. This material is used to produce parts subjected to light stress but is not suitable for parts that might be subjected to an internal pressure due to its poor transverse properties. The most common applications for this steel are bolts, studs, nuts, and cycle parts (29).

Table 5 - Modulus of elasticity and rigidity and chemical composition of 9S20 steel.

Modulus of elasticity (GPa)	Modulus of rigidity (Gpa)	Ladle analysis chemical composition				
		C max. (%)	P max. (%)	S max. (%)	Mn max. (%)	Si max. (%)
208	81,1	0,13	0,11	0,15 - 0,25	0,6 - 1,2	0,05

Table 6 - Minimum yield strength and tensile strength for different shaft diameters of cold worked 9S20

Treatment condition	Diameter d (mm)	Min. yield strength (Mpa)	Tensile strength (Mpa)
Cold worked	$d \leq 16$	390	490 - 790
	$16 < d \leq 40$	355	440 - 740
	$40 < d \leq 63$	295	390 - 690
	$63 < d \leq 100$	235	360 - 610

2.7.4 *Iglide*® G300

Iglide® is designed to produce high-performance polymer bearings, being composed by strengthening fibres, a thermoplastic base material, and solid lubricants. This material makes the bearings extremely versatile given their overall performance, once they run at low to average speeds on different shaft materials, operate at a wide range of temperatures (40°C to 130°C), and allow oscillating movements.

Table 7 - *Iglide* G300 properties (30).

	Unit	Value
Density	g/cm ³	1,45
Dynamic coefficient of surface friction (steel)	μ	0,08 - 0,15
Modulus of elasticity	GPa	7,798
Tensile strength at 20°C	MPa	210
Compressive strength	MPa	78
Max. long term temperature	°C	130
Min. long term temperature	°C	-40
Short-term oscillating max. speed (for ø 8mm shaft)	rpm	3335



Figure 2.29 - IGUS® bearing made of *Iglide* 300 (30).

2.7.5 Polyoxymethylene (POM)

Polyoxymethylene (POM), also called acetal, is an engineering thermoplastic recurrently used to replace metals, given its stiffness, resistance to corrosion and dimensional stability. For those reasons, this material is ideal to produce precision parts needing high stiffness and low friction. Acetal is typically aspect is opaque white given its high crystalline composition, but all colors are available.

POM's mechanical properties can be compared to nylon and polypropylene (PP), as on Table 8.

Table 8 - Comparison between mechanical properties of POM, PP and nylon.

	POM	PP	Nylon
Density (g/cm³)	1,38 - 1,44	0,9	1,13 - 1,15
Tensile strength (MPa)	34,5 - 62	27,6 - 34,5	96,5
Tensile modulus (GPa)	3,08	0,86 - 1,55	1,59 - 3,79
Tensile elongation (%)	10% - 76%	100% - 600%	15% - 80%

POM components can be manufactured by plastic injection or machined manually or by CNC depending on the application and part requirements. When injected, a shrinkage of around 2% (same as PP) must be considered.

This thermoplastic is used on a panoply of applications, such as mechanical gears, guiding elements, fan wheels, and even parts of the car's fuel sending system (31).

3 Current Solution Analysis

The retractable cam is a component that is no longer installed on new elevator models, nevertheless it will continue to be supplied as spare part in elevators with manual hinged landing doors and no car door. In this chapter the solution currently used by *Schmitt + Sohn*, presented on Figure 3.1, will be fully analysed to understand which main issues have to be adjusted, by developing a new model.

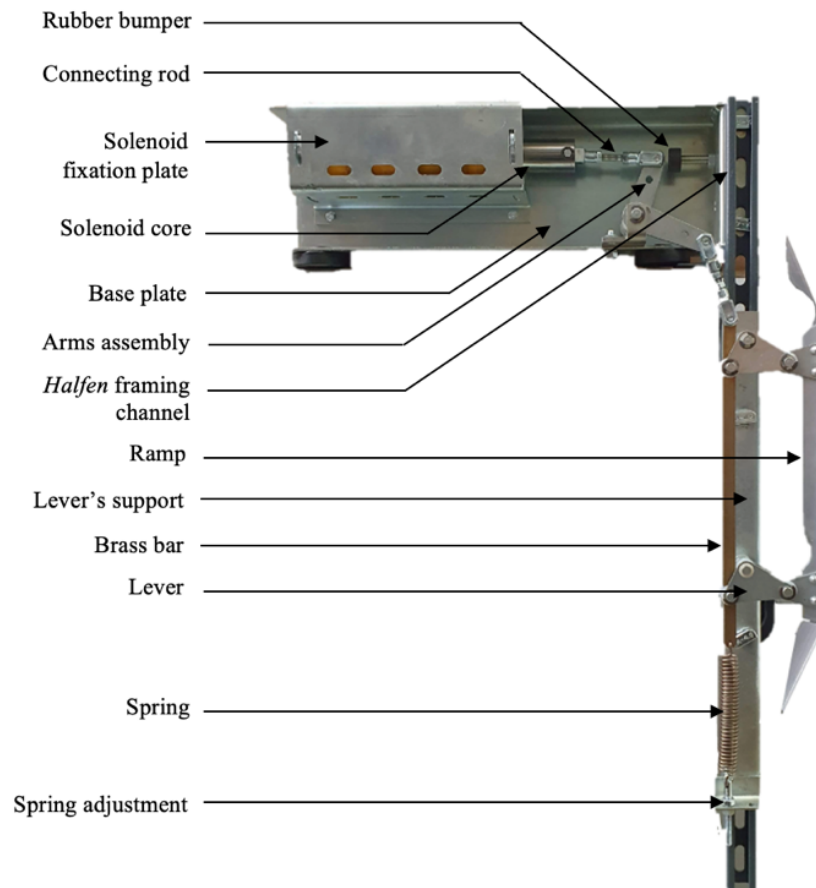


Figure 3.1 - The currently used Retractable Cam and its main parts.

Two issues arise from the “voice of the customer”. Relatively to the internal customers, the fitters, require a purpose to improve access to the spring for adjustment. On the other hand, common elevator users, representing the external costumers, would like to have a more comfortable solution in terms of noise, in particular when the ramp reaches its end of travel (completely open or completely closed). This problem is due to a large amount of kinetic energy, that is not dissipated smoothly, in a proper way. To solve the problem, the force produced by the solenoid and the spring will be calculated and compared to the minimum force needed to cause the movement. The current model analysis will then start by the kinematic study of the mechanism.

3.1 Kinematic Study

In this section, the geometry of the actual solution is studied, and the position vectors are calculated to understand the mechanics behind the retractable cam mechanism.

The sketch presented on Figure 3.2 was drawn on *Autodesk Inventor 2015* to illustrate the mechanism more clearly, where some points and angles have been indicated. The bodies were numbered to ease the analysis. Bodies 1 and 3 are connecting rods, body 2 is the arms assembly, 4 and 5 correspond to the levers, body 6 is the bar and 7 is the ramp.

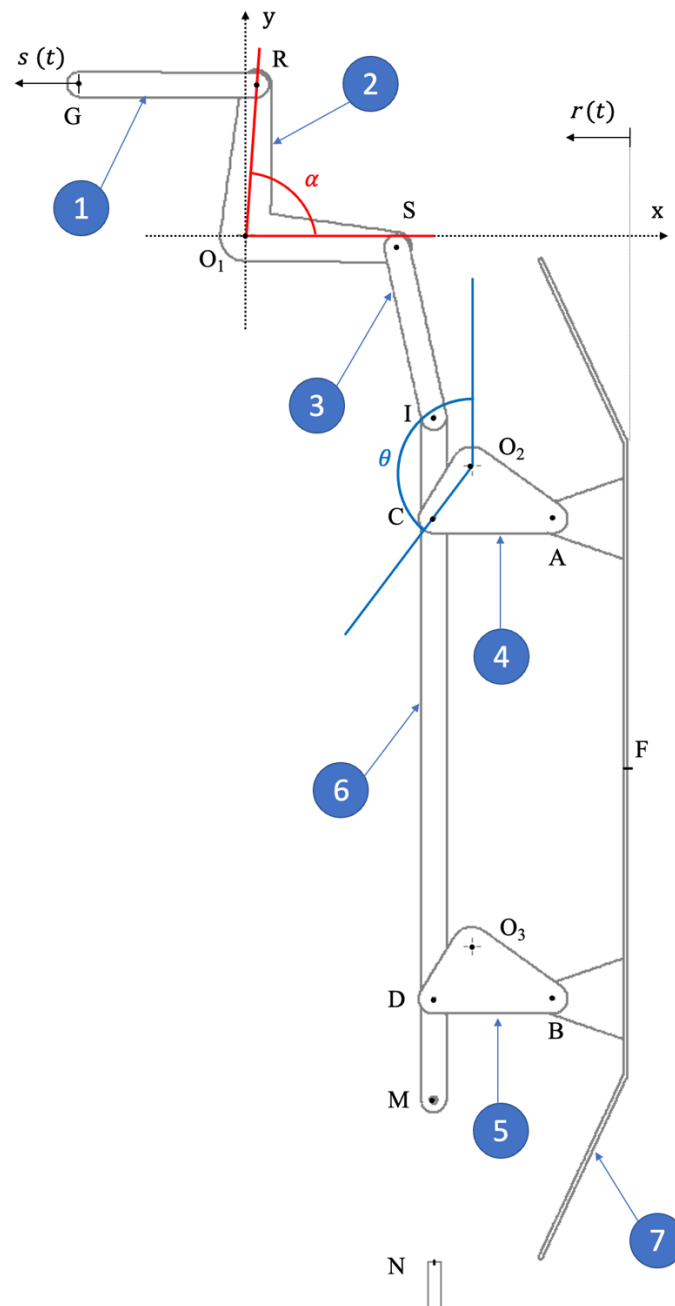


Figure 3.2 - Current solution diagram, indicating relevant points, angles, and bodies.

To proceed with kinematic analysis of the mechanism, two angles were defined: α and θ (annotated in red and blue, respectively, on Figure 3.2). α , varies directly with body 2 once it is defined between x axis and O_1R . The angle formed between the y axis and O_2C is named θ .

The points O_1 , O_2 , O_3 are points fixed to the retractable cam's structure that only allow rotation of the elements attached to them and are considered the origins of cartesian coordinate system one, two and three respectively. The spring links to body 6 through point M, and its other extremity is connected to point N, which is physically a threaded spindle fastened to the structure for spring pre-charge adjustment. Point G is where the solenoid armature is attached, so where its force is exerted, and the movement it describes is exclusively horizontal.

The dimensions have been taken from the technical drawings and the distances between two points from different parts were measured on the CAD assembly file, using the "Distance" tool, found on "Inspect" tab, on *Autodesk Inventor 2015*. The dimensions are presented on Table 9.

Table 9 - Distances between relevant points of the mechanism.

Dimension	Value /mm
$\overline{GR}$	71,99
$\overline{RO_1} = \overline{O_1S}$	60,00
$\overline{SI}$	70,72
$\overline{IC} = \overline{DM}$	40,00
$\overline{CD} = \overline{O_2O_3} = \overline{AB}$	190,00
$\overline{O_2C} = \overline{O_3D}$	25,40
$\overline{O_2A} = \overline{O_3B}$	38,00
$\overline{CA} = \overline{DB}$	47,00
$\overline{O_1O_2} - x$	89,50
$\overline{O_1O_2} - y$	91,25

Note: Point F is located on the surface of the ramp, right on its middle.

Physically, the distances $\overline{GR}$ and $\overline{SI}$ are defined by an assembly of two fork heads (DIN71752 -M6x12) and an M6 threaded rod, as illustrated on Figure 3.3, making them adjustable.



Figure 3.3 - Connecting rod.

The connecting rods adjusted lengths considered assure a proper functioning of the retractable cam and were measured physically, using a recently calibrated digital caliper – *Mitutoyo Digimatic* – featuring a resolution of 0,01 mm and an accuracy of $\pm 0,02$ mm. Three measurements were made, and the average distance can be found on Table 9.

The fact of having adjustable parts, results on an infinite number of configurations of the component, each with a different performance and amplitude of movements. The analysis performed is based on the values of Table 9.

Independently from the configuration, the normal operation cycle is the following:

1. The ramp is extended, while the solenoid is not energized.
2. When electrical current powers the solenoid, its core pushes point G to the left, causing the “L” shape part (arms assembly) to rotate counterclockwise, increasing angle α . The connecting rod pushes the brass bar up, to make the levers rotate (both levers rotate at a same rate), and consequently retract the ramp
3. The movement stops when the solenoid armature reaches its end-of stroke.

Having defined the angles and dimensions, the position vectors were calculated, and can be consulted on Annex A.

3.1.1 Relations between variables and mechanism amplitude

The mechanism in question has one degree of freedom, while the position vectors of Annex A are function of two different angles. Therefore, a relation between the two angles α and θ should be presented.

The method used to obtain that solution is through the vector $\vec{SI}$, which is the sum of the vectors $\vec{SO_1}$ and $\vec{O_1I}$ that depend on θ and α . Considering the $\vec{SI}$, norm fixed (using the value of Table 9), and attributing a value to θ , it is possible to calculate the corresponding α through the $\vec{SI}$ norm expression.

Using *Excel 2016*, values were attributed to θ , in intervals of 3° , and the corresponding angle α was calculated. Then polynomial regression fit of first and second order were calculated, and the equation presenting lower relative error will serve as an approximation of the theoretical expression of θ function of α .

The second order regression presented an inferior relative error of 0,3% so its expression (7) will be considered to relate α and θ (in radians).

$$\theta(\alpha) = 0,6189 \alpha^2 + 4,6749 \alpha - 8,1989 \quad (7)$$

NOTE: Given that the expression is based on a specific norm value for $\overrightarrow{SI}$ (0,07072 mm), it is only valid for this value.

The interval of values that an angle can assume is also fundamental to understand the retractable cam's movement. By observation, it was realized that the maximum absolute value that θ could assume is when the bar touches the nylon bushings supporting points O2 and O3, as shown on Figure 3.4. That position is adopted when the ramp is advanced and was named extended position.

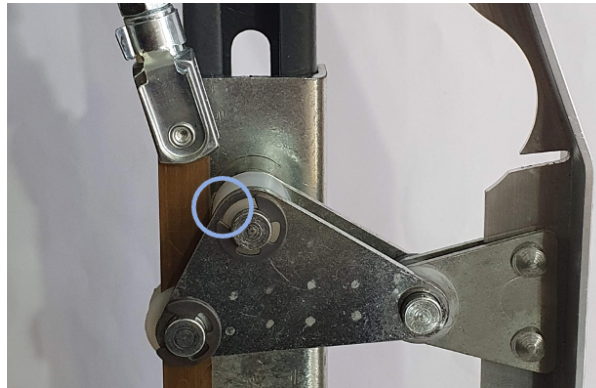


Figure 3.4 - Bar touching the bushing, ramp advanced position. The contact zone is highlighted in blue.

However, the successive impact of the bar on the bushings located in O₂ and O₃, will eventually cause damage and produce noise. For that reason, the ramp is considered fully extended, when the bar finds itself 2 mm from the bushing surface, by limiting the movement with the end-of-travel element that actuates near point R.

The angle θ for the extended position is named θ_e and can be calculated through a simple geometric relation according to Figure 3.5.

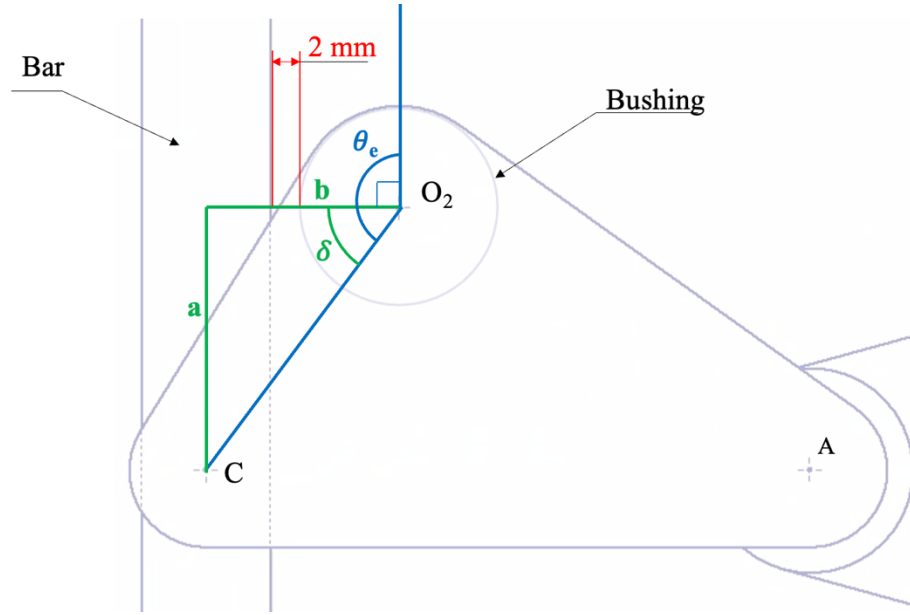


Figure 3.5 - Geometry used to calculate θ_e (2mm gap annotated in red).

Considering the bar's width equal to 10 mm, the radius of the bushing 8 mm and $\overline{O_2C}$ is 25,40 mm (see Table 9), the following equation allows to calculate θ_e :

$$\theta_e = -\frac{\pi}{2} - \delta = -\frac{\pi}{2} - \cos^{-1}\left(\frac{b}{\overline{O_2C}}\right) = \cos^{-1}\left(\frac{10/2 + 2 + 8}{25,40}\right) = -0,799\pi \text{ rad} \quad (8)$$

Note: The angle can be reduced by adjusting the cushion that actuates close to point R, when the ramp does not have to extend to its maximum to unlock the landing door locking system.

Using equation (7), α_e which is the value of α when the ramp is extended, can be calculated.

Considering θ_e equal to $-0,799\pi$ ($-143,80^\circ$) and $\overline{SI}$ measuring 0,07072 m, α_e measures $0,487\pi$ ($87,57^\circ$).

On the other hand, α and θ are also limited when the ramp is retracted (when the solenoid is actuated) by the same reason: the nylon washers located in points A and B might break because of recurrent impacts. The angles in the retracted position are α_r and θ_r . The procedure to calculate the two angles in this case is similar, using the geometry represented on Figure 3.6.

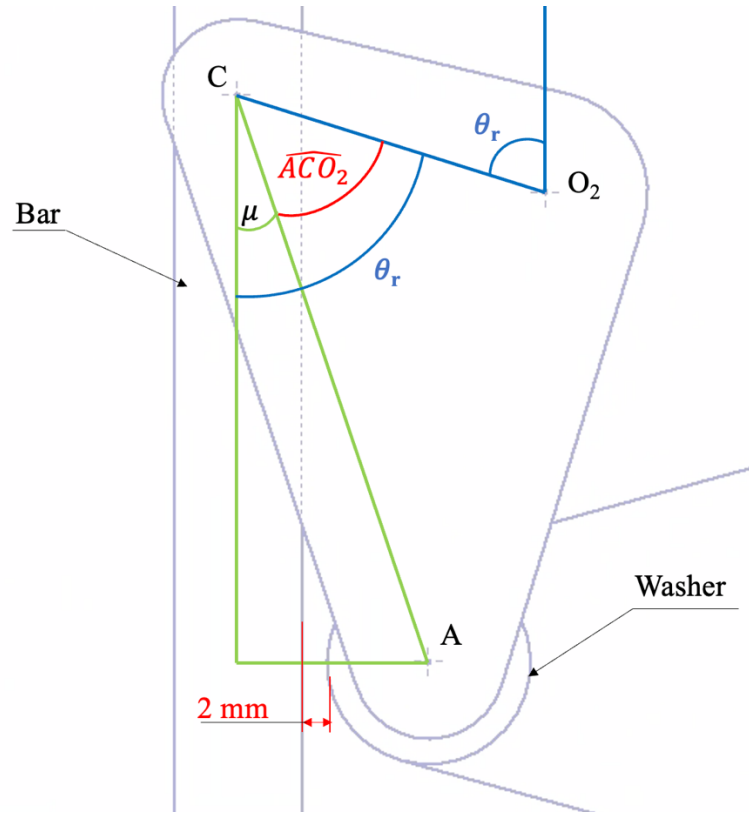


Figure 3.6 - Geometry used to calculate θ_r (2mm gap annotated in red).

Considering that washer's radius is 8 mm, the gap is 2 mm, the bar width is 10mm, $\widehat{ACO_2}$ is $0,293\pi$ rad, $\overline{CA}$ is 47,00mm and that the sum of μ , $\widehat{ACO_2}$ is equal to θ_r , equation (9) allows to calculate θ_r .

$$\theta_r = \mu + \widehat{ACO_2} (=)\theta_r = \sin^{-1}\left(\frac{5 + 2 + 8}{\overline{CA}}\right) + 0,293\pi = 0,402\pi \text{ rad} \quad (9)$$

Once θ_r is negative, its value is $-0,402\pi$ ($-72,42^\circ$), and the corresponding α , named α_r , is $0,644\pi$ ($115,93^\circ$), for $\bar{SI}$ measuring 0,07072 m.

The end-of-stroke element that limits α_e and θ_e angles is a rubber damper, while the opposite extreme position (α_r and θ_r) is limited by the end of the solenoid stroke.

One of the most relevant mechanism parameters is the ramp position and its movement amplitude. Since it is fixed to points A and B, its position, r , depends directly on θ angle, as shown on equation (10):

$$r(\theta) = \overline{O_2A} \times [\sin(\theta_e - \beta^1) - \sin(\theta - \beta)] \quad (10)$$

Having calculated θ limits, it is now possible to calculate the ramp maximum travel, by replacing θ by θ_r , as written on equation (11). This value is given by the difference of x coordinates of point A on both extended and retracted positions.

$$\Delta r_{max} = \overline{O_2A} \times [\sin(\theta_e - \beta) - \sin(\theta_r - \beta)] = 41,2mm \quad (11)$$

Therefore, the ramp position varies from $r_e = 0$ mm when extended and $r_r = 41,2$ mm when retracted.

3.1.2 The spring movement and elongation

Before characterizing the spring, it is relevant to ascertain how it interacts with the mechanism, considering that one extremity is fixed (point N), after being adjusted, and the other, point M, is part of the bar so describes a circular movement.

As determined on Annex A:

$$\overrightarrow{NM} = \begin{pmatrix} \overline{O_2C} (\sin(\theta) - \sin(\theta_e)) \\ (l_0 + \varepsilon_0) + \overline{O_2C} (\cos(\theta) - \cos(\theta_e)) \\ 0 \end{pmatrix} \quad (12)$$

Where:

l_0 , is the distance between spring hooks, with the spring being unloaded.

ε_0 , is the spring pre-charge elongation.

An angle named ρ is described by the spring axis (considered to be NM) and the y axis. When the ramp is extended the spring is vertical, when not, it forms an angle ρ . The variation of ρ angle for the extended and retracted positions can be observed on Figure 3.7, and its value calculated through equation (13).

¹ β is equivalent to angle $\widehat{CO_2A}$.

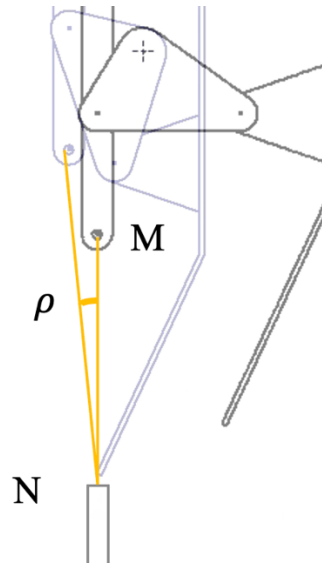


Figure 3.7 - The variation of ρ , depending on the mechanism position is highlighted in orange. The mechanism represented in blue is retracted, and in black is extended.

$$\rho = \tan^{-1} \left(\frac{NM^x}{NM^y} \right) \quad (13)$$

Where:

NM^x , is the component of $\overrightarrow{NM}$ on x axis.

NM^y , is the component of $\overrightarrow{NM}$ on y axis.

ρ , is the angle between the spring axis and the y axis.

Considering a length inside hooks (ℓ_0) of 87,62mm and a pre-charge elongation (ε_0) of 15,31mm (as measured), during normal operation, ρ assumes values between 0 and $-0,027\pi$ rad ($-4,82^\circ$).

Regarding the spring elongation (Δl), its value can be calculated for a given mechanism position by calculating the norm of vector $\overrightarrow{NM}$ and subtracting l_0 , as illustrated on Figure 3.8

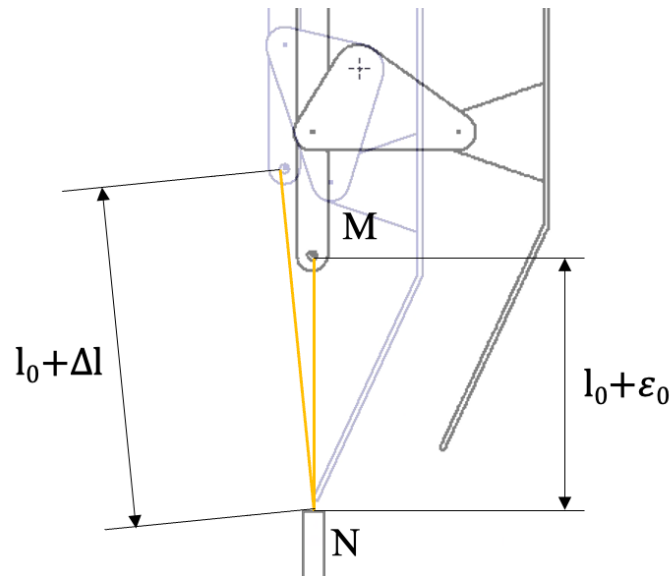


Figure 3.8 – Diagram representing spring elongation for fully extended and retracted positions.

The minimum elongation of the spring is on the extended position and is equal to the pre-charge elongation, when the ramp is extended, and the maximum can be found by calculating the norm with $\theta = \theta_r$ and subtracting l_0 . Thus, Δl varies from 15,31 mm to 43,8 mm.

3.2 Spring Characterization

The spring is a key element in the retractable cam, given that it is responsible to extend the ramp and actuate the landing door locking system. Therefore, it is essential to obtain accurate information about its characteristics such as dimensions, and stiffness that will be essential to calculate its force on section 3.3.

In this specific case the spring produces elastic force when subjected to traction, meaning it is an extension spring. This choice of this type of spring is due to its stability, given that buckling does not happen, unlike on compression springs.

To measure the spring dimensions, a calibrated digital caliper – *Mitutoyo Digimatic* – featuring a resolution of 0,01 mm and an accuracy of $\pm 0,02$ mm was used. The measured dimensions were the wire diameter, inner diameter, outer diameter, and length inside hooks. The number of total coils was also a factor to consider.

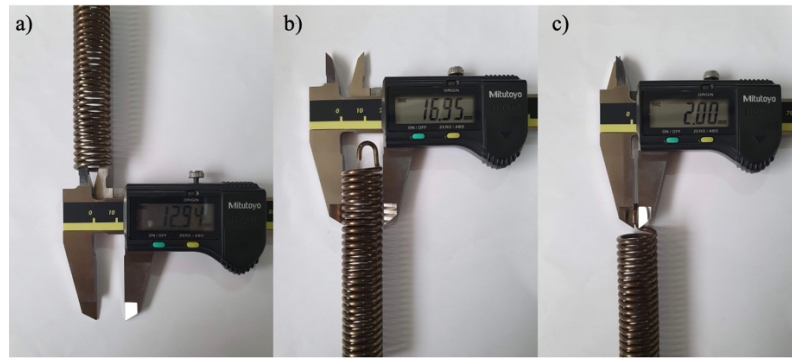


Figure 3.9 - Measurements of spring dimensions with a digital caliper. a) inner diameter measurement; b) outer diameter measurement and c) wire diameter measurement.

The results are presented on Table 10:

Table 10 – Currently used spring dimensions.

Dimension	Symbol	M1	M2	M3	Average Value
Number of total turns	n	-	-	-	30
Wire diameter /mm	ϕ_w	1,99	2	2	2
Outer diameter /mm	ϕ_o	16,95	16,96	16,95	16,95
Inner diameter /mm	ϕ_i	12,97	12,94	12,95	12,95
Length inside hooks /mm	ℓ_0	87,63	87,6	87,62	87,62

The most relevant spring property to the current solution study is its stiffness coefficient, k , which allows to calculate the force exerted by the spring when multiplied for a given elongation. To obtain this value the spring was subjected to an axial traction test where the force was measured with a digital dynamometer (*Chantillon DXF - 200* with an accuracy of $\pm 0,5\%$ of full scale) and its elongation with a linear gage (*Mitutoyo EH – 102 D* featuring a range of 50 mm and an accuracy of 0,01 mm) connected to a two-axis display with two decimal places (*Mitutoyo LGD – 1050L*).

The test material was mounted carefully in a manner the spring is only tractioned in the direction of its axis. Firstly, a hook was attached to the dynamometer and then it was fixed in the movable part (car) of a support that allows stable and controlled linear movement. The support moves by turning a worm screw, and by dislocating the car, the spring is elongated. One of the spring endings was fixed in a static part of the support, and the other attached to the dynamometer's hook. To measure the spring's elongation, the linear gage was fixed on a structure, with its measurement tip in contact with the movable part of the support and connected to the display.



Figure 3.10 - Axial traction test installation with all the measurement devices and test sample in place.

The test was performed following the subsequent steps:

1. In a position where the spring is not exerting force, turn on the measuring devices, and set the dynamometer's value to zero.
2. By turning the support's worm screw, move the support's car to a position where the spring is just about to exert force.
3. Set the linear gage to zero.
4. Start elongating the spring by dislocating the car. Register the force value displayed on the dynamometer every interval of 2,5 mm in elongation.
5. Once the elongation reaches 45 mm (close to linear gage's end of travel), return to step 2. and do the procedure three time.

With the results obtained a linear regression was calculated to determine the stiffness coefficient. The values are displayed on Table 11:

Table 11 - Spring elongation and force measurements to determine stiffness coefficient, k .

Δl (mm)	F_s (N) - experimentally				All data		17,5 mm $\leq \Delta l \leq 45$ mm	
	m1	m2	m3	Average	F_{Spring} theoretical	Rel. Error	F_{Spring} theoretical	Rel. Error
0	0	0	0	0,00	0	0%		
2,5	3	3	4	3,33	4,01	16,97%		
5	7	7	7	7,00	8,03	12,82%		
7,5	11	11	11	11,00	12,04	8,66%		
10	15	15	15	15,00	16,06	6,59%		
12,5	19	19	19	19,00	20,07	5,34%		
15	23	23	24	23,33	24,09	3,13%		
17,5	27	27	28	27,33	28,10	2,73%	28,17	2,97%
20	31	31	32	31,33	32,12	2,44%	32,19	2,67%
22,5	36	35	36	35,67	36,13	1,28%	36,22	1,52%
25	40	39	40	39,67	40,15	1,19%	40,24	1,43%
27,5	44	44	44	44,00	44,16	0,36%	44,27	0,60%
30	48	48	48	48,00	48,17	0,36%	48,29	0,60%
32,5	52	52	52	52,00	52,19	0,36%	52,32	0,60%
35	56	57	57	56,67	56,20	0,82%	56,34	0,58%
37,5	60	60	61	60,33	60,22	0,19%	60,36	0,05%
40	65	65	65	65,00	64,23	1,20%	64,39	0,95%
42,5	69	69	69	69,00	68,25	1,10%	68,41	0,86%
45	73	73	73	73,00	72,26	1,02%	72,44	0,78%
						3,50%		1,14%

The values “m1”, “m2” and “m3” represent the three measurements for each spring elongation value.

The first linear regression, named “all data” was calculated considering the force for all elongation values, from 0,0 to 45,0 mm and is illustrated on the upper graphic on Figure 3.11. On Table 11 the spring force was calculated using the regression expression as well as the relative error. For this case, the medium relative error was 3,50%, and the value for the spring’s stiffness coefficient was 1,6058 N/mm.

