

Master Thesis in Mechanical Engineering

Development of hole hemming tool for creating dissimilar joints in high performance vehicle structures

Author:

Adriano Gilberto Casal Conceição

Supervisor:

Dr. Mohammad Mehdi Kasaei

Co-Supervisor:

Prof. Dr. Lucas F. M. da Silva

Dr. Ricardo J. C. Carbas

Dr. Eduardo A. S. Marques

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Abstract

Nowadays, the transportation industry is receiving increasing pressure to reduce the weight of vehicle structures for the improvement of fuel consumption and air pollution performances while maintaining collision safety. To this end, new materials, e.g. advanced high-strength steels, aluminium alloys, magnesium alloys, and composites have been continuously developed and applied to manufacture lightweight structures. The realization of such multi-material vehicle structures depends on the processes used to join dissimilar materials. Newly, a novel joining process, termed 'hole hemming', has been developed by *Kasaei and da Silva* [1] for attaching sheets made from dissimilar materials in which only one of the sheets should be sufficiently ductile. The hole hemming process consists in producing a mechanical interlock between pre-drilled holes which can be made anywhere on the sheets. The process is carried out in a two-stage operation including flanging the hole of an outer sheet and bending the flange over the hole of an inner sheet.

The main objective of this dissertation is the development of a flexible tool, which allows the realization of this new process for different materials, with different thicknesses and geometries, under laboratory conditions. This dissertation includes the design, fabrication and assembly of the structural, active and passive components of the hole hemming tool and the evaluation of their performance. The tool is designed to have adjustable structural and passive tool components as well as easily interchangeable active tool components. The tool is evaluated for joining the AA6082-T4 aluminium sheet to the AZ31 magnesium sheet by experimental tests. The experimental data is also compared to the numerical results obtained from the finite element analysis of an ideal design. The results in both stages allow concluding the developed tool meets the considered requirements and can be efficiently used for creating hole-hemmed joints between dissimilar materials.

Resumo

Atualmente, o setor de transporte está a receber uma crescente pressão para reduzir o peso das estruturas dos veículos, de forma a melhorar o consumo de combustível e a poluição do ar, mantendo a segurança contra colisões. Para este fim, novos materiais, ex.: aços avançados de alta resistência, ligas de alumínio, ligas de magnésio e compósitos têm sido continuamente desenvolvidos e aplicados no fabrico de estruturas leves. A realização de tais estruturas de veículos multimateriais depende dos processos usados para unir materiais diferentes. Recentemente, um novo processo de união, denominado "hole hemming", foi desenvolvido por *Kasaei and da Silva* [1] para unir chapas de diferentes materiais, em que apenas uma das chapas deve ser suficientemente dúctil. O processo de hole hemming consiste em produzir uma ligação mecânica entre furos pré-perfurados que podem ser feitos em qualquer lugar das chapas. O processo é realizado através de uma operação dividida em duas fase, incluindo a criação da flange do furo de uma folha externa e o dobrar o flange sobre o furo de uma folha interna.

Como tal, o principal objetivo desta dissertação é o desenvolvimento de uma ferramenta flexível, que permita a realização deste novo processo para diferentes materiais, com diferentes espessuras e diferentes geometrias, em condições laboratoriais. Esta dissertação inclui o desenho, fabricação e montagem dos componentes estruturais, ativos e passivos da ferramenta de hole hemming. A ferramenta foi desenhada para ter componentes estruturais e passivos ajustáveis, bem como componentes ativos facilmente substituíveis. A ferramenta é avaliada pela união da chapa de alumínio AA6082-T4 à chapa de magnésio AZ31 por testes experimentais. Os dados experimentais são comparados com os resultados numéricos obtidos a partir da análise de elementos finitos de um processo ideal. Os resultados em ambas as fases permitem concluir que a ferramenta desenvolvida atende aos requisitos considerados e pode ser utilizada de forma eficiente para a criação de juntas entre materiais dissimilares.

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1.Introduction

1.1. Background and motivation

Because of high productivity, low cost and the ability to connect dissimilar materials, mechanical joining by plastic deformation such as clinching and self-pierce riveting, has received much attention in recent years [1].

Strict climate goals make the trend towards the use of lighter materials and, consequently, the increased need for flexible and versatile joining processes for joining multi-material systems.

Reducing vehicle weight is an important factor in achieving climate goals, particularly in the mobility sector, as it reduces emissions and energy requirements.

Currently, there are several different joining processes, however, their use is limited for joining different material systems as these processes are too restrictive and inflexible to react quickly to changing material requirements. In addition, other problems can be caused due to uneven stiffness, chemical incompatibility and different coefficients of thermal expansion of different materials. Another challenge for conventional joining processes is often the poor weldability of dissimilar metals.

To combat these problems, efforts have been made to create new joining processes and to develop existing joining processes that, due to their versatility, allow changing process conditions or batch variations [2].

To contribute to this new field of research, the Advanced Joining Processes Unit (AJPU) at FEUP, needs an appropriate tool to correctly perform a new joining process, called hole hemming, making it possible to test and evaluate this new process under development.

1.2. Objectives

With the development of a new process for joining materials, there is also the need to develop a tool that allows to carry out and test this new process. There are several parameters that influence the hole hemming process. So, in order to be able to test different parameters and conditions of the process under development, this tool must allow the joining of different combinations of materials, and for this to be possible in a practical and simple way, it is necessary to have:

- Adjustable structural and passive tool components.
- Easily interchangeable active tool components.

Under these circumstances, the main objective of this dissertation is to develop a flexible tool for the hole hemming process that allows joining dissimilar materials with very different mechanical properties. The presentation of this flexible tool is supported by experimental, numerical and analytical methods.

1.3. Methodology

First, a review of the literature on the design of tools was carried out. At the same time, the hole hemming process developed by *Kasaei and da Silva et.al.* [1] was studied and analysed to propose a proper concept for the required tool.

After analysing the hole hemming process by finite element simulations and understanding the necessary requirements, the study and analysis of the testing machine where the tool is mounted was carried out in order to understand its restrictions, dimensions and fixing methods.

Then, the structural components of the tool were designed using the principles of mechanical design and finite element simulations. After the validation of the design of these components, the respective orders of materials and their machining were carried out.

The same procedure was followed for the passive components of the tool, and after the end of this phase, the procedure was repeated for the active components of the tool.

Finally, the tool was mounted on the testing machine, and the experimental tests for joining lightweight materials were performed to validate the developed tool.

1.4. Dissertation outline

This dissertation is organized in five chapters, each one covering a different topic. In the present chapter, a brief introduction to the project is presented.

The second chapter addresses the literature review about joining by plastic deformation, tool design, and hole hemming process.

The third chapter corresponds to the design of the hole hemming tool, which is divided into 6 subchapters. The first subchapter addresses the requirements needed for the hole hemming process. The second subchapter corresponds to the restrictions, dimensions and fixing methods of the testing machine. The third subchapter refers to the first idealized settings for the tool. The fourth subchapter presents the final configuration adopted for the hole hemming tool. The fifth subchapter describes the development and design of the tool's structural components. In the sixth subchapter, the development and design of the passive components of the tool is presented. The seventh subchapter corresponds to the development and design of the tool's

active components. Finally, in the last subchapter, the tool assembly process on the testing machine is presented.

In the fourth chapter, the tool is tested to evaluate its performance, including the test specifications and the experimental and numerical results.

Finally, in the fifth chapter, this dissertation is concluded, and some suggestions are proposed for the future work on the hole hemming tool.

2. Literature review

2.1 Joining by plastic deformation

Due to the increasing pollution of the atmosphere, the automotive industry has been pressured to reduce the weight of automotive bodies in order to reduce the consumption of fuels. However, it is necessary to ensure that the decrease in automotive bodies is not associated with a decrease in collision safety. As a solution, new materials such as advanced high strength steels, magnesium alloys, aluminum alloys and composites have been employed for the manufacturing of automotive body parts. However, the development of lightweight automotive bodies requires new joining processes that have the ability to connect such parts made from dissimilar materials [1].

Plastic deformation, although is generally used in forming processes, can also be employed for joining materials, especially in cases of dissimilar materials where conventional joining methods are ineffective. Joining processes by plastic deformation are divided into two categories: metallurgical joining and mechanical joining [2].

Cold welding and friction welding are some examples of metallurgical joining processes, in this type of joining, first a break occurs between the contaminants at the interface and the oxide films due to a severe plastic deformation. After this phenomenon the resulting clean surfaces are joined through high interfacial pressure. However, to obtain this type of bond, a large plastic deformation is required. This type of process can also be called solid-state welding [2].

In mechanical joining by plastic deformation a mechanical interlock is produced between sheets by creating material plastic flow, which connects them without the need for melting and metallurgical bonding [2]. This type of joint, due to its low heat input, robustness to metallurgical incompatibilities and other process variants, is the joint that provides the creation and development of versatile joining processes [3].

Due to the large number of processes by mechanical joining, the need arises for a classification of these processes. There are numerous classifications for these processes, however, the alternative classification that *Hahn et.al.*[4] developed was considered. In its classification, essential distinguishing features are used in the sense of an ordering scheme [3].

The use or non-use of an auxiliary element for joining is a good example of a distinctive feature, and it is still possible to distinguish this type of process through the need of a pre-hole or the necessary accessibility. As for the processes where an auxiliary joining element is not required,

the distinction can be made through the performance or not of a cutting operation or the shape of the joint [3].

In figure 1 a scheme of this classification is presented. In this scheme recent developments are added, and non-round joint shapes are combined, however multi-stage processes and function carriers are not considered [3].

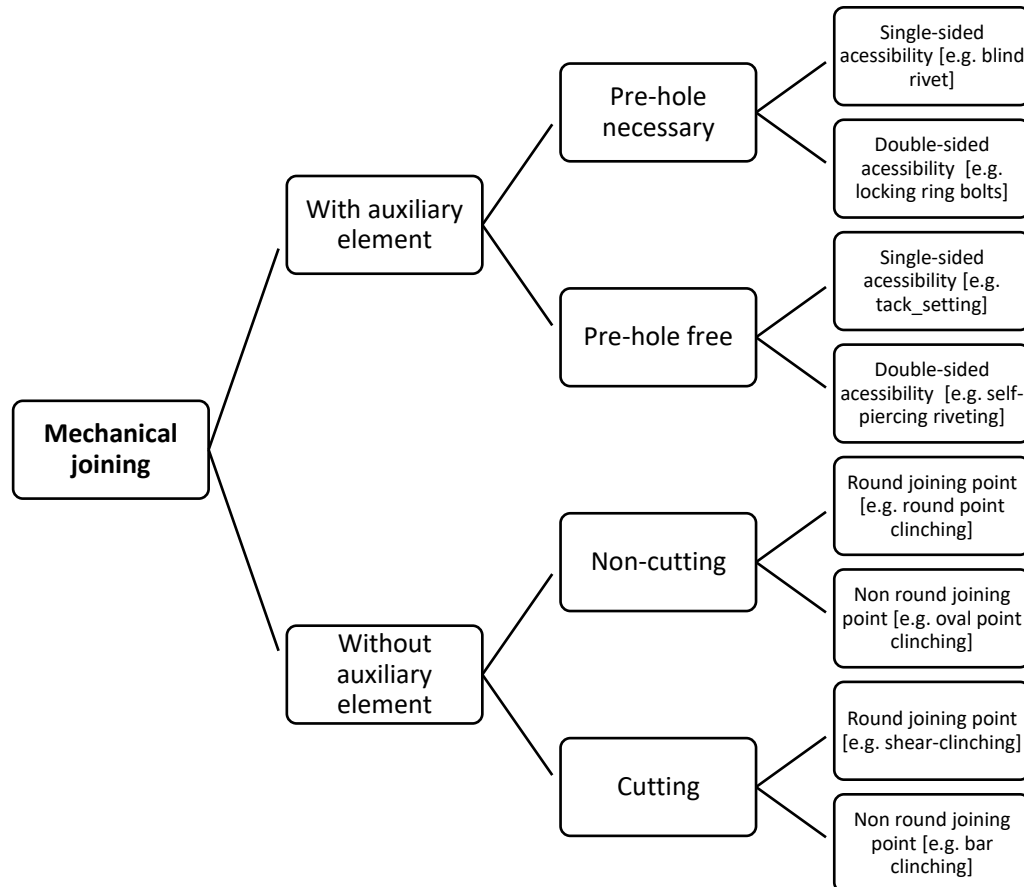


Figure 1- Classification scheme of mechanical joining processes [3].

In these processes, in order to obtain a proper interlock, either both materials must be adequately ductile, or an auxiliary element must be used, which increases the joining cost. Among all the processes, clinching and self-pierce riveting stand out due to their high productivity, low cost and ability to join different materials.

In self-piercing riveting, the sheets are pierced by a tubular rivet without pre-drilling. Two or more sheets are joined by creating force-fit and form-fit joints using rivets as mechanical fasteners. Compared to the more traditional methods of sheet material joining, the advantages offered by self-piercing riveting include the ability of joining dissimilar materials. Self-piercing riveting can principally be used for the joining of aluminum alloys with mild steels, with copper alloys, with titanium and magnesium alloys. These joints are also believed to have good mechanical strength, in particular, fatigue strength. However, the joining becomes more difficult

when there are large differences regarding the strength of the materials to be joined and the arrangement of the high strength steel has an impact on the joinability [5]–[7].

Clinching is a joining process without additional or consumable joining elements, where only plastic deformation of both workpieces leads to the joining. In this process, two sheets either metallic or non-metallic, stack up, and experience a locally severe plastic deformation applied by clinching equipment, without auxiliary elements. The joining mechanism is similar to that of self-piercing riveting, while no rivet is used. Although the advantages and disadvantages of mechanical clinching are similar to those of self-piercing riveting, joining cost and time of mechanical clinching is less than that of self-piercing riveting because of no rivet, whereas the strength of the joint is lower. Today, clinching is a state of the art joining method in the sheet metal processing industry and can be used for materials with thicknesses from 0.1 mm for single sheets and up to total thicknesses of 12 mm. However, this method is only applicable to materials of relatively high formability due to the locally severe plastic deformation of the workpieces, which may lead to cracking of the workpieces of lower formability [6]–[8].

It is still difficult to join materials with low ductility or high strength, especially when joining aluminum alloys to advanced high strength steels or to magnesium alloys that are often required for the manufacturing of automotive bodies. The materials with low ductility tend to fracture due to plastic deformation while it is difficult to pierce high strength steel sheets in self-pierce riveting [9].

For joining this type of material combinations (a material with high strength and low ductility and a material with good ductility) the hole clinching process was proposed [10]. In this process, the high strength sheet with low ductility on the die side is punched and the sheet with good ductility on the punch side is indented into the hole. However, because the die side sheet cannot support the punch side sheet, the sheet with good ductility is damaged more quickly than in the conventional clinching process. Thus, a shorter forming stroke should be selected in the hole clinching process. In addition, the punch-sided material, which is weaker, is responsible for the strength of holed-clinched joints. The placing of a high strength and low-ductile material on the punch side not only increases the possibility of fracture, but also poses a serious challenge to the process of forming a mechanical interlock [1].

To try to solve this problem, punching and clinching operations were combined in a single-stage process called shear clinching. Although the process requires a complex tool setup, it reduces operation times and eliminates the difficulties of positioning the two sheets. Because of the formation of two necks in the punch sided sheet, the sheet with good ductility is more severely

deformed and, consequently, the load capacity of joints decreases as compared to that in the hole clinching process [11].