With the purpose of reducing the error, a second linear regression, “17,5 $\leq \Delta l \leq 45$ ”, was calculated, only considering the elongation values where the relative error value is lower than 3%, this means considering elongations from 17,5 to 45 mm. The spring force values on the 8th column of Table 11 were calculated using the expression of the regression of the lower graphic of Figure 3.11. The medium relative error obtained was 1,14% and the stiffness coefficient 1,6097 N/mm.

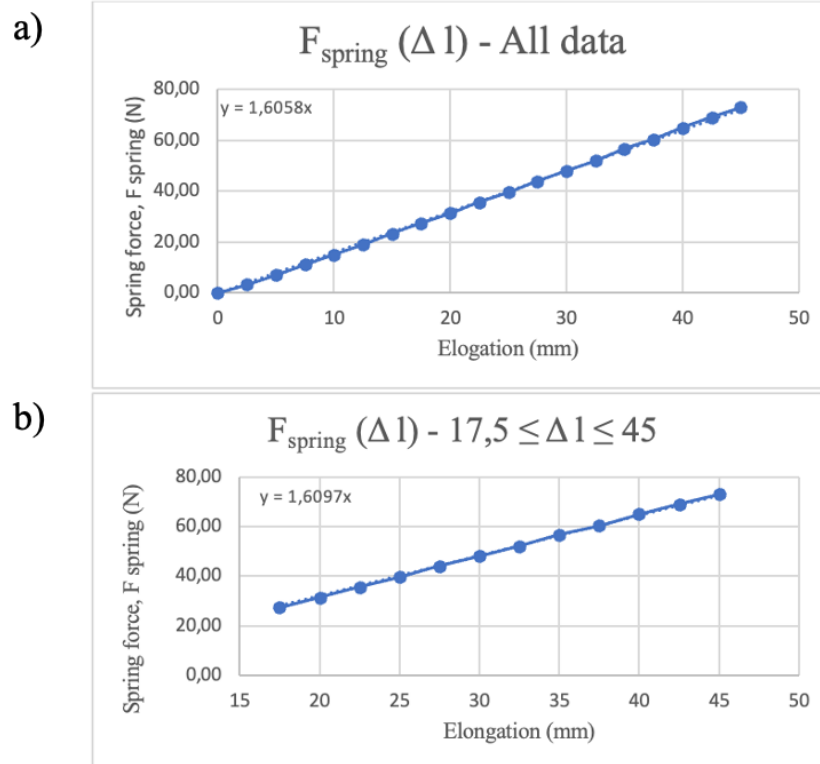


Figure 3.11 - Graphics of spring force function of elongation. a) considering all data; b) considering elongation values from 17,5 to 45,0 mm.

The value of 1,6097 N/mm will be used for all the studies related to that specific spring.

To adjust the spring force, a pre-charge can be imposed. This means that in the less extended position, the spring is already elongated. In the case of the current solution, the pre-charge can be adjusted, by fastening or unfastening the threaded pin where the spring is fixed, on the lower part of the retractable cam. The minimum possible is 0 mm and the maximum pre-charge is 15,31 mm, having been measured with a caliper.



Figure 3.12 - Spring pre-charge adjustment.

3.3 Static Equilibrium

The cinematic analysis is not enough to characterize the mechanics behind the retractable cam. The study of the forces involved on the extension and retraction of the ramp are fundamental.

The extension force (force exerted by the ramp), F_{ramp} , is caused by the spring's elastic force and is a major factor regarding a retractable cam once it must be enough to push the landing door system lever to unlock the door. The force produced by the spring, F_{spring} , depends on two factors, its elongation and stiffness coefficient. The elongation was analysed previously on 3.1.2 and the stiffness coefficient on 3.2. Equation (14) allows to calculate F_{spring} :

$$F_{\text{spring}} = \Delta l \cdot k \quad (14)$$

Where:

Δl , is the spring elongation, including pre-charge elongation [mm]

k , is the spring's stiffness coefficient [N/mm].

The force exerted by the spring varies from 24,6 N while the ramp is fully extended, and 70,5 N when retracted (considering a pre-charge elongation of 15,31 mm)

Regarding the retraction force, this one is provided by the solenoid and aims to counter the spring force causing the ramp to retract.

As it is expected, the force exerted by the ramp depends on its position, giving that it is influenced by the spring elongation. Therefore, also the solenoid must exert a sufficient force throughout the whole movement when the ramp is to be retracted.

The forces involved on the movements are calculated in three relevant situations, described in the following sub-sections.

3.3.1 Situation 1: Ramp extended; solenoid not actuated

The first important situation is when the ramp is extended, pushing the locking mechanism lever to its maximum, achieving mechanical equilibrium. In this case, the locking mechanism lever force exerted in the ramp (F_{ext}) is the reaction of the force that the ramp exerts on it (F_{ramp}). Therefore, the modulus is the same but with opposite direction.

In this scenario, once the solenoid is not energized, only two forces are relevant (excluding the reactions on the support shafts): The spring force and the external force (F_{ext}) exerted by the locking system lever on the ramp.

As there are many locking system models, it can happen that the ramp does not need to fully extend to completely push the locking system lever to its maximum.

Another relevant aspect that must be considered is since when the ramp applies force in the landing door locking system lever. This question arises from the fact that a minimum gap has to be defined so when the car is traveling the retractable cam's ramp (retracted) does not touch the locking system levers of each floor. This gap must be at least 10mm, as illustrated on Figure 3.13.

For this reason, on the first 10 mm of extension (since the retracted position), the only forces that the spring must overcome are the inertias and friction between parts, so there is extension movement. Having said that, only from $r = 0\text{mm}$ and $r = 31,2\text{mm}$ the ramp force is truly relevant to push the locking system lever.

Therefore, the ramp force was calculated for the whole range of positions where force is needed to push the lever (from $r = 0\text{ mm}$, ramp fully extended, to $r = 31,2\text{ mm}$), as displayed on Figure 3.15. The ramp position is given by r and is considered fully extended when $r = 0$, and fully retracted when $r = 41,2\text{mm}$.

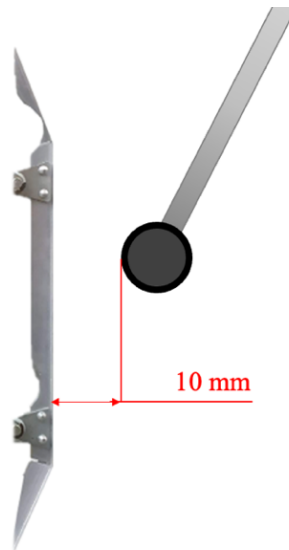


Figure 3.13 - Minimum distance between retracted ramp and unactuated lever of landing door locking system.

The values were obtained considering the maximum pre-charge elongation of 15,31mm and solving the static equilibrium equations (15) to (23):

$$\sum F^x = 0(=) R_{O2}^x + R_{O3}^x + F_{spring}^x + F_{ext} = 0 \quad (15)$$

$$\sum F^y = 0(=) R_{O2}^y + R_{O3}^y + F_{spring}^y = 0 \quad (16)$$

$$\sum M_{O2}^z = 0(=) O_2 M^x \cdot F_{spring}^y - O_2 M^y \cdot F_{spring}^x - O_2 F^y \cdot F_{ext} - O_2 O_3^y \cdot R_{O3}^x = 0 \quad (17)$$

$$\sum F^x = 0(=) R_{O2}^x + R_C^x + \frac{F_{ext}}{2} = 0 \quad (18)$$

$$\sum F^y = 0(=) R_{O2}^y + R_C^y = 0 \quad (19)$$

$$\sum M_{O2}^z = 0(=) O_2 C^x \cdot R_C^y - O_2 C^y \cdot R_C^x - O_2 A^y \cdot \frac{F_{ext}}{2} = 0 \quad (20)$$

$$\sum F^x = 0(=) R_D^x + R_C^x + F_{spring}^x = 0 \quad (21)$$

$$\sum F^y = 0(=) R_D^y + R_C^y + F_{spring}^y = 0 \quad (22)$$

$$\sum M_{O2}^z = 0(=) -CD^y \cdot R_D^x - CM^y \cdot F_{spring}^x = 0 \quad (23)$$

The reactions on point I, are not relevant, once this it is a static and this point is indirectly connected to the solenoid's core, which is not actuated at this moment. The diagram associated to equations (16) to (23) is on Figure 3.14.

Equations (15) to (17) are based on diagram a) on Figure 3.14. and the diagram b) and c), corresponding to bodies 4 and 5, are associated to equations (18) to (20) and (21) to (23), respectively.

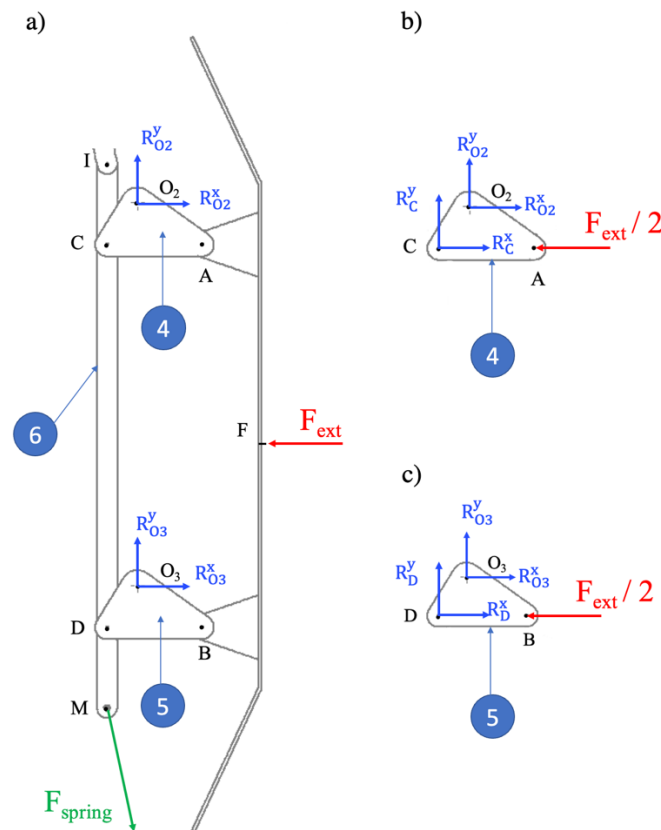


Figure 3.14 - Diagrams used to formulate equilibrium equations to calculate F_{ext} . The reactions are represented in blue, the external force F_{ext} in red, and the spring force F_{spring} in green.

In this system of equations, the unknowns are the reactions in points O_2 , O_3 , C and D, and the external force (F_{ext}).

The results are represented graphically on Figure 3.15. F_{ramp} has the opposite direction, so the opposite signal of F_{ext} .

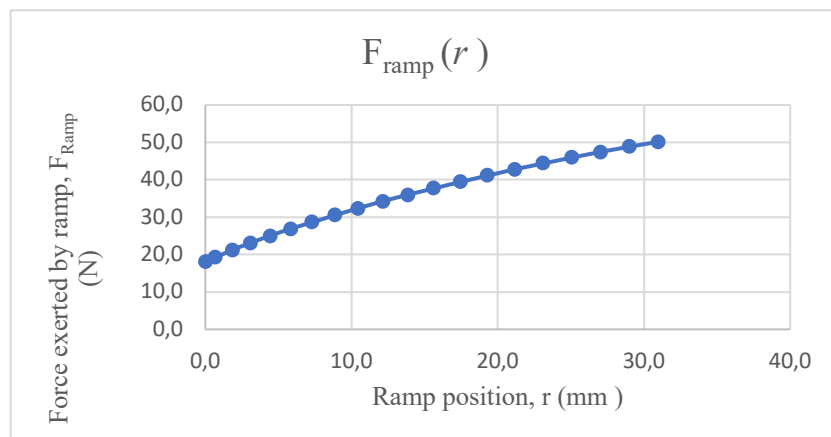


Figure 3.15 - The force exerted by the current solution's ramp as function of its position, from $r = 0$ mm to $r = 31,2$ mm.

3.3.2 Situation 2: Ramp extended, beginning of retraction movement

The second situation succeeds the first and occurs when the ramp is extended and suddenly the solenoid is actuated to initiate the retraction movement. To start the movement, the solenoid must be able to push the bar (body 6 on Figure 3.2, on page 43) up, by exerting a greater force than the sum of spring's, inertias, and static friction forces, at the first instant when the solenoid is actuated. In practice, the landing door locking system lever force would be favorable for this movement, nevertheless, as its value is not possible to estimate, it is considered zero, making this the worst-case scenario (in terms of solenoid force needed). The ramp can be in various positions and should be able to retract in all.

The first step was to calculate the minimum value of the reaction on point I (R_I) on the bar, required to start the retraction movement (not considering inertia and static friction) for several ramp positions. Thus, R_I in x and y components was calculated using the mechanical equilibrium equations (24) to (34):

$$\sum F^x = 0(=) R_{O2}^x + R_{O3}^x + F_{spring}^x + R_I^x = 0 \quad (24)$$

$$\sum F^y = 0(=) R_{O2}^y + R_{O3}^y + F_{spring}^y + R_I^y = 0 \quad (25)$$

$$\sum M_{O2}^z = 0(=) O_2 M^x \cdot F_{spring}^y - O_2 M^y \cdot F_{spring}^x - O_2 O_3^y \cdot R_{O3}^x + O_2 I^x \cdot R_I^y - O_2 I^y \cdot R_I^x = 0 \quad (26)$$

$$\sum F^x = 0(=) R_{O2}^x + R_C^x = 0 \quad (27)$$

$$\sum F^y = 0(=) R_{O2}^y + R_C^y = 0 \quad (28)$$

$$\sum M_{O2}^z = 0(=) O_2 C^x \cdot R_C^y - O_2 C^y \cdot R_C^x = 0 \quad (29)$$

$$\sum F^x = 0(=) R_{O3}^x + R_D^x = 0 \quad (30)$$

$$\sum F^y = 0(=) R_{O3}^y + R_D^y = 0 \quad (31)$$

$$\sum M_{O3}^z = 0(=) O_3 D^x \cdot R_D^y - O_3 D^y \cdot R_D^x = 0 \quad (32)$$

$$R_I \cos \omega^2 - R_I^y = 0 \quad (33)$$

$$R_I \sin \omega - R_I^x = 0 \quad (34)$$

² ω , is the angle between y axis and vector $\vec{IS}$.

The diagrams related to the equilibrium equations (24) to (34) are illustrated on Figure 3.16.

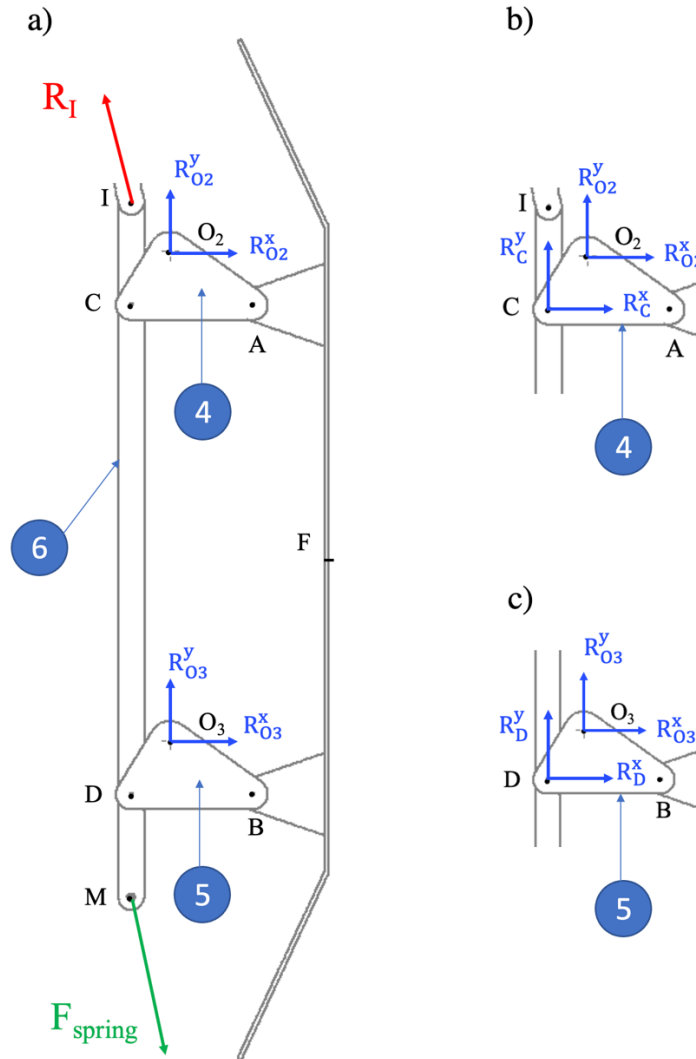


Figure 3.16 - Diagrams used to formulate equilibrium equations to calculate reaction in I. Reactions in I are represented in red, the remaining reactions are represented in blue, and the spring force F_{spring} in green.

Equations (24) to (26) refer to diagram a) of Figure 3.16, considering bodies 4,5 and 6. Equations (27) to (29) and (30) to (32) correspond to diagrams b) and c) of Figure 3.16, respectively.

In the previous equations, ω and the spring force are known, and the unknowns are the reactions. Having calculated R_I^x and R_I^y another mechanical equilibrium must be established on the arms assembly, which transmits the solenoid force to the SI connection rod. The reaction in S has the same modulus than the reaction in I, but in opposite direction. The mechanical equilibrium equations of the assembly arms are equations (35) to (38):

$$\sum F^x = 0(=) R_{O_1}^x + R_R^x + R_S^x = 0 \quad (35)$$

$$\sum F^y = 0(=) R_{O_1}^y + R_R^y + R_S^y = 0 \quad (36)$$

$$\sum M_{O_1}^z = 0(=) O_1 R^x \cdot R_R^y - O_1 R^y \cdot R_R^x + O_1 S^x \cdot R_S^y - O_1 S^y \cdot R_S^x = 0 \quad (37)$$

$$\frac{R_R^x \cdot \tan \varphi^3}{R_R^y} - 1 = 0 \quad (38)$$

The diagram related to the equilibrium equations (35) to (38) are illustrated on Figure 3.17.

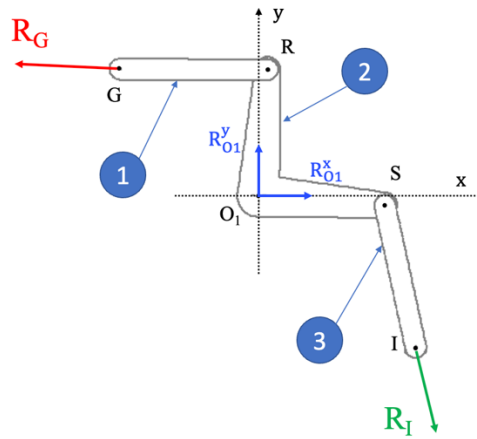


Figure 3.17 - Diagram used to formulate equilibrium equations to calculate the reactions in G. Reactions in G are represented in red, the remaining reactions are represented in blue, and reactions in I in green.

Equations (35) to (37) refer to the diagram presented on Figure 3.17, considering the bodies 1, 2 and 3.

Once the solenoid armature is connected to point G and only exerts force horizontally, R_G^x corresponds to the minimum force that the solenoid must exert to start the retraction movement ($F_{Solenoid}$). The minimum solenoid force required to start the movement in a certain position is described by Figure 3.18.

³ φ , is the angle between x axis and vector $\overrightarrow{GR}$.

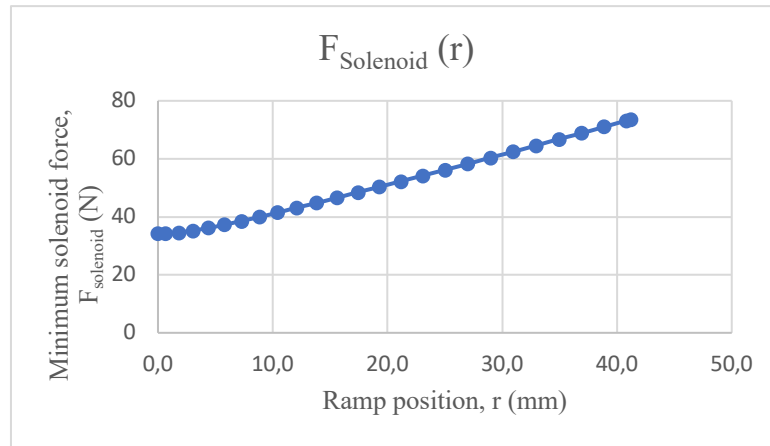


Figure 3.18 - Minimum solenoid force required to start retraction movement as function of the ramp position.

By observing Figure 3.18, the solenoid must exert a force of approximately 35N to retract the ramp, not considering the inertia and friction effects.

3.3.3 Situation 3: Solenoid holds the ramp retracted

After the retraction movement being completed the force produced by the solenoid is responsible for maintaining the retracted position.

The force the solenoid must exert in order to maintain the ramp retracted, can be consulted on Figure 3.18 for $r = 41,2$ mm. 73,57 N are needed to maintain the retracted position.

3.4 Solenoid Characterization

The solenoid is a crucial element of the retractable cam once it is responsible for producing the force that will cause the ramp to retract. Therefore, it will be subjected to a performance and a power consumption analysis.

3.4.1 Solenoid Force Test

After having calculated the force exerted by the spring and having performed the static study of the forces in the mechanism, it is important to measure experimentally the force produced when the solenoid is actuated.

To perform the test, the first step was to develop an electric circuit composed by a diode rectifier bridge and a switch. This device's purpose is to convert the 230V AC from the electrical grid to the nominal current of the solenoid, which is 180V DC, through a full-wave

rectification. To power the solenoid, its plug must be connected to diode rectifier bridge, which is already connected to the electrical grid. When power switch position is “ON”, the solenoid is actuated, and its core is retracted.

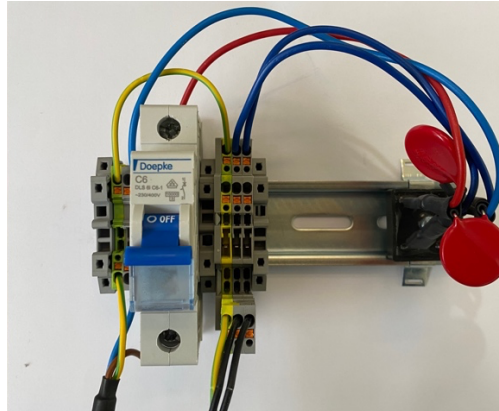


Figure 3.19 – Diode rectifier bridge with switch.

Secondly, the solenoid’s core was attached to a digital dynamometer (*Chantillon DXF* - 200 with an accuracy of $\pm 0,5\%$ of full scale), using a steel cable. Both the retractable cam and the dynamometer were fixed to a structure, so there is no relative movement. Thereby, when the solenoid is actuated, the core retracts, pulling the steel cable and transmitting the force to the dynamometer’s hook. Consequently, the force value displayed on the dynamometer is the pulling force of the solenoid’s core.

Once the testing material is set, the dynamometer is turned on, while the cable is not under tension. The following step is to actuate the solenoid and annotate the force value, then turn the switch off. This last step is repeated three times, resulting in three measurements. The obtained values correspond to the end of stroke force and are displayed on Table 12.

Table 12 - Solenoid force measurement values.

	Measurement 1	Measurement 2	Measurement 3	Average
F_m (N)	148	149	149	148,67

Comparing those values to the minimum force required for the solenoid to maintain the retracted position, calculated on sub-section 3.3.3, even not considering the losses caused by friction of moving parts (such as sleeve nuts) and inertia, it is possible to conclude that the solenoid used is over dimensioned for this purpose.

3.4.2 Solenoid Power Consumption

In order to compare the solution currently used to the one that is to be developed, the electrical power consumption was also calculated.

When installed on an elevator, the solenoid's electrical supply occurs the same way: a diode rectifier bridge converts the 230 V from the electrical grid to direct current that powers the solenoid. Therefore, the power consumption was measured before and after the full-wave rectification, in other words, before and after the diode rectifier bridge.

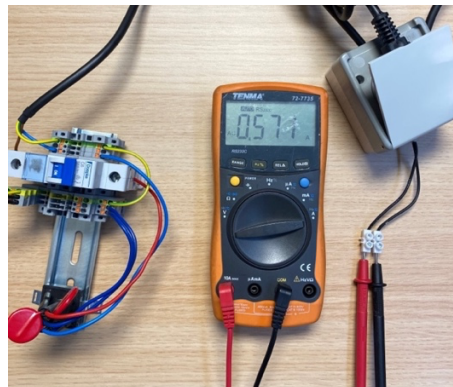


Figure 3.20 – Multimeter measuring the electric current before rectification (AC).

The electric power was calculated using equation (39):

$$P = U \cdot I \quad (39)$$

Both electric tension and current were measured using a multimeter (*Tenma 72-7735*), three times, as shown on Table 13.

Table 13 - Electric tension and current measurements for solenoid power consumption

		Measurement 1	Measurement 2	Measurement 3	Average
AC	U [V]	234	234	234	234
	I [A]	0,574	0,578	0,575	0,576
DC	U [V]	207,5	207,3	207,7	207,5
	I [A]	0,530	0,535	0,533	0,533

Using the data available on Table 13 and equation (39), it is possible to calculate the power consumed from the grid (AC) as well as the power the solenoid consumes (DC). The results are exposed on Table 14.

Table 14 – Power consumed from the grid and power consumed by the solenoid.

	P [W]
AC	134,784
DC	110,598

As indicated on Table 14, it is noticeable that the power consumed from the grid is higher than the power that the solenoid consumes. This difference is due to the fact that the diode rectifier bridge's electric efficiency is lower than 100% and can be calculated using equation (40).

$$\eta_{Diode\ rectifier\ bridge} = \frac{P_{consumed\ by\ the\ solenoid}}{P_{consumed\ from\ grid}} = \frac{110,598}{134,784} = 82,056\% \quad (40)$$

3.5 Production

The whole Retractable Cam assembly is composed by of 118 components. Aiming to organize all the parts used by the company, Schmitt + Sohn decided to categorize them in three groups:

- Production Parts
- Standardized Parts
- Purchase Parts

Production parts, refers to parts developed by the “Research and Development” department in *Schmitt + Sohn* itself. The production can occur internally in two different facilities, or, in case the production technology needed is not available there, is outsourced, based on the documents and drawings provided by *Schmitt + Sohn*.

Standardized parts comprise bolts, washers, nuts, and other components, normally described by the German standards (DIN).

The parts acquired from external suppliers but are neither standardised nor developed by Schmitt + Sohn are considered purchase parts.

This section consists of identifying the materials, the manufacturing technologies and the constructive solutions used to produce the retractable cam. The information needed about the parts production was acquired from the technical drawings, and by talking with qualified personal working on the company.

3.5.1 Retractable cam's structure

The component's structure is responsible for holding the ramp mechanism as well as the solenoid and the spring, therefore robustness is a must, and the ramp position should be easily adjustable. The structure is basically an assembly of four parts, excluding fasteners, described below. Table 15 summarizes the information relative to the raw material used and technologies adopted to produce each.

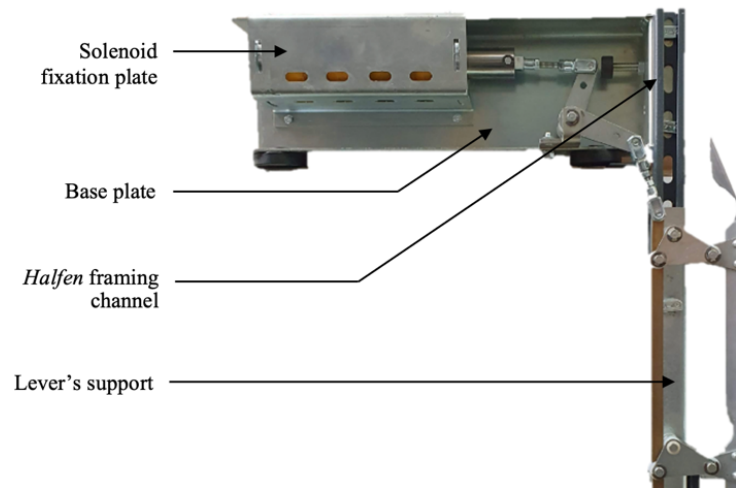


Figure 3.21 - Retractable cam's structure elements, excluding fasteners.

The retractable cam base plate, which is where the component is fixed to the elevator's car, is composed by two laser cut and bent EN 10130 DC01 2,5mm steel sheets, welded together by manual MAG welding. Two other parts are fixed to that assembly: the solenoid fixation sheet and a *Halfen* framing channel.

The solenoid fixation plate is produced recurring to the same technologies used to produce the base sheets. It is fixed with 4 DIN 933 M4x10 bolts.

The *Halfen* framing channel is a purchase item, and has originally one meter long, so it must be cut using an industrial metal cutting circular saw, to be mounted on the retractable cam with the correct length. Afterwards, the part is coated with electrostatic paint for protection against corrosion, and then fixed to the base plate with two M8 *Halfen* bolts. This *Halfen* framing channel provides an adjustable fixation for the lever mechanism support.

The lever's support is a laser cut 2 mm EN 10130 DC01 sheet, that is then bent in an "U" shape, recurring to a hydraulic bending machine. Two 9S20 shafts, machined on a lathe, are secured to it with a chisel, whose function is to hold the levers of the mechanism. An "L" shape steel sheet is also welded to the lever's support to fix the spring support.

Table 15 - Structure component's material and production technologies (fasteners and other standardized parts were not considered).

Part	Material	Production technologies					
		Laser cut	Welding	Bending	Zinc coated	Electrostatic paint	Saw cut
Base plate	EN 10130-DC01 ($t=2,5$ mm)	x	x	x	x		
Solenoid fixation plate	EN 10130-DC01 ($t=2,5$ mm)	x		x	x		
<i>Halfen</i> framing channel	N/A	No details from supplier				x	x
Lever's support	EN 10130-DC01 ($t=2$ mm)	x	x	x	x		

As it can be observed on Table 15, all components are coated to protect against corrosion. The parts that are laser cut and bent are produced on the *Schmitt 2* facility, nevertheless they are then galvanized using zinc electrodeposition technology, described on standard ISO 2081:2008. The coating thickness should be between 5 and 10 μm .

3.5.2 The mechanism

The mechanism is composed mainly by five moving elements: the levers, the ramp, the arms assembly, the brass bar, and the connecting rods. The part's movement is caused either by the solenoid and the spring, which are connected to.

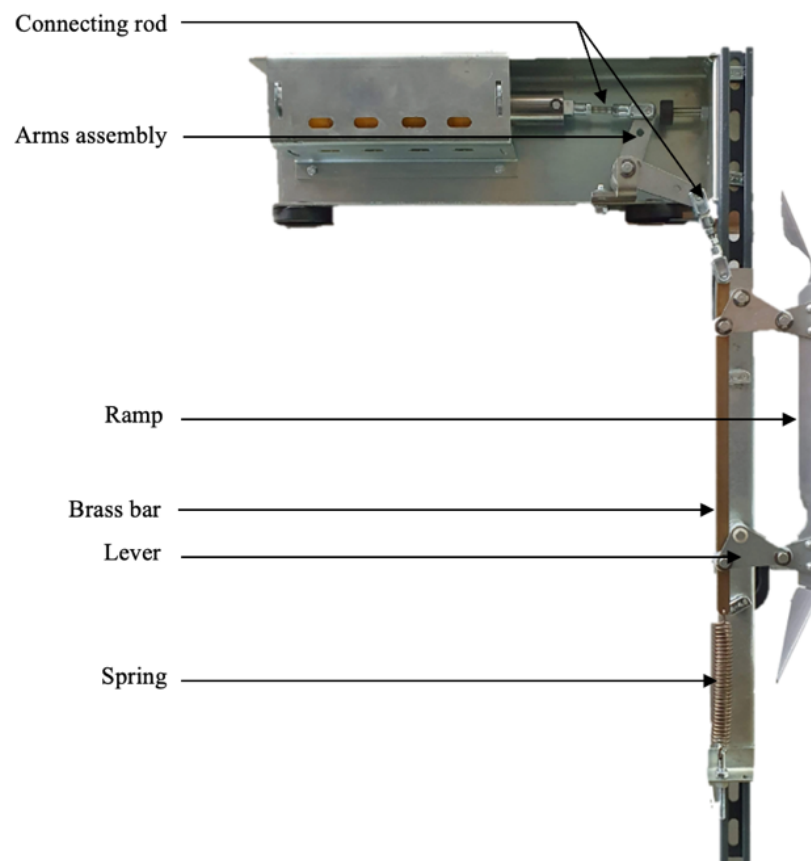


Figure 3.22 - Retractable cam's mechanism main elements.

The levers are basically two identical laser cut triangular pieces of 2mm EN 10130 DC01, separated by a lathe machined nylon bushing. The ramp and the bar are fixed to the levers through 9S20 steel machined spindles, locked by retaining washers (DIN 6799). While the levers are produced internally, the bushings and machined shafts are furnished by an external supplier.

The bar has a considerable impact on the retractable cam's production cost once it is produced in brass for machining (CuZn39Pb2) and is totally produced by milling by an external supplier. The reason why this alloy has been chosen to this application is its low friction coefficient, and the fact that at the time of the part development, this material had a much lower price. It is an essential element for this solution given that it directly transmits the spring force and the solenoid force to the levers.

A simple element that corresponds to more than 11,5% of the production cost of the currently used retractable cam is the ramp. This crucial part is produced from a “T” shape aluminium extruded profile, that is then cut by craft in a complex shape, and then bent by hand. Also, four holes are drilled to fix four plates that connect the ramp to the levers. The plates are permanently fixed to the ramp by mean of carnations applied manually and hold to the 9s20 spindles.

To transfer the horizontal movement of the solenoid’s core, to a vertical pushing force on the bar, capable of retracting the ramp, a welded assembly formed by two identical arms and a shaft is used. The two arms are produced from a laser cut steel sheet (EN10025-S235JR) and the shaft is manufactured in 9S20 steel by turning. The three parts are then welded together in a manner that the two arms form a right angle, and then the assembly is zinc coated. The assembly is fixed to the retractable cam base plate, by a support made from an “U” shape bent sheet (EN 10025-S235JR) with 4 mm thick.

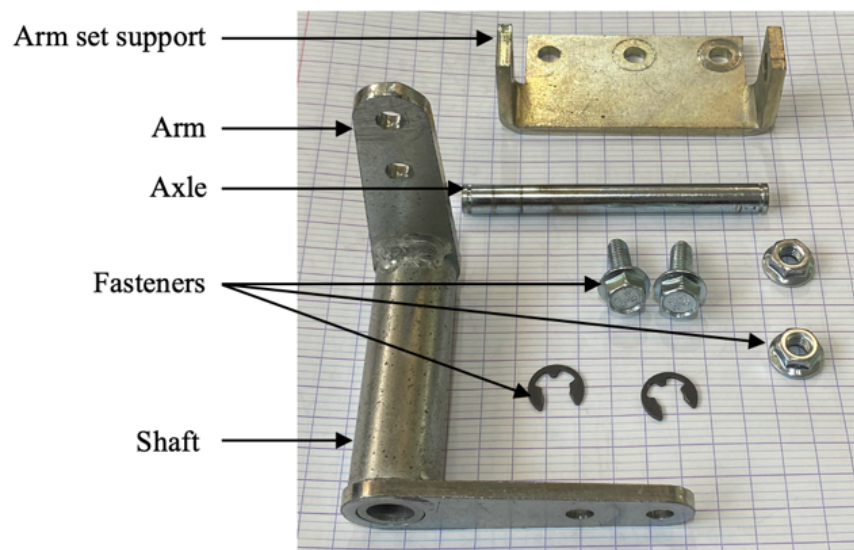


Figure 3.23 – Arms assembly parts.

The connecting rods are adjustable elements in length, given that they are an assembly of five parts: two DIN 71752 M6 fork joints, a cut DIN 976 M6 threaded rod and two DIN 934 M6 nuts to fix the assembly length. The adjustment range is limited by the length of the threaded rod.



Figure 3.24 - Connecting rod.

Table 16 – Mechanism component's material and production technologies (fasteners and other standardized parts were not considered).

Part	Material	Production technologies						
		Laser cut	Welding	Bending	Galvanization (Zinc)	Turning (machining)	Milling (machining)	Cut /Bend by craft
Set of arms (Sub-assembly)	EN 10025-S235JR /9S20	x	x		x	x		
Levers	EN 10130-DC01 ($t=2$ mm)	x			x			
Lever's spacers	Nylon					x		
Lever's shafts	9S20				x			
Brass bar	CuZn39Pb2						x	
Ramp	Aluminum extruded profile							x

3.5.3 Solenoid production

The currently used solenoid is not supplied by a specialized electromagnets company but by a small local manufacturer, therefore the constructive solution is far from ideal, and might not assure a constant performance from unit to unit.



Figure 3.25 - Currently used solenoid.

Furthermore, this solenoid model is produced in purpose for the retractable cam. As illustrated on Figure 3.25, the winding is covered on a yellow fabric strip, and the two covers are machined on a lathe and made of 9S20 steel, machined. Regarding the electric enclosure related to the solenoid, its degree of protection respects the EN 81-20 section 5.10.1.2.2 a), featuring an Ingress Protection rating IP54 (defined in IEC 60529). The core material is not specified by the manufacturer, but it is obviously in a ferromagnetic material, manufactured by turning.

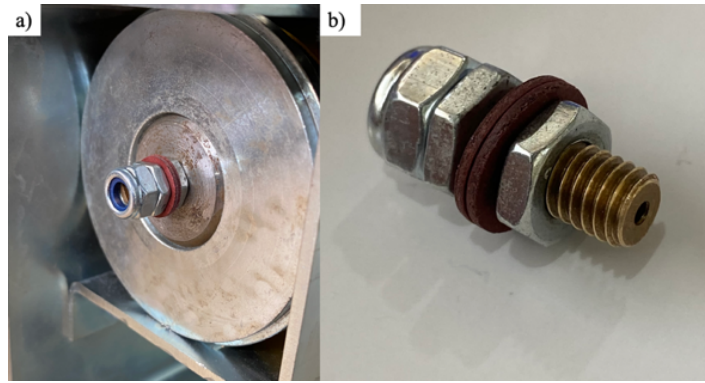


Figure 3.26 - Pneumatic damping element. a) location of the element; b) the element assembly.

A relevant feature of this solenoid is the pneumatic damping element (Figure 3.26), to reduce the noise produced when the solenoid core reaches the end of travel. It is composed by a stud with a hole in the center, nuts and two washers, so when the solenoid is actuated, the core pushes the air contained inside out through the stud's hole, which acts like a flow regulation valve, depending on how tight the nuts are fastened.

The inefficient production of this element has a huge impact on the total production cost of the retractable cam, once the complete solenoid represents approximately 67% of it.

3.6 Overall analysis

This section aims to resume the analysis of the currently used retractable cam and is divided in two parts: strengths and weaknesses of the current solution. This overall review will be considered when redesigning the component, by trying to eliminate the weaknesses and preserving the strengths.

3.6.1 Strengths

The principal strength regarding this solution is its versatility in terms of performance. The combination of a ramp extension amplitude of 41mm and the force applied by it on the locking system is unique on the market.

Also, its compact dimensions allow the component to be installed on every elevator requiring a retractable cam.

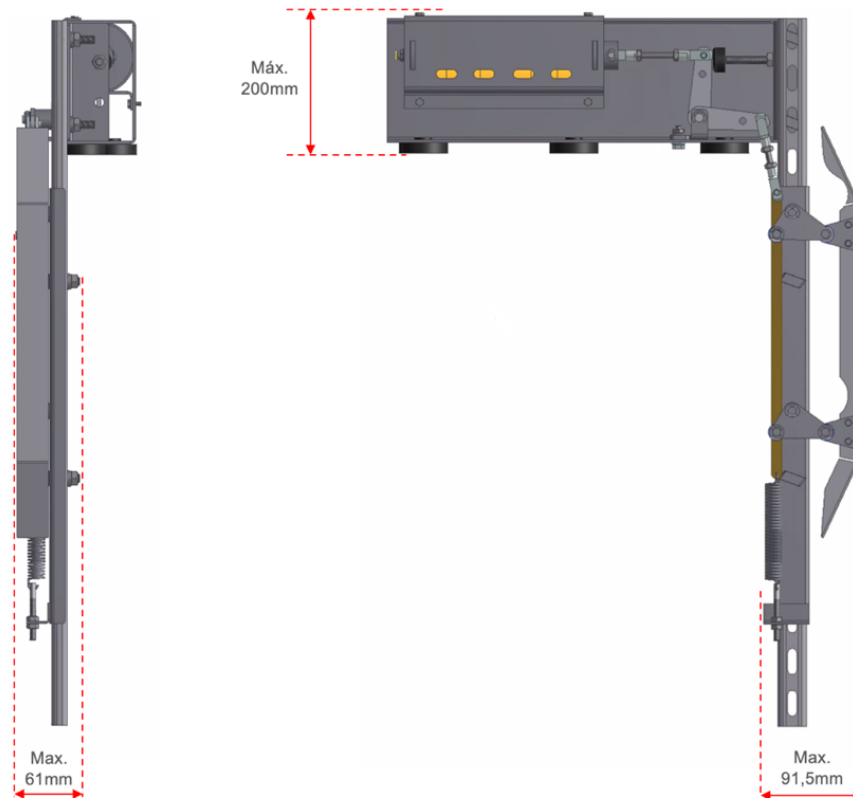


Figure 3.27 - Current solution dimensions annotated on CAD image.

Both these advantages avoid using different models of retractable cams depending on the case.

3.6.2 Weaknesses

On the other hand, there are many topics which could be improved, mostly regarding production, but also in terms of performance.

- Expensive raw materials and production processes: the raw material chosen to fabricate some elements is not ideal given its extremely high cost, which is the case of the ramp and the bar. The production processes do not contribute for a production cost reduction either. The substitution of these components may be an opportunity to reduce drastically the production cost
- Coarse tolerances: some tolerances are too coarse for some parts resulting in excessive gap between parts or can prevent the correct assembly of some components. Consequently, undesirable noise and incorrect functioning can occur. For instance, the slot for the retaining washers on the shafts that hold the

levers can be wrongly positioned, and so the retaining washer does not fit if the lever is in place.

- The solenoid: this element represents an issue both in terms of performance and production, once its energy consumption is high, and its manufacture represents an important part of final production cost. A replacement of the model would probably reduce the cost, force unpredictability and consumption.
- Welding: the incorporation of welding makes the production process more complex once it implies qualified manpower and adds one more stage on the production line.
- The threaded rod: A major problem that can occur during the installation of the retractable cam on an elevator is caused by the threaded rod used to link the fork joints in the connecting rods. Given the large range in height where the ramp can be located, the rod mounted can be too short for the required adjustment. In those cases, a new threaded rod with a specific length must be asked to the production, and then shipped to the retractable cam is to be mounted. This procedure delays the installation time and increases cost.
- Spring pre-load adjustment: As previously referred, the noise produced by the retractable cam is purpose for the elevator users to complain. One of the possible solutions to reduce noise is the correct adjustment of the spring's pre-charge. The problem is that the part that allows to do so is inaccessible when the retractable cam is mounted. For that reason, to adjust it, the component must be uninstalled so the worker can increase or decrease the spring's pre-load. A spring adjusting point with easy access would allow the installation personal to better adjust the component.
- End-of-travel: instead of limiting the end of travel using the solenoid end of stroke itself, the use of end-of-travel rubber bumpers on the two movement limits would also dissipate kinetic energy, reducing noise.

4 The New Retractable Cam

The solution to be developed aims to avoid the weaknesses detected on the previous model, described in 3.6, and accomplish the objectives defined on 1.3, in order to create a product easier to install and adjust, reliable, silent, and with a lower production cost as possible, not compromising the robustness. On the other hand, the currently commercialized model analysis was also important to define some guidelines in terms of dimensions and performance, more specifically ramp travel and force.

The first step to develop the new solution was to define objectively the requirements.

4.1 New Solution Requirements

When producing a component that is to be commercialized, many restrictions and constraints must be considered for the proper functioning of the product. Also, the company can impose some restrictions, related to the production of the component for instance. In this section, the requirements are defined:

1. The force the ramp exerts on the landing door locking system must be at least equal to the force that the currently commercialized solution exerts.
2. The ramp extension amplitude must be, at least, 41 mm, the same comparing to the current solution.
3. The length while the ramp is extended or retracted, and the width of the retractable cam's part where the ramp is fixed must be reduced or maintained (see Figure 3.27).
4. Any part of the component should not pass 200 mm from the top of the elevator's car (see Figure 3.27).
5. The solenoid's ingress protection rating, IP (defined in IEC 60529), must be at least IP2X, according to the standard EN 81-20 5.10.1.2.2 a).
6. In case the solenoid is not actuated, the ramp must remain extended.
7. The spring elongation, and the two end of travel stoppers adjustments must be located above the elevator's car.
8. The solenoid's nominal voltage must be 180V DC.

4.2 New Model Description

The solution developed follows the same principle as the previous one: the ramp is extended by spring action and retracted when the solenoid is energized. Also, the mounting position of the retractable cam remains on the top of the elevator's cabin. Nevertheless, most of currently used retractable cam weaknesses no longer exist. *Autodesk Inventor 2015* was the CAD software used to model the new solution and producing the technical and assembly drawings.

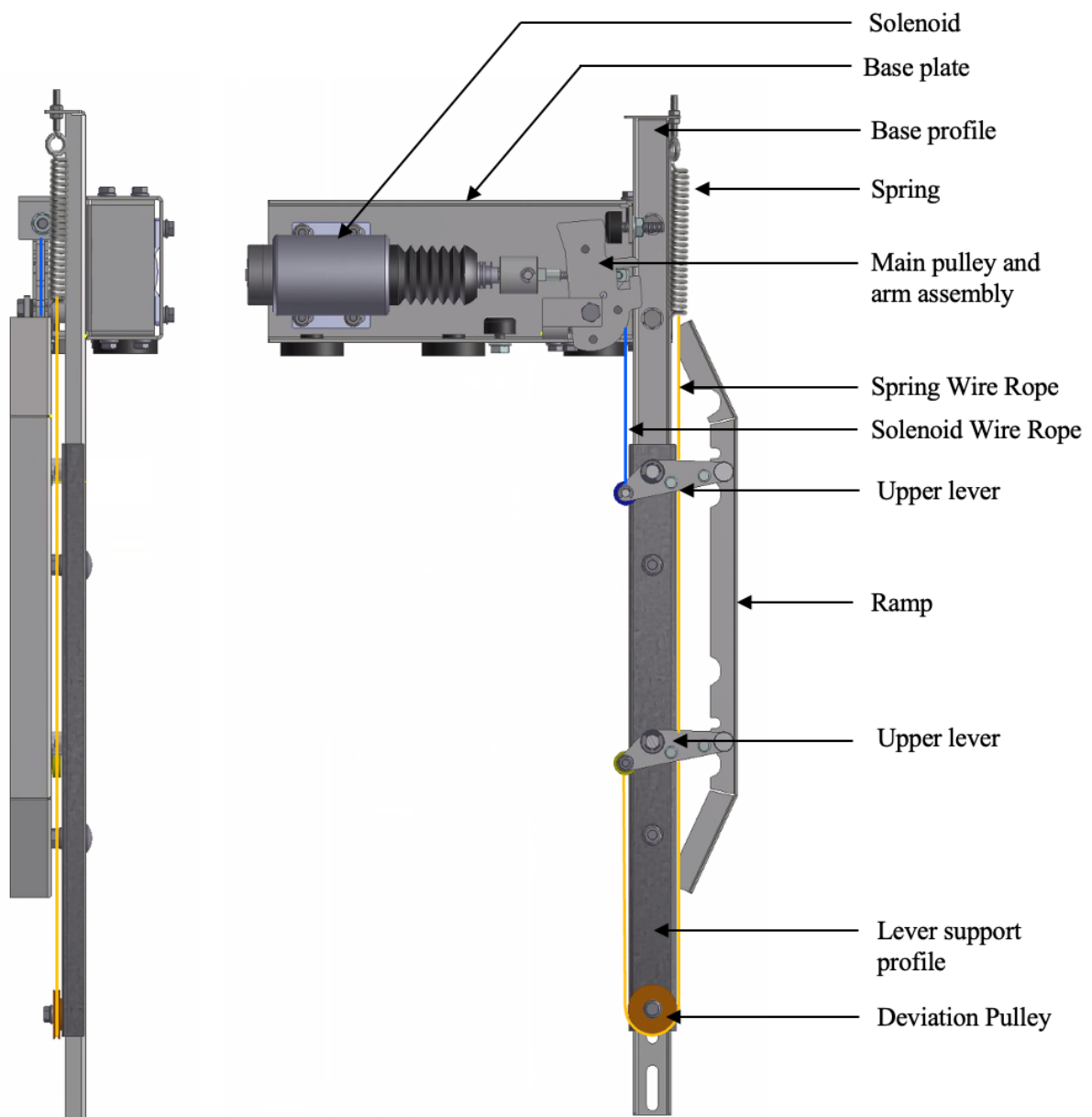


Figure 4.1 - The developed solution CAD model, and the main parts.

By observing Figure 4.1, and with a brief description it is possible to understand how the developed solution operates.

Probably the most notorious difference comparing to the currently commercialized is the use of pulleys and steel wire ropes to replace the connecting rods. Two distinct wire ropes integrate this solution and are represented in blue and orange on Figure 4.1.

The orange one, named spring cable, connects one spring extremity to a pulley located on the lower lever, passing through a deviation pulley (represented in orange), to deviate approximately 180° the spring force direction. This way, the orange cable, is under tension by the spring force, forces the lever to rotate counterclockwise and extend the ramp. The deviation pulley plays an essential role in the solution once it allows to allocate the spring adjustment element on the top of the component.

The blue wire rope indirectly transmits the solenoid force to the upper lever, where one of its extremities is connected. The other is fixed on a quarter pulley, called main pulley, which in turn, rotates jointly with an arm. This assembly basically changes the direction of the solenoid's force, given that it is horizontal, and the pulling force is vertical. In case of wire rope being damaged, the process of removing from the pulley is simple, due to the fixation type.

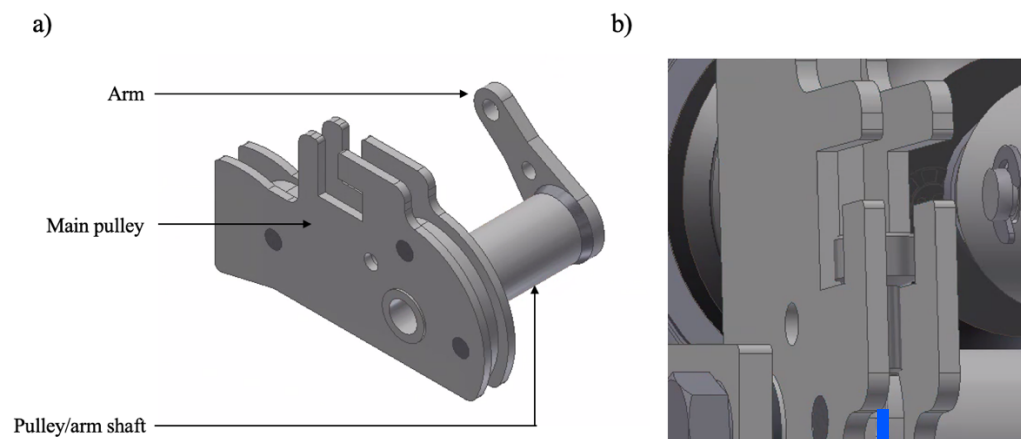


Figure 4.2 - a) Main pulley / arm assembly; b) Solenoid wire rope fixation on the main pulley.

It is also relevant to refer that the main pulley is, as well, the component where the end-of-travel damping elements actuate, as represented on Figure 4.3.

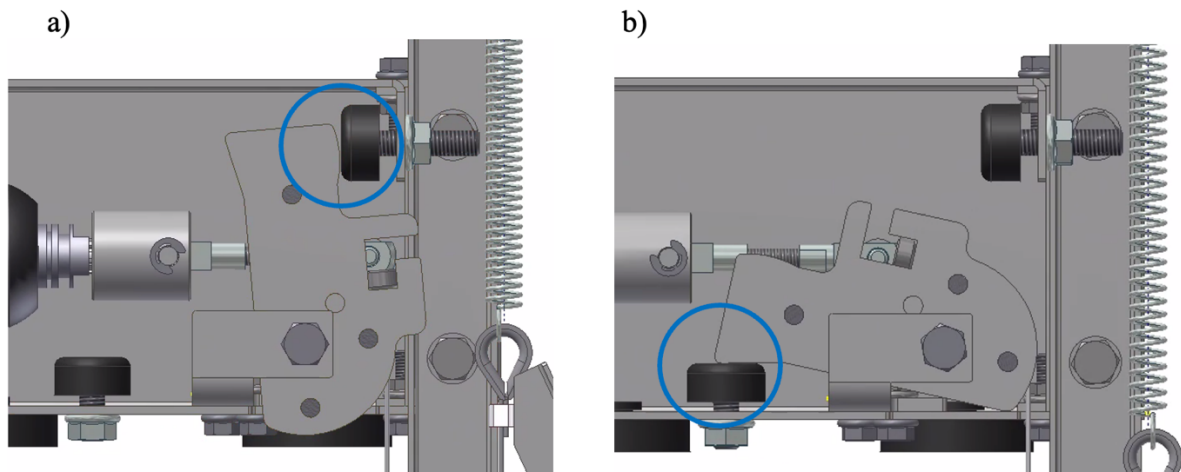


Figure 4.3 - a) End-of-travel element limiting extension movement by actuating on the main pulley; b) end-of-travel element limiting retraction movement by actuating on the main pulley.

On the other hand, the solenoid force is transmitted to the pulley / arm assembly through a hinged connection, linking the solenoid's armature to the arm. The solenoid's fixation is a simple M6 threaded hole on the extremity of the armature, having that, the solution adopted to elaborate a hinged connection was an adapter, that is fixed to the solenoid's core through an M6 hexagon socket head cap screw (DIN 6912), and holds a shaft, where a connecting rod composed by an M6 threaded rod, and two fork joints can be fixed.

With this brief overview of the components that compose this retractable cam solution and Figure 4.1, the description of the retracting and extending operation is clear:

1. The ramp is on its extended position, forced by the lower lever where the spring force is actuating (through the spring wire rope represented in orange).
2. When the solenoid is actuated, its core retracts, causing the main pulley/arm assembly to rotate counterclockwise, until the pulley touches the end-of-travel element. During this movement the upper lever is causing the ramp to retract, by action of the solenoid wire rope.
3. When the end-of-travel is reached, the solenoid remains energized, as long as the ramp needs to be retracted, being the solenoid and spring wire rope constantly under tension.
4. As soon as the solenoid is turned off, the spring force, transmitted to the lower lever, causes levers to regain the extended position of the ramp.

The use of steel rope wires and pulleys showed to be a good answer for the problem related to the adjusting elements, previously, the connecting rods. The need for adjusting the length of connecting elements remain, although, it is much simpler, safer, and fast to cut the wire rope with a specific length and locking it with end-fixings than sawing an M6 threaded rod on the job site.

This modification also facilitated the accomplishment of another requisite: to move the spring adjustment element to a position higher than the elevator's car cabin. The use of a wire rope and the deviation pulley allowed to position the spring in a position where it is possible to adjust its pre-charge without uninstalling the component, and at the same time forcing the levers to adopt the extended position.

The blue cable indirectly transmits the solenoid force to the upper lever, where one of its extremities is connected. The other is fixed on a quarter pulley, which in turn, rotates jointly with an arm connected to the solenoid's movable core by an assembly of two fork joints and a threaded rod (similar to the connections on the currently used solution).

Basically, the assembly of the main pulley and the arm, replace the arms assembly on the current retractable cam model.

Instead of having the brass bar to link the two levers, in this design the solenoid force pushes the upper lever, causing the ramp to retract, and the spring force acts in the lower lever, causing the ramp extension. For this reason, this component was removed.

4.3 Spring and solenoid design

The extension spring and the solenoid are the two elements producing force responsible for the movement of the retractable cam, so must be carefully dimensioned to assure the correct functioning of the component.

Firstly, the movement was analyzed to define the ramp position as function of the levers angle, as well as the spring elongation, which is essential to calculate the spring force, and so the ramp force, depending on its position.

4.3.1 Kinematic study

The dimensions and angles adopted on the levers, as on other parts were limited by the dimensional requirements imposed on 4.1 (topics 3 and 4). For this section, the most important dimensions are the ones related to the mechanism. To simplify the analysis, two diagrams comparing the two solutions are presented on Figure 4.4, with relevant points and angles for the study. From now on, it is the new solution that will be studied (Figure 4.4 a))

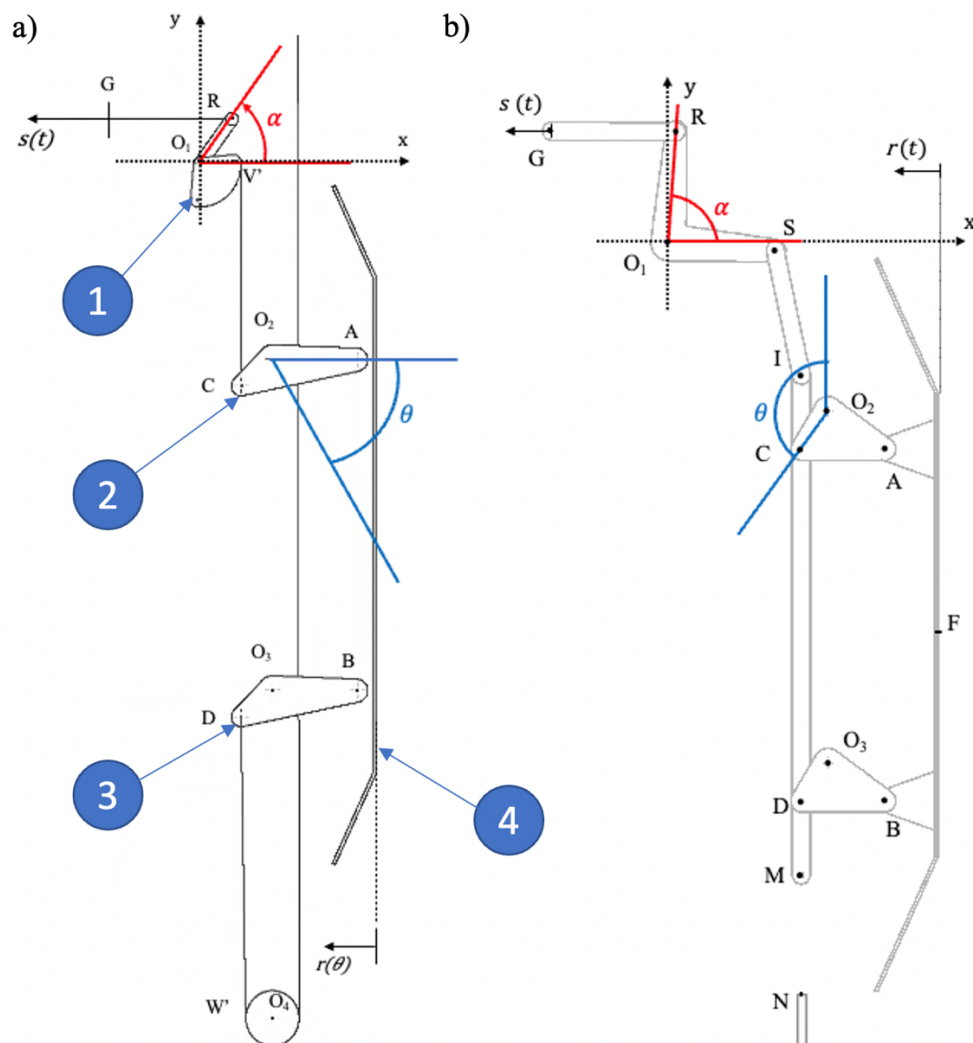


Figure 4.4 – a) New geometry diagram, with the bodies numbered; b) Current solution diagram.

The main angles considered for the new solution study are θ and α . θ is the angle that O_2A does with x axis, and α is the angle described by x axis and O_1R arm.

The dimensions to consider are resumed on Table 17.

Table 17 - Distances between relevant points of the new solution's mechanism

Dimension	Value /mm
$\overline{GR}$	63,00
$\overline{RO_1}$	34,10
$\overline{O_1O_2} - x$	46,36
$\overline{O_1O_2} - y$	118,00
$\overline{CD} = \overline{O_2O_3} = \overline{AB}$	200,00
$\overline{O_2C} = \overline{O_3D}$	25,40
$\overline{O_2A} = \overline{O_3B}$	53,00
$\widehat{CO_2A} = \widehat{DO_3B}$	143,43°
$\overline{O_3O_4} - y$	198,00

Points W' and V' are the points where the wire rope leaves the deviation pulley and main pulley, consequently the distances DW' and CV' are variable during the movement. Also, the angles that the rope wires describe with y axis are not constant.

The levers geometry was designed to maintain the wire rope transmitting the solenoid's force vertical when the ramp is on the extreme positions. The same happens with the connecting rod GR.

Firstly, given that one of the requirements is the ramp travel ($r_{\max}$) to be 41 mm (41,2 mm precisely), and this is directly related to θ , it possible to calculate θ 's amplitude. Considering $r = 0$ mm when the ramp is extended, equation (41) returns $r(\theta)$:

$$r(\theta) = -\overline{O_2A} \cdot \cos(\theta) + \overline{O_2A} \cdot \cos(\theta_e) = -\overline{O_2A} \cdot \cos(\theta) + 53 \quad (41)$$

When the ramp is extended, O_2A is horizontal($\theta_e=0$) so the lever of the landing door locking system cannot push the ramp back, even if the spring is exerting a small force. The value of θ when the ramp is retracted, θ_r is calculated using equation (42):

$$r_{max} - 53 = \overline{O_2A} \cdot \cos(\theta_r) (=) 53 - 41,2 = 53 \cdot \cos(\theta_r) (=) \theta_r = -77,14^\circ \quad (42)$$

So θ is negative, varying from 0° to $-77,14$ degrees when the ramp is retracted.

It is important to refer that the levers were design in a manner that when the ramp is on fully extended or retracted position, point C and D are on the same coordinate in x . Consider C and D when the ramp is totally extended and C' and D', respectively, when the ramp is retracted.

By calculating the displacement of C and to C' or D to D', in y , the travel that the solenoid wire rope has to pull is also determined:

$$\overline{CC'} = \overline{DD'} = \overline{O_2C} \cdot [\sin(\theta_r - \widehat{CO_2A}) - \sin(-\widehat{CO_2A})] = 31,65 \text{ mm} \quad (43)$$

Considering now the radius of the main pulley (quarter pulley) equal to 25mm, in order to pull the solenoid wire rope 31,65mm, the angle described by the pulley (α amplitude) can be calculated through equation (44):

$$31,65 = 25 \cdot \Delta\alpha (=) \Delta\alpha = 1,266 \text{ rad} \approx 72,54^\circ \quad (44)$$

The angle described by the arm that connects to the solenoid between the extended and retracted positions is critical, once it was conceived in a manner that its vertical position is the bisectrix of angle α as represented on Figure 4.5.

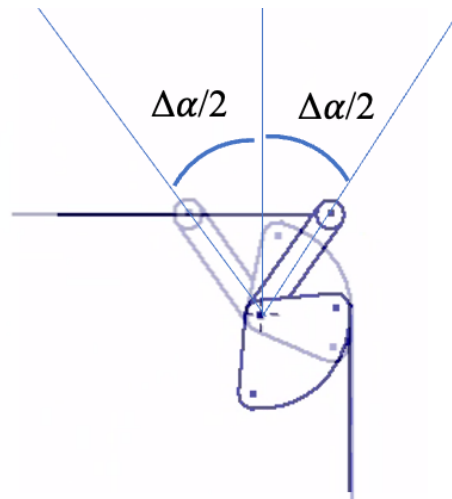


Figure 4.5 - Amplitude of angle α , with its vertical bisectrix highlighted.