There is a process of joining by plastic deformation, called hemming, which is capable of joining dissimilar materials beyond the limitations explained above. In the hemming process, the edge of an outer sheet is bent over an inner sheet to join two sheets together. The process is carried out in a multistage operation including flanging, pre-hemming and final hemming. This way the inner sheet is not deformed, and no additional joining elements are required. Thus, only by controlling the plastic deformation of a sheet, the mechanical interlock can be formed [1].

According to the type of equipment, the hemming processes can be categorized into:

- **Die hemming**, this process is suitable for mass production.
- **Roll hemming**, this process has high flexibility for complicated parts.

The hemming process is widely employed in the automotive industry to assemble the inner and outer panels of automobile doors, hoods, or deck lids [12].

The occurrence of cracks on the outside surface of the outer sheet that may lead to fracture is the main defect of hemmed joints. The possibility of fracture in the roll hemming process is less than that of the die hemming process because the sheet is gradually deformed in the former. Despite the capability of the joining of dissimilar materials, the hemming process has attracted less attention than the other joining processes by plastic deformation that goes back to the fact that it can be only applied to sheet edges [13]–[15].

Under these circumstances, *Kasaei and da Silva et.al.* [1] have been developing a new hemming-based joining process, called “hole hemming”. The hole hemming process enables to join dissimilar materials with different mechanical properties, and without limitation in the joint position. In this process, the hole edge of an outer sheet is bent over an inner sheet to join two sheets together. The process is carried out in a two-stage operation including Hole Flanging and Hemming so that the inner sheet is not deformed, and no additional joining elements are required. Thus, only by controlling the plastic deformation of a sheet, the mechanical interlock can be formed. In this process, unlike the hole clinching process, the high strength steel sheet is formed to create a mechanical interlock.

2.2. Hole hemming

In the hole hemming process, there is no limit to the joint position. This is because the sheets are connected through pre-drilled holes, these holes can be drilled anywhere on the sheets as long as they are concentric. Although concentric, these holes must have different radius, with the sheet with the smaller radius being called 'outer sheet', and the sheet with the larger radius as 'inner sheet'. The 'outer sheet' must have sufficient ductility to prevent fracture, while the inner sheet has no restrictions [1].

The process is divided in two stages. The first stage is called 'Hole Flanging', as the name implies this stage is a flange operation. At this stage, the components that are in contact with the sheet are the flanging punch, the lower die and the upper die. The 'Hole Flanging' begins with the descent of the upper die, so that the dies close, initially deforming the 'outer sheet' (for future purposes we will call this movement as 'stage 0'). Then the edge of the hole is bent to a 90° angle by moving up the flanging punch. The dies are designed to make an indent next to the flange. This causes the hole edge of the inner sheet can be placed at the closest distance to the flange in the next stage. At this stage the parameters that need more care and attention are the hole radius of the outer sheet, the die radius, the fillet radius of the upper die, the indent depth and the nominal length of the flange [1].

The second stage is called 'Hemming', at this stage the components that are in contact with the sheet are the hemming punch, the upper die and the blank holder (which is employed to fix the position of the sheets during the plastic deformation). Therefore, it is necessary to change the active components from one stage to the other. First, the 'inner sheet' is assembled under the 'outer sheet', then the flange is formed over the hole edge of the inner sheet by means of a hemming punch to produce a mechanical interlock. In order for the flange to fit inside the hole and the inner sheet to be completely under the outer sheet, the hole radius of the inner sheet must be equal to or slightly larger than the radius of the die hole. The hole edge of the inner sheet will also be slightly deformed and placed inside the indent, this indent having a depth equal to the thickness of the inner sheet, ensuring that no protrusion will be created. Better mechanical locking can be obtained by increasing the contact surface between the hole edges of the outer and inner sheets. At this stage the most important parameter is the fillet radius of the hemming punch [1].

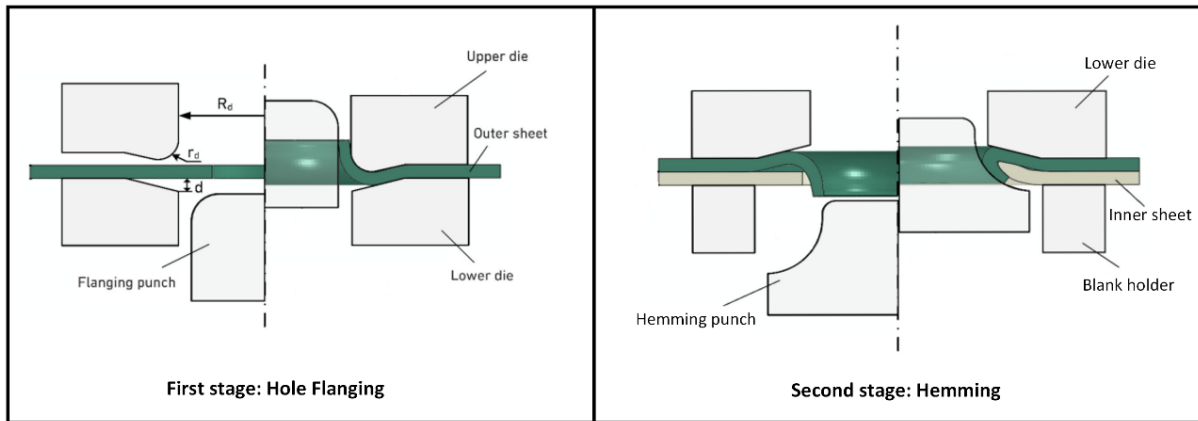


Figure 2-Schematic representation of the hole hemming process [1].

2.3. Tool design

The tool design, more properly die design in our case, is a complex and fascinating subject. It is one of the most exacting of all the areas of the general field of tool design [16]. The design and construction costs can be significant and may easily reach several tens of thousands of Euros, therefore, a lot of time and attention should be spent on this section, in order to avoid errors that can significantly increase the final costs. There are many factors to be considered when planning a production process and designing the tools:

- Batch size and production rate;
- Complexity of the part to be formed and number of intermediate forming stages (preforms);
- Critical functional surfaces and required level of geometrical and dimensional accuracy;
- Forming temperature;
- Equipment (presses, drop hammers) to be used;
- Tool materials, heat and surface treatments;
- Preheating die temperature and procedure;
- Lubrication system [17].

Tools can be classified according to a variety of elements and in keeping with the diversity of designs of tools. It is possible to classify tools according to:

- **Manufacturing processes:** blanking, punching, bending, deep drawing, etc;

- **Number of operations:** single-operation (simple), and multi-operation (combination) dies;
- **Number of stations:** single station and progressive dies [18].

The components that are generally incorporated in a tool are shown in Figure 3. This figure shows the tool in the conventional closed position.

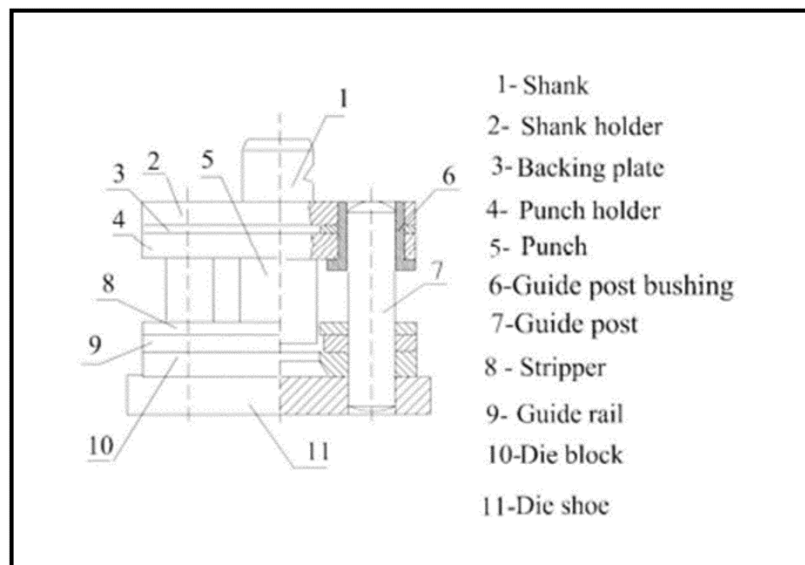


Figure 3-Components of a tool [18].

The upper subset is made up of the following components: the punch holder, the blanking plate, the punch plate, the punch, and the shank. The lower subset is made of the following components: the die shoe, the die block, the guide rails, and the guide plate [18].

In this case, it is intended to design tools to act in plastic deformation processes. This area has already been explored, so it was possible to find several tools sets similar to the one we intend to apply.

Reichel et.al. [19] design a flexible tool demonstrator capable of replicating material flow in the lancing, bending and compression stages of a new joining by forming process without auxiliary elements. The flexible tool demonstrator is defined by its modular concept that allows the active tool components to be easily interchanged for testing and exploring different materials and thickness combinations, surface conditions and cross-section areas of the strips under laboratory conditions. The flexible tool demonstrator is shown in Figure 4a.

Lambiase et.al. [20] developed a numerical model describing the evolution of ductile damage to predict the onset of fracture during the clinch joining of thin aluminum AA6082-T6 sheets. The damage model was calibrated and validated using instrumented punch-out tests in order to

increase the reliability and robustness of the damage parameter. The authors used a prototypal punching frame, shown in Fig. 4b, which was clamped on an MTS model 322.31 Universal Testing machine. In the tests performed, dies with three different diameters were used in order to produce different punch-die clearances. K340 hardened steel material was used for the manufacture of punches and dies in order to prevent smoothing and fractures on the cutting edges. The specimens are centered in the die using an external ring and the clamp of the specimen was secured by the blank-holder (with a preload of 6 kN and spring stiffness of 1.2 kN/mm). The test was carried out with a constant punch stroke speed of 10 mm/min.

Wiesenmayer *et.al.* [21] used the tool set shown in Figure 4c to test the joinability of the high-strength alloy AA7075 in shear-clinching processes in dependance of its temper. The die-sided tool consists of a base plate to which the die is mounted. The inner and the outer punch are each installed to individual plates, which are connected via springs with a pre-load of 29 kN. Both plates are guided relative to the base plate by pillars, allowing the exact positioning of the tools. The blank holder is also mounted to an individual plate, which is connected via springs with the outer punch plate. The pre-load is 1 kN.

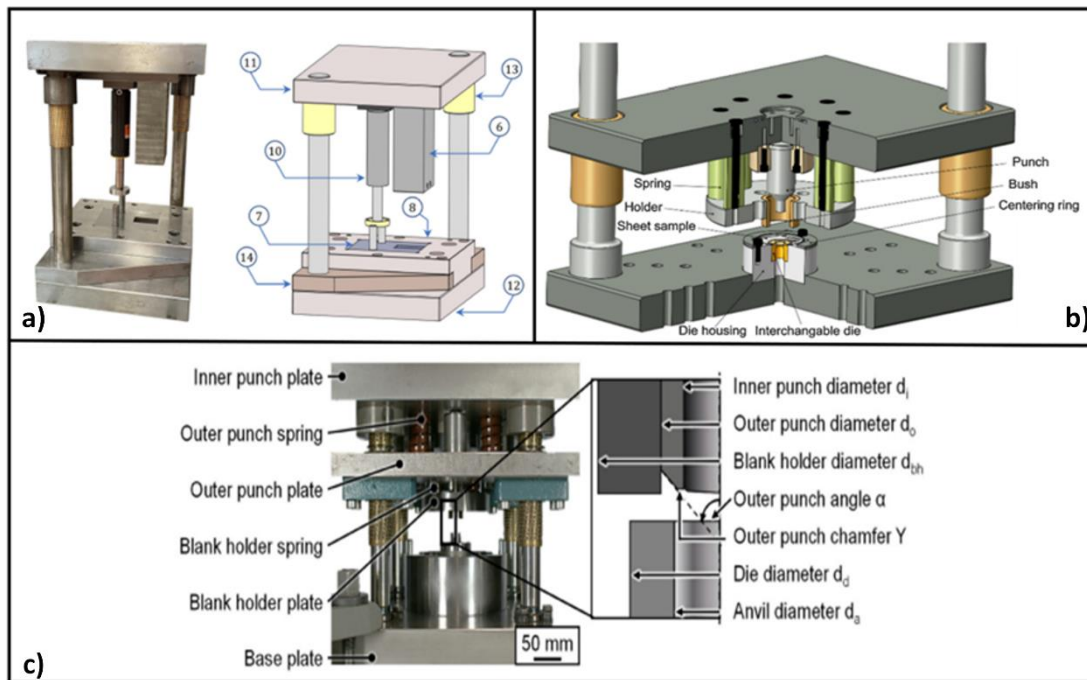


Figure 4-Examples of tools designed for perform mechanical joining processes [19][20][21]

3. Tool design

This chapter describes the design process of all components that were incorporated in the tool to perform the hole hemming process, as well as the selection process of some standardized components.

3.1. Requirements

With the development of a new joining process, there is also the need to develop a tool that allows to carry out and test this new process. As seen earlier, there are several parameters that influence the hole hemming process. So, in order to be able to test different parameters and conditions of the process under development, this tool must allow the joining of different combinations of materials, and for this to be possible in a practical and simple way, it is necessary to have:

- Adjustable structural and passive tool components.
- Easily interchangeable active tool components.

Before starting the design of the tool components, it is necessary to analyze the forces involved in this new process for joining a material combination. To this end, the joining of AA6082-T4 aluminum and AZ31 magnesium alloy sheets, which are newly applied to manufacture lightweight structures in the automotive industries, was selected and investigated using finite element analysis. This task was performed by *Pereira et.al.* [22] and the forces applied to the active components of the tool, namely the dies and punches, were provided for the design of the tool. The AA6082-T4 sheet is the 'outer sheet', presenting a greater formability. The AZ31 sheet is the 'inner sheet' and only assembled in the second stage, 'Hemming'.

Summary of the mechanical properties of these materials is listed in Table 1. Moreover, their stress-strain curves are shown in Figure 5.

Table 1- Mechanical properties of outer sheet and inner sheet [22].

Material	Thickness [mm]	Young modulus [GPa]	Yield stress [MPa]	Ultimate tensile stress [MPa]	Elongation [%]
AA 6082-T4	2	70	173	278	24.5
AZ31	0.9	44	142	259	16.6

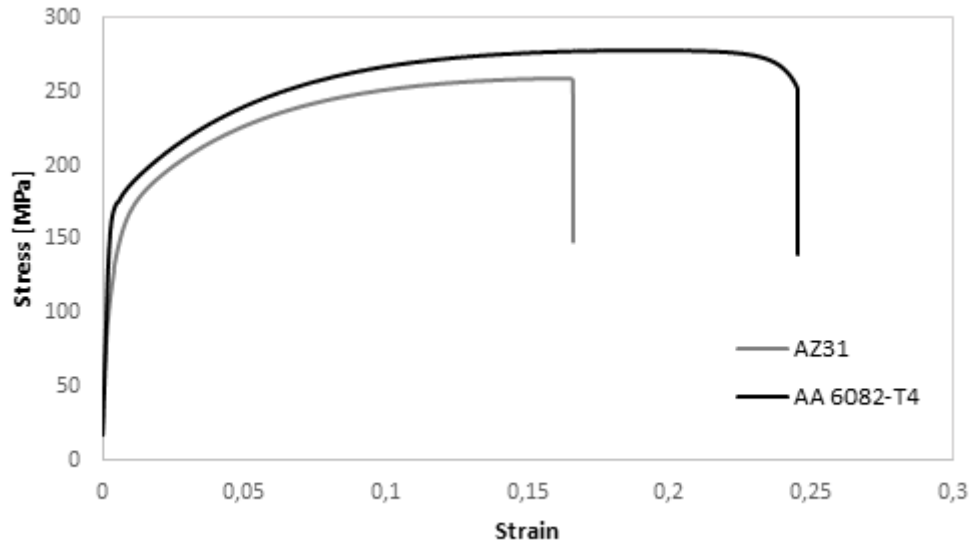


Figure 5-Strain vs stress of outer sheet and inner sheet [22].