Therefore, considering the arm length of 34,10 mm, the solenoid stroke (Δs) needed is calculated using equation (45):

$$\Delta s = 34,10 \cdot 2 \cdot \sin\left(\frac{\Delta\alpha}{2}\right) = 40,35 \text{ mm} \quad (45)$$

The solenoid chosen has to have a stroke larger than 40 mm, but a margin should be considered. Thus, the minimum stroke acceptable should be at least 43 mm.

The next step is to select a spring that will allow the ramp to exert a greater, or equal force than the previous solution, when actuating the locking system arm.

4.3.2 Spring design

In order to remember the ramp force provided by the spring, in different positions, on the currently used solution, Figure 3.15 is repeated as Figure 4.6. The spring chosen for the new solution has to assure that the ramp exerts at least the same force on the locking system than the current.

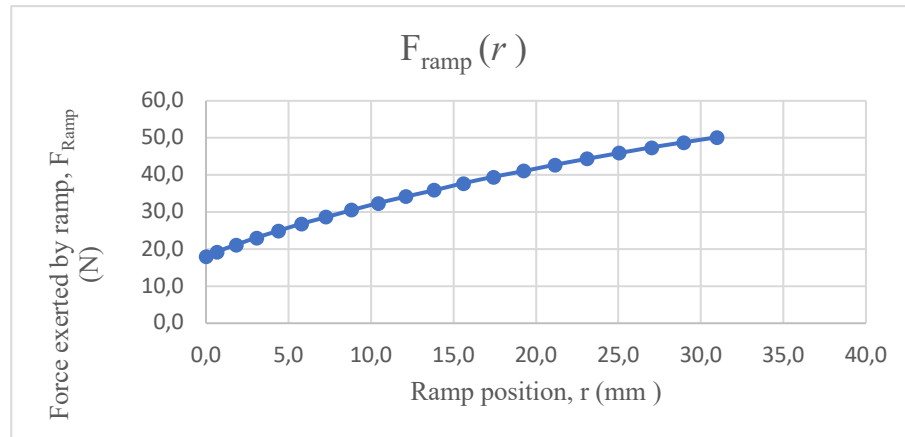


Figure 4.6 - The force exerted by the current solution's ramp in function of its position, from $r = 0$ mm to $r = 31,2$ mm.

To compare with the current solution, the same procedure will be performed with the recently developed geometry, using the equations of the static balance. It is considered that the solenoid is turned off, so no forces are applied on point C and the external force (referent to the landing door locking system lever) is applied on the centre of the ramp (named point F), as illustrated on Figure 4.7. This situation is comparable to the one described in 3.3.1.

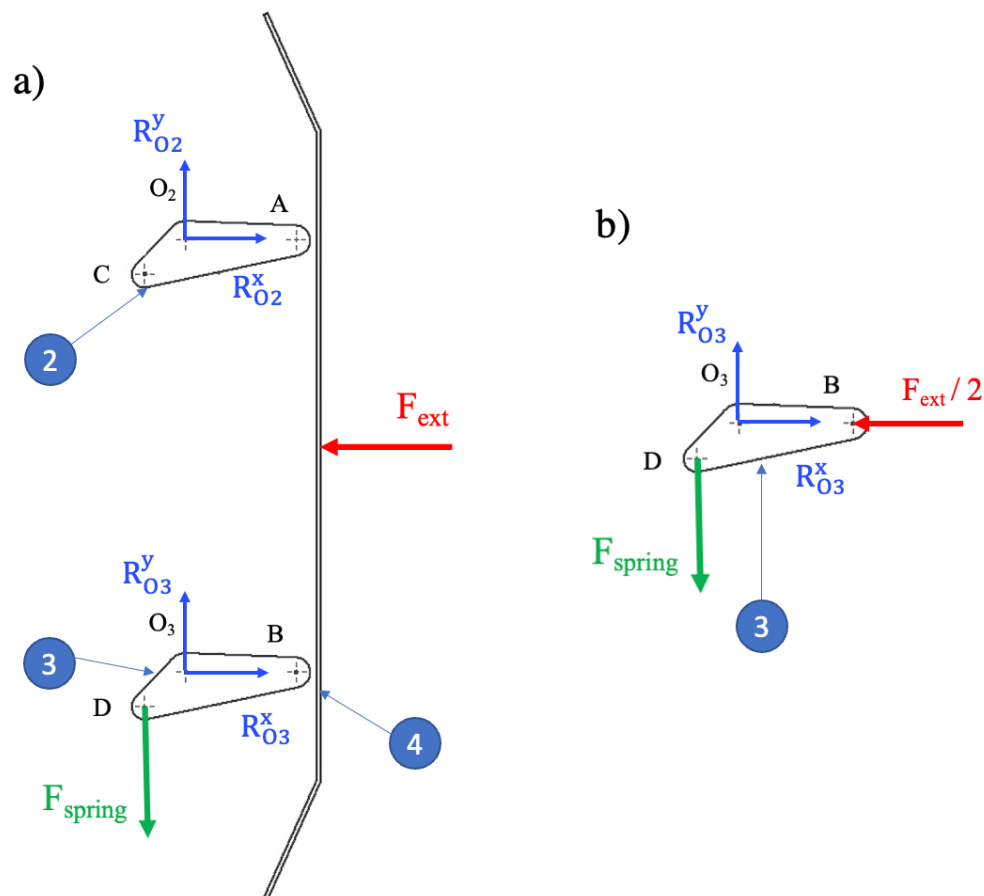


Figure 4.7 - Diagrams used to formulate equilibrium equations. The reactions are represented in blue, the external force F_{ext} in red, and the spring force F_{spring} in green.

The static balance equations to dimension the spring are equations (46) to (52):

$$\sum F^x = 0 (=) R_{O2}^x + R_{O3}^x + F_{spring}^x + F_{ext} = 0 \quad (46)$$

$$\sum F^y = 0 (=) R_{O2}^y + R_{O3}^y + F_{spring}^y = 0 \quad (47)$$

$$\sum M_{O2}^z = 0 (=) O_2 F^y \cdot F_{ext} - O_2 O_3^y \cdot R_{O3}^x - O_2 D^y \cdot F_{spring}^x + O_2 D^x \cdot F_{spring}^y \quad (48)$$

$$\sum F^x = 0 (=) R_{O3}^x + F_{spring}^x + \frac{F_{ext}}{2} = 0 \quad (49)$$

$$\sum F^y = 0 (=) R_{O3}^y + F_{spring}^y + R_B^y = 0 \quad (50)$$

$$\sum F^y = 0 (=) O_3 D^x \cdot F_{spring}^y - O_3 D^y \cdot F_{spring}^x + O_3 B^x \cdot R_B^y - O_3 B^y \cdot \frac{F_{ext}}{2} = 0 \quad (51)$$

$$\tan(\gamma^4) = \frac{F_{spring}^x}{F_{spring}^y} \quad (52)$$

The equations (46) to (48) were written based on diagram a) of Figure 4.7, considering bodies 2, 3 and 4 of the new solution. Equations (49) to (51) refer to body 3.

As previously said the ramp force is the reaction of the external force (F_{ext}) provided by the locking system, therefore, it has the same modulus but opposite direction.

For several positions of the ramp, (varying θ from 2° to 2°) the required spring force was calculated, in order to equate the current ramp force. This spring force value is displayed in green on the scatter chart of Figure 4.8.

Also, a simulation was made, to determine what would be the spring force of a spring featuring a stiffness coefficient of $3,3 \text{ N/mm}^5$ and considering a pre-charge elongation of 8mm. The result is displayed in orange on Figure 4.8.

⁴ γ , is the angle described by the wire rope from point D to W' and the vertical.

⁵ This spring model is intended to be used, once it is already installed in other *Schmitt + Sohn* components.

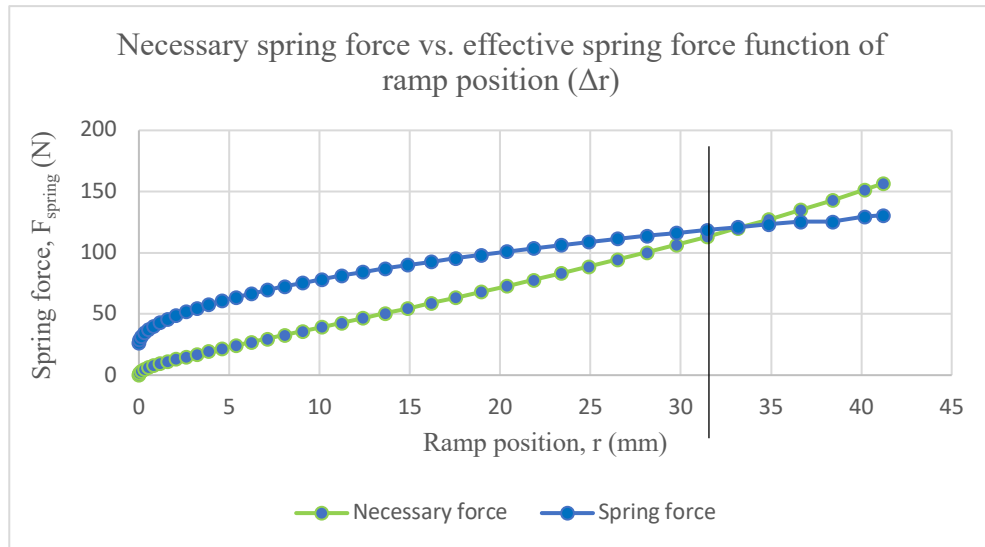


Figure 4.8 - Scatter chart representing the spring force required to match the current solution's ramp force, for a given ramp position (blue), and simulation of the spring force for a given ramp position using a spring with $k=3,3$ N/mm and a pre-charge elongation of 8mm.

It is noticeable that the relation between the ramp position and the spring force is not linear, and that is due to the fact that the spring elongation is not linearly proportional to the ramp's position. Not considering the weight of the parts, no spring force is needed to balance the external force of the locking system, given that the levers are horizontal.

When this spring ($k = 3,3$ N/mm) with 8 mm of pre-charge elongation is used, the maximum force the ramp can exert on the landing door locking system arm, for a given position is illustrated on Figure 4.9.

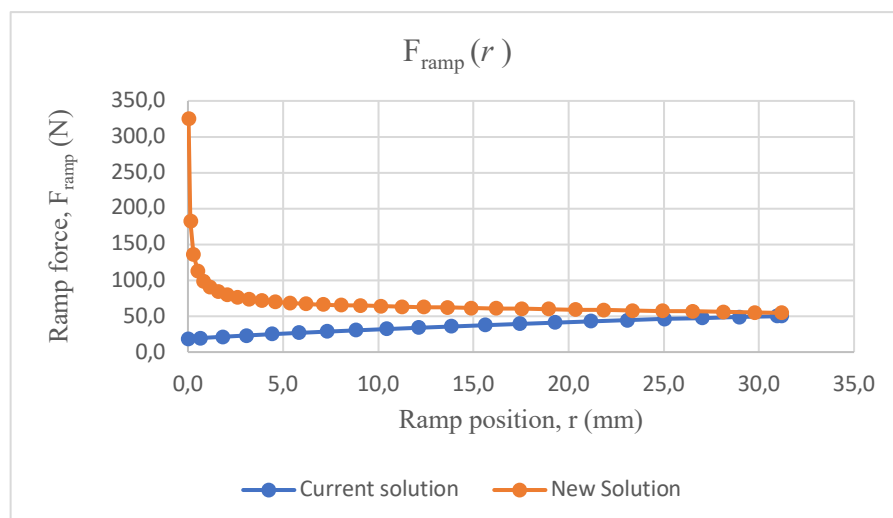


Figure 4.9 - The force exerted by the new solution's ramp as function of its position, from $r = 0$ mm to $r = 31,2$ mm, using a spring with $k = 3,3$ N/mm and a pre-charge elongation of 8 mm, comparing to the force exerted by the current solution's ramp.

As it is possible to observe on Figure 4.9, considering all parts infinitely rigid, it is impossible for the external force (provided by the locking system) to force the ramp to retract, when it is fully extended, as it is supposed. This is due to the fact that the levers are horizontal on this position, as it is possible to see on Figure 4.7. For the whole range of positions, the new solution's ramp can provide a greater force than the previous, using this spring model.

On Figure 4.8 and Figure 4.9, it is clear that the spring used for the simulation is theoretically over dimensioned. Nevertheless, the inertia of bodies, and static friction between moving parts were not considered in calculations. This means that a margin must be left to compensate those energy dissipations.

After a prototype version of this new retractable cam solution being produced, a series of experiments with different springs should be performed, to correctly select the spring to use.

4.3.3 Solenoid dimensioning

The solenoid itself must be dimensioned, indicated for this application, and should be supplied by a specialized solenoids manufacturer. The nominal voltage should be 180V DC, so it can be energized by electricity from the grid transforming it from 230V AC to 180V DC using a simple diode rectifier bridge. This requirement restricted the search for suppliers once most of them just supply solenoids with nominal voltages of 12 or 24 V DC.

After some research, a supplier presented two solenoid models that showed to be ideal candidates to integrate the final solution.

Note: Despite being commercially available solenoid models, in this document they will be referred as solenoid A and B.

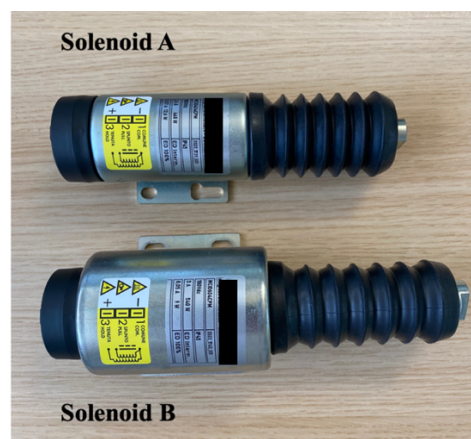


Figure 4.10 - Photography of solenoids A and B.

After having asked the supplier quotation for the two models, for different order volumes, the conclusion was that both have a lower acquisition price than the one used presently. For the same order volume that is typically ordered lately, the saving facing the actual solenoid is approximately 34% for solenoid A and 15% for solenoid B. The saving margins can be increased by ordering more units per order, as expected.

Table 18 - Solenoid A and B characteristics.

	Solenoid A	Solenoid B
Nominal voltage	180V DC	180V DC
Power - holding (W)	12,6	9
IP	IP45	IP45
Force - beginning of stroke (N)	25	75
Force - end of stroke (N)	210	250
ED (%)	100%	100%
Stroke (mm)	45	45

By analyzing Table 18, the most significant differences between the two models are the force produced, and the power consumption. The only information relative to pulling force provided by the supplier is its value on the end and on the beginning of stroke, and as expected the value is grater, when it is fully retracted. Therefore, only the solenoid force required to initiate the ramp retraction, and the force to maintain it retracted will be calculated, through a static study where equations (53) to (62) where developed, based on the diagrams of Figure 4.11.

Once the solenoid force required depends on the spring used, and on its pre-charge elongation, for this study it is considered the same spring used in the simulation ($k=3,3$ N/mm) with a pre-charge elongation (ϵ_i) of 8 mm. Also, the retractable cam will be adjusted in order to prevent the solenoid core to reach its end of stroke, but stay as close as possible, so noise is not produced.

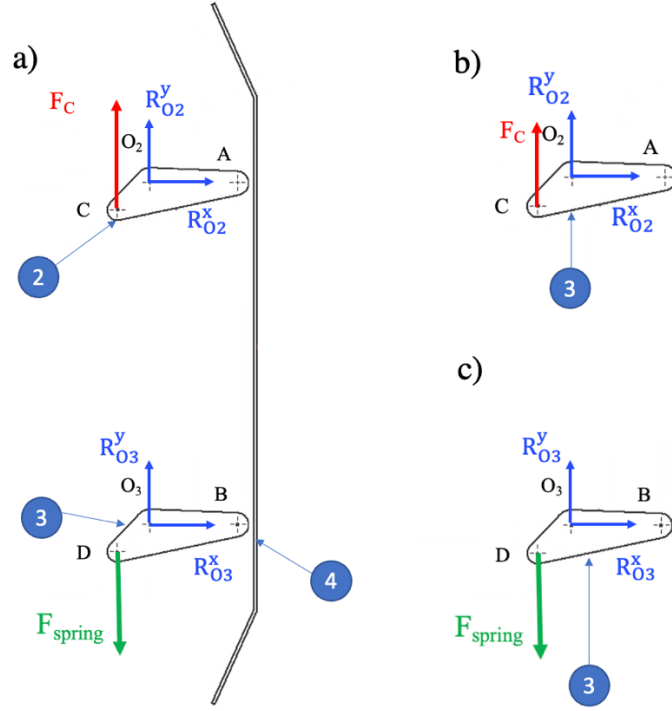


Figure 4.11 - Diagrams used to formulate equilibrium equations to calculate reaction in C. Reaction in C is represented in red, the remaining reactions are represented in blue, and the spring force, F_{spring} , in green.

The first step to calculate the retraction force is to establish the static balance equations:

$$\sum F^x = 0 \quad (=\Rightarrow) R_{O2}^x + R_{O3}^x + F_{spring}^x + F_C^x = 0 \quad (53)$$

$$\sum F^y = 0 \quad (=\Rightarrow) R_{O2}^y + R_{O3}^y - F_{spring}^y + F_C^y = 0 \quad (54)$$

$$\sum M_{O2}^z = 0 \quad (=\Rightarrow) O_2C^x \cdot F_C^y - O_2C^y \cdot F_C^x - O_2O_3^y \cdot R_{O3}^x + O_2D^x \cdot F_{spring}^y - O_2D^y \cdot F_{spring}^x = 0 \quad (55)$$

$$\sum F^x = 0 \quad (=\Rightarrow) R_{O3}^x + F_{spring}^x + R_B^x = 0 \quad (56)$$

$$\sum F^y = 0 \quad (=\Rightarrow) R_{O3}^y + F_{spring}^y + R_B^y = 0 \quad (57)$$

$$\sum M_{O3}^z = 0 \quad (=\Rightarrow) O_3B^x \cdot R_B^y - O_3B^y \cdot R_B^x + O_3D^x \cdot F_{spring}^y - O_3D^y \cdot F_{spring}^x = 0 \quad (58)$$

$$\sum F^x = 0 \quad (=\Rightarrow) R_{O2}^x + F_C^x + R_A^x = 0 \quad (59)$$

$$\sum F^y = 0 \quad (=\Rightarrow) R_{O2}^y + F_C^y + R_A^y = 0 \quad (60)$$

$$\sum M_{O2}^z = 0 \quad (=\Rightarrow) O_2A^x \cdot R_A^y - O_2A^y \cdot R_A^x + O_2C^x \cdot F_C^y - O_2C^y \cdot F_C^x = 0 \quad (61)$$

$$\tan(\omega^6) = \frac{F_C^x}{F_C^y} \quad (62)$$

⁶ ω , is the angle described by the wire rope from point C to V' and the vertical.

Note: The levers were designed in a manner that when the ramp is fully extended or fully retracted, CV' is vertical ($\omega=0$).

Equations (53) to (55) were calculated using diagram a) of Figure 4.11. Equations (56) to (58) and (59) to (61) were calculated using diagrams c) and b), respectively, from Figure 4.11.

After solving this equation for the fully extended position, the result indicates that the wire rope that transmits the solenoid force needs to push the lever with a minimum force of 26,8 N. But this value is not the force that the solenoid must exert (F_{solenoid}), once on the arm/main pulley assembly, the arm's length and the pulley radius are different. Further, the connecting rod that transmits the solenoid force to the arm is only horizontal on the movement limits, as well as the solenoid wire rope is vertical.

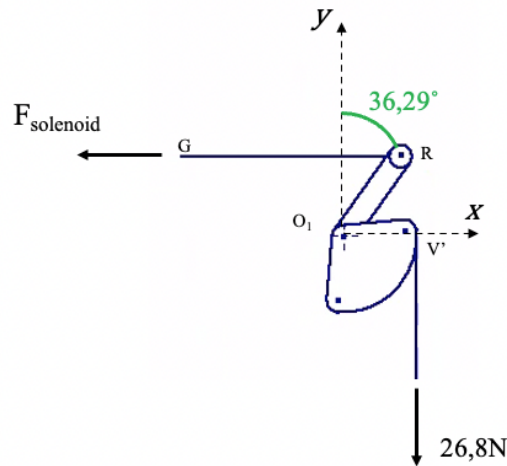


Figure 4.12 - Diagram of forces in the arm/main pulley assembly in fully extended position.

The solenoid force (F_{solenoid}) required to retract the ramp from its fully extended position is then calculated taking as base Figure 4.12, using equation (63):

$$\sum M_{O1}^Z = \begin{Bmatrix} O1V'^x \\ 0 \\ 0 \end{Bmatrix} \times \begin{Bmatrix} 0 \\ -26,81 \\ 0 \end{Bmatrix} + \begin{Bmatrix} 0 \\ O1R^y \\ 0 \end{Bmatrix} \times \begin{Bmatrix} F_{\text{solenoid}} \\ 0 \\ 0 \end{Bmatrix} = 0 \quad (=) \quad F_{\text{solenoid}} = -24,39 \text{ N} \quad (63)$$

To start the ramp retraction movement, at least 24,39 N are needed, not considering friction between moving parts and inertia.

The same process was repeated to determine the solenoid's holding force needed to maintain the ramp retracted. Using the same static balance equations, (53) to (62), the value determined for the rope wire transmit to the upper lever is 129,13 N.

The solenoid force required, was calculated through equation (63), taking as base the diagram presented on Figure 4.13 was used.

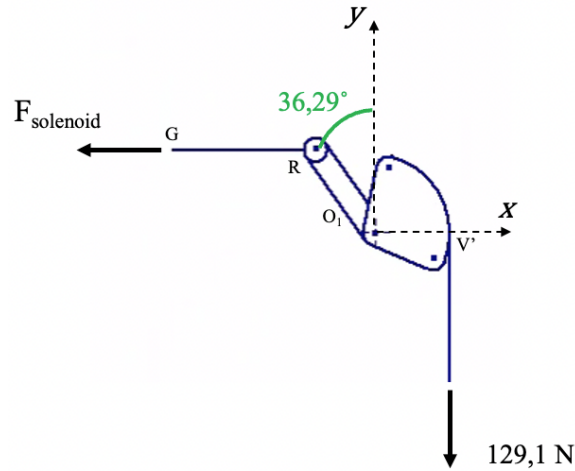


Figure 4.13 - Diagram of forces in the arm/main pulley assembly in fully retracted position.

$$\sum M_{O_1}^Z = \begin{Bmatrix} O_1 V'^x \\ 0 \\ 0 \end{Bmatrix} \times \begin{Bmatrix} 0 \\ -129,13 \\ 0 \end{Bmatrix} + \begin{Bmatrix} 0 \\ O_1 R^y \\ 0 \end{Bmatrix} \times \begin{Bmatrix} F_{solenoid} \\ 0 \\ 0 \end{Bmatrix} = 0 \quad (64) \quad (\Rightarrow) \quad F_{solenoid} = -94,67 \text{ N}$$

Considering equation (64) the solenoid must exert a force of 94,67 N to maintain the retracted position.

From the two solenoid models proposed, one will be selected to integrate the final solution. The main factor to choose is the force produced, once it must be enough on the beginning of stroke (approximately) to start the retraction process (higher than 24,39 N), and the force exerted on almost end of stroke must be higher than 96,67 N. It has to be taken into account that those results were calculated ignoring the friction and inertia's effects, and the extreme positions of the ramp do not represent exactly the solenoid's ends of stroke.

So, by those values, solenoid A should be enough, although a retractable cam prototype must be created, in order to experiment empirically. In case solenoid A does not provide enough force, it must be replaced for solenoid B.

Two other factors shall be considered, in case both solenoids have enough force: the price of the component and the power consumption. As written on Table 18, solenoid A has a higher energy consumption (12,6 W) comparing to solenoid B (9 W), although, solenoid B is approximately 22,5% more expensive than solenoid A, for the same order volume.

So, in case both models are suitable in terms of force, the company must choose, if it is pretended to produce a more expensive model, using solenoid B with lower energy consumption, and larger margin in terms of force, or use solenoid A, producing a cheaper, but with higher power consumption model.

4.4 New Model Manufacturing

Not only the design, but also the manufacturing technologies used to produce the retractable cam have a significant impact on the result in terms of performance, durability, and production cost.

The target is to produce a robust and efficient solution without compromising the cost, given that one of the objectives established was a 10% production cost reduction.

To attain that, the solenoid chosen must not be the only “source of saving”. To reduce costs in other components, an effort was made to produce as many parts as possible in-house. Consequently, the technologies available at *Schmitt 2* production facility are a restriction in terms of how each part can be produced. Also aiming to save costs, small parts such as fasteners that were selected are all standard, and when possible, chosen from the already existing company’s inventory.

The production facility is divided in four sections: sheet metal forming, welding, painting, and assembly.

Sheet metal forming section contains two laser cutting machines, a punch press, two bending machines, and two folding machines. Regarding the laser machines, one of them is a CO₂ laser, which is mostly used to cut metal sheet with thicknesses over 2 mm, and the other is a fibre laser that uses N₂ as assistance gas, typically used to cut thin metal sheets. The punch press is less and less used nowadays, so it will not be considered on the new retractable cam solution as useful production technology.

Welding is a process, that in this case is to be avoided, although it will be used to produce a specific component. The welding section comprises an automated arm for MAG, but this type of welding is also done manually by qualified workers in stations. A spot welder is also available.

The painting technology used is electrostatic powder painting and features an automatic carousel to transport the parts to be painted through the various stages of the process: impurities washing, water washing, drying, painting, and polymerization tunnel.

The assembly section at *Schmitt 2* is basically to perform assembly of elevator cars and doors to be transported to elevator installation building.

The retractable cam, although the steel sheet parts are produced at *Schmitt 2*, the final assembly is to be done at *Schmitt 1*, on the electrical production line.

To describe the production behind the different elements of the recently developed retractable cam, these were divided in sub-groups, associated to the following sub-chapters.

4.4.1 Structure

The first part to be designed was the solenoid structure, which mainly made of laser cut and bent metal sheet parts. In contrast to the current solution the *Halfen* framing channel is no longer used, to save costs. This way, all parts composing the new retractable cam structure are made “in-house”, excluding the coating process.

The welding process was completely removed from this assembly, once it is a process that requires skilled labour and takes time. Further, the parts can only be coated (zinc or cataphoresis) after welding, so the exclusion of this process will expedite the production process.

Four sheet metal parts now form the retractable cam’s structure: the retractable cam base plate, an angle bracket, the base profile, and the lever support profile, as illustrated on Figure 4.14

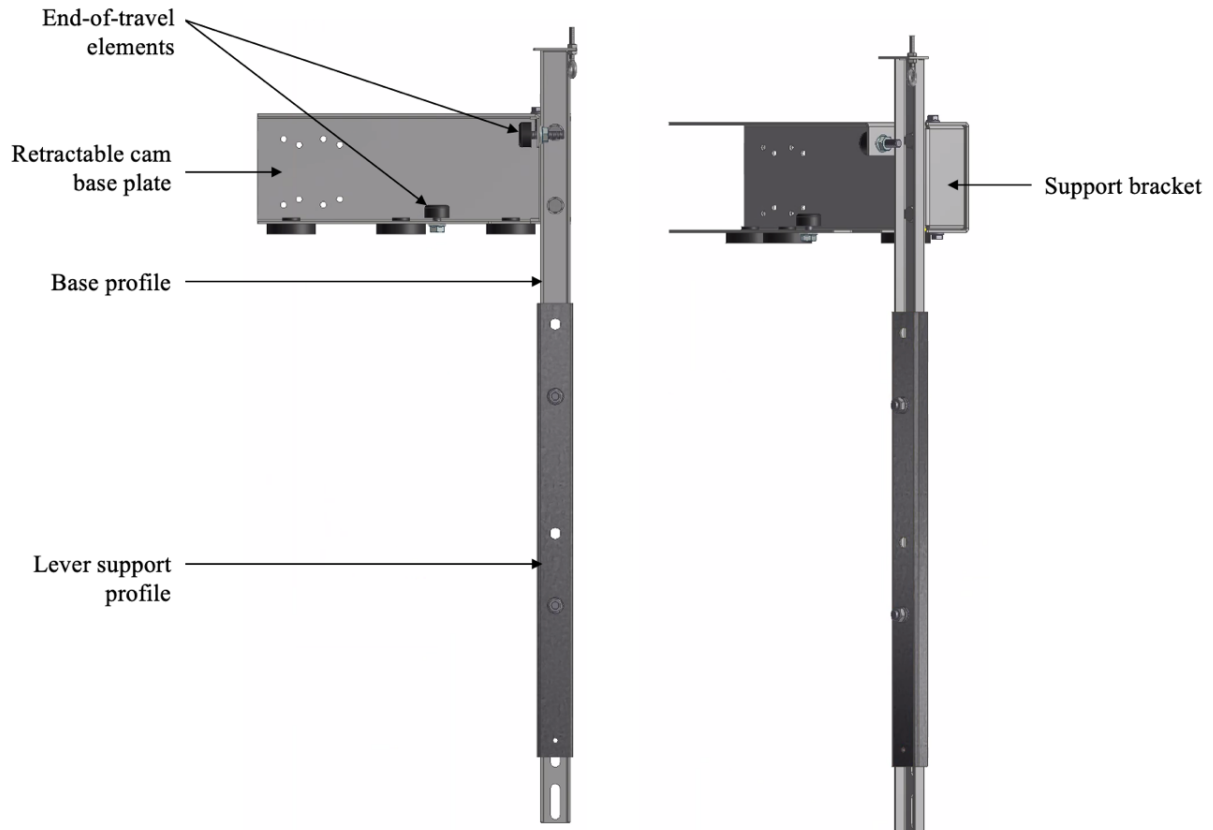


Figure 4.14 - Redesigned structure parts.

The retractable cam base, as the currently produced, uses as raw material 2,5 mm thick EN10130-DC01 steel sheet, that is subsequently cut in the CO₂ laser machine and bent in an “U-shape”, with bending angles of 90°. Two M8 threaded holes were drilled to place the adjustable end-of-travel elements. This part technical drawing corresponds to Appendix B.2.1.

The support bracket is made from the same steel sheet of the base and cut on the CO₂ laser machine. Nevertheless, the is slightly different, given that there are two bending directions instead of one. This part technical drawing corresponds to Appendix B.2.5.

This bracket is then fixed to the base using 4 DIN 6921 M6 x 20 bolts, and DIN 6923 nuts located on the crossed ovals on the two parts. The bending process, as well as the laser tolerances can cause some problems of circular holes alignment. To prevent that to happen the holes have oval shape, crossed, to that misalignment does not occur, and the bolts can be inserted without any problem.

The third part composing the structure, the base profile, replaces the *Halfen* framing channel, and has two purposes: fix the lever support profile, allowing it to be adjustable in height, and serves as spring adjustment system support. This “U-shaped” profile is made of the

same type of sheet than the base and the bracket, but thinner, with 2 mm thickness, also laser cut and bent.

As the bracket, this part also demands a specific bending sequence to avoid punch and die partialization: the first flange to bend is number 1, which is on the top of the profile, and only then the two others can be bent.

The base profile is attached to the bracket using DIN 933 M8 x 20 bolts, DIN 125 washers, as well as spring washers (DIN 127) to guarantee a fixed and stable fixation between two parts, and DIN 6823 M8 nuts, as shown on Figure 4.15

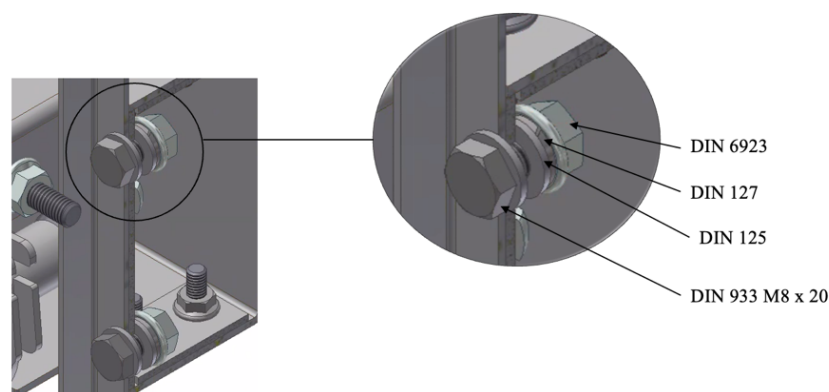


Figure 4.15 - Fixation between base profile and bracket. Image from a cut.

The base profile's drawing is Appendix B.2.2.

The last structure's part, corresponding to drawing #3 on Appendix B.2.3, page 108, is a simple "U-shaped" profile, made of CO₂ laser cut EN10130-DC01 steel sheet with 2mm thick, and with five circular holes on the central zone. The holes with a diameter of 8,5 mm serve to pass the DIN 603 M8 x 30 bolts that fix this part to the base profile, while the ones with 11 mm diameter serve as housing to the lever's support shafts (Appendix B.2.15) that are fixed to this part with a chisel. The threaded hole secures the M6 bolt that will serve as axis for the deviation pulley.

4.4.2 Levers

The levers are a crucial element on this solution as they are responsible for securing the ramp and assure its movement, but also is where the spring force and solenoid force actuate. Unlike on the current model, the upper lever and the lower lever are not identical, even if their production process is similar.

Both are composed of two identical exterior plates, made of steel sheet EN10130-DC01 with 2 mm thick and a central plate whose raw material is EN 10111-DD11 (6 mm thick). The

three parts are cut on the CO₂ laser machine. Then, they are assembled in “sandwich” and revited, so the three plates get aligned and fixed to each other’s.

The difference on the two levers is on the pulley position. The upper lever supports the pulley where the solenoid cable is attached, in between the two exterior plates, through a 9S20 spindle. In order to axially fix the levers, retaining washers (DIN6799 made of spring steel) are located on two grooves on the spindle extremities. The pulley of the lower lever is offset from the lever centre plane, and its axle is a DIN 7991 M6x30 bolt. For the fact that one of the pulleys is in the centre of the lever and the other is offset, the centre plate does not have the same geometry.

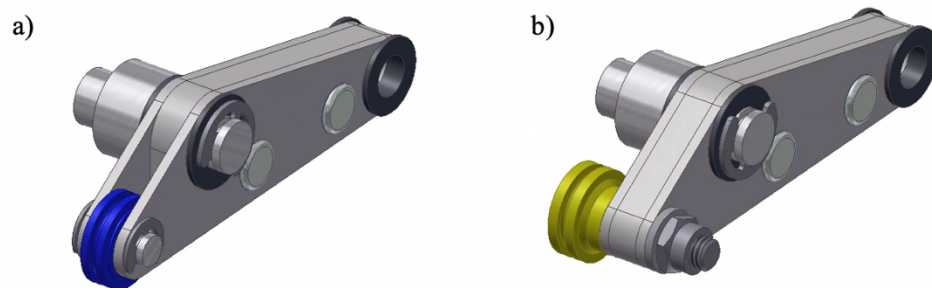


Figure 4.16 - a) Upper lever assembly; b) Lower lever assembly

Table 19 - Production technologies used to manufacture the parts composing the redesigned retractable cam structure.

Part	Material	Production technologies	
		Laser cut	Turning (machining)
Lever exterior plate	EN 10130-DC01 (t=2mm)	x	
Lever interior plate	EN 10111-DD11 (t=6 mm)	x	
Pulley spindle	9S20		x
Upper lever pulley	POM		x
Lower lever pulley	POM		x

As it is possible to observe on the technical drawings, the minimum gap between the shafts that hold the pulleys (spindle and DIN 7991 M6 x 30 bolt) and the hole on the exterior plate of the lever is very tight. This can probably be the cause of a problem, considering that the hole is laser cut. Given the piercing operation, the hole may not be completely circular, but with a small mark that can prevent the bolt or spindle to enter. This phenomenon has already occurred in some parts manufactured at *Schmitt 2*, so a solution was developed by qualified personal through a series of experiments.

The solution found was the modification of the hole shape, and reposition of the piercing point, as illustrated on

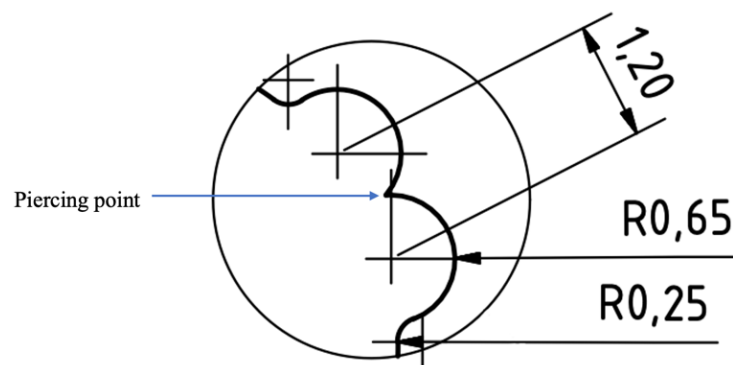


Figure 4.17 - Modification on laser cut hole shape to prevent shafts not to enter.

The two levers are fixed to the spindles (technical drawing on Appendix B.2.15) nailed to the lever support profile (technical drawing is on Appendix B.2.3). To prevent excessive gap and friction IGUS® flange plastic bearings were used. Those components are made of *Iglide®* G300, described on 2.7.4, and assure a tight contact between hole and shaft, as well as a low friction between surfaces. To hold the lever in place retaining washers DIN 6799 are inserted on a groove located on the spindle extremity.

Alternatives:

Other manufacturing technologies were also weighted, namely plastic injection molding, and milling.

The first factor to be considered when selecting a production technology is the number of parts to be produced. In this case the demand for retractable cams is around 50 yearly. This represents a strong cause not to produce the levers by plastic injection molding, no matter the polymeric material. This technology involves a very high initial cost due to the molds manufacturing, reason why this process is normally adopted for large to very large productions.

Consequently, this technology was considered not suitable for any specific part (except standard or purchased parts) that integrates this retractable cam solution.

The other option is to manufacture machined levers, once this production process is ideal for small to medium productions of parts requiring good mechanical properties. In the case of being produced by machining the lever would be composed by a unique part only, made of aluminum 6061 – T6, which is a very common aluminum grade for CNC machining and featuring a good strength. The lever would be totally manufactured by milling on a CNC machine. The reasons why this solution is not the chosen, is because it represents a higher cost per part, and is not possible to be produced internally on *Schmitt 2* facility. Nevertheless, this aluminum machined lever concept is a considerable solution, and a CAD model was created, as on Figure 4.18.



Figure 4.18 - Machined aluminum lever.

Note: The part illustrated on Figure 4.18 does not have a corresponding technical drawing, once it is not part of the final solution.

4.4.3 The ramp

As previously realized, the ramp has a considerable impact on the final production cost of the retractable cam. Mainly because of its complex production method, that forces *Schmitt + Sohn* to order it to an external supplier. Therefore, the aim was to develop a ramp that could be produced internally with a widely internally used material. The result is illustrated on Figure 4.19 b) and the corresponding technical drawing is Appendix B.2.4.

To reduce cost, the raw material chosen was steel sheet EN 10130-DC01, with a thickness of 1,5 mm. The sheet is cut on the fibre laser machine and then bent. Three bending operations are needed, following a specific sequence, annotated on Figure 4.19 a).

The bend 1 bending angle is 90° , and the others are approximately 25° . Even if the bending angle is different, the CAD software has access to tools available and managed to choose a tool set that is able to do the three bend processes, without the need of changing.

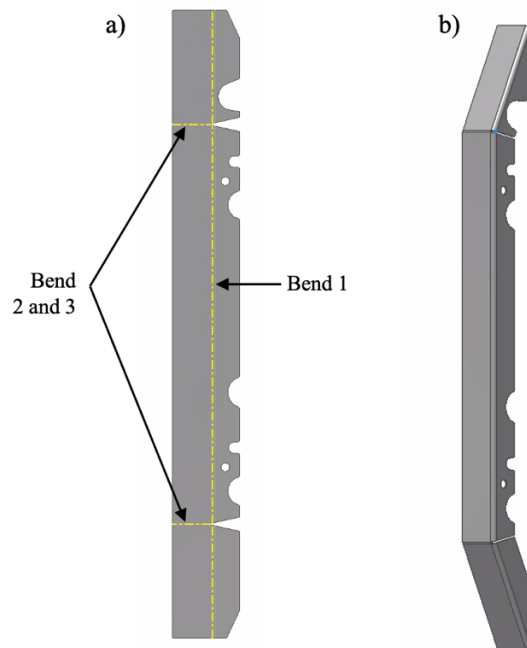


Figure 4.19 - a) The flat pattern of the developed ramp; b) The developed ramp (already bent)

The ramp is fixed to the levers thanks to orbital revits, which plastically deformed zone is close to the ramp. The orbital revits are made of 9S20 steel and manufactured by turning on a lathe. The technical drawing can be consulted on Appendix B.2.16.

Rotation must be possible between the levers and the ramp, given this last component is always vertical, and the levers rotate, but this movement must be smooth and tight. Therefore, also here IGUS® flange bearings were used in between the orbital revit and the ramp. Once the bearing has a flange, this prevents ramp and levers undesirable contact.

4.4.4 Main pulley and arm assembly

This sub-assembly, whose technical drawing is Appendix B.2.22, is the most complex part in terms of production of the new retractable cam solution, once it involves four laser cut steel sheet parts and a lathing part, all welded together.

Basically, as it is possible to observe on Figure 4.20 this is an assembly of an arm, a lathing part (pulley/arm shaft) and the main pulley.

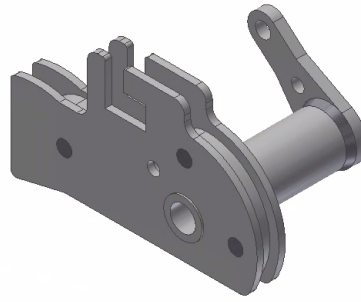


Figure 4.20 - Main pulley and arm assembly.

The main pulley, in fact, is far from being a normal pulley considering its geometry. Further from guiding a cable, this pulley also holds the cable terminal, and is where the retractable cam's end-of-travel elements actuate, making it a crucial part of this solution mechanism. This pulley, as the levers is composed by three CO₂ laser cut steel sheets assembled in "sandwich", two external and a central plate.

The central plate (technical drawing on Appendix B.2.8) is a 6 mm thick EN 10111-DD11 and its circular part, with 25 mm radius supports the solenoid wire rope. The external plates (technical drawing on Appendix B.2.7), whose raw material is 2,5mm EN 10130-DC01 steel sheet, aim to guide the solenoid wire rope, as well as holding in place the wire rope's terminal.

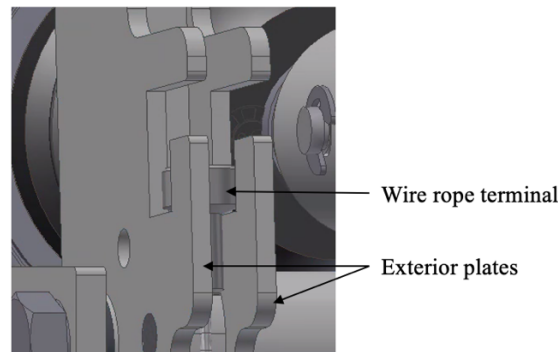


Figure 4.21 - Wire rope terminal fixation between the two exterior plates of the pulley.

It is also these two parts that stop the retraction or extension of ramp movements by colliding with the end-of-travel elements. The three plates are permanently fixed together by manual MAG welding. The external plates have three holes each, whose objective is to be filled with weld, once the three plates are aligned and in place to be fixed.

Note: To align the three plates, the pulley/arm shaft must be in place, and on the auxiliary hole, a 6 mm shaft, such an M6 bolt, must be placed to assure that the parts are aligned.

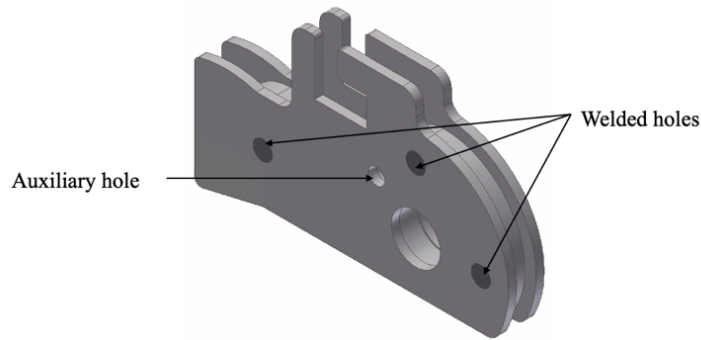


Figure 4.22 - Main pulley assembly

The arm is a simple part to produce, requiring only one operation: laser cut. Its raw material is 4mm EN 10111-DD11. The corresponding technical drawing is Appendix B.2.9.

In order to connect the pulley and the arm, a lathing part made of 9S20 is welded to both. This lathing part is named “Pulley /arm shaft” and its technical drawing is on Appendix B.2.20. When welding the three parts together, a specific angle between the pulley and the arm must be guaranteed by passing through both auxiliary holes (from the pulley and the arm) a 6 mm shaft, as an M6 bolt. Representation on Figure 4.23.

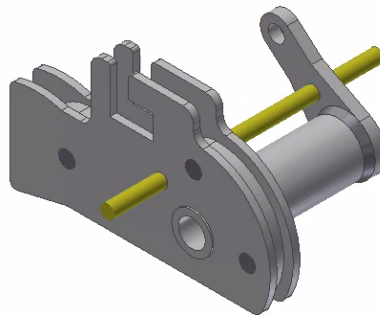


Figure 4.23 - Main pulley and arm assembled and welded, with the auxiliary shaft to assure the correct angle.

4.4.5 Force transmission elements

Not less important than the elements that exert force, such as the spring and the solenoid, are the elements that transfer it to the different parts of the mechanism. In the present solution there are three groups of force transmission elements to characterize: the steel wire ropes, the POM pulleys and the assembly that transmits the solenoid force to the arm of the “main pulley and arm assembly”. Steel wire ropes are propitious to damage when they are subjected to bends with a small radius, reason why pulleys and thimbles are adopted.

This solution integrates two steel wire ropes, with 1,5 mm in diameter and tensile strength grade of 1770 MPa, which are already used on some other *Schmitt + Sohn* components. One of them has on its extremity a terminal made of 9S20, in a lathe, and the other is fixed around the pulley with a wire rope clip.

The second wire rope has the same characteristics and links the lower lever through a pulley, where it is attached with a clip to the spring. To prevent the cable part linked to the spring from being damaged, a thimble was used, as illustrated on Figure 4.24.

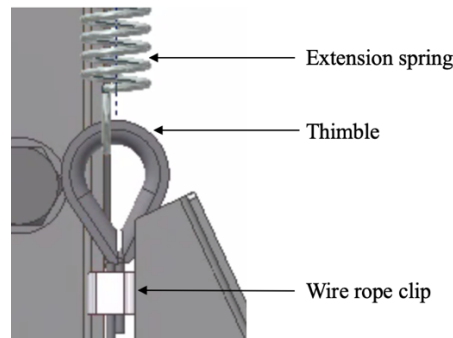


Figure 4.24 - Spring and wire rope fixation.

The three pulleys share the same production technology and material, although they are different. The pulleys that will not be subjected to significant rotation relative to their shaft are directly in contact with it, more specifically, the two pulleys fixed on the levers. The main difference between both is the fact that one is symmetrical, and the other is not, once it is fixed offset on the lever. The symmetrical is represented in blue and the corresponding technical drawing on Appendix B.2.13, the second one is Appendix B.2.14. Both are represented on Figure 4.25.

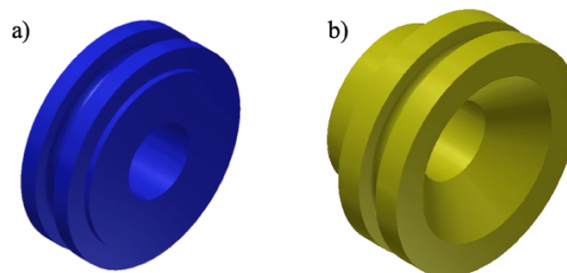


Figure 4.25 - a) Pulley assembled in the upper lever, where is fixed the wire rope transmitting the solenoid's force; b) pulley assemblies on the lower lever, where the spring wire rope is fixed.

The deviation pulley (technical drawing on Appendix B.2.17) is the one with higher diameter to reduce angular speed during retraction and extension movements, and consequently

reducing wear. Also, to reinforce its support and reduce gap, a 9S20 lathing bushing (technical drawing on Appendix B.2.18) lays in between the pulley and the bolt serving as axle, being impeded from rotating due to bolt fastening.

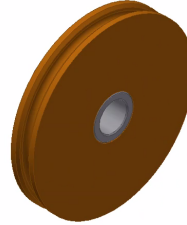


Figure 4.26 - Deviation pulley and 9S20 bushing at its center.

4.4.6 Tolerances

In this solution, as on the previous one, there are several parts, with different connections between them, where some are static, and others move relatively to others. For the good functioning, and durability of the mechanism, some gaps should be adjusted, otherwise excessive wear and impediment of movement of some parts can occur.

Currently, to produce parts internally and externally the general tolerance class chosen is typically medium (m), according to ISO 2768, although in some cases this must be adjusted. This is the case, for instance of the parts contacting with IGUS® bearings, where the manufacturer indicates what are the tolerance of the housing bore, as well as the shaft tolerance.

- Housing bore tolerance for IGUS® bearing: H7
- Shaft tolerance for IGUS® bearing: h9

Another case where the tolerance is not controllable is the steel sheet thickness. In this case the tolerance associated is defined by standards, which is EN 10131:2006 (Cold rolled uncoated and zinc or zinc-nickel electrolytically coated low carbon and high yield strength steel flat products for cold forming – Tolerances on dimensions and shape) and EN 10051:2010 (Continuously hot-rolled strip and plate/sheet cut from wide strip of non-alloy and alloy steels – Tolerances on dimensions and shape).

On some other cases, the tolerance was defined depending on the situation, if a bigger gap is acceptable or not.

The tolerances were calculated radially and axially, as demonstrated on the following example: tolerance Lever – Lever shaft.

- Axial tolerance:

In this case, the shaft is considered to be the sum of the thickness of the three plates composing the lever, plus the IGUS® flange height.

The hole is the section of the shaft where the lever settles, between the larger diameter part and the retaining washer.

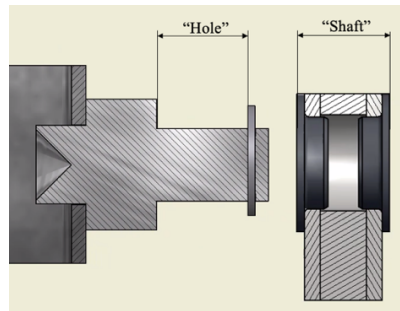


Figure 4.27 - Hole and shaft for axial tolerance calculations between lever shaft and lever

The objective is to have a minimum gap larger than 0 mm, and the maximum gap as tight as possible. In this case the only tolerance that is controllable is the “hole”, corresponding to the shaft’s length. To calculate the gaps, an *Excel* spread sheet was used, originating Table 20. In light gray are the values that cannot be changed and in red, the ones that must be changed to obtain the maximum and minimum gaps desired. With the values used on Table 20, the minimum gap is 0,09 mm and the maximum is 1,59 mm.

Table 20 - Minimum and maximum axial gap calculation between lever shaft/ retaining washer and the lever.

		Ei	Nominal	Es	min.	max.
Shaft	Igus 1	-0,14	1	0	0,86	1
	Steel sheet 2mm	-0,15	2	0,15	1,85	2,15
	Steel sheet 6mm	-0,21	6	0,21	5,79	6,21
	Steel sheet 2mm	-0,15	2	0,15	1,85	2,15
	Igus 2	-0,14	1	0	0,86	1
	TOTAL		12		11,21	12,51
Hole	Lever shaft	0	12,6	0,2	12,6	12,8
	TOTAL		12,6		12,6	12,8

Min. Gap [mm]	0,09
Max. Gap [mm]	1,59

- Radial tolerance:

Once the contact between the lever plates and the lever's shaft is through an IGUS® GFM 1012-04, the tolerances required by the manufacturer need to be respected. Therefore, the housing hole (on the exterior lever plate) must be 12H7. That is achieved by under dimensioning the hole when cut by laser, and adjusting the diameter by machining. The shaft is machined and must be 10h9.

The minimum and maximum gaps were calculated using Table 21.

Table 21 - Minimum and maximum radial gap calculation between lever shaft/IGUS and lever housing hole /IGUS.

		Ei	Nominal	Es	min.	max.
Housing hole	Furo veio (12H7)	0	12	0,15	12	12,15
IGUS	D		12		-	-
	d (10E10)	0,025	10	0,083	10,025	10,083
Shaft		-	10	0	9,964	10
	Veio (10h9)	0,036				

IGUS housing hole./D IGUS	Min. Gap [mm]	0
	Max. Gap [mm]	0
d IGUS/ shaft	Min. Gap [mm]	0,025
	Max. Gap [mm]	0,119

According to Table 21, the gap between the housing hole and the IGUS® bearing is inexistant (= 0 mm). The gap between the IGUS® inner diameter and shaft varies from 0,025 and 0,119 mm.

4.5 Comparison between currently used and new solution.

After having read the detailed description of the two solutions, the differences are clear in terms of adjustment facility, power consumption and production cost.

Adjustment:

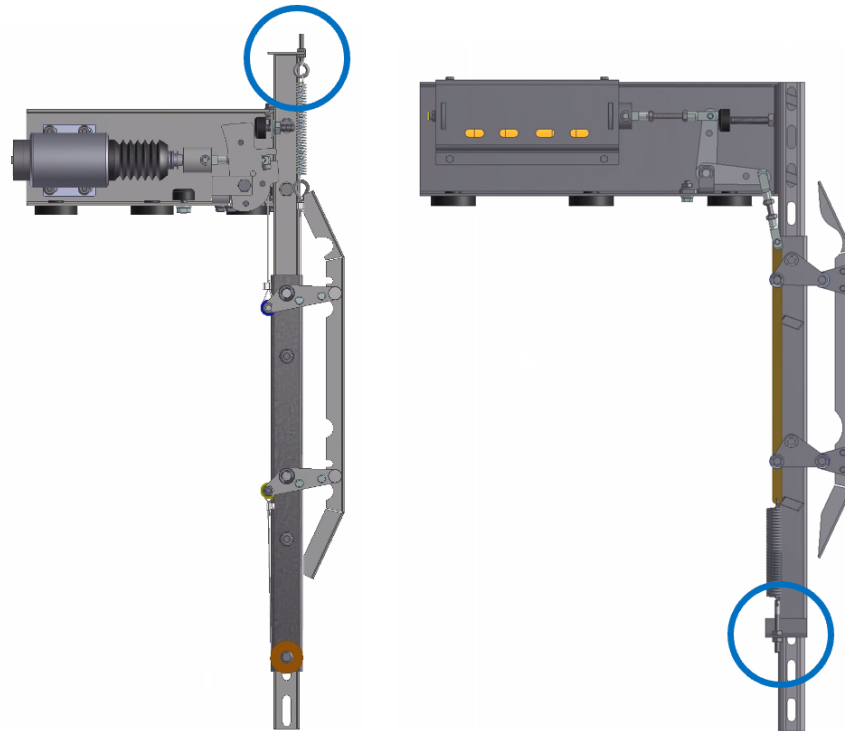


Figure 4.28 - Comparison between the position of spring pre-charge adjustment element on the new solution (on the left) and on the currently commercialized (on the right).

As clearly illustrated on Figure 4.28, the spring adjustment element on the new solution is easily accessible once it is located above the elevator's car roof in contrast to the current retractable cam, where the threaded stud is in a tight space, turning the mounting personal's job harder and more time consuming. Also, in case of spring replacement necessity, the operation is facilitated, given that there is no necessity of uninstalling the whole component to do the procedure.

Energy consumption:

Relatively to energy consumption, the objective of reducing in 10% was accomplished by far, no matter the solenoid model chosen. As calculated on 3.4.2, the currently used solenoid has a holding power of 110,6W, what is vastly higher than the two possible options to integrate on the developed retractable cam.

Table 22 - Power consumption comparison, considering the actual in use, solenoid A and B.

	Current solution Solenoid	Solenoid A	Solenoid B
Power - Holding (W)	110,6	12,6	9
Relative reduction (%)	-	88,6%	91,9%

Table 22 reveals that even choosing the most power consuming model (Solenoid A) a reduction of almost 89% is attained. By integrating solenoid B, the power saving is more relevant, reaching approximately 92%.

Production cost:

The new solution's production cost is also drastically lower compared to the previous solution, and that is due to two factors: solenoids A and B have a lower acquisition cost than the currently in use, and the constructive solution itself, that allows to produce more components *in-house*, in a more economical way.

Even having reduced the cost related to the solenoids, those still represent a significant part on the final retractable cam's production cost. Maintaining the order volume, and integrating solenoid A, this component cost will represent around 73% of the new solution's total production cost, and the percentage increases to 77% if solenoid B is the one selected. On the other hand, the solenoid now used only represents around 67% of the total production cost of the currently commercialized component. Therefore, an increase on the percentage of the total production cost occupied by the solenoid price, having reduced it means one thing: the production cost savings associated to the constructive solution itself were higher than the savings related to the solenoid.

Finally, the new solution with solenoid A is 38% less expensive to produce than the current solution. If solenoid B is integrated instead, the savings decrease to 25%.

Aiming to reduce even more the costs associated to the solenoid, a quotation simulation was done, considering only one order yearly, instead of various, as it is done until now. The quantity discount proposed by the manufacturer, results in a price reduction of more than 31% per unit. Using this supply approach, the saving associated to the new solution with solenoid A is 52%, and 43% integrating solenoid B. To resume, Table 23 displays the total production cost savings of the new solution relative to the current one, depending on the solenoid used and on the ordering method.

Table 23 – Total production cost savings of the new solution compared to the current one, depending on the solenoid used and on the ordering method. The worst saving value is highlighted in red, and the best in green.

	Maintaining ordering method		1 order yearly	
	Solenoid A	Solenoid B	Solenoid A	Solenoid B
Total production cost saving face to the current solution (%)	38%	25%	52%	43%

Note: As already explained a solenoid model will be selected to integrate the solution permanently, after the prototype has been produced and tested. Once the prototype has not yet been produced, and consequently the solenoid model was not selected, the total production cost was calculated to every scenario as on Table 23. The ordering method may be decided by the company.

Welding:

The number of welded assemblies was also reduced in 66,7% given that the current solution integrated three sub-assemblies needing welding, and the new solution only has the main pulley/arm assembly that must be welded.

Performance:

The new retractable cam solution was developed to perform equally or better than the current one, although the force produced by the ramp to actuate the locking system's arm may vary with the spring used and the pre-charge selected.

In terms of maximum ramp travel both present the same and are adjustable to adapt the component to the elevator where it is to be mounted.

5 Conclusions and future work

5.1 Conclusions

The present dissertation describes the analysis of a crucial component installed in elevators with a manual landing door and details the development of a new model of the same component. Aiming at modernizing this component and solving some issues inherent to the solution, *Schmitt + Sohn* proposed the project to *Faculdade de Engenharia da Universidade do Porto* as a theme for master's dissertation.

The contextualization was essential before the beginning of the project itself, where the importance and general functioning of an elevator, and more specifically of the retractable cam took place.

The analysis of the currently used retractable cam began observing the component physically and operating in a functioning elevator. The mechanism geometry was then studied as well as the production process, evaluating the weaknesses of the solution, which became improvement opportunities.

Based on what could be improved and on requirements imposed for the project, the development of a new component began. The focus was to reduce production costs, replace the solenoid for a more efficient one and ease the spring adjustment. To do so, some elements were replaced, others modified, changing the production technology and material associated and others even removed. The solenoid was selected from a few models available on the market, considering the specific requirements. The CAD model was designed on *Autocad Inventor 2015*, from where the technical drawings were drawn.

The final solution accomplished all objectives and fulfilled the imposed requirements, resulting in a component easier to adjust, with a lower power consumption (approximately less 90%) and representing a significantly lower production cost for the company (between 25% and 52% lower, depending on the solenoid model and order volume).

5.2 Future work

The next phase of the project will consist of performing the functional test of the components, as well as industrialization.

Functional test of the components will be essential to confirm if the model operates as expected, but also to confirm what spring and what solenoid model are to be installed in the final solution.

Industrialization refers to all the process of adapting the prototype production to mass production. Both internally produced components and outsourced ones can be improved to ease the production and ensure a proper supply chain process efficiency and effectiveness (lead time and cost).

In order to achieve the above-mentioned aims, a prototype must be produced, and a series of tests on it, such as measuring the ramp force, are required.

Depending on the tests results, given that some components are subjected to considerable strains, a structural analysis using finite elements method (FEM), can be required.

Finally, a test on the electrical grounding of the solenoid must be carried out.