In the finite element simulations, the punches and dies were considered as rigid bodies and the applied forces were extracted from their reference point. The forces applied to the active components in the vertical direction are shown in Figures 6, 7, 8, 9, 10, 11 and 12.

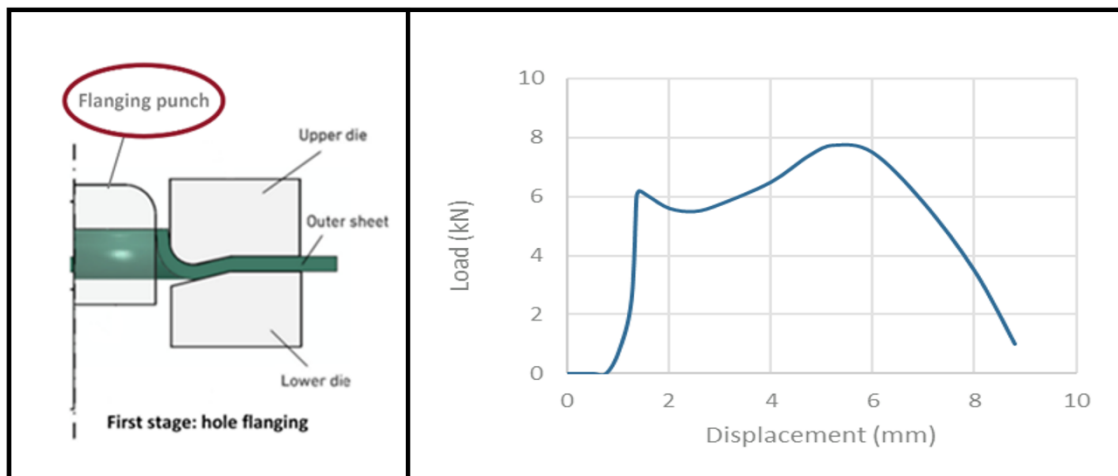


Figure 6-Load vs displacement of flanging punch [22].

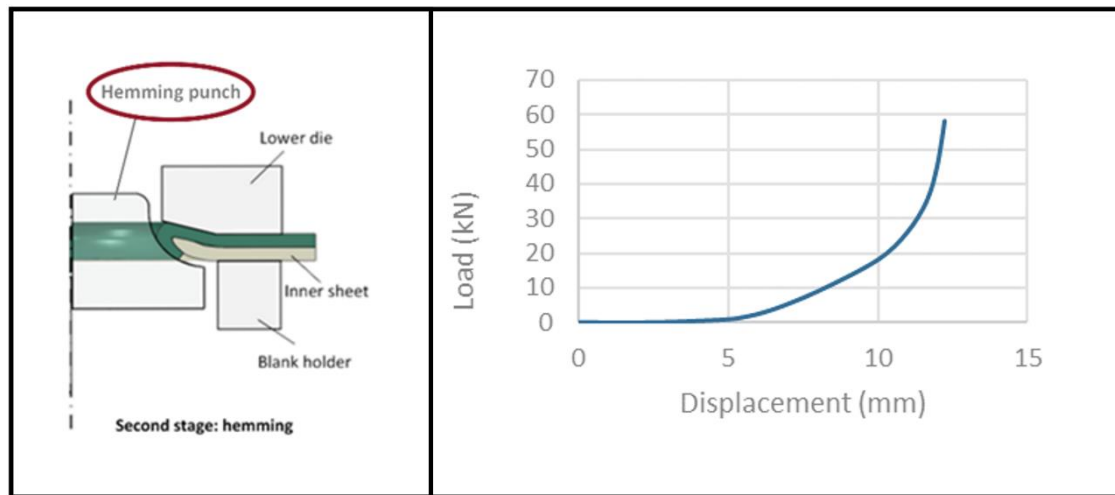


Figure 7-Load vs displacement of hemming punch [22].

As seen in Figure 6 and 7, the punch used for the first stage, 'flanging punch', is subjected to a maximum load of 8 kN, while in the punch used for the second stage, 'hemming punch', the maximum load is about 60 kN. This higher load in the second stage is explained by the fact that in the second stage the mechanical interlock is created through a hemming operation, involving the compression of the flange over the hole edge of the inner sheet.

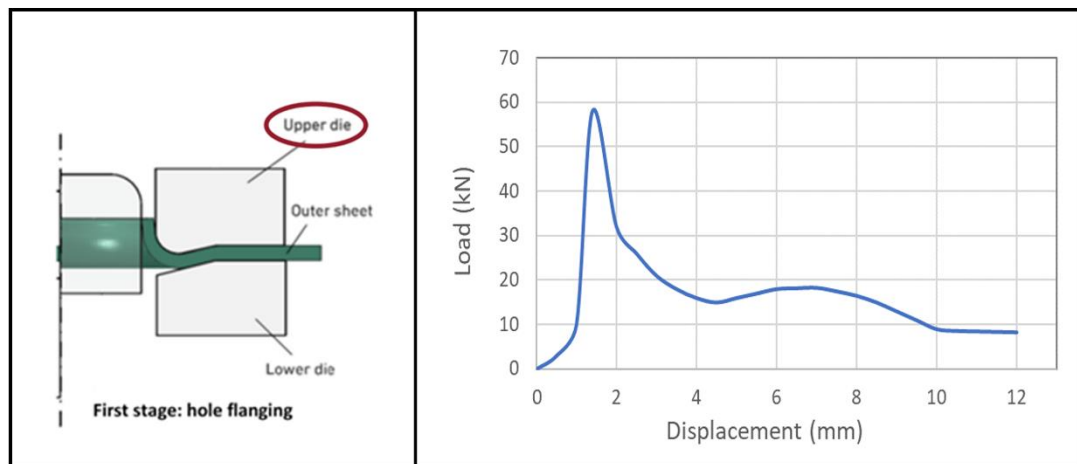


Figure 8-Load vs punch displacement, upper die [22].

As for the upper die, a peak of force is observed in the displacement of 1.45mm. This instant corresponds to the end of the step 1 of the flanging stage, where the upper die exerts force on the 'outer sheet', deforming it before the punch comes into contact with the outer sheet. It can be seen that for the upper die the maximum force exerted is approximately 60 kN.

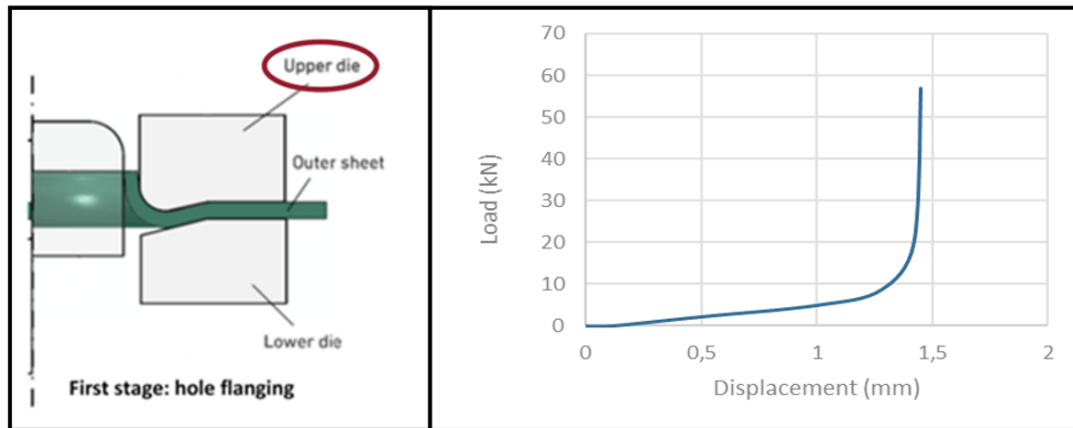


Figure 9-Load vs punch displacement, upper die (initial phase) [22].

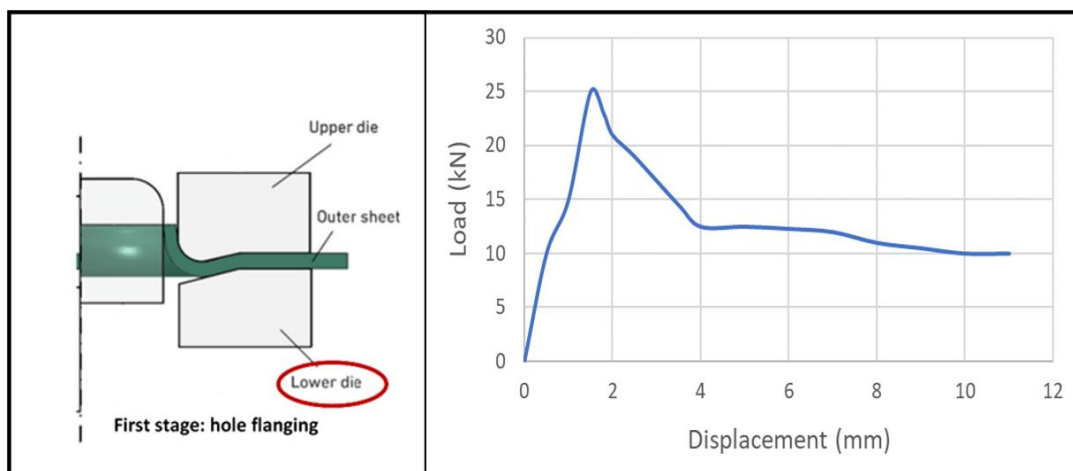


Figure 10-Load vs punch displacement, lower die in the first stage [22].

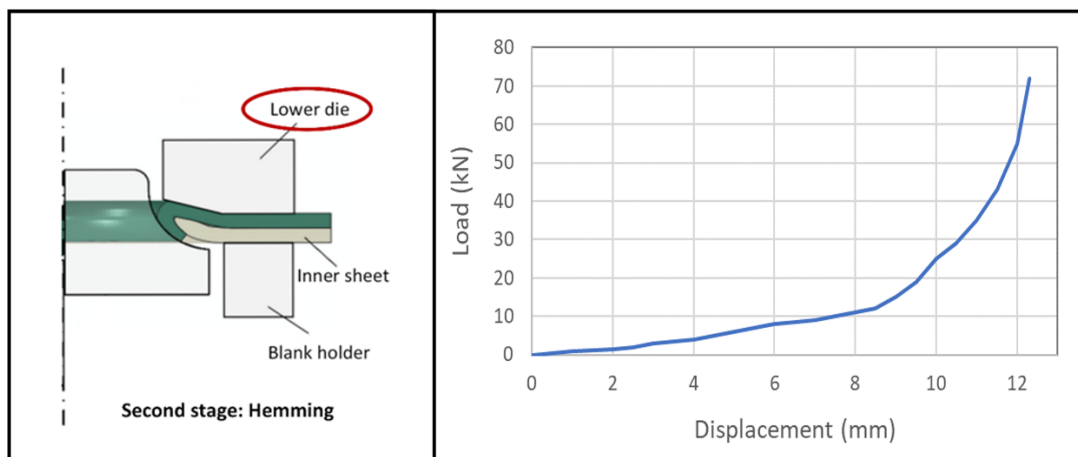


Figure 11-Load vs punch displacement, lower die in the second stage [22].

Regarding the lower die, it interacts in the two stages of the hole hemming process. In the first stage, 'hole flanging', the maximum force applied to this component is approximately 25 kN while in the second phase, 'hemming', this value reaches 70 kN.

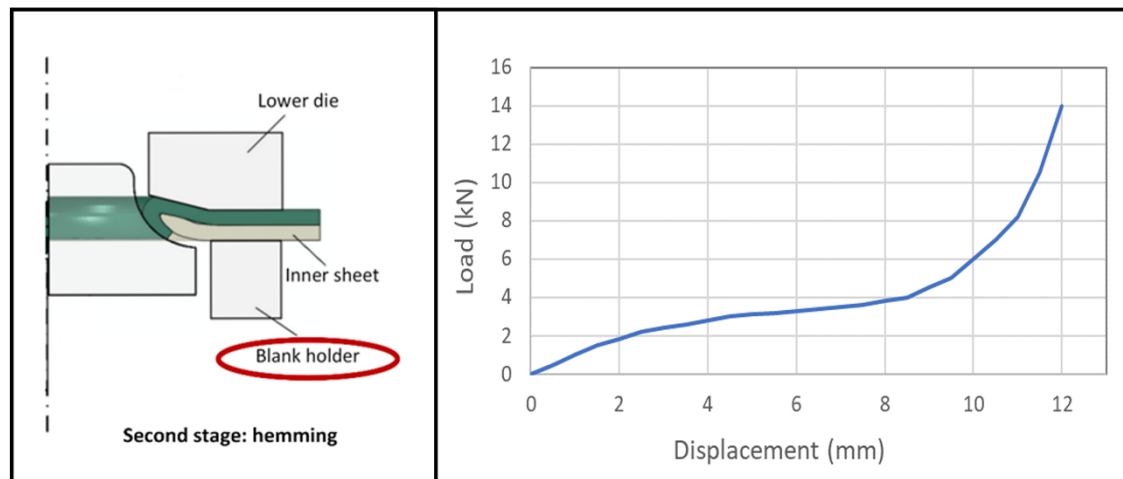


Figure 12-Load vs punch displacement, blank holder [22].

Finally, the blank holder is the die that is subjected to the least amount of force, about 17 kN, as its function is solely to fix the sheets.

Thus, through the forces extracted from the simulations, it was possible to know the forces required for the hole hemming process.

3.2. Testing machine

Another aspect that needs to be carefully analyzed, before starting the design of the tool for the hole hemming process, is the testing machine where it will be mounted. The machine used was an Instron model 8801 with a load cell of 100 kN, as can be seen in Figure 13.

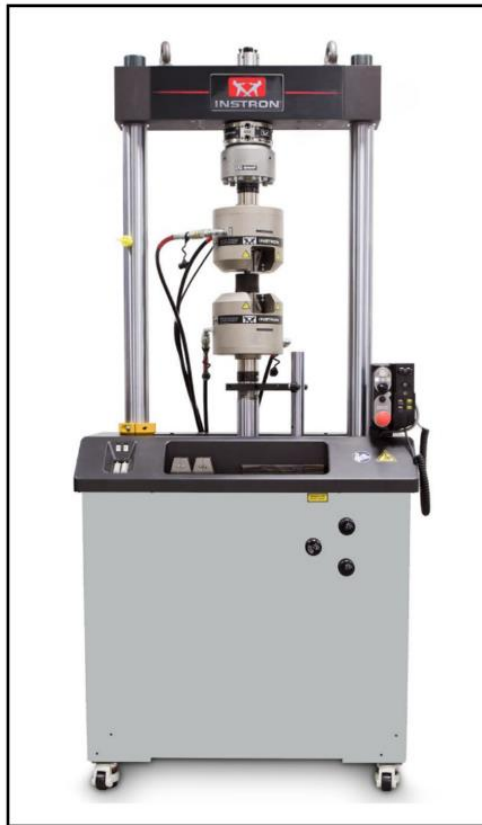


Figure 13-Instron 8801 SERVOHYDRAULIC FATIGUE TESTING SYSTEM (UP TO 100 kN) [23].