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APPENDIX A

Position vectors of current solution

APPENDIX A – Position vectors of current solution

$$\overrightarrow{O_1 R} = \begin{pmatrix} \overline{O_1 R} * \cos(\alpha) \\ \overline{O_1 R} * \sin(\alpha) \\ 0 \end{pmatrix}$$

$$\overrightarrow{O_1 S} = \begin{pmatrix} \overline{O_1 S} * \cos\left(\alpha - \frac{\pi}{2}\right) \\ \overline{O_1 S} * \sin\left(\alpha - \frac{\pi}{2}\right) \\ 0 \end{pmatrix}$$

$$\overrightarrow{O_1 O_2} = \begin{pmatrix} 0,0895 \\ -0,09125 \\ 0 \end{pmatrix}$$

$$\overrightarrow{O_2 C} = \begin{pmatrix} \overline{O_2 C} * \sin(\theta) \\ \overline{O_2 C} * \cos(\theta) \\ 0 \end{pmatrix}$$

$$\overrightarrow{O_2 I} = \begin{pmatrix} \overline{O_2 I} * \sin(\theta) \\ \overline{O_2 I} * \cos(\theta) + \overline{CI} \\ 0 \end{pmatrix}$$

$$\overrightarrow{O_1 I} = \begin{pmatrix} \overline{O_2 I} * \sin(\theta) + 0,0895 \\ \overline{O_2 I} * \cos(\theta) + \overline{CI} - 0,09125 \\ 0 \end{pmatrix}$$

$$\overrightarrow{SI} = \overrightarrow{O_1 I} - \overrightarrow{O_1 S}$$

$$\overrightarrow{O_2 M} = \begin{pmatrix} \overline{O_2 I} * \sin(\theta) \\ \overline{O_2 I} * \cos(\theta) - \overline{CM} \\ 0 \end{pmatrix}$$

$$\overrightarrow{O_2 A} = \begin{pmatrix} \overline{O_2 A} * \sin(\theta - \beta) \\ \overline{O_2 A} * \cos(\theta - \beta) \\ 0 \end{pmatrix}$$

$$\overrightarrow{NM} = \begin{pmatrix} \overline{O_2 C} * (\sin(\theta) - \sin(\theta_e)) \\ l_0 + \overline{O_2 C} * (\cos(\theta) - \cos(\theta_e)) \\ 0 \end{pmatrix}$$

APPENDIX B

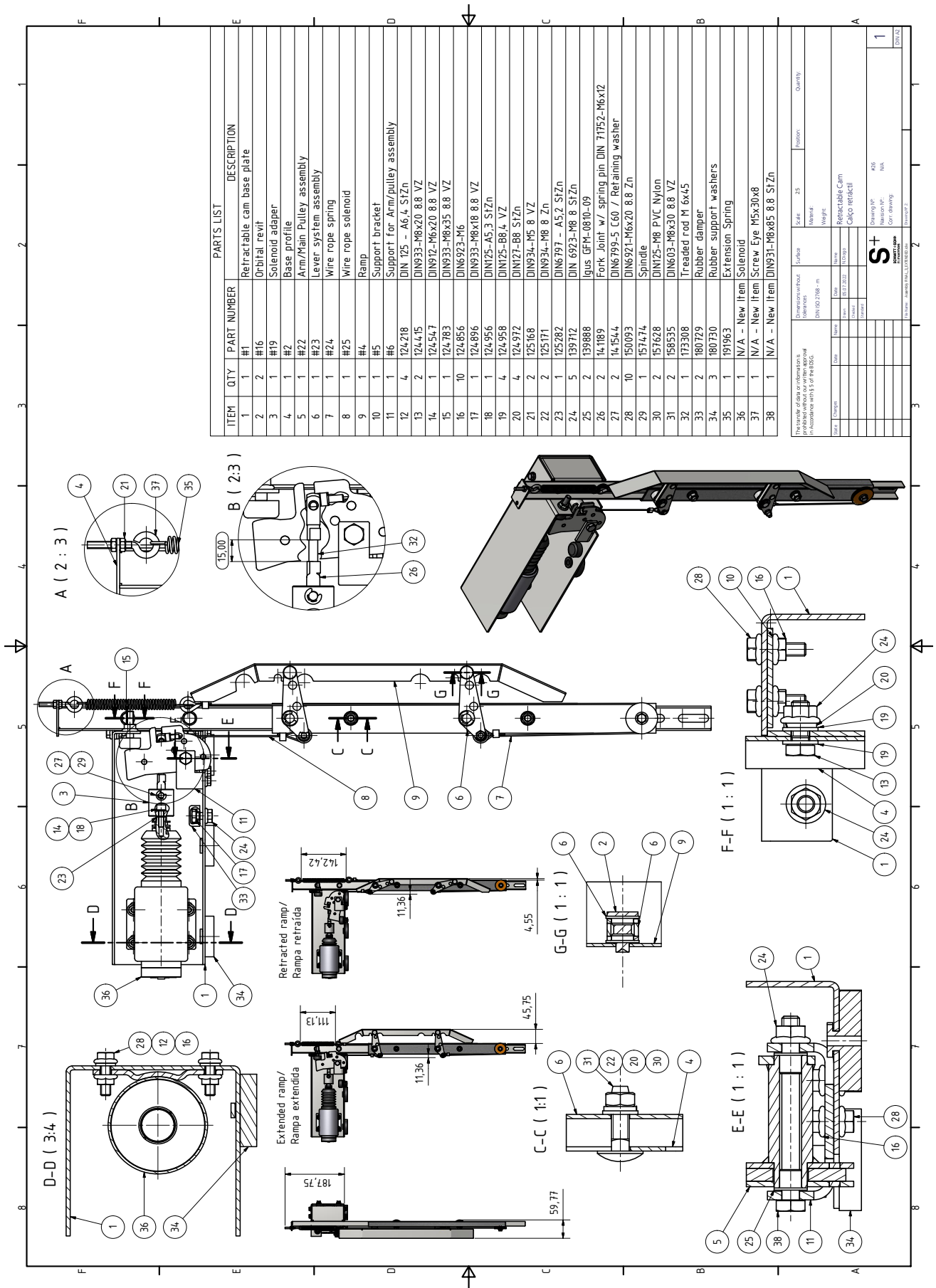
Technical Drawings

APPENDIX B.1 - Technical drawing considerations

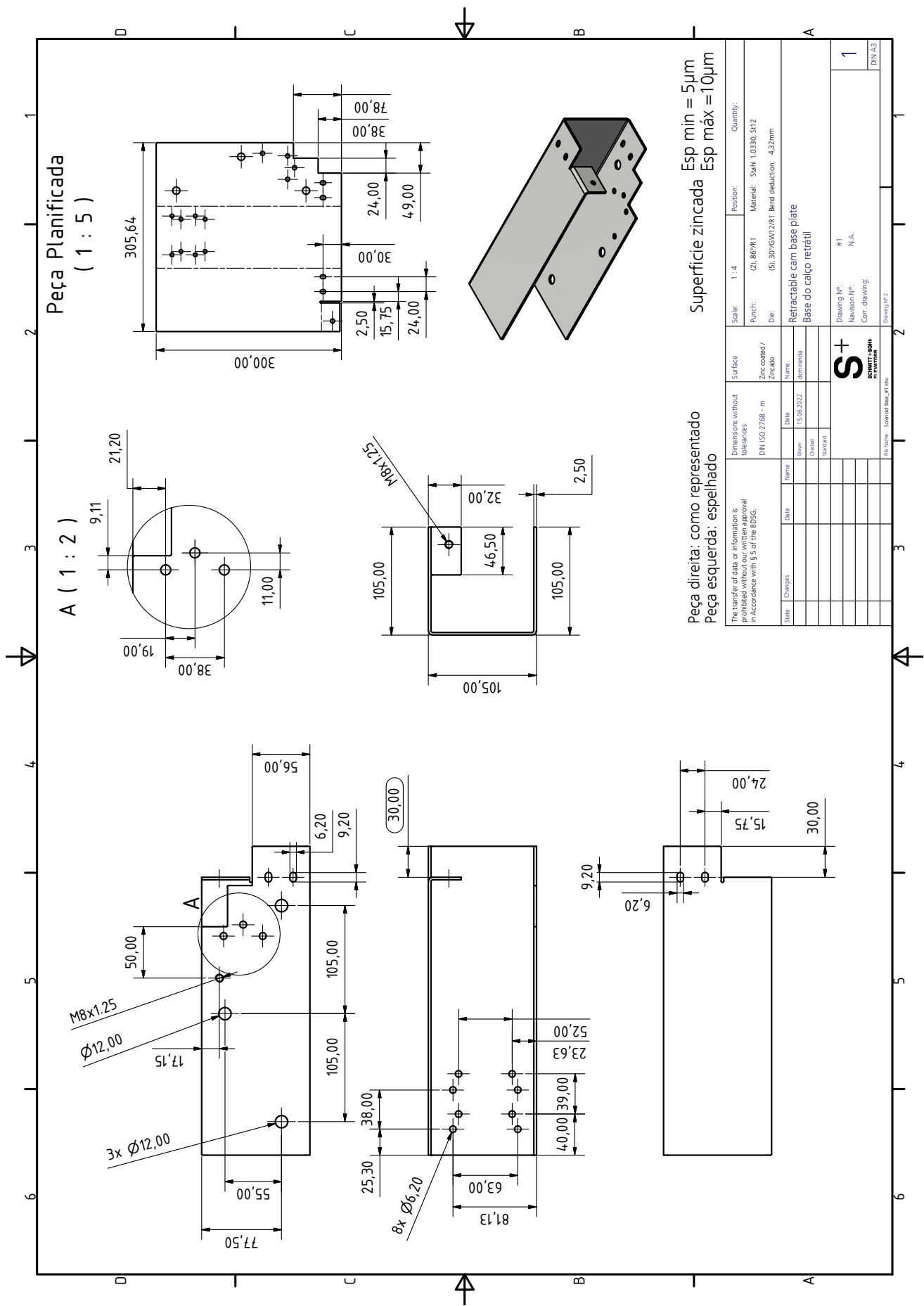
Considerations:

- The technical drawings have important annotations for plant operators that should be considered by, when a part or assembly is being produced. To facilitate the operator's interpretation, the notes are written in Portuguese.
- The technical drawing sheet format and layout is the one used officially by the company and may not be changed.

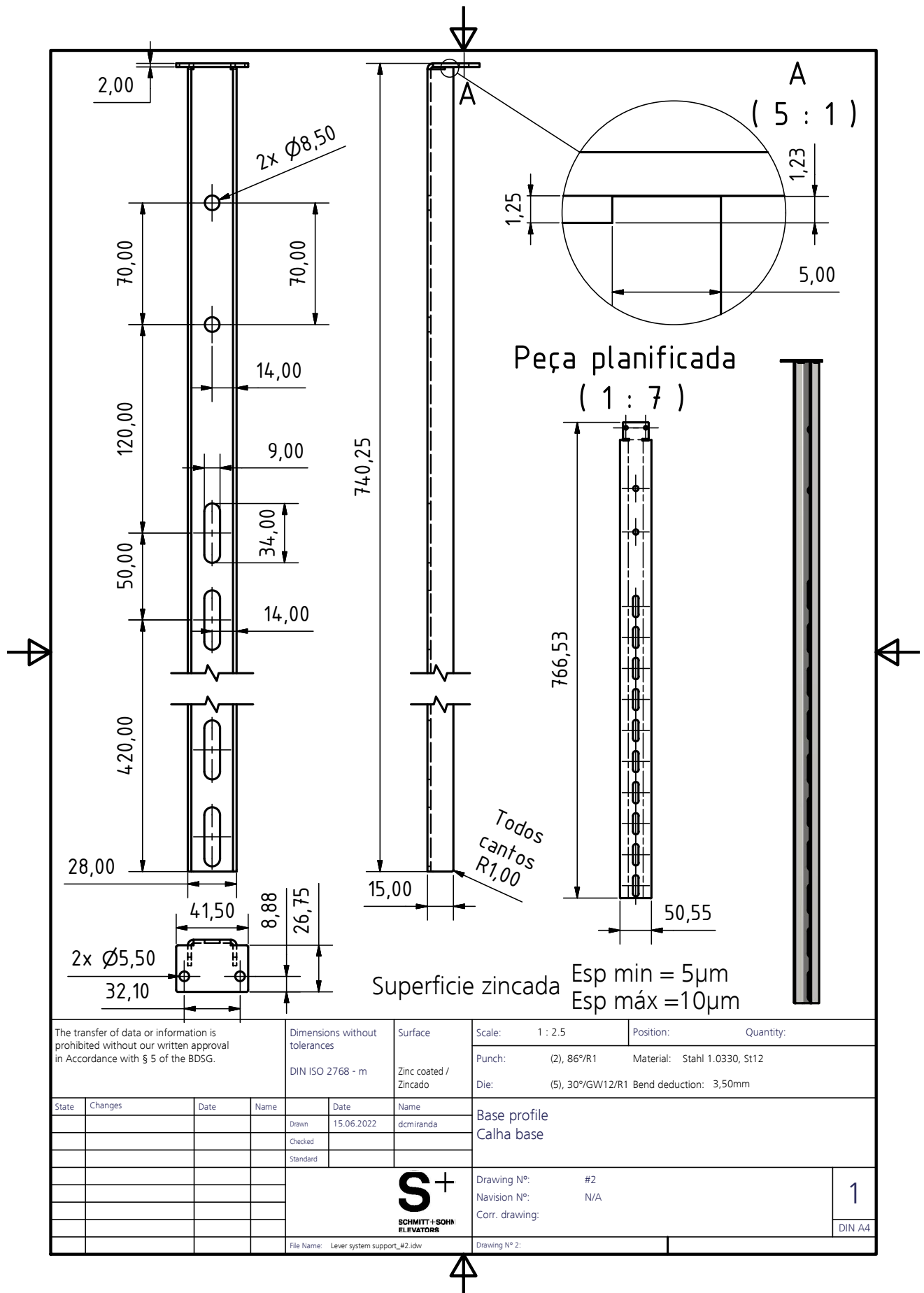
APPENDIX B.2 – Final Solution Assembly



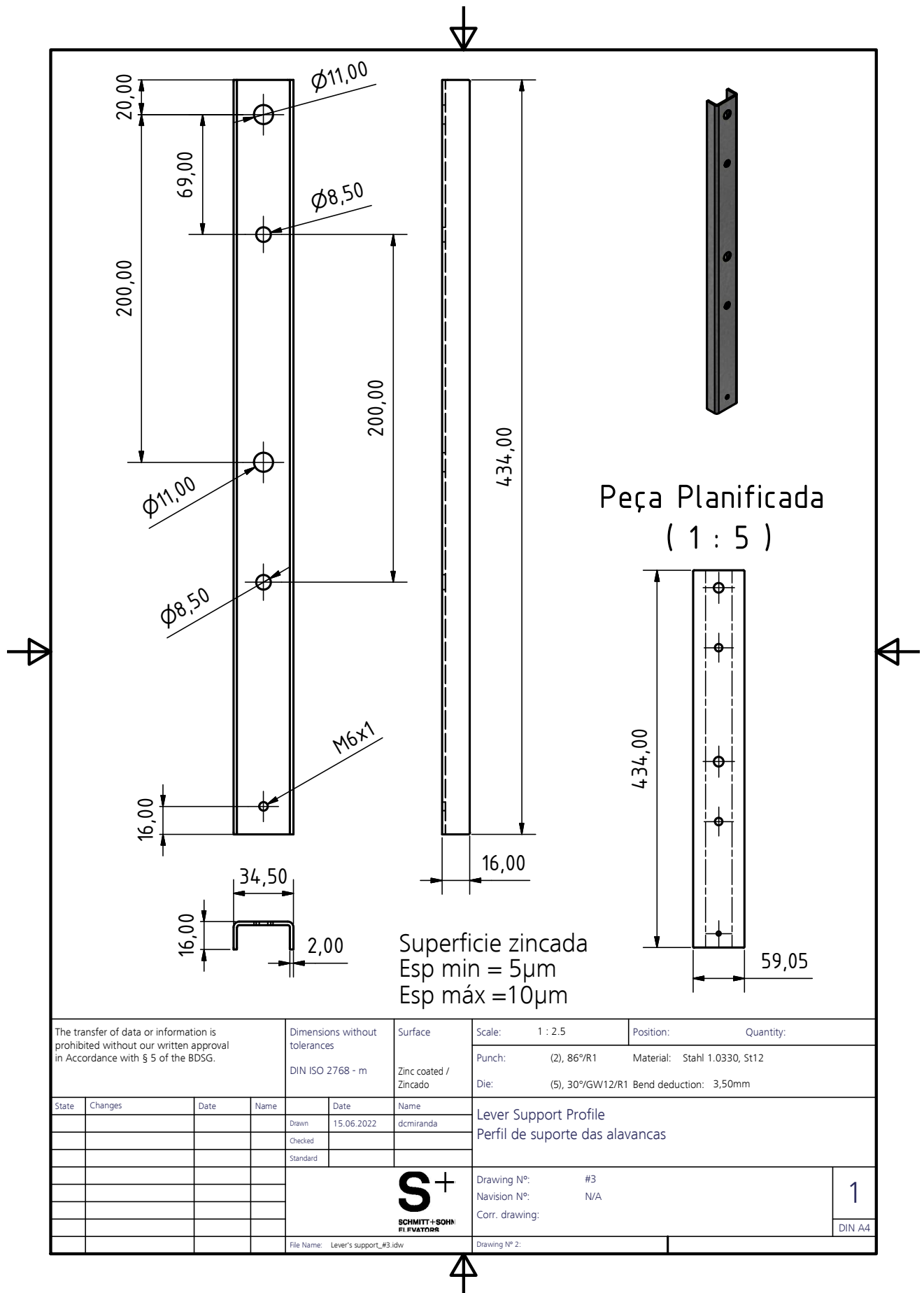
APPENDIX B.2.1 - Retractable Cam Base Plate



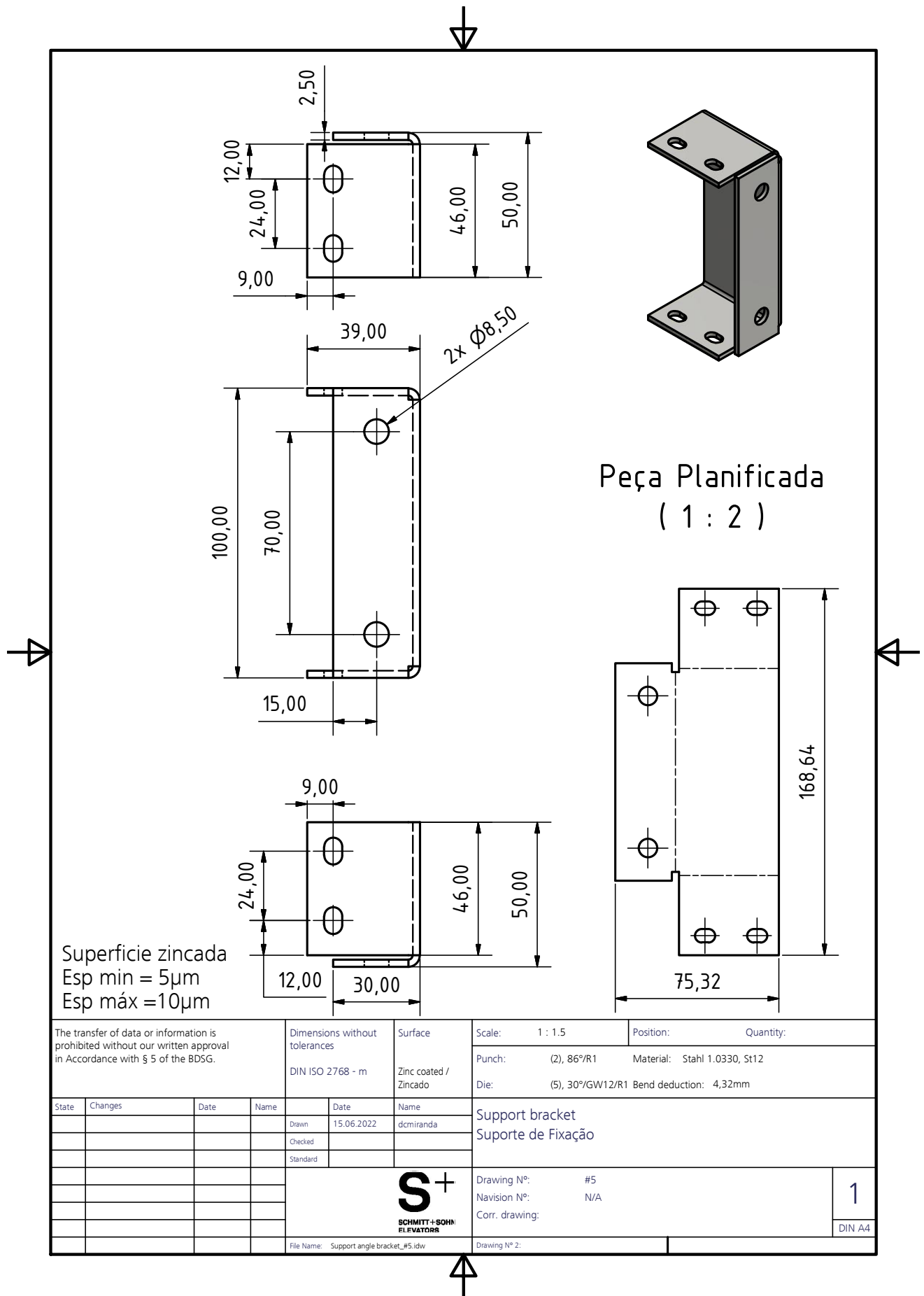
APPENDIX B.2.2 - Base Profile



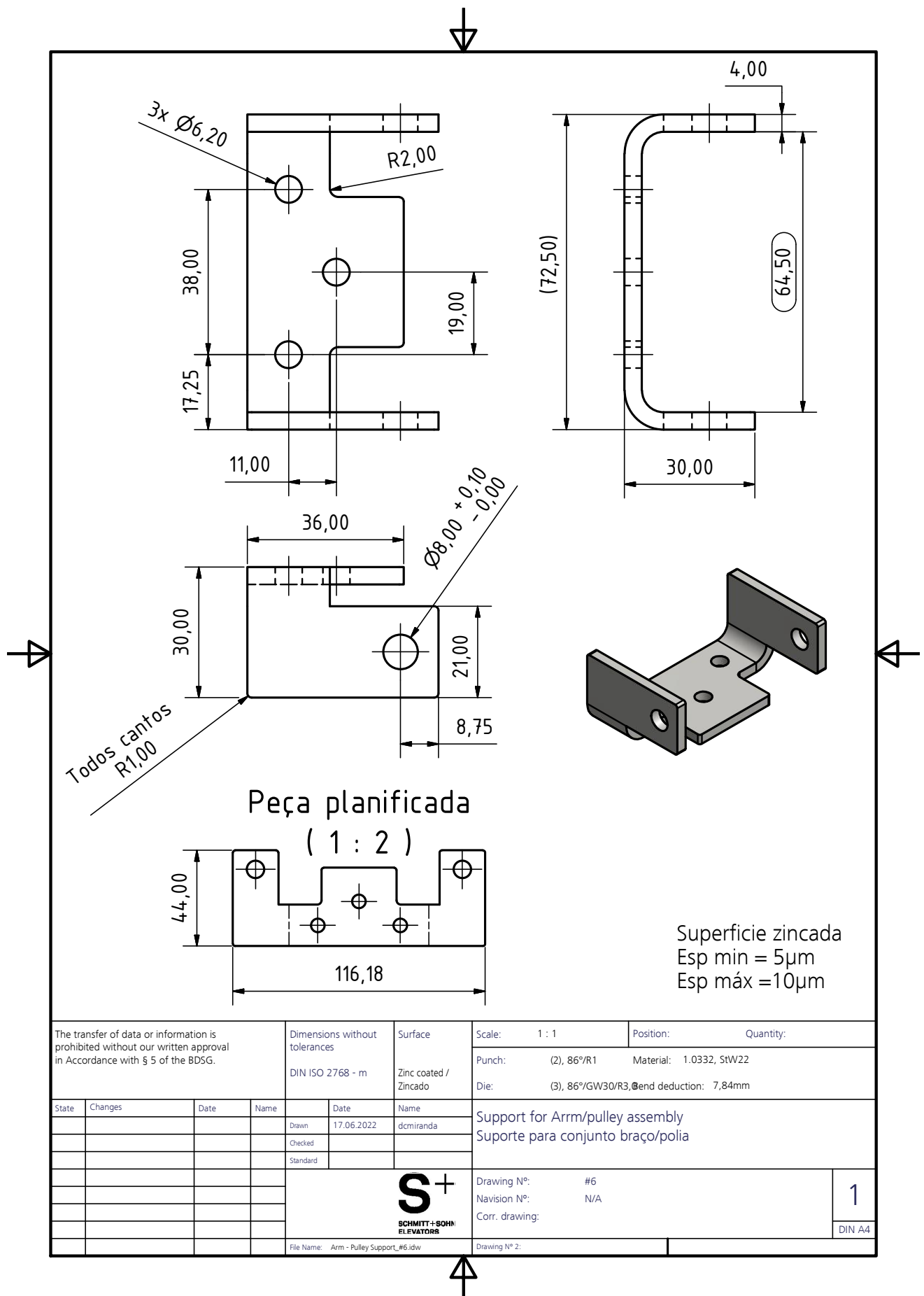
APPENDIX B.2.3 - Lever Support Profile



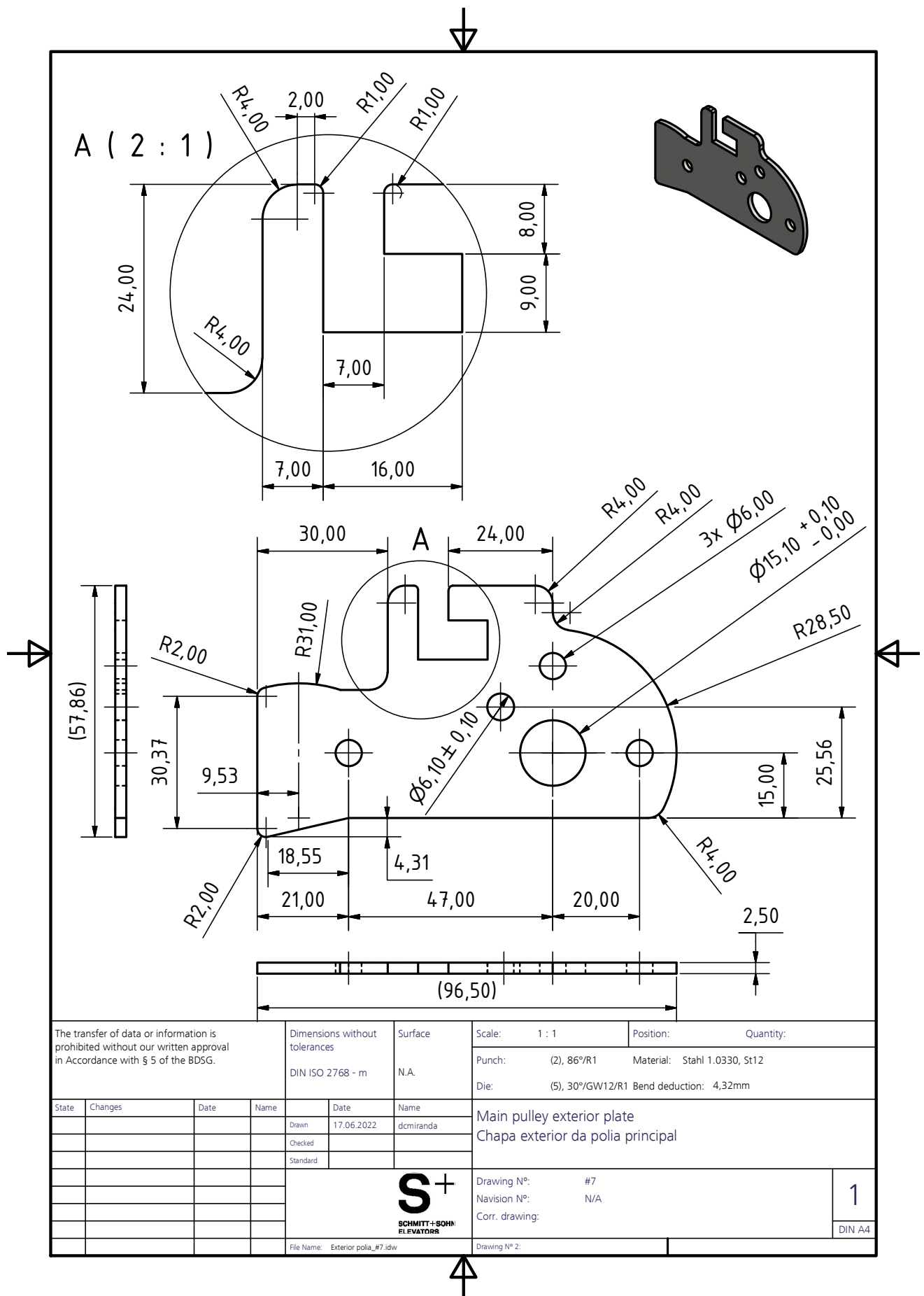
APPENDIX B.2.5 - Support Bracket



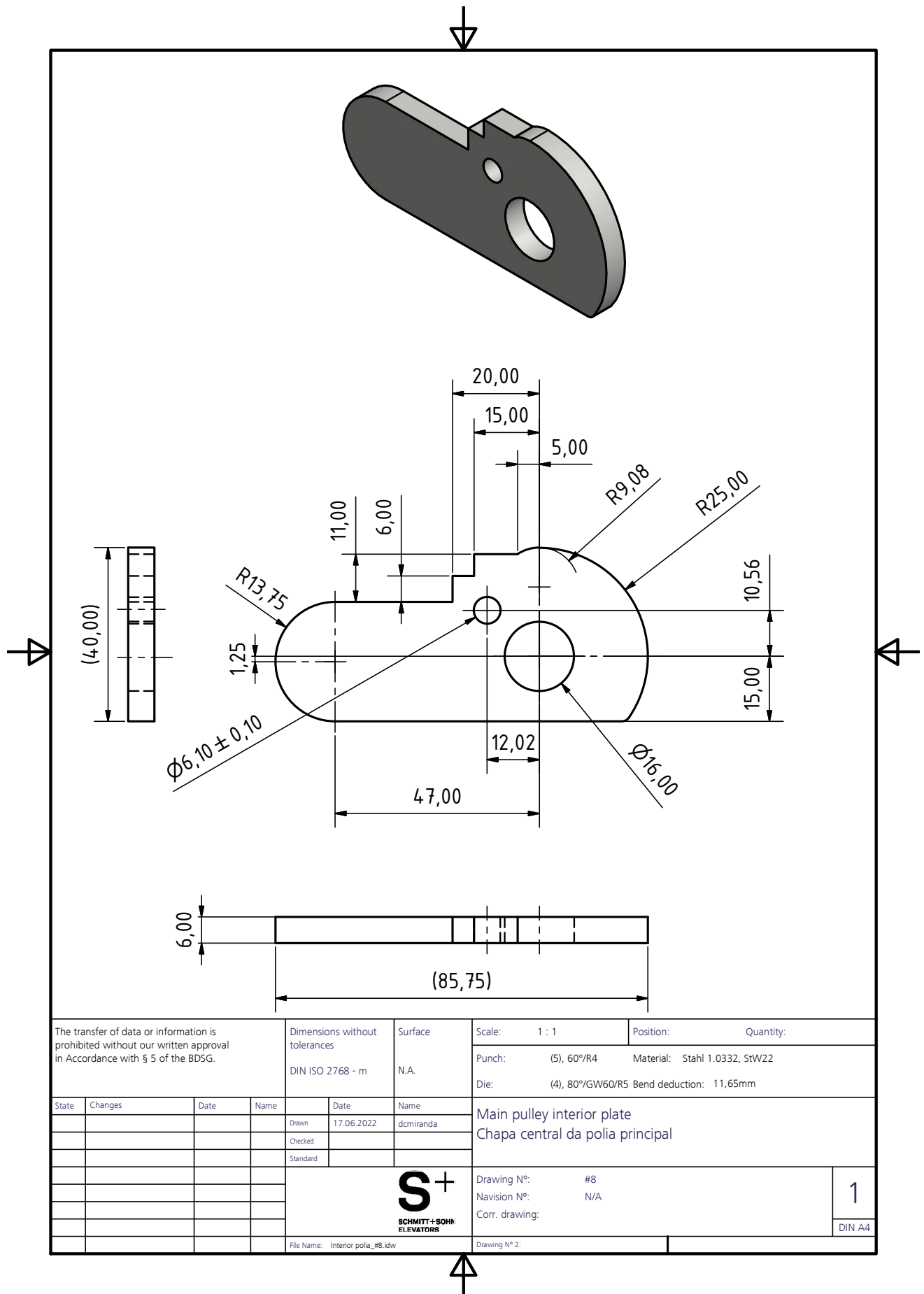
APPENDIX B.2.6 - Support for Arm/Main Pulley Assembly



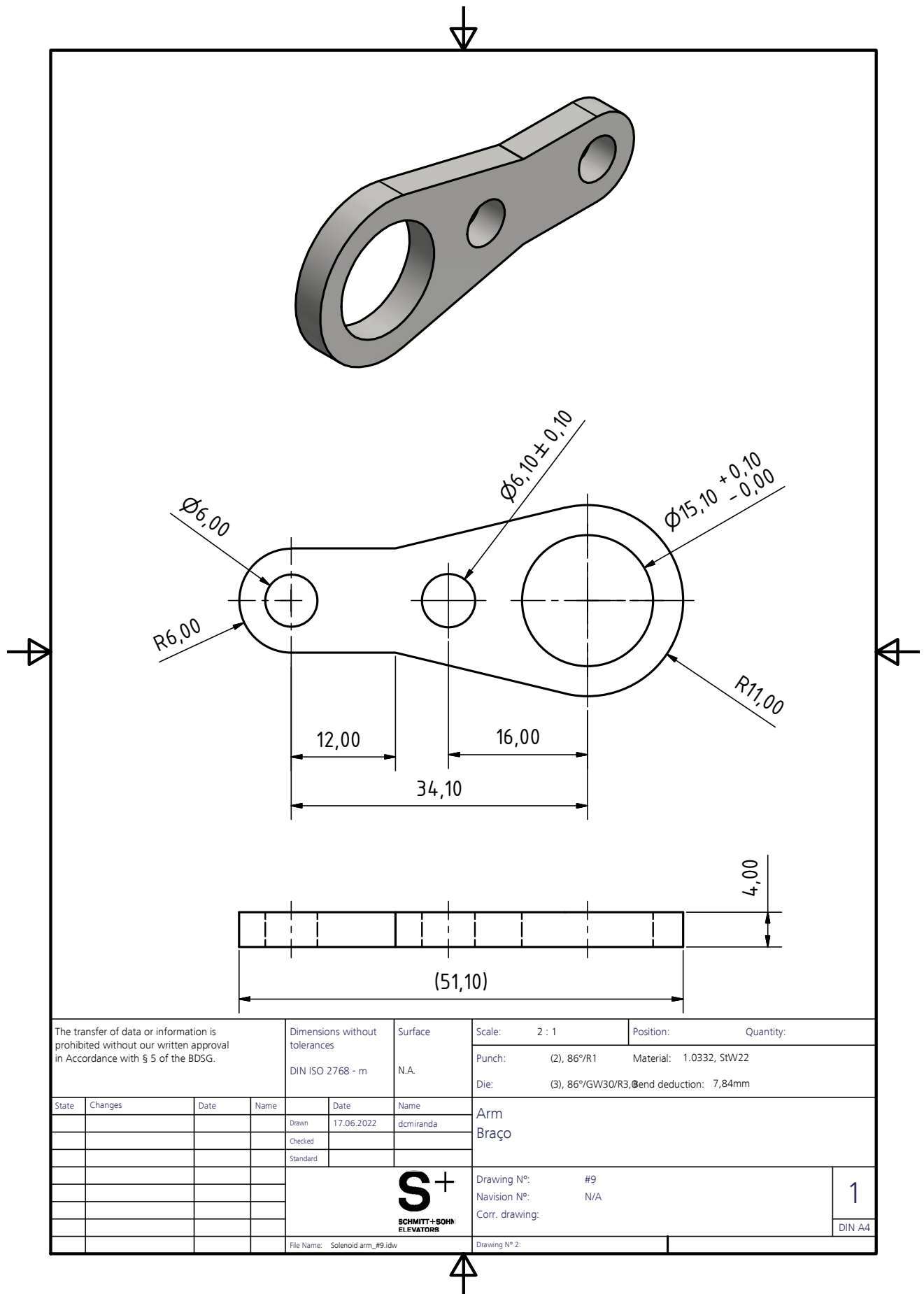
APPENDIX B.2.7 - Main Pulley Exterior Plate



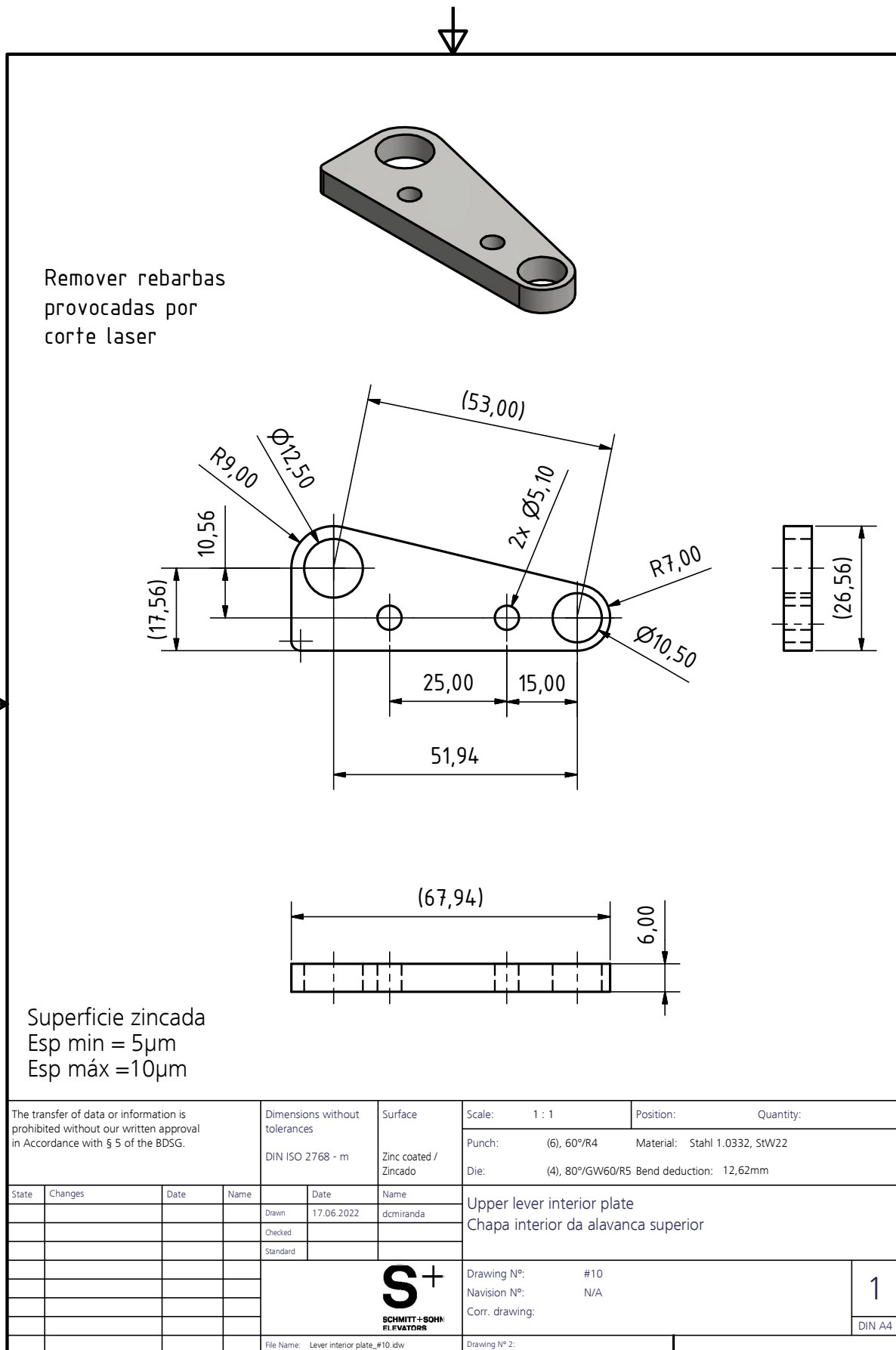
APPENDIX B.2.8 - Main Pulley Interior Plate



APPENDIX B.2.9 - Arm

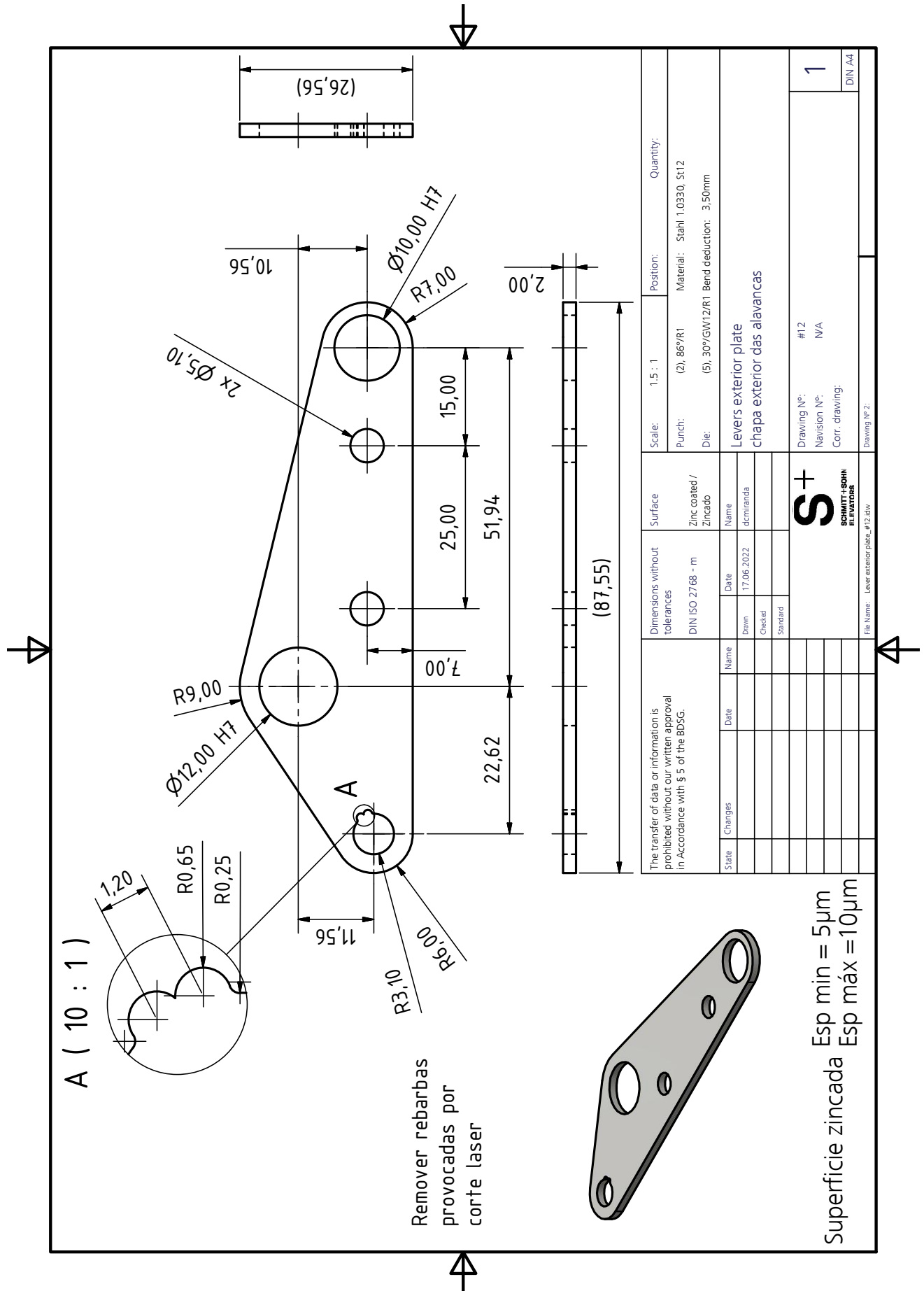


APPENDIX B.2.10 - Upper Lever Interior Plate

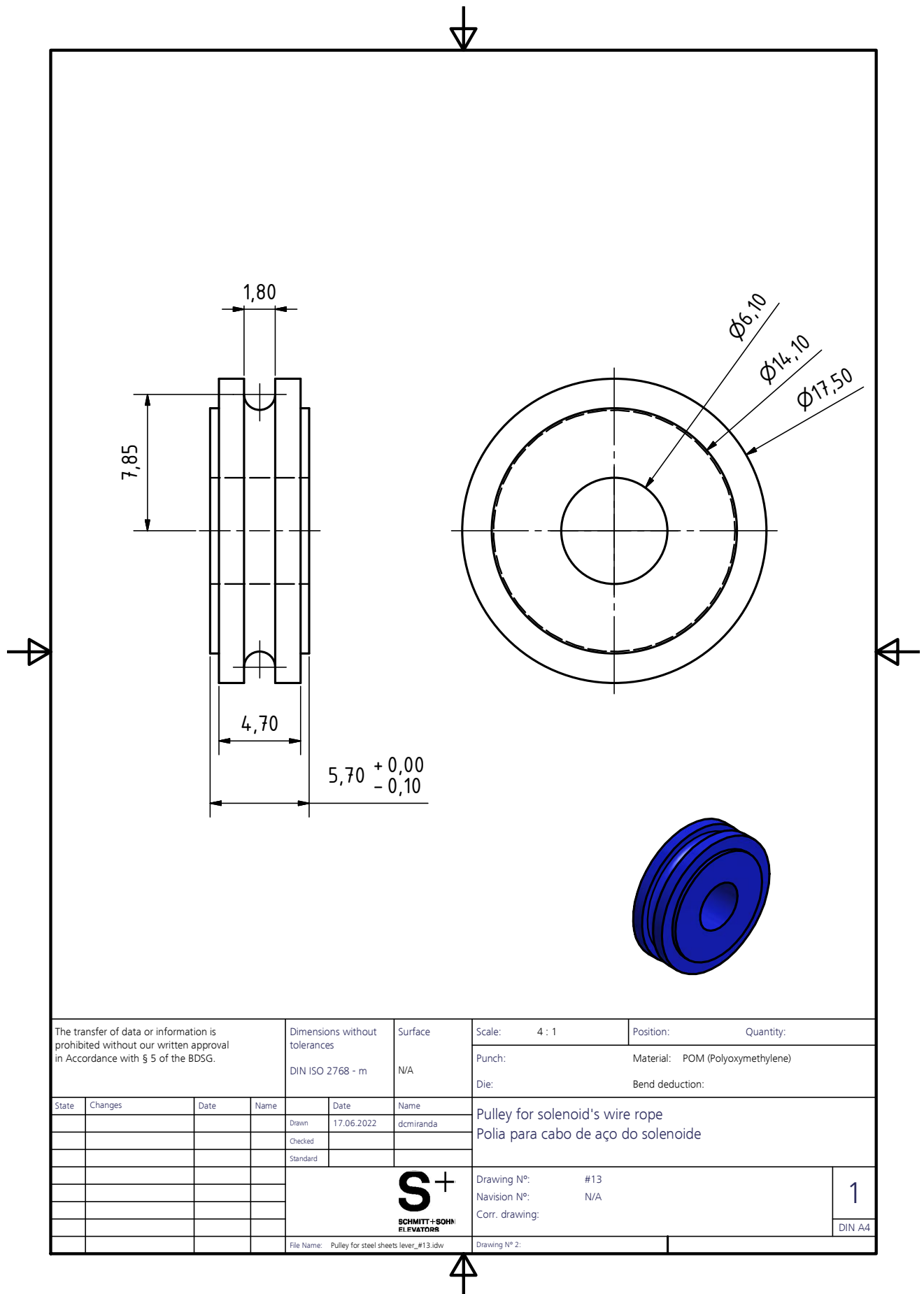




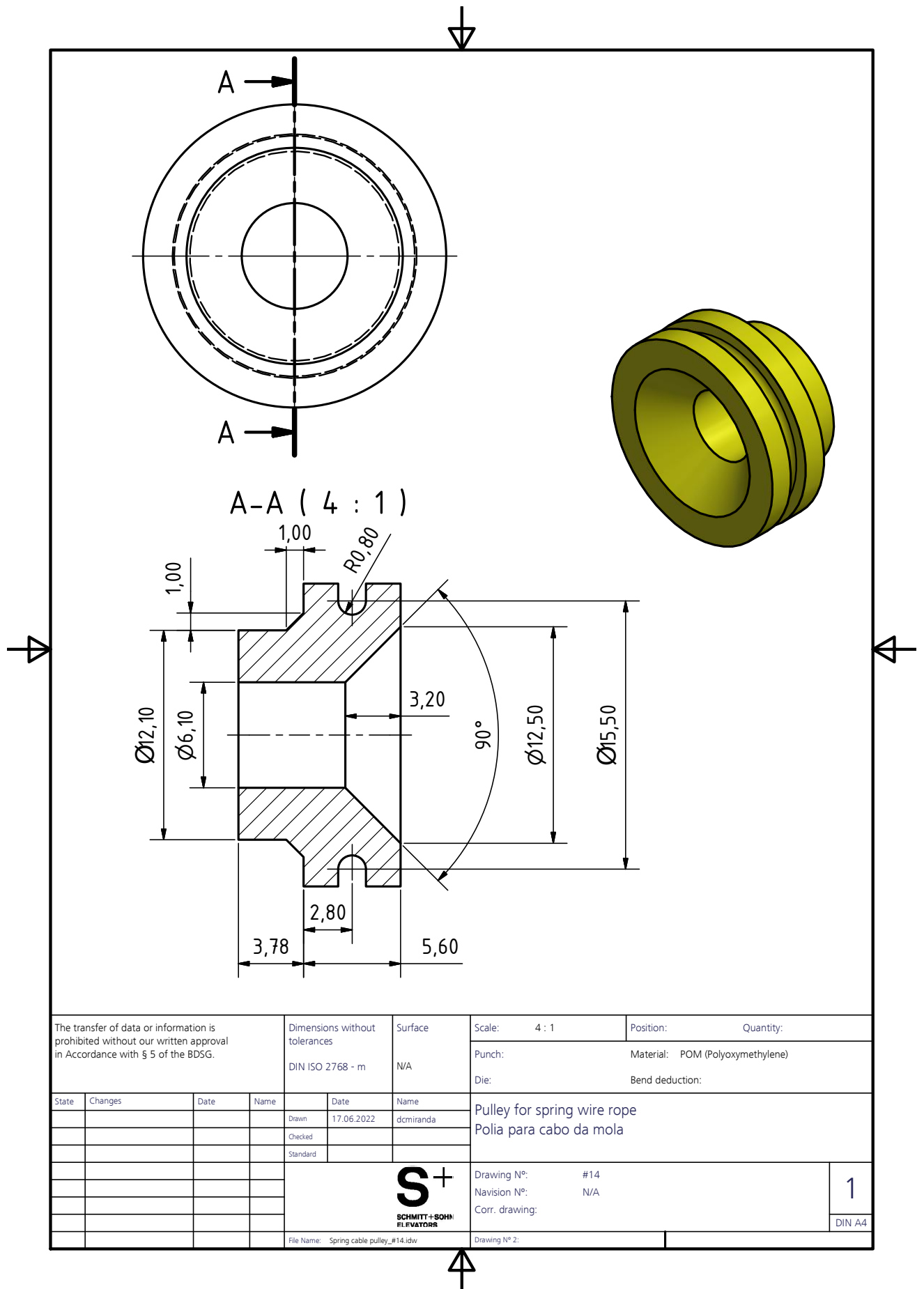
APPENDIX B.2.12 - Levers Exterior Plate



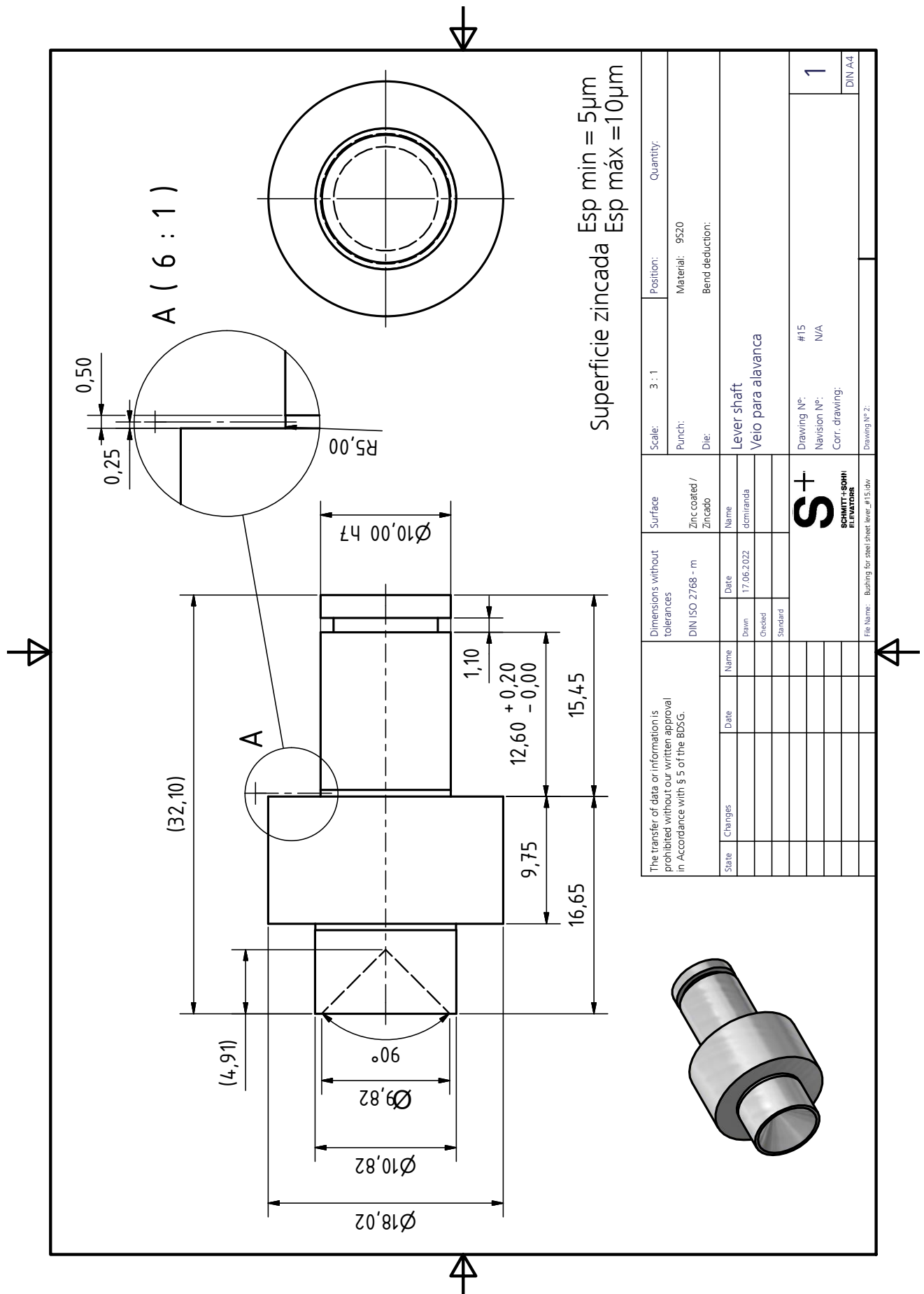
APPENDIX B.2.13 - Pulley for Solenoid's Wire Rope



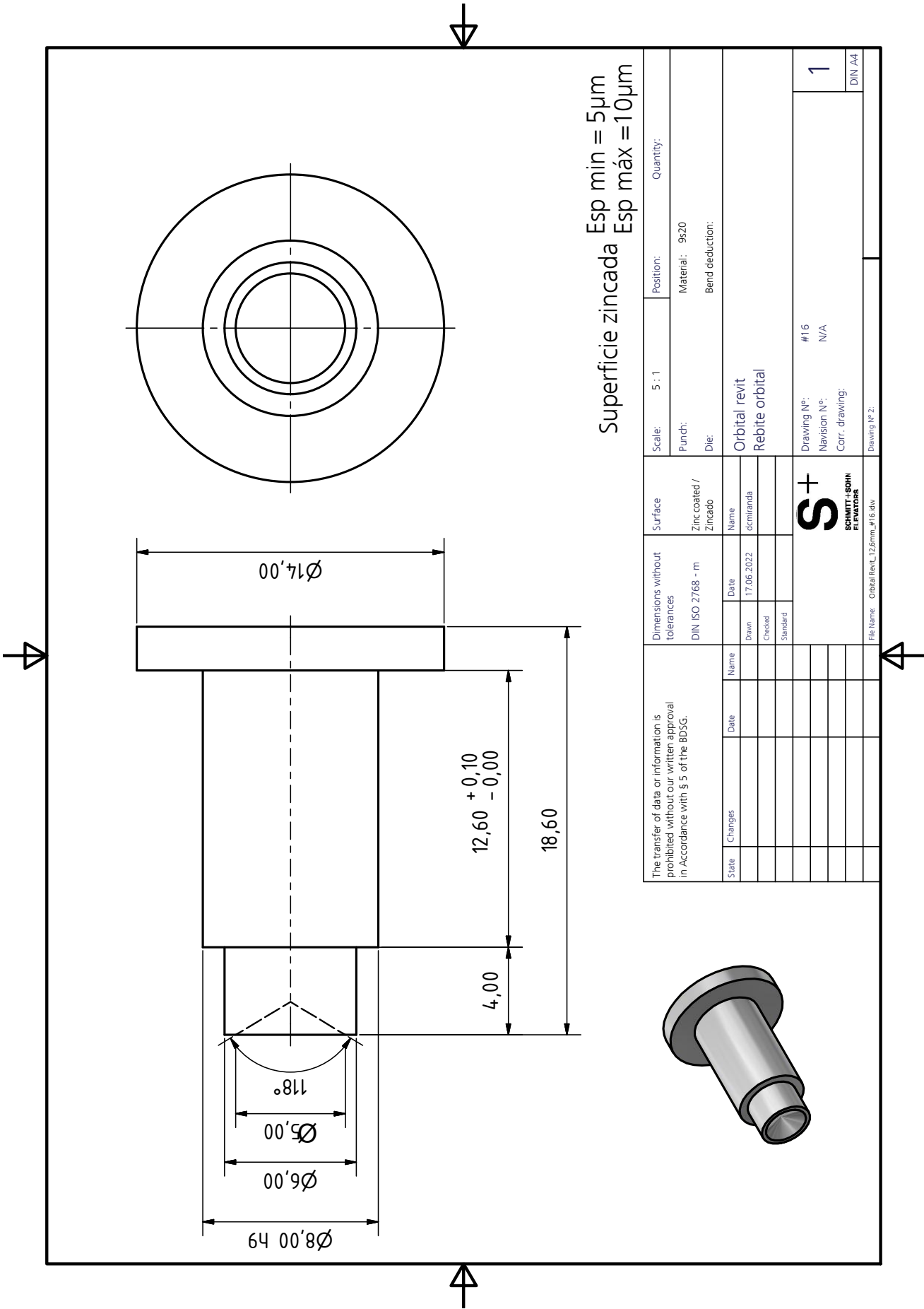
APPENDIX B.2.14 - Pulley for Spring Wire Rope