The Instron® 8801 is a compact servo hydraulic fatigue testing system that meets the challenging demands of various static and dynamic testing requirements. 8801 systems provide complete testing solutions to satisfy the needs of advanced materials and component testing and are ideally suited for fatigue testing and fracture mechanics. The compact design of the 8801 frame makes it ideal for installation within any laboratory environment, generally without the need for strengthened floors or raised ceiling heights. This machine presents the following features:

- A double-acting servo hydraulic actuator with force capacity up to ± 100 kN;
- High-stiffness, precision-aligned load frame with twin columns and actuator in lower base;
- 150 mm (6 in) of usable stroke;
- Designed for both dynamic and static testing on a variety of materials and components
- Choice of hydraulic configuration and dynamic performance to suit application;
- Extra-height frame option for testing longer load strings;
- Adjustable upper crosshead with hydraulic lifts and locks fitted as standard for easy adjustment of daylight;
- Patented Dynacell™ advanced load cell technology for faster testing and reduction of inertial errors;

- Compact servo hydraulic fatigue test system – frame requires less than 0.5 m² (5.4 ft²) of floor space;
- Hydrostatic bearing actuators for higher side-load resistance or material critical applications, such as low cycle fatigue (LCF);
- Designed to be used with the 3520 Series of Hydraulic Power Units;
- Compatible with a large range of grips, fixtures, chambers, video extensometers, protective shields, and other accessories;
- Patented stiffness based tuning algorithm that enables users to tune a variety of specimens in second [23].

After analyzing the features, it was found that the machine has sufficient capacity to meet the requirements of the forces previously analyzed.

In order to be able to correctly assemble the tool on the machine, it was necessary to check the mechanical interfaces and dimensions of the machine, to ensure that the hole hemming tool will be properly fixed to the machine and that the testing machine will be able to accommodate it.

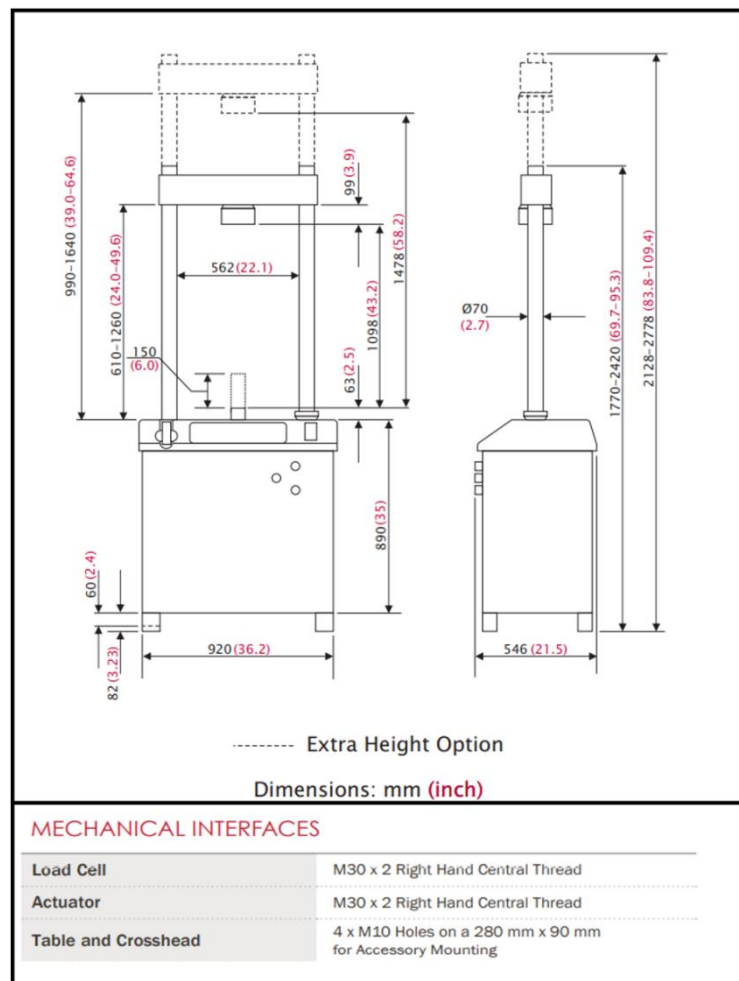


Figure 14-Dimensions of Instron 8801 SERVOHYDRAULIC FATIGUE TESTING SYSTEM (UP TO 100 KN) [24].

Through the technical sheet of the testing machine, it was found that the flexible tool must be fixed through an M30x2 right hand central thread. Regarding the dimensions, as can be seen in Figure 14, the width and height of the tool should not exceed 562 mm and 1098 mm, respectively [24].

3.3. Concepts

The first step to start the tool design was the conceptualization of tools to realize the hole hemming process. The objective of this step was to obtain the concept that provides the best conditions for the process under study.

Considering the testing machine where the tool will be installed, three different concepts were formed.

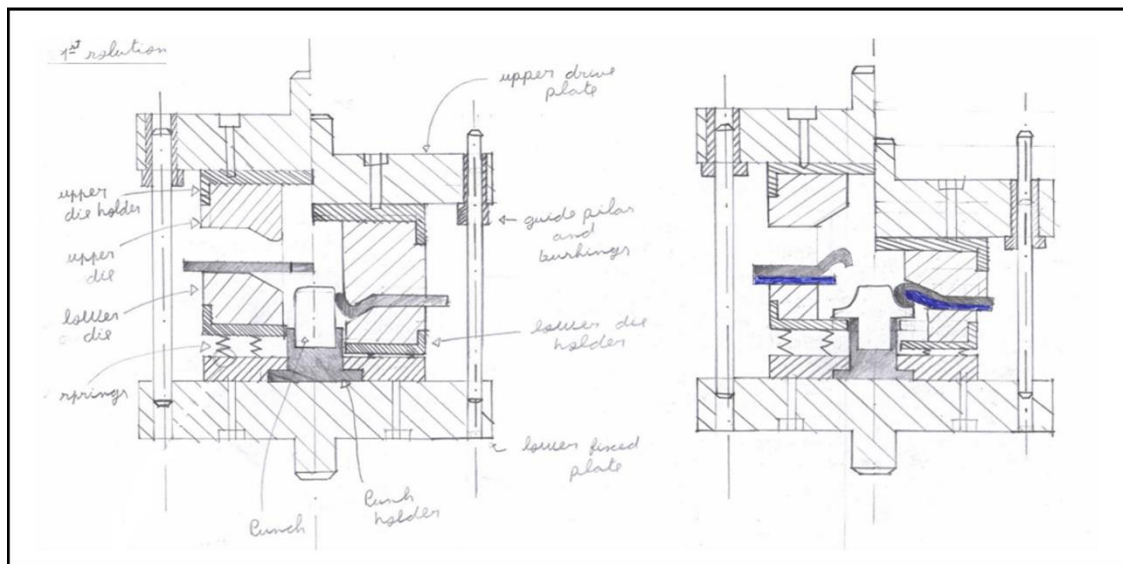


Figure 15-First concept for the hole hemming tool.

The first concept developed is shown in Figure 15. This solution consists of:

- Upper drive plate, this being assembled to the testing machine, and being one of the components that presents movement;
- Lower plate, which is also attached to the testing machine;
- Upper die holder, which is attached to the upper drive plate and is responsible for holding the upper dies;
- Lower die holder, which is attached to the lower plate through springs;
- Die holder, which is attached to the lower plate and is responsible for holding the punches;
- Guide pillars and bushings, responsible for guiding;
- Dies and punches [25][18][16].

In this concept, the relative movement between the punches and the dies is achieved through the use of a set of springs in the lower plate. So, when the upper part of the testing machine moves downwards, the springs are compressed allowing contact between the punch and the sheet.

To use this concept, it is necessary to change the active components of the tool (punches and dies) from one stage to the other. Another requirement of this concept is that the punches have to be housed in the same plate (in this case in the lower plate).

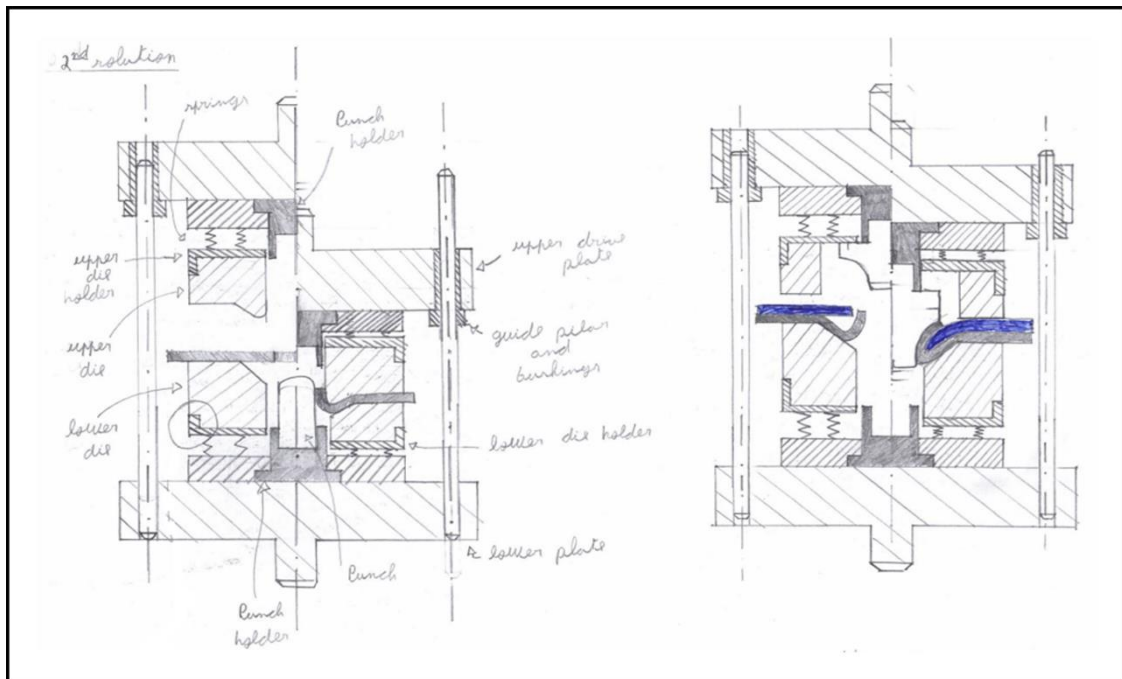


Figure 16- Second concept of the hole hemming tool.

The second concept developed, shown in Figure 16 has practically the same components as the concept previously presented, except that in this solution two punch holders and two sets of springs are used. By using this set up, it is possible to place the punches at the top and at the bottom, so it is only necessary to change the upper die and the punch from one stage to the other. However, for this concept it would be necessary to control the force exerted by the springs both at the top and at the bottom.

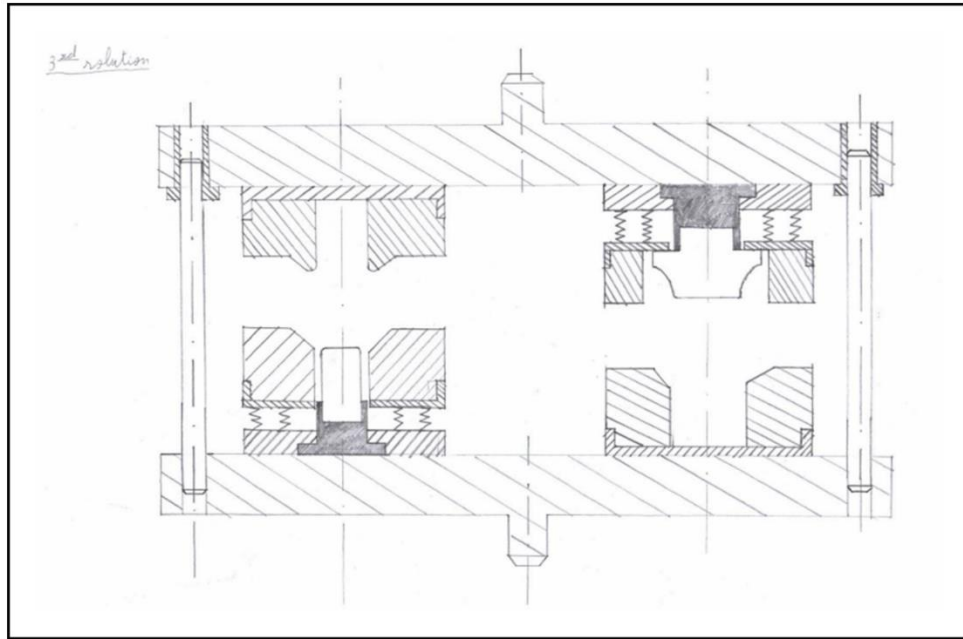


Figure 17- Third concept of the hole hemming tool.

The last concept is a progressive die, shown in Figure 17. Progressive dies are made with two or more stations. Each station performs an operation on the workpiece or provides a rest station so that the workpiece is completed when the last operation has been accomplished. After the first part has ran through all the stations, each subsequent stroke of the press produces another finish part [18]. For this study, the die consists of two stations, the first station corresponding to the first stage 'Hole Flanging' and the second station corresponding to the second stage 'Hemming'. Using this solution, it is not necessary to change any active components from one stage to the other, however this solution requires a very careful adjustment, due to the contact between the dies and the sheets in the different stages simultaneously. This solution also requires greater force on the part of the press, as it performs both stages at the same time.

In order to be able to choose the most suitable solution for the hole hemming process, Table 2 with the advantages and disadvantages of each of the concepts is presented.

Table 2- Advantages and disadvantages of the different concepts of the hole hemming tool.

	Advantages	Disadvantages
First concept	- Only needs springs at the bottom side - Only needs one punch holder - Easy control of the forces applied	- Necessary to change all the active tools from one stage to another - The punches have to be housed in the same plate (in this case in the lower plate)
Second concept	- Only needs to change one punch and one die from one stage to another - Easy positioning of the plates in the two phases	- Necessary to use springs in both sides - Difficult to control the forces applied
Third concept	- No need to change the active tools from one stage to another - Performs both stages at the same time	- Necessary to provide a bigger force - Requires a very careful adjustment

Of the concepts presented above, the one that provides greater productivity is the third concept. However, as this process is in a study phase, a concept that provides maximum flexibility along with ease of adjustment should be used. Bearing in mind the requirements of the tool and advantage and disadvantage of the concepts, the most suitable solution for this case is the first concept.