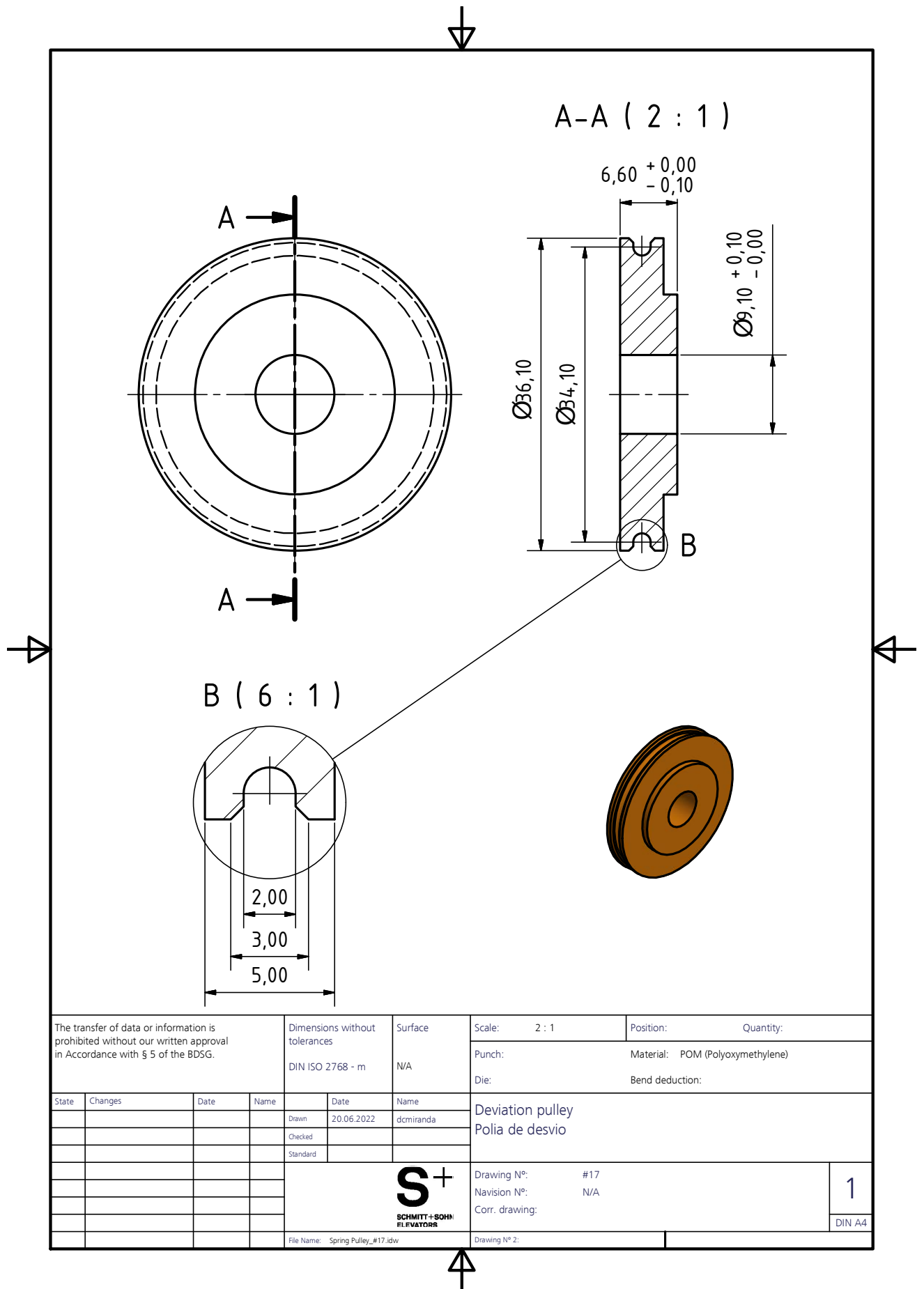
APPENDIX B.2.15. - Lever Shaft



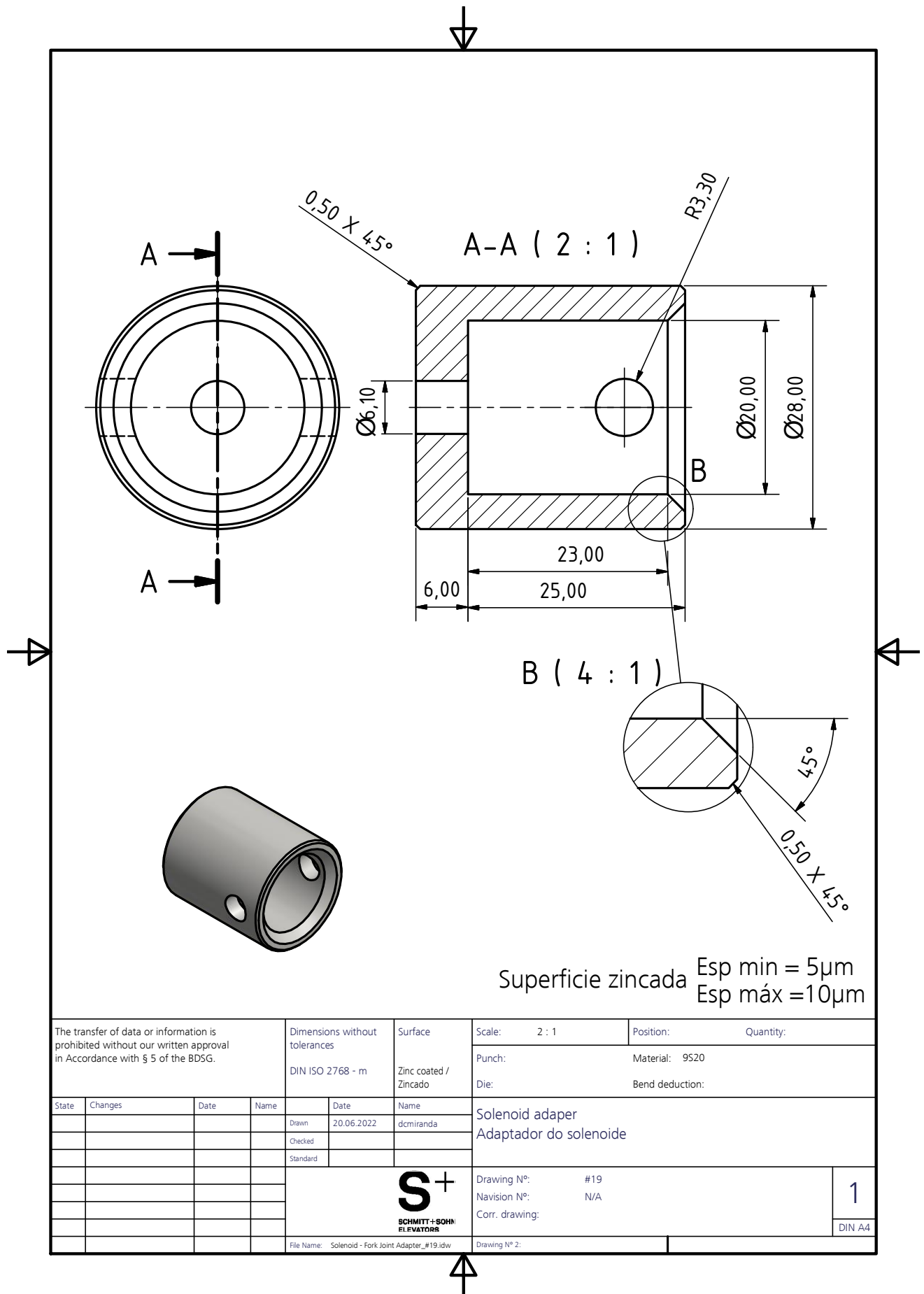
APPENDIX B.2.16 - Orbital Revit



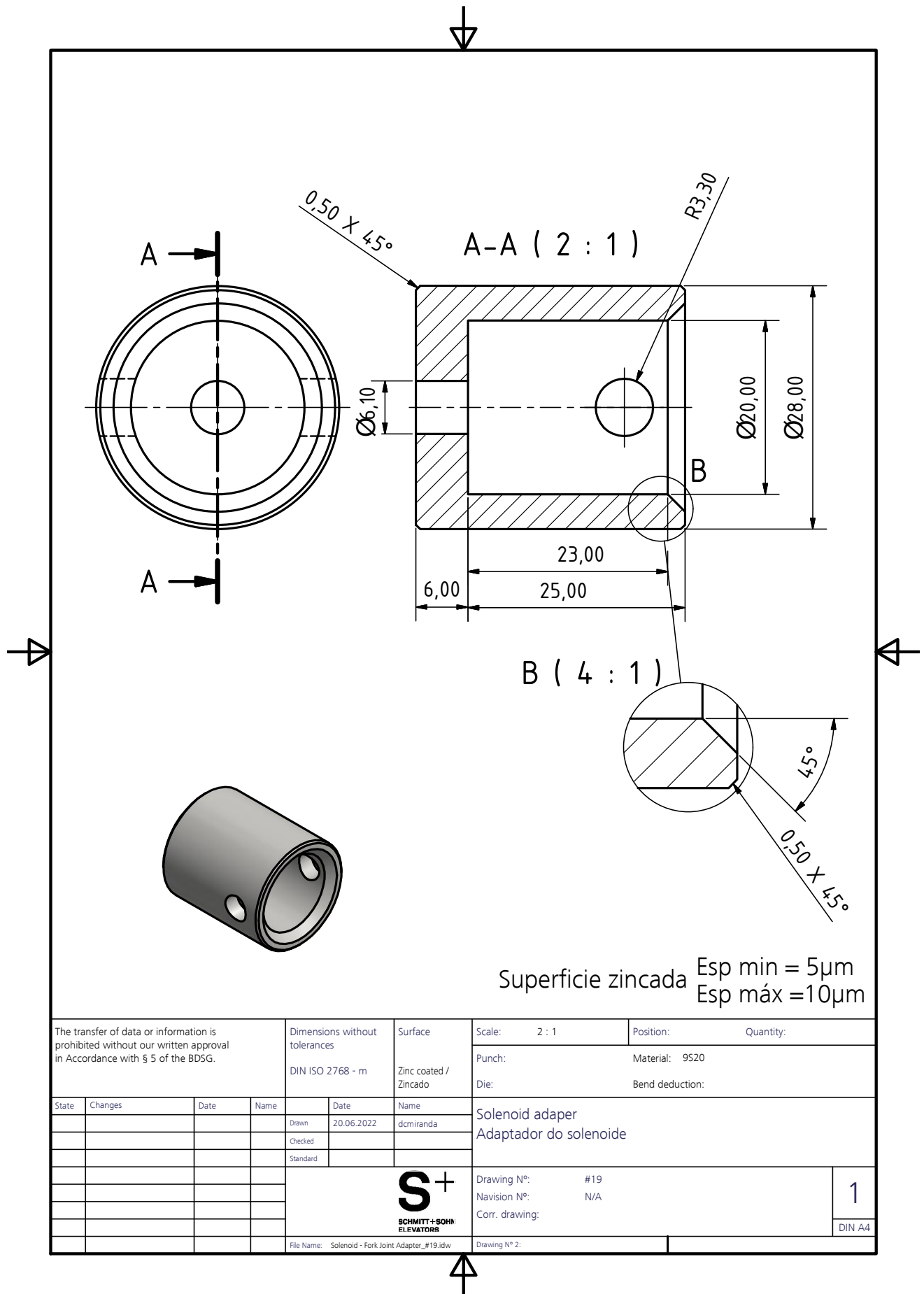
APPENDIX B.2.17 - Deviation Pulley



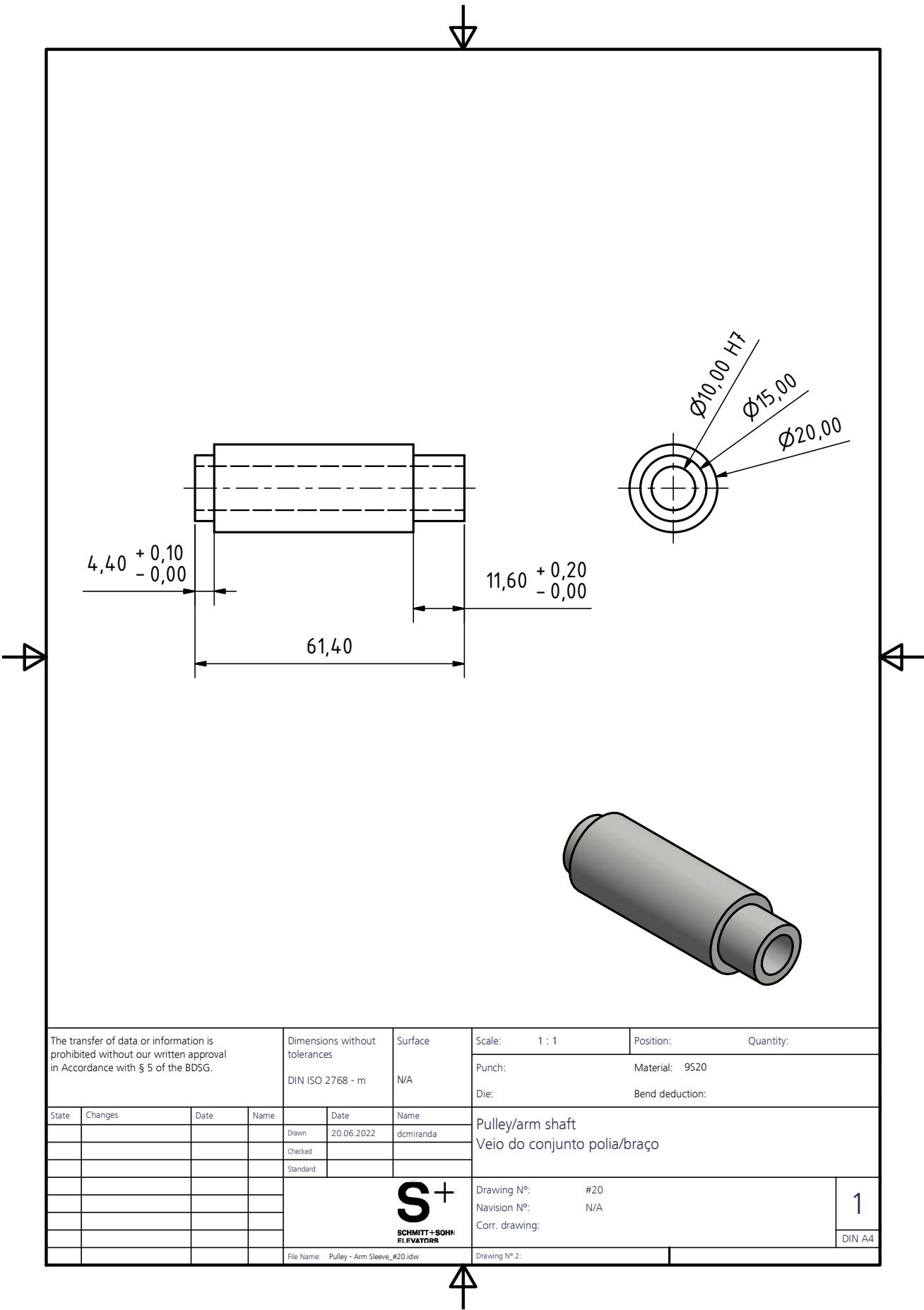
APPENDIX B.2.18 - Bushing for Deviation Pulley



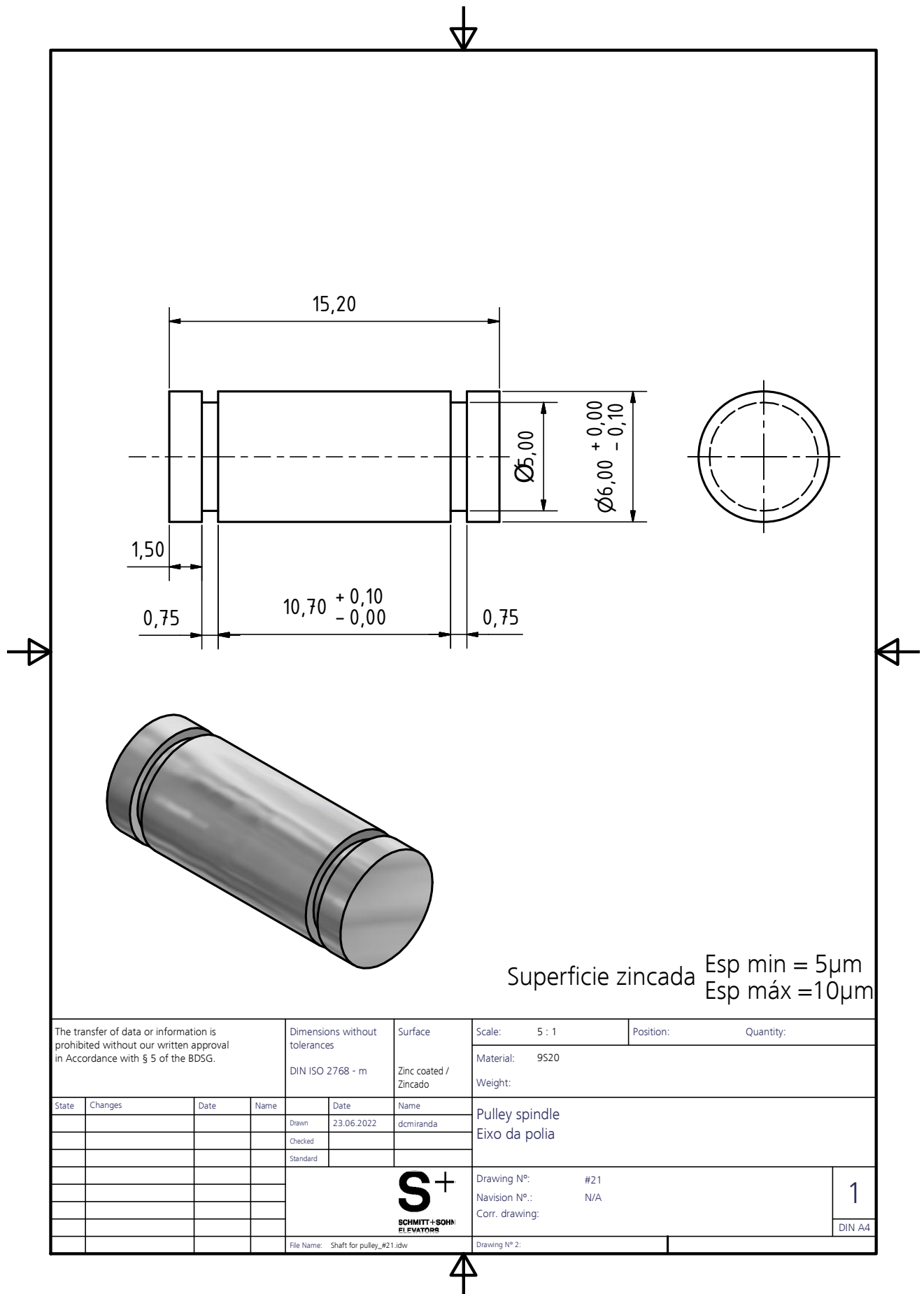
APPENDIX B.2.19 - Solenoid Adapter



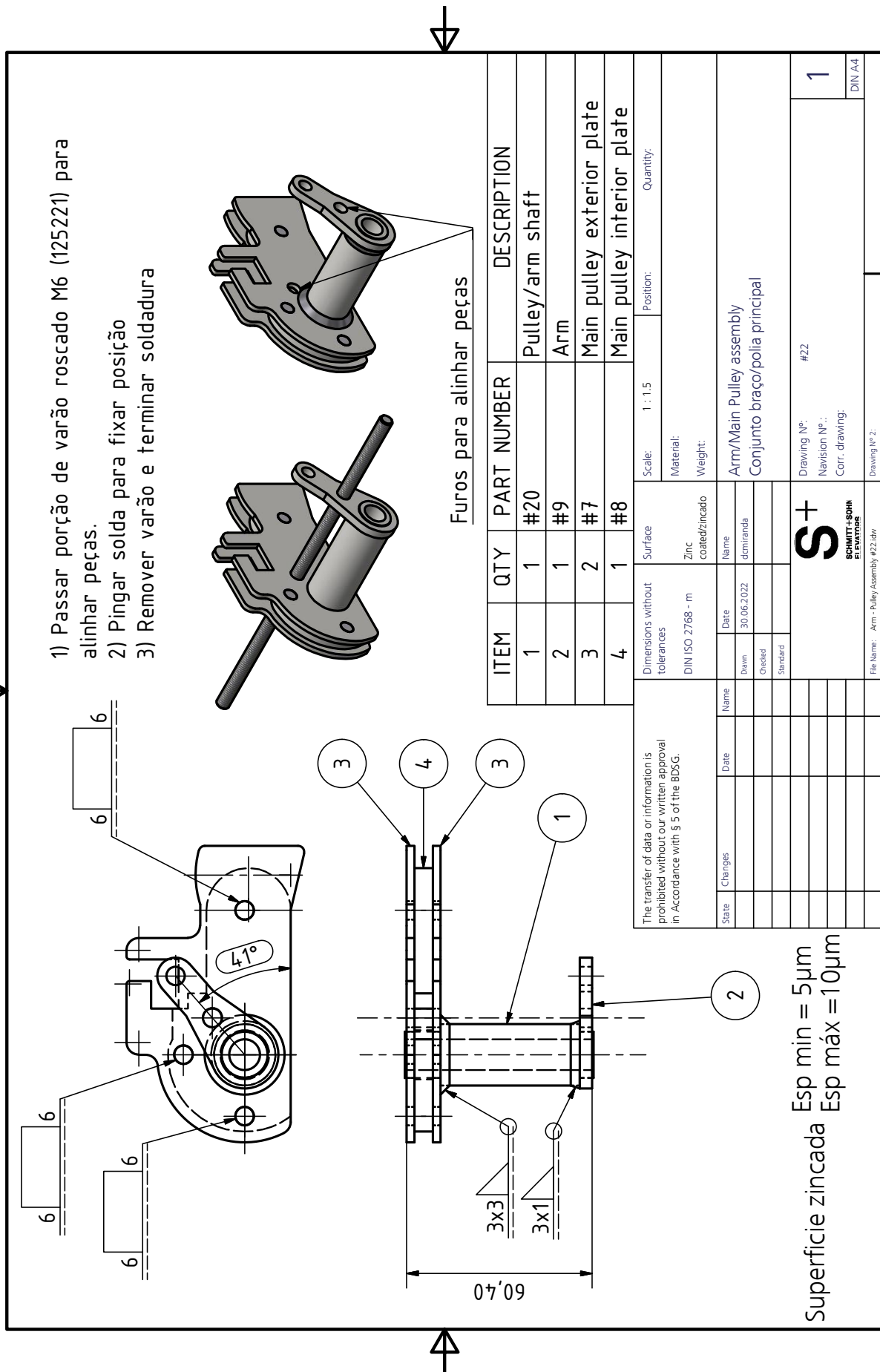
APPENDIX B.2.20 - Pulley Arm Shaft



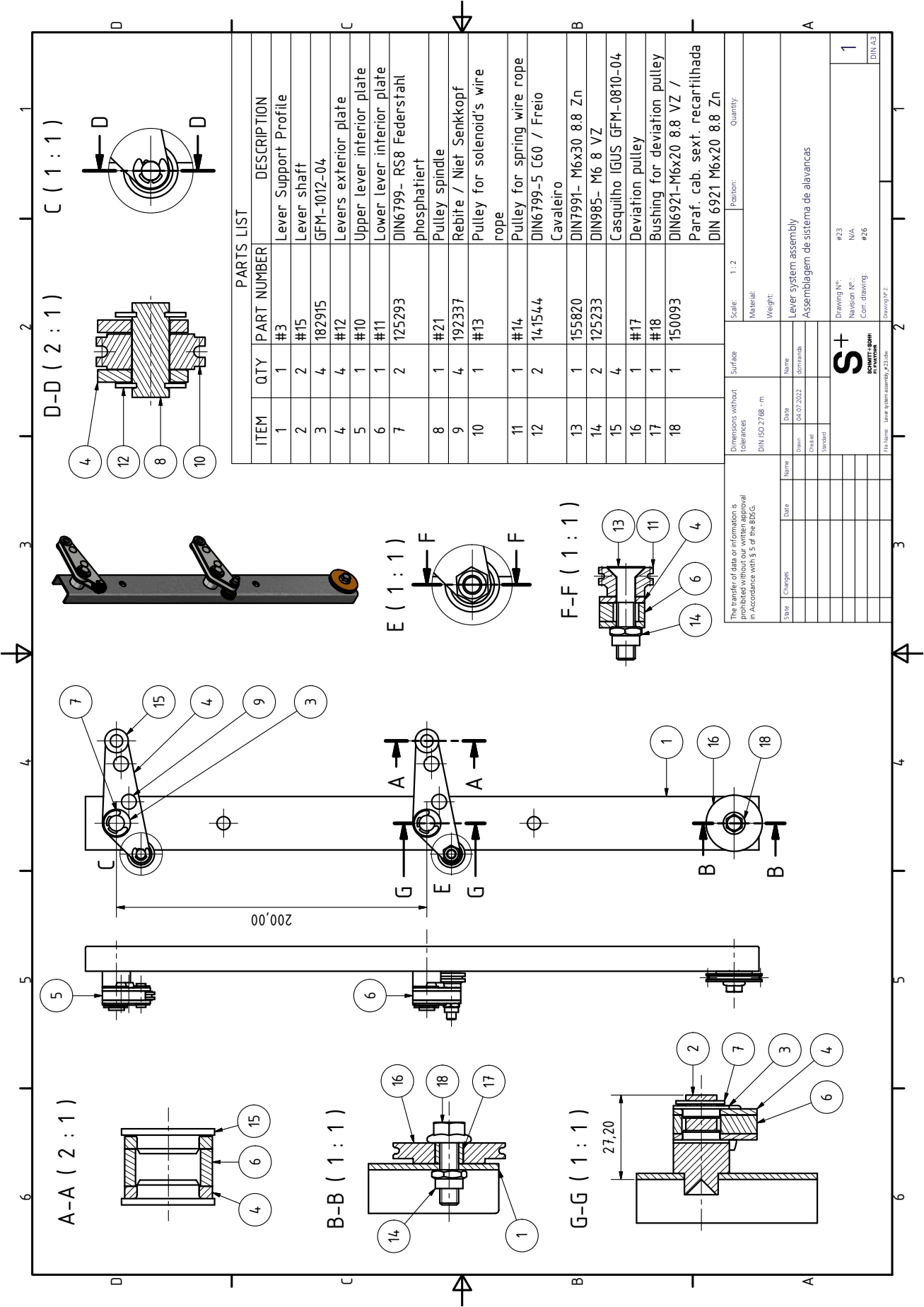
APPENDIX B.2.21 - Pulley Spindle



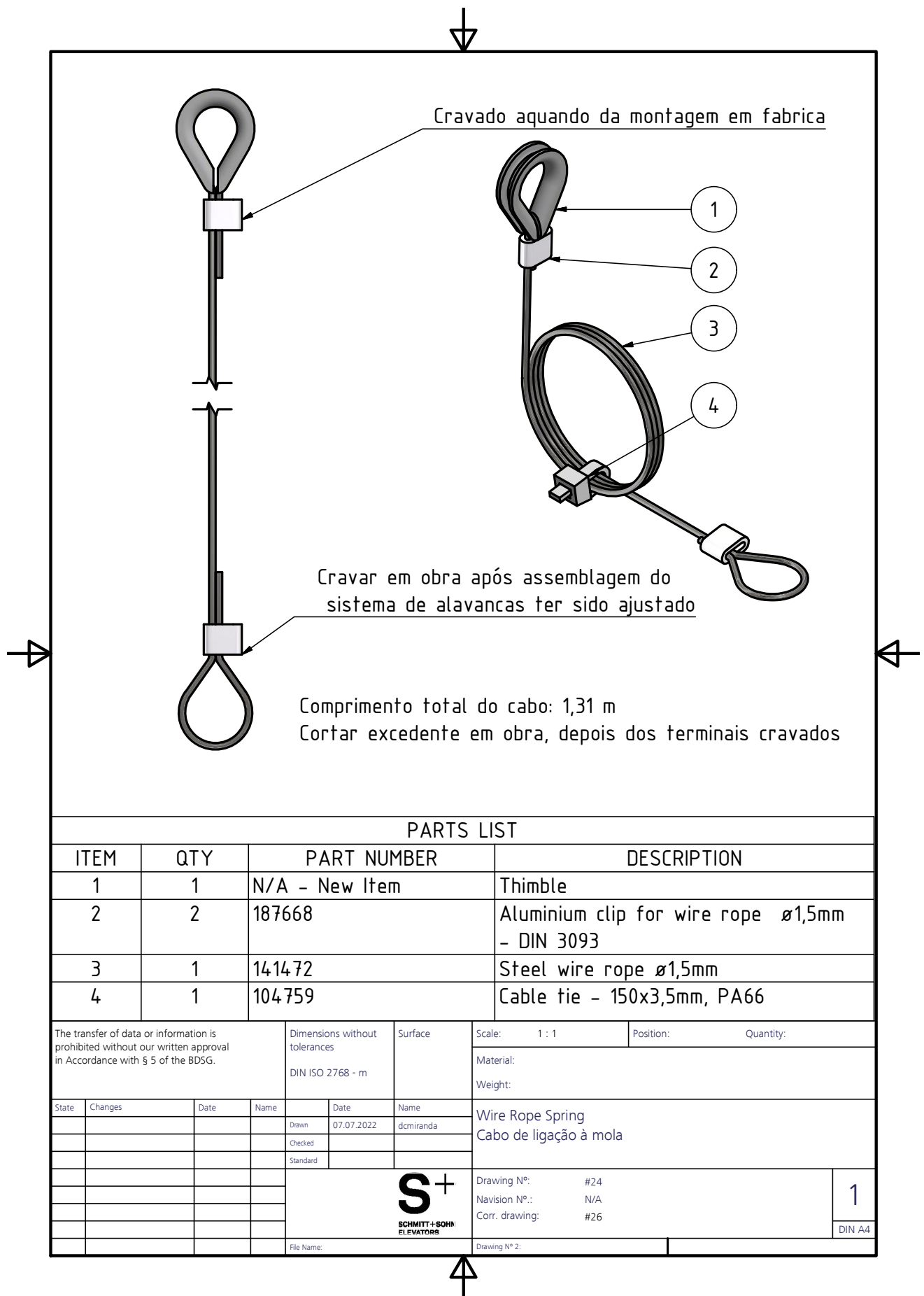
APPENDIX B.2.22. - Arm/Main Pulley Assembly



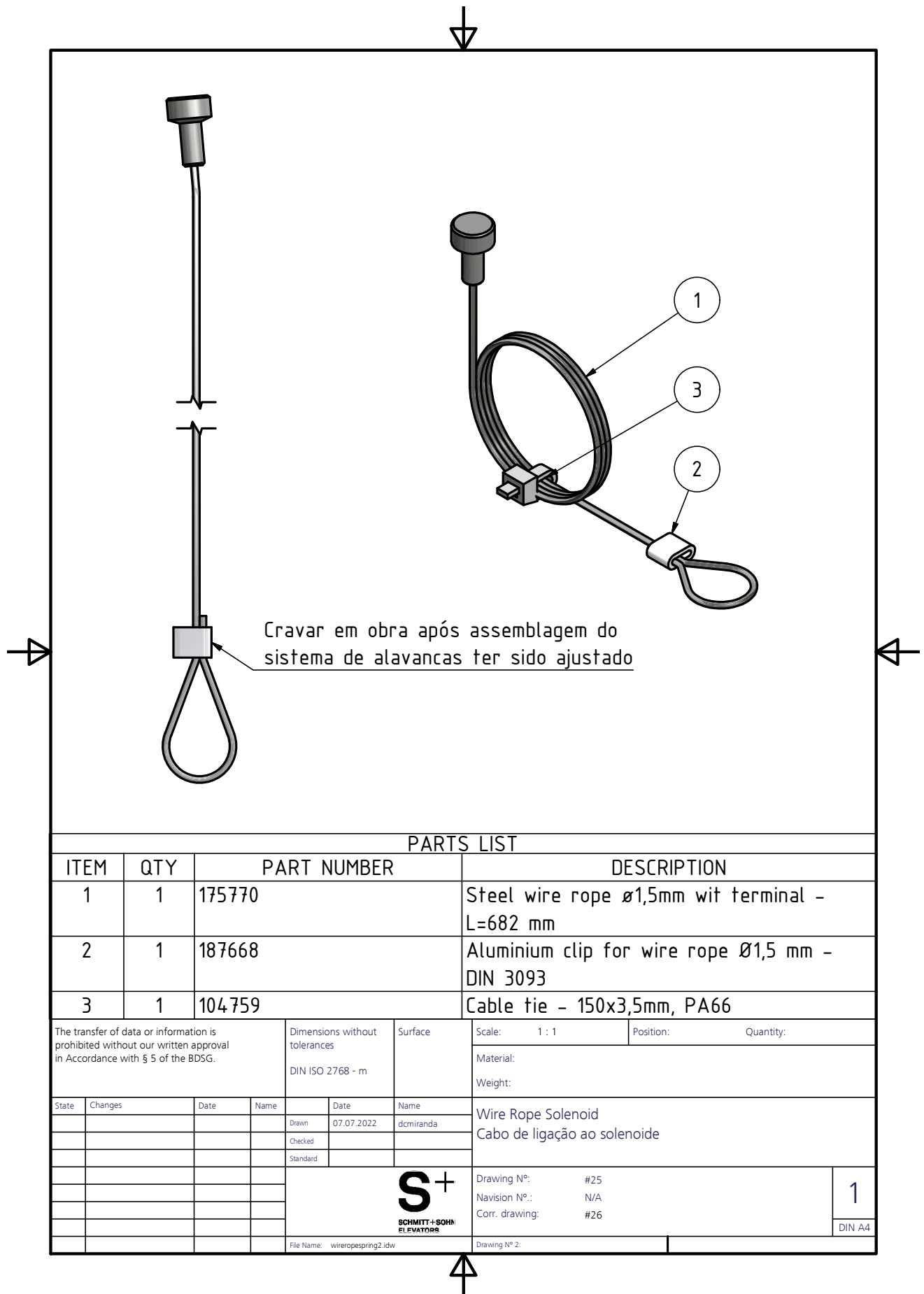
APPENDIX B.2.23 - Lever System Assembly



APPENDIX B.2.24 – Wire Rope Spring



APPENDIX B.2.25 – Wire Rope Solenoid



APPENDIX C

Bill of Material of Final Solution

APPENDIX C – Bill of Material of Final Solution

Component	Material	Standard/corresponding drawing	Qty. (un./m/m ²)
Solenoid MCI45/MCI60	Various	N/A	1
IGUS GFM1012-04	Iglide G300	N/A	4
IGUS GFM0810-04	Iglide G300	N/A	4
IGUS GFM0810-09	Iglide G300	N/A	2
Solenoid adapter	9S20	#19	1
Spindle 9S20	9S20	N/A	1
Spindle for pulley	9S20	#21	1
Pulley/arm shaft	9S20	#20	1
Lever shaft	9S20	#15	2
Orbital revit 12,6mm	9S20	#16	2
Bushing for deviation pulley	9S20	#18	1
Deviation pulley	POM	#17	1
Pulley for solenoid's wire rope	POM	#13	1
Pulley for spring wire	POM	#14	1
Revit	St Zn	DIN 7337 A-5*16	4
Screw eye M5x30*8	St Zn	N/A	1
Hexagon nut	St Zn	DIN 934 M5	2
Hexagon head screw	St Zn	DIN933 M8x18	1
Hexagon flange bolt with ribbed flange	St Zn	DIN6921 M6x20	10
Hexagon head bolt	St Zn	DIN 931 M8x85	1
Hexagon head screw	St Zn	DIN 933	2
Hexagon nut with serrated flange	St Zn	DIN 6923 M8 Zn	5
Hexagon nut with serrated flange	St Zn	DIN 6923 M6 Zn	6
Washer	St Zn	DIN 125 A6,4	5
Threaded rod	St Zn	DIN 913 M6x45	1
Fork joint with spring type	St Zn	DIN 71752 M6x12 St Zn	2
Hexagon head screw	St Zn	DIN 933 M8x35	1
Retaining washer	Spring steel	DIN 6799 5mm	4
Washer	St Zn	DIN 125 A5,3	1
External tooth lock washer	St Zn	DIN 6797 A5,3	1
Hexagon socket head cap screw	St Zn	DIN 6912	1
Spring lock washer	Spring steel	DIN 127 B8	4
Washer	St Zn	DIN 125 A8,4	2

Carriage bolt	St Zn	DIN 603 M8x30	2
Hexagon nut	St Zn	DIN 934 M8	2
Retaining washer	Spring steel	DIN 6799 8mm	2
Washer	St Zn	DIN 125 B8,4	4
Prevailing torque type hexagon nut with non-metallic insert	St Zn	DIN 985 M6	2
Hexalobular socket countersunk head screw	St Zn	DIN 7991 M6x30	1
Wire ropes clip	Aluminum	DIN 3093	2
Extension spring	Spring steel	DIN 2097	1
Rubber damper	Rubber	N/A	2
Rubber support washers	Rubber	Z004198	3
Spring wire rope with terminal (Ø 1,5mm)	N/A	N/A	1m
Spring wire rope (Ø 1,5mm)	N/A	N/A	1,31 m
Retractable cam base plate	DC01, t=2,5 mm	#1	0,0916 m ²
Base profile	DC01, t=2,0 mm	#2	0,0387 m ²
Lever support profile	DC01, t=2,0 mm	#3	0,0256 m ²
Ramp	DC01, t=1,5 mm	#4	0,0207 m ²
Support bracket	DC01, t=2,5 mm	#5	0,0127 m ²
Support for arm/pulley assembly	DD11, t=4 mm	#6	0,0051 m ²
Main pulley interior plate	DD11, t=6 mm	#8	0,0034 m ²
Main pulley exterior plate	DC01, t=2,5 mm	#7	0,0112 m ²
Arm	DD11, t=4 mm	#9	0,0011 m ²
Upper lever interior plate	DD11, t=6 mm	#10	0,0012 m ²
Lower lever interior plate	DD11, t=6 mm	#11	0,0023 m ²
Levers exterior plate	DC01, t=2,0 mm	#12	0,0093 m ²