3.4. Hole hemming tool

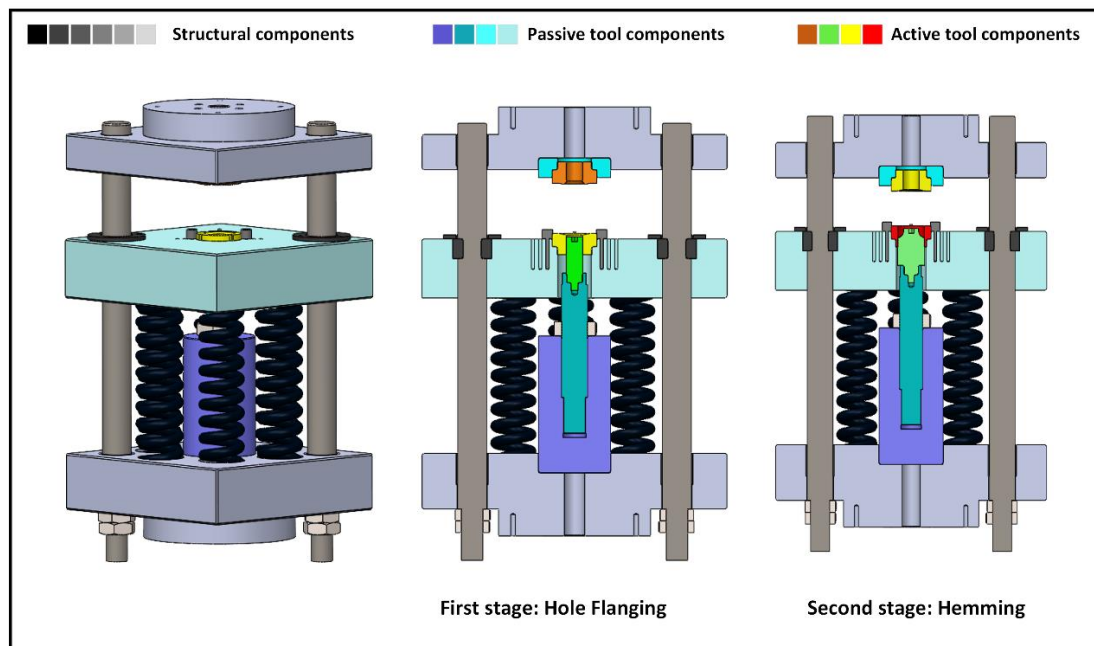


Figure 18-Schematic representation of the hole hemming tool.

The quality and reliability of the process is highly dependent on the hole hemming parameters, as previously seen in section 2. It is necessary to build a flexible tool to be able to explore and test the different combinations of materials, so that it is possible to easily change the parameters of this process and easily change active tool components. Thus, the tool components can be classified into three main categories (Figure 18): active tool components, passive tool components and structural components [19].

The active components of the hole hemming tool are those that come into direct contact with the sheets during joining by plastic deformation, that is, the flanging punch (shown in green), the hemming punch (shown in light green), the upper die (shown in brown), the lower die (shown in yellow), and the blank holder (shown in red). These components only serve to manufacture hole-hemmed joints with specific geometries and sizes and need to be replaced from one stage to the other stage as they are the most stressed components. However, it is possible to easily replace these components with others with different geometries and sizes to produce a wide range of hole-hemmed joints due to the modular concept of the hole hemming tool.

As for the passive components of the hole hemming tool, these surround the active components of the tool and support them directly or indirectly. These components include the punch holder (shown in navy blue) that fixes the different types of punches, the height adjuster (shown in dark blue) that allows to change the height of the punches, the die holder (shown in light blue) and the middle plate (shown in turquoise) that accommodates the different dies. Distribute the forces through the sheets and the tool structure without damaging the hole-hemmed joints and overloading the active tool components, accommodate the active tool components, provide the guidance for the positioning of the sheets, and allow the adjustment of the position of the active tool components are the main functions of the passive tool components.

Finally, the structural components of the hole hemming tool are those that accommodate the passive components of the tool and are independent of the process. These components have as main functions to provide orientation and centering between the upper, middle and lower plates, to enable the installation of the tool in the testing machine and to allow the variation of the preload to test different process conditions. Furthermore, as structural components are process independent, they can be used to accommodate active and passive tool components from other processes. These components include the upper drive plate, the lower plate, the guide pillars, the bushings, the pins, the keys, the plates, the nuts, and the springs.

3.5. Structural components

This chapter describes the design and validation process for building the structural components of the hole hemming tool, Figure 19. The structural components of the hole hemming tool were designed based on the following requirements:

- Accommodate the passive tool components.
- Provide guidance and centering between the upper, middle, and lower tool parts.
- Enable the installation in the testing machine.
- Allow preload variation.

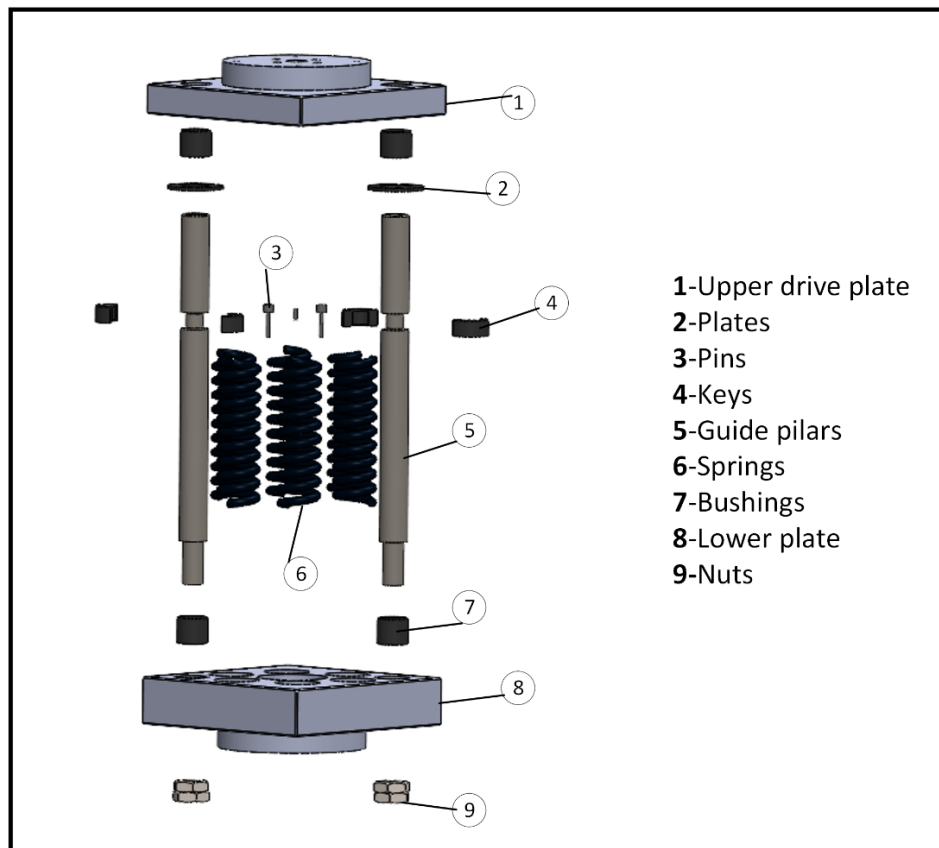


Figure 19- Schematic representation of the structural components of the hole hemming tool.

3.5.1. Preload adjustment

Using the results obtained through finite element simulations, it was found that there is a peak in the force exerted during the first stage of the process, which occurs before it starts, that is, before the flanging punch starts contacting the plate (step 1 of stage 1). This force is due to the compression of the upper die against the strip, in order to deform the outer sheet before it starts contacting the flanging punch, this force will be called preload.

The preload exerted in this process has a maximum value of approximately 60 kN, so for the study of the conditions of this process it was considered interesting to be able to switch the preload between 40 and 60 kN.

The solution found to allow the existence and variation of the preload is shown in Figure 20:

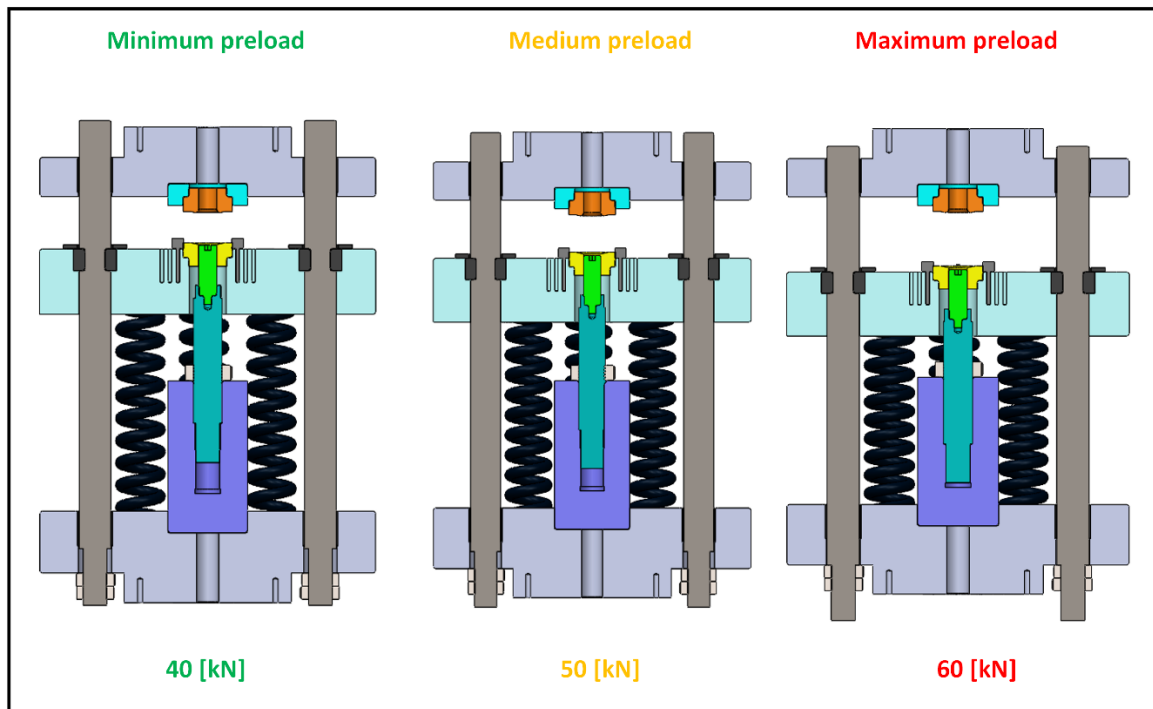


Figure 20- Schematic representation of the preload variation of the hole hemming tool.

This solution involves the use of springs, guide pillars, nuts, and keys. The preload is ensured by the springs, because after the upper die comes into contact with the outer sheet, it is only possible for the punch to move upwards when the force exerted by the springs is overcome. In this way, by controlling the pre-load existing in the springs, it can control the existing preload in the process. When screwing or unscrewing the nuts on the respective guide pillars, the middle plate will approach or move away from the lower plate (since middle plate is fixed to the guide pillars through the keys), which causes the springs to be compressed or expanded, thus allowing to regulate the preload.

3.5.2. Springs

Springs play a very important role in preload adjustment, therefore they should be carefully selected [25]. For the selection of springs, the following requirements were considered:

- Total preload should vary between 40 and 60kN
- Operation deflection should be at least 10mm (according to the distance that the punches move)

- Stiffness should be selected so that the increase in force per 10 mm deflection, which is required for flanging the hole of an outer sheet and bending the flange over the hole of an inner sheet, is not significant.

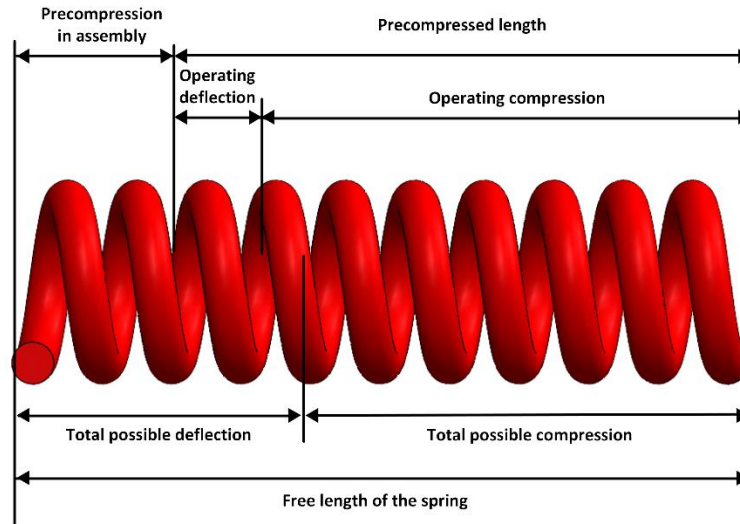


Figure 21- Compression spring and its properties.

Figure 21 presents a compression spring and its most important properties. When sizing the springs, it is essential to ensure that the value of the maximum shear stress applied does not exceed the value of the admissible shear stress [26].

$$\tau_{max} < \tau_{adm} \quad (1)$$

The admissible shear stress can be defined by [27]:

$$\tau_{adm} = \frac{B \cdot d^a}{2} \quad (2)$$

“B” and “a” being constants dependent on the material chosen for the springs. The value of “d” is the diameter of the spring wire.

The maximum shear stress can be calculated by the following expression [26]:

$$\tau_{max} = \frac{8 \cdot \frac{F_{mola}}{n} \cdot D}{\pi \cdot d^3} \cdot \left[\frac{4 \cdot C - 1}{4 \cdot C - 4} + \frac{0.615}{C} \right] \quad (3)$$

Where “n” is the number of springs, “D” is the diameter of the spring, “ F_{mola} ” is the load applied by each spring and the dimensionless constant “C” is defined as the ratio between the diameter of the spring and the diameter of the wire.

First, the value of the constant “C” must be chosen within a range of admissible values from 4 to 12, after general design considerations. This range must take into account production difficulties and undesirable behavior of springs outside the considered values[26]:

$$C = \frac{D}{d} \quad (4)$$

With the help of this coefficient, it is then possible to calculate the Bergsträsser factor[26]:

$$K_B = \frac{4C + 2}{4C - 3} \quad (5)$$

It is still necessary to choose the number of springs. This choice must be made in order to obtain a symmetrical distribution in the tool, not being a large enough number to cause constraints in the production and organization of the body, nor so small as to promote exaggerated tensions in each spring.

$$F_k = \frac{F_{molas}}{n} \quad (6)$$

According to *Budynas et.al.* [26], the design process of a spring in static service can start by determining the minimum wire diameter, d, for a given application, according to the formula:

$$\frac{S_{sy}}{n_s} = K_B \cdot \frac{8 \cdot \frac{F_{mola}}{n} \cdot C}{\pi \cdot d^2} \quad (7)$$

In the previous equation, S_{sy} , according to the same source, can be considered approximately equal to:

$$S_{sy} = 0.45 \cdot \frac{A}{d^m} \quad (8)$$

Where A and m are coefficients that depend only on the material. In the previous equation, n_s is the safety factor applied to the calculation.

The behavior of a spring can be described through the following equation:

$$F = k \cdot \delta = \frac{G \cdot d^4}{8 \cdot D^3 \cdot N_a} \cdot \delta \quad (9)$$

Where “G” is the nominal value of Young's modulus for the chosen material, N_a is the number of active turns and δ is the spring displacement.

However, for our case, as the springs in question need to have a high stiffness to be able to exert the necessary force, so the selection of standard compression springs was chosen.

Using the *UNCETA* supplier, the recommendations, shown in table 3, were obtained for the selection of their springs [28]:

Table 3- Recommendations for operating deflection [28].

Glossary

Total work deviation advised for +3.000.000 cycles Mm= Lo X 25% N= Lo X 25% X R
Total work deviation advised for approximately 1.500.000 cycles Mm= LO X 30% N= Lo X 30% X R
Total work deviation advised for 300.000-500.000 cycles Mm= LO X 35% N= Lo X 33.75% X R
Total work deviation advised for 100.000-200.000 cycles Mm= LO X 40% N= Lo X 37.50% X R

Considering the requirements mentioned above and following the recommendations to obtain a total work deviation for approximately 1.500.000 cycles, the selected springs were ISO 10243 XM198. The properties of these springs are presented in table 4 [28]:

Table 4- Properties of the springs selected [28].

Selected Spring	Iso 10243 xm198
Stiffness	0,128 [kN/mm]
Lo (free lenght)	305 [mm]
Section	11,6 x 12,3 [mm]
Outer diameter	63 [mm]
Inner diameter	38 [mm]
Max deflection [LO x 30%]	91,5 [mm]
Max load [Lo x 30% x R]	11,712 [kN]
Number of springs [Ns]	6
Max load [Lo x 30% x R x Ns]	70,272 [kN]

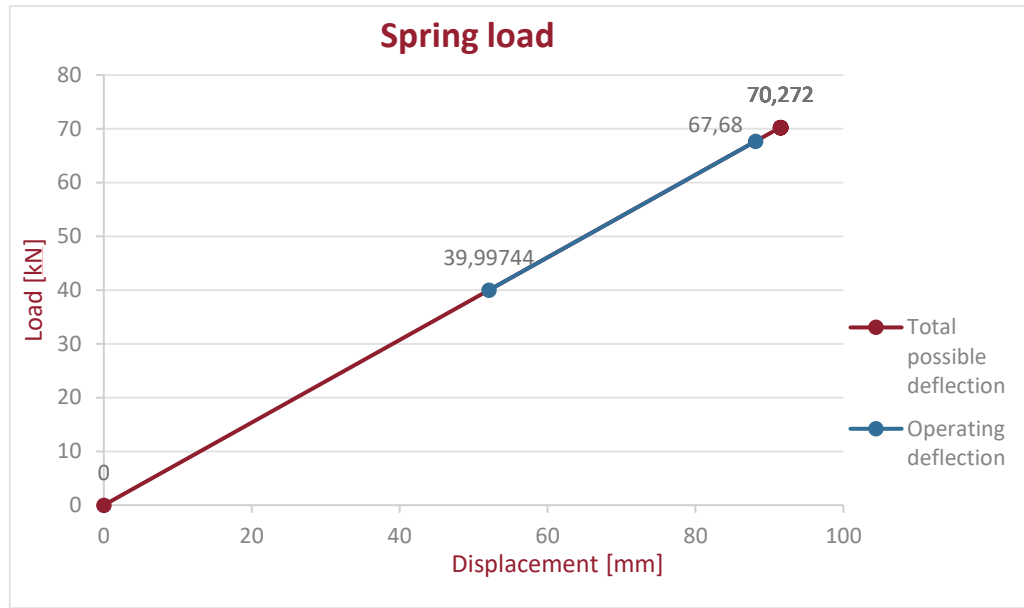


Figure 22- Load vs displacement of the springs set.

With the use of this set of springs it is possible to satisfy all the requirements mentioned above.

Starting with the first and second requirements, it is possible to use a preload of either 40 kN or 60 kN, always ensuring a minimum working length of at least 10 mm, as it is possible to see in Figure 22. For a preload of 40 kN an initial compression of 52 mm corresponding to 17 % of the free length is required. In the case of a preload of 60 kN an initial compression of 78 mm corresponding to 25 % of the free length is required.

As for the second requirement, the increase in spring force can be calculated by the expression:

$$F = R * N_s * \delta \quad (10)$$

Considering a displacement of 10 mm, the increase in spring force over the course of the process will be only 7.68 kN.

It should be noted that other spring ranges were investigated, but either they did not meet the requirements, or an excessive number of springs was needed. The solution presented is the best compromise between satisfying the requirements and the number of springs to be used.

3.5.3. Upper drive plate and lower plate

Concerning accuracy, alignment, supporting the passive tool components and fixing the tool to the test machine, the most important components were the upper drive plate and the lower drive plate.

These two components guarantee the fixing of the hole hemming tool to the testing machine. This fixation is carried out through a bolt that threads simultaneously in the test machine and in the upper drive plate/lower plate, as shown in Figure 23.

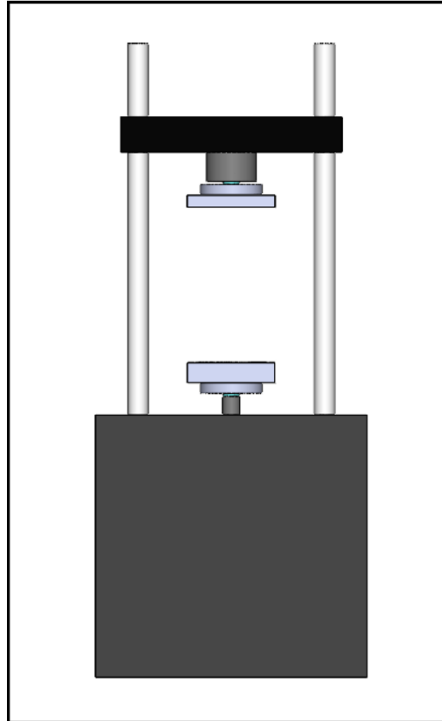


Figure 23- Schematic representation of the assembly of the lower plate and upper drive plate in the testing machine.

The first step in the design of these components was the consultation of catalogues, in order to analyze the dimensions that these components should take. According to V. *Boljanovic et.al.* [29], it was possible to obtain a table that relates the dimensions of the plates with the strength of the testing machine, this table is shown in Figure 24.

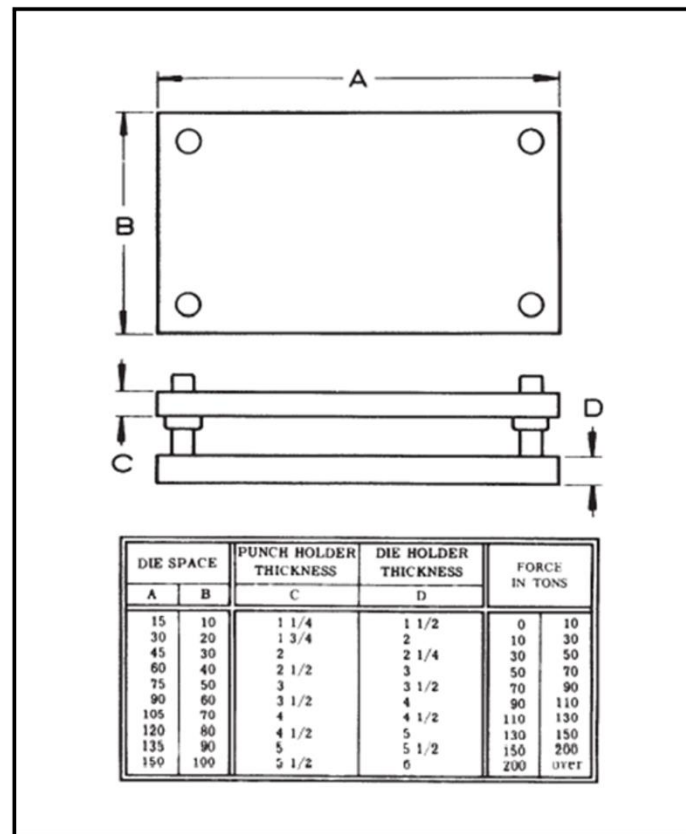


Figure 24- Table for determining plates thickness [29]

Knowing that our testing machine has a force greater than 100 kN, the recommended dimensions are presented in table 5:

Table 5- Recommendation thickness for lower plate and upper drive plate.

	Lower plate	Upper drive plate
A	381 [mm]	381 [mm]
B	254 [mm]	254 [mm]
Thickness	38.1 [mm]	31.75 [mm]

However, these dimensions serve only as a first indication because to obtain the final dimensions of the plates, there are different aspects to be taken into account.

The first dimension that should be defined was the width and length of the plates. To define these dimensions, attention was paid to the recommendations shown above, the dimensions of the testing machine and the space required for the hole hemming process. It was decided to use square plates. To guarantee that the plate could rotate freely inside the testing machine, the maximum possible value for its length was determined:

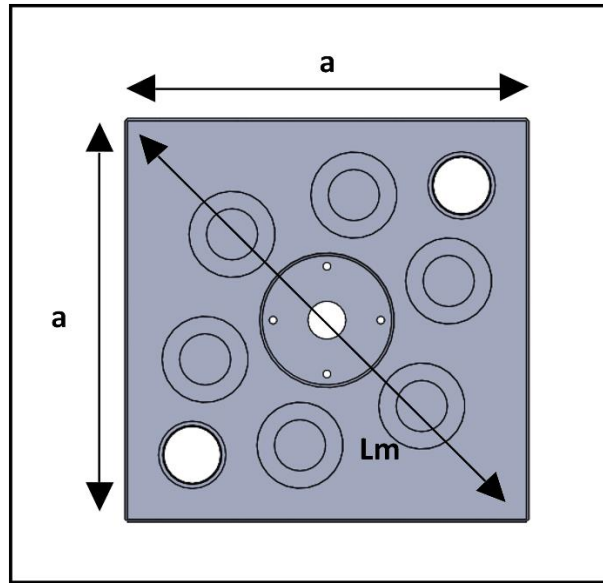


Figure 25- Upper view of the lower plate.

$$L_m^2 = a^2 + a^2 \quad (11)$$

Where L_m is the distance between the pillars of the testing machine, Figure 25.

$$a = 397.394 \text{ mm} \quad (12)$$

Taking into account this maximum value for the length of the plates and the values previously recommended, a width of 300 mm is assigned to the plates.

The next step in determining the dimensions was to choose the number and location of the guide pillars. Through the review of some literature, it was possible to find different styles of positioning the guide pillars.

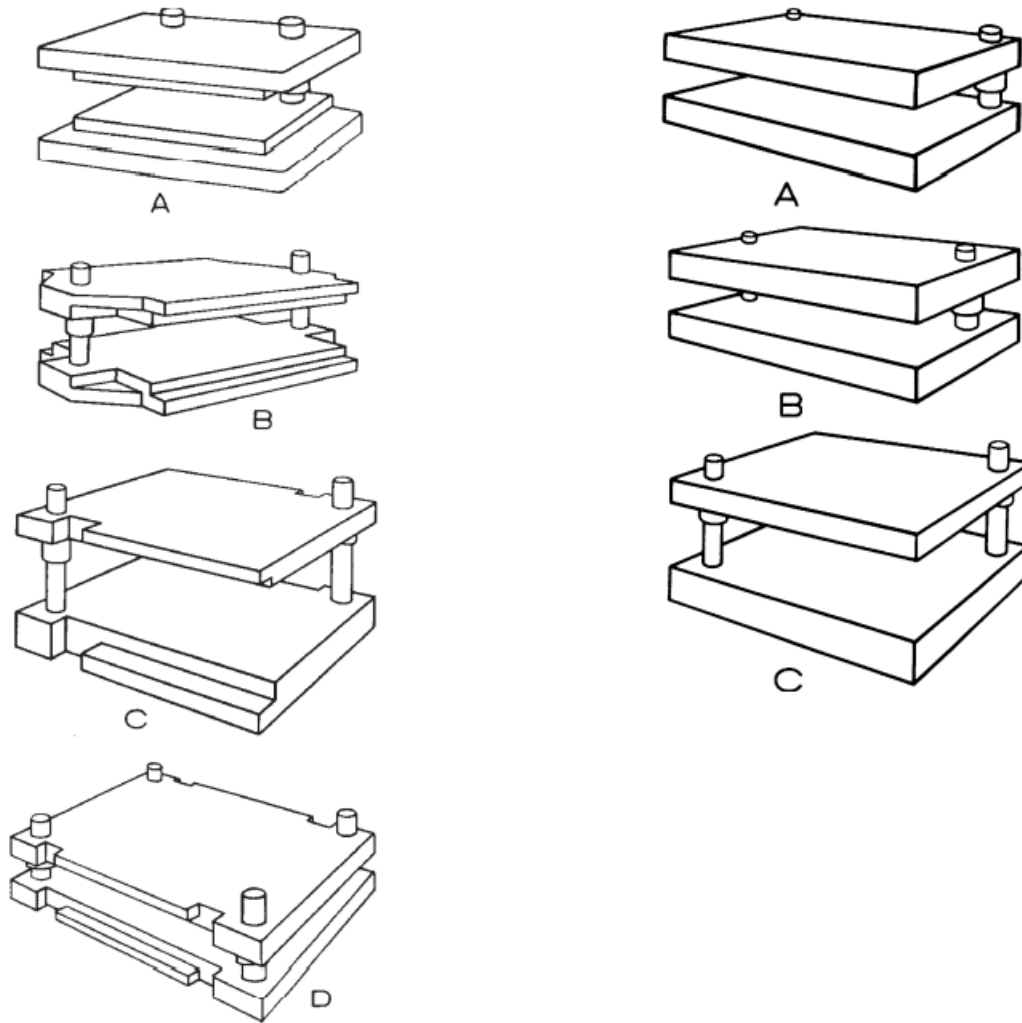


Figure 26- Different styles of the guide pillars [29].

First, it was decided to use only two guide pillars, because the use of four guide pillars can make the tool assembly very difficult. Regarding their positioning, it was considered that the best option would be to place them diagonally, allowing greater space and accessibility to the remaining components.

As for the distance between the guide pillars, enough space was left for the accommodation of the springs and other components, and the values presented in the catalogs, Figure 27, were considered [28].

Portamatrices dos placas al centro												
Montaje: Casquillo con valona - 12.												
Nota: Portamatrices tres placas solicite precio y plazo.												
Bajo demanda suministramos otras medidas												
Tamaño A x B mm	XM118	a ₁ mm	a ₂ mm	C ₁ mm	C ₂ mm	C ₃ mm	d ₁ mm	d ₂ mm	L mm	c mm	l ₁ mm	l ₂ mm
225 x 100	Consultar precio	163	138	37	32	23	24	25	140	48	45	18
325 x 100	Consultar precio	263	238	37	32	23	24	25	140	48	45	18
235 x 125	Consultar precio	173	148	37	32	23	24	25	140	48	45	18
275 x 165	Consultar precio	213	188	37	32	23	24	25	140	48	45	18
325 x 165	Consultar precio	263	238	37	32	23	24	25	140	48	45	18
350 x 200	Consultar precio	273	242	42	32	28	30	32	160	58	50	18
450 x 200	Consultar precio	373	342	42	32	28	30	32	160	58	50	18
450 x 250	Consultar precio	362	321	47	37	32	40	42	180	72	50	18
500 x 300	Consultar precio	412	371	47	42	32	40	42	180	72	50	18
500 x 350	Consultar precio	412	371	47	42	32	40	42	180	72	50	18

Figure 27- Catalogue of the plates. [28]

The process of obtaining this dimension was an iterative process with the process of determining the position of the springs and with the simulations presented later in order to predict the bending in these plates. The final value obtained was a diameter of 282,843 mm, Figure 28.

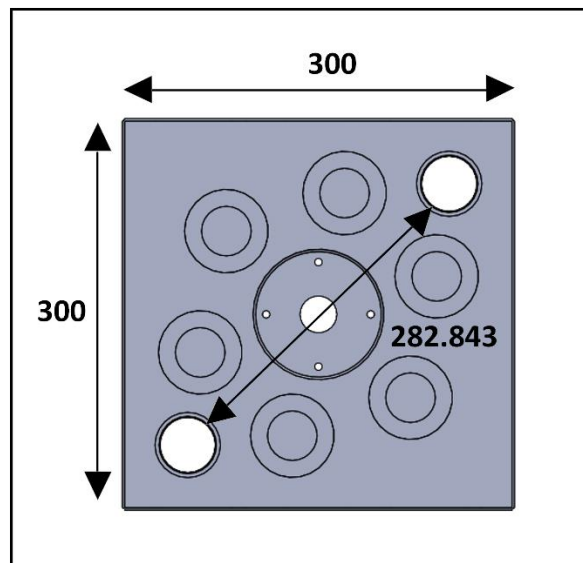


Figure 28- Dimensions of lower plate.

3.5.4. Lower plate

The lower plate is responsible for housing the springs, bushings and height adjuster, fixing the tool to the testing machine and aligning the plates, Figure 29.

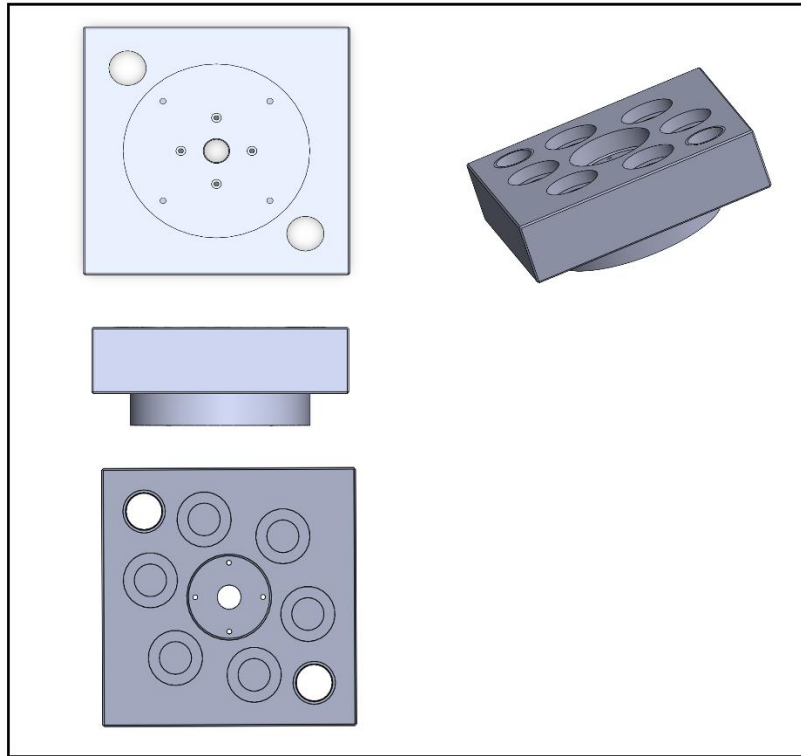


Figure 29- Schematic representation of the lower plate.

As for the housing of the springs, it is intended that they are distributed symmetrically, they were then distributed by a circle of $\varnothing 190$ mm presenting an angle of 60° between them. The supplier's recommendations regarding the assembly of the springs were also followed, Figure 30 [28].

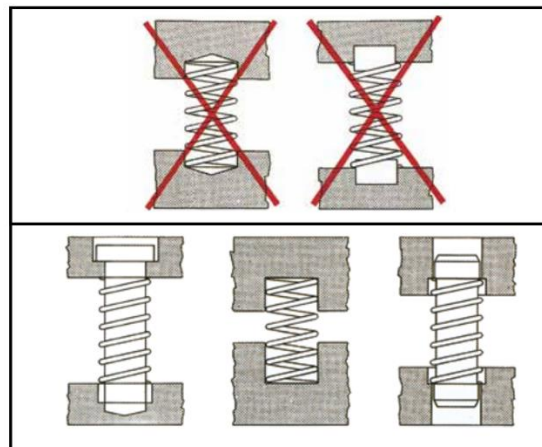


Figure 30- Recommendations for the assembly of the springs [28].

Since easy assembly of the springs is essential for the hole hemming tool, it was chosen to use only cavities. It was considered that when following the supplier's recommendations regarding the maximum compression of the springs, the springs are safe regarding the buckling type of

instability, which can occur in compression springs when the deflection exceeds the critical deflection.

Regarding the housing of the height adjuster in this component, this is achieved through an H7/g6 adjustment, a sliding adjustment, to guarantee the precision and alignment of the active components [30].

As previously mentioned, the clamping of the tool to the test machine is carried out through a bolt that threads simultaneously in the testing machine and in lower plate. According to *Morais et.al.* [31], it's possible to calculate the resistance of the connection through:

$$R_{p0,2} = \frac{F_{max}}{A_{sf}} \quad (13)$$

Where F_{max} is the maximum force that can be applied to the connection, A_{sf} is the resistant sectional area and $R_{p0,2}$ is the value of the nominal yield point.

Since the connection is made through an M30x2 thread, the following value for the resistant section area is obtained [31]:

$$A_{sf} = 621 \text{ mm}^2 \quad (14)$$

The lower plate is manufactured in F10 from *Ramada Aços*, so its nominal yield stress value is as follows [32]:

$$R_{p0,2} = 330 \text{ MPa} \quad (15)$$

Thus, the maximum force possible to apply can be obtained:

$$F_{max} = 204.930 \text{ kN} \quad (16)$$

Since the strength of the testing machine is just over 100 kN [23], the connection between the testing machine and the tool is secure, with a safety factor of approximately 2.

As for the bolt that threads simultaneously in the testing machine and in the tool, it is not necessary to carry out a verification, as it is made of a material whose nominal elastic limit is higher than that of the lower plate, PM300 from *Ramada Aços*, so it supports a higher load [32].

Care was also taken to follow the recommendations of the *Morais et.al.* [31] regarding the implantation length, j , and the thread length, B .

$$j \geq 1.5 * d \geq 45 \text{ mm} \quad (17)$$

$$B \geq j + 4P \geq 53 \text{ mm} \quad (18)$$

Where d is the thread diameter and P is the respective pitch. In order to satisfy the required thread length, a through hole was made, this component has a thickness of 105 mm, so the required thread length is guaranteed.

3.5.5 Upper drive plate

The upper drive plate has the same responsibilities as the lower plate, being responsible for the assembly of bushings and die holder, fixing the tool to the testing machine and aligning the plates, Figure 31.

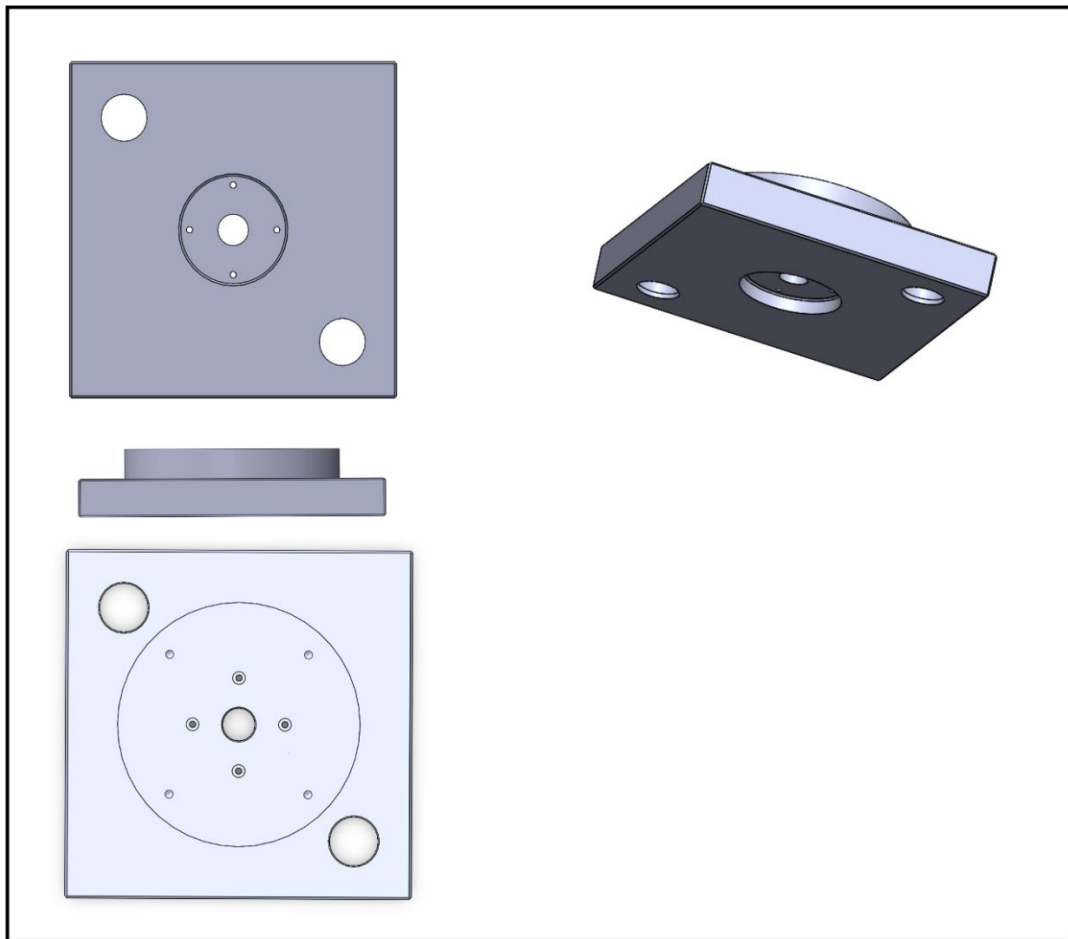


Figure 31- Schematic representation of the upper drive plate.

Regarding the housing of the die holder in this component, this is achieved through an H7/g6 adjustment, a sliding adjustment, to guarantee the precision and alignment of the active components [30].

As for fixing this component to the testing machine, since this component is made of the same material as the lower plate, the fixing method is exactly the same. The calculations performed for the lower plate are identical to those for the upper drive plate. So, the conclusion is also the same, the connection between this component and the testing machine is secure.

Note that in the calculations performed for the connection, the weight of these components was not considered, this is justified because the force caused by the weight of these components is insignificant when compared to the maximum force of the test machine.

3.5.6. Simulations

As for the accuracy and alignment of the lower, middle and upper parts, these two components play a very important role, as it is on these two components that all other components are based, so any misalignment or deformation in these two components can cause misalignment between the punches and the dies, poorly distributed forces or even jamming of the tool's operation [29]. To avoid these problems, special care was needed in the quotation, geometric tolerance and dimensional tolerance of these two components. It was also necessary to carry out several simulations using the *SOLIDWORKS2021's* software in order to be able to evaluate the deformations and stresses existing in these components in a critical case of operation.

Another of the reasons that encouraged the performance of simulations for these two components was the fact that the testing machine used only has a fastening system through a pin (as mentioned above) that fixes a small area in the center of the plates. This fastening system can facilitate the bending of the plates.

To carry out the simulations, as previously mentioned, the *SOLIDWORKS2021's* software was used in Static Simulation mode. Both components were manufactured in material F10 from *Ramada Aços*, so the properties of this material were assigned to the simulations [32]. Regarding the applied forces and the fixing method, the fixing method is the same in both components (fixed through the M30x2 thread), Figure 32 [23]. Regarding the applied forces, these are different in the two components. Starting with the upper drive plate, in this component the only force that was considered was the force that the die holder exerts on it. As for the lower plate, in this component only the force exerted by the springs was considered, in order to obtain the most critical case of bending (note that in reality this force is applied by the springs and the height adjuster, but in order to study the critical case of bending, it was chosen to consider all the force applied to the springs). In both cases, a force of 100 kN was considered, corresponding to the maximum capacity of the testing machine assured by the manufacturers [23], this value

was considered, because in this way the use of the structural components in any other process involving this testing machine is guaranteed.

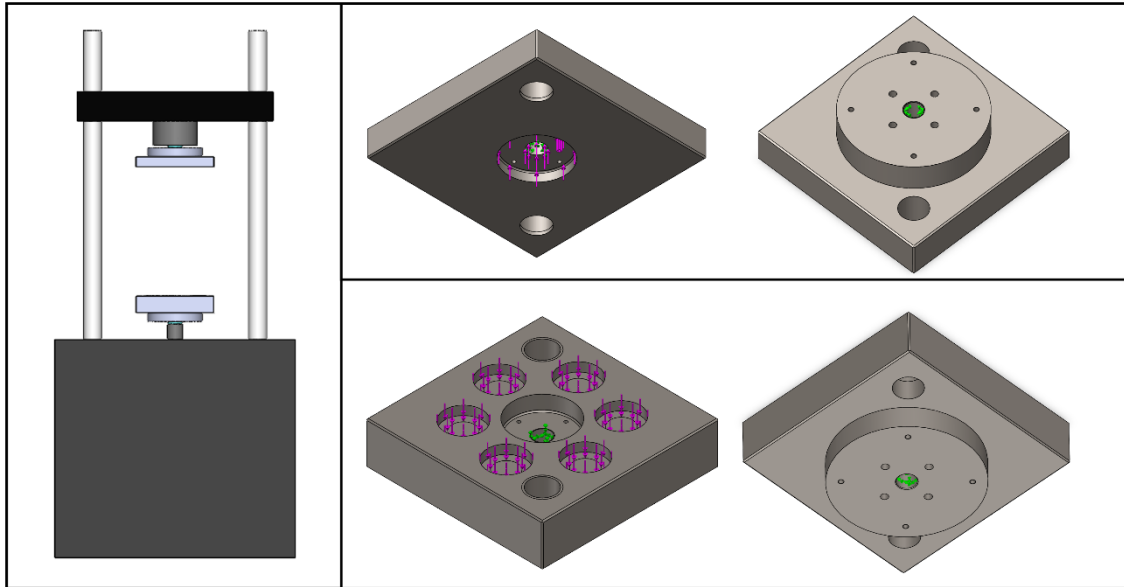


Figure 32- Fixtures and forces applied on the upper drive plate and lower plate.

To guarantee alignment and centering, it was necessary to ensure that the maximum displacement of the guide pillar holes was less than a certain value.

For precision die sets, tolerances between bushings and guide pillars are maintained from 5 μm to a maximum of 10 μm (Figure 33). This tolerance assures extremely accurate alignment between punches and corresponding holes in die blocks. For commercial die sets are given more liberal clearances between bushings and guide pillars. These range from 10 μm to 23 μm [29].

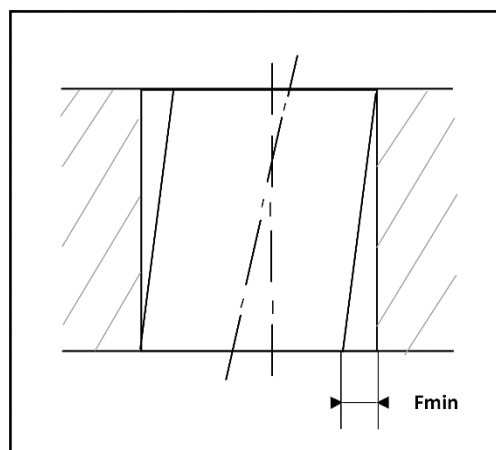


Figure 33- Schematic representation of the clearance between bushing and guide pillar.

Considering the tightest tolerance, it was defined that the admissible value for the displacement of the holes of the guide pillars, δ_{adm} , must be less than 5 μm .

Due to the distribution of the springs, the maximum displacement due to bending in the holes of the guide pillars occurred in the diagonal direction.

The obtained results from simulations are presented in Table 6 and 7 and in Figure 34:

Table 6- Results obtain for lower plate.

Lower plate	
Max von Mises stress	100 [MPa]
Max displacement in holes	> 5 [μm]

Table 7- Results obtain for upper drive plate.

Upper drive plate	
Max von Mises stress	56 [MPa]
Max displacement in holes	> 5 [μm]

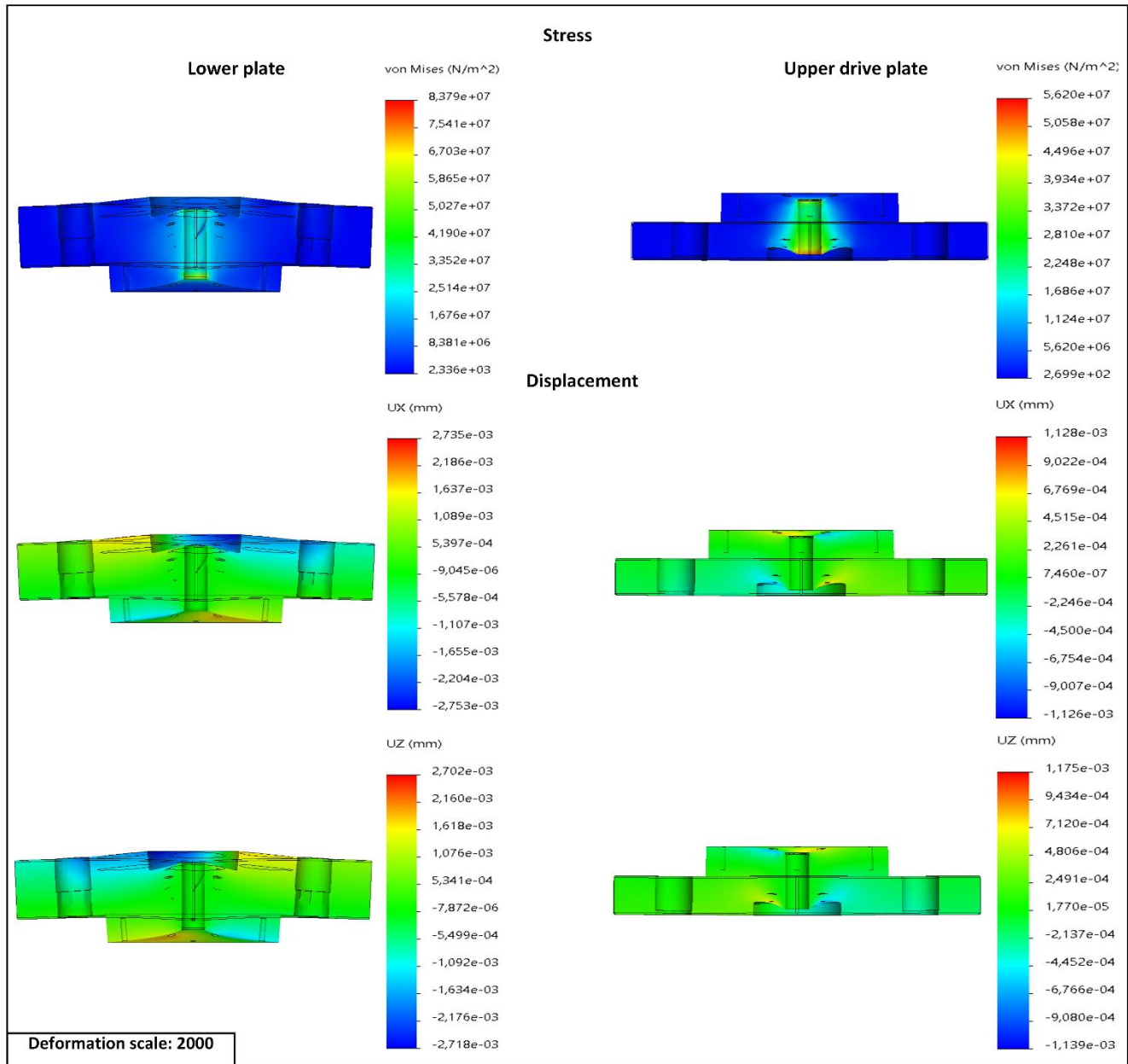


Figure 34- SOLIDWORKS simulations of upper drive plate and lower plate.

To ensure that the displacement of the holes in the guide pillars did not exceed 5 μm , it was necessary to increase the thickness of the plates beyond the values observed in the catalogues. This resulted in an increase in the weight of these plates. To reduce this weight, the thickness is only increased in a disc where the forces are applied, thus allowing to reduce the bending effect. The final dimensions of these components are shown in Table 8:

Table 8- Final dimensions of lower plate and upper drive plate.

	Lower plate	Upper drive plate
Width	300 [mm]	300 [mm]
Length	300 [mm]	300 [mm]
Thickness	105 [mm]	80 [mm]

Due to the high minimum preload required for this tool, 40 kN, it was considered interesting to analyze the functionality of these components, using 4 springs instead of 6 springs. This makes it possible to use a minimum preload of 26 kN.

Regarding the functioning of the springs, this is practically all the same, allowing exactly the same length of deformation. The only differences lie in the absolute value of the force and stiffness of the spring set.

In the case of the upper drive plate, it only suffers a reduction in the value of the applied force, not being necessary to carry out any additional verification.

Considering now the lower plate, it is necessary to carry out new simulations in order to ensure that the displacement of the holes of the guide pillars remains less than 5 μm . In carrying out the simulations, all parameters equal to the case presented above were considered, with the exception of the applied forces. For the applied forces, attention had to be paid to the positioning of the springs, and the reduction of the force applied by them.

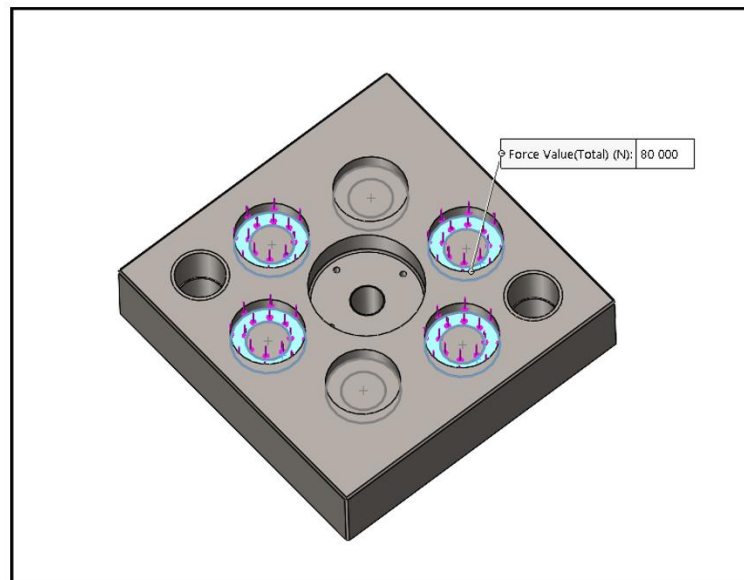


Figure 35-Forces applied by the set of springs.

As for the positioning of the springs, this was done to obtain a deformation due to bending similar to that of the previous case, so the distribution shown in Figure 35 was used. Regarding the force applied by these, a total force of 80 kN was considered.

The results obtained from the simulations are presented in table 9 and Figure 36:

Table 9- Results obtain for lower plate with a set of 4 springs.

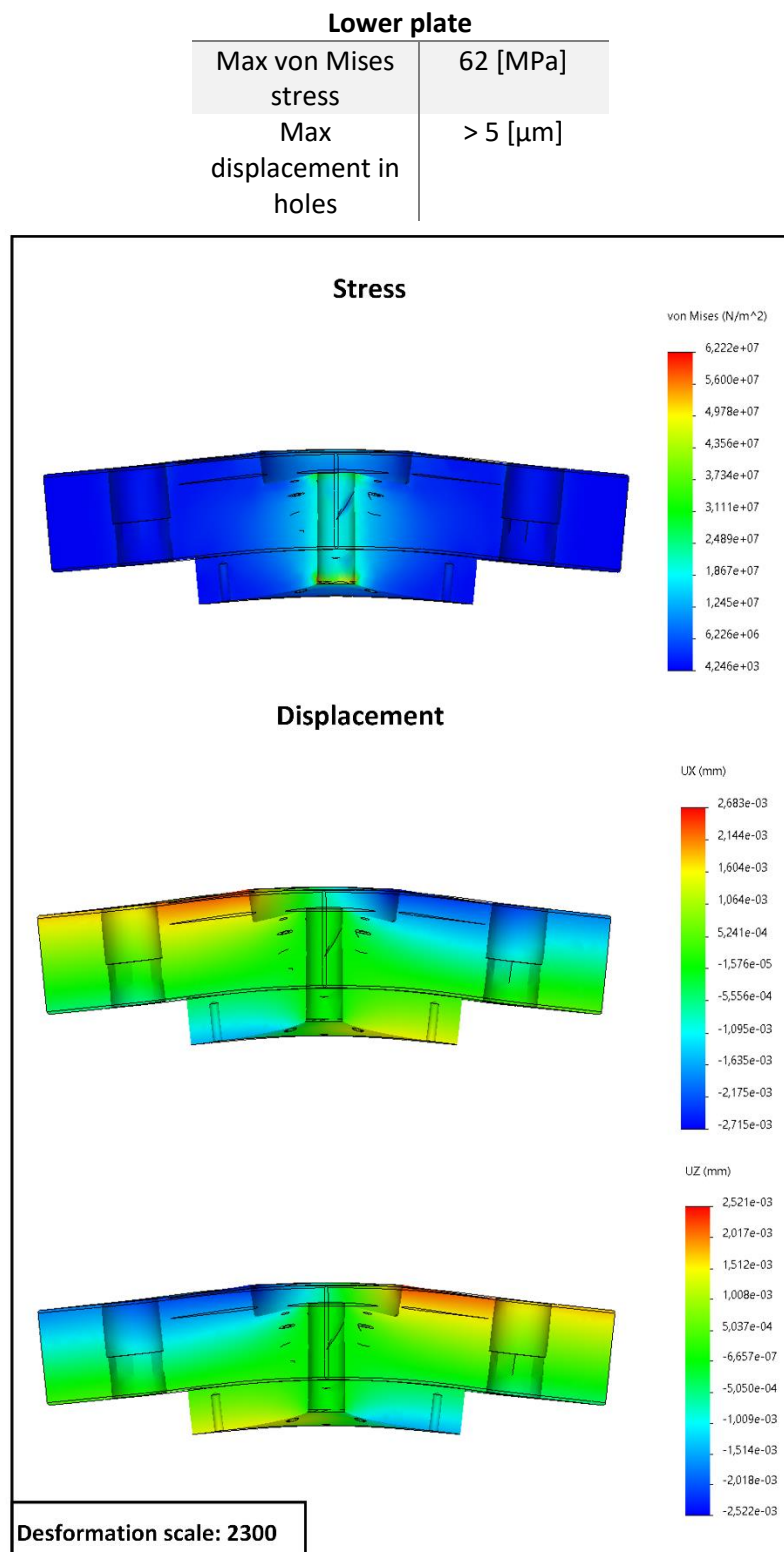


Figure 36-Solidworks simulation of the lower plate for a set of 4 springs.

3.5.7. Guide pillars

Guide pillars are precision-ground pins that are fitted into accurately bored holes in the plates. They engage guide bushings to align punch and die components with a high degree of closeness and accuracy [29].

In the developed tool, the guide pillars, in addition to providing the alignment, also provide the preload variation. This component must contain a threaded part and an entry for housing the keys.

To guarantee a load variation of 20 kN it was necessary to guarantee a certain thread length. As seen earlier, the force exerted by the springs can be calculated using the Expression 10. So, the displacement required to apply a certain force can be given by:

$$\delta = \frac{F}{R * N_s} \quad (19)$$

Knowing that the resistance of the springs, R , is 0.128 kN/mm, the number of springs, N_s , is 6 and the force variation required is 20 kN, the distance is needed to compress the springs is:

$$\delta = 26 \text{ mm} \quad (20)$$

Therefore, the thread length must allow this distance variation, Figure 37.

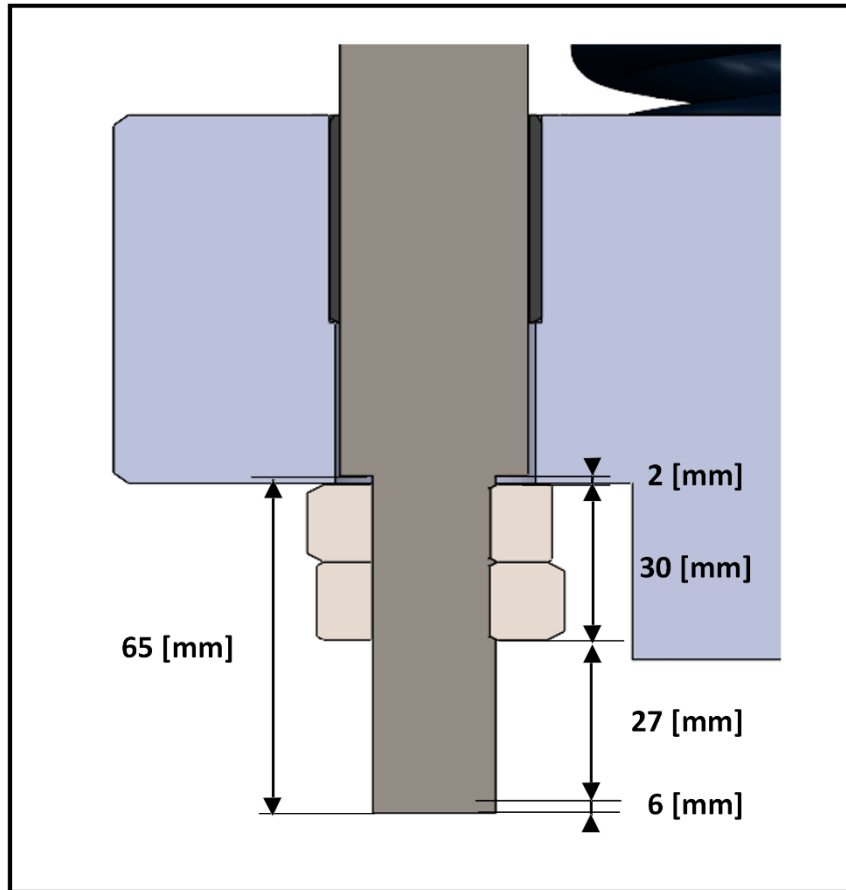


Figure 37-Dimensions of the guide pillar's thread.

The thickness of the selected nut and lock nut was considered and the recommendation of *Morais et.al.* [31] on the length of the protrusion, n , was taken into account:

$$n \geq 2 * P \quad (21)$$

Another aspect that was considered, was to prevent the corner of the guide pillars from entering the bushings, in order to avoid a rapid degradation of these components or even the tool jamming [29]. To this end, in the design of the lower plate, it was necessary to ensure that the distance between the projection on which the bush is located and the lower part of the lower plate was greater than 27 mm.

The last aspect to pay attention to in the length of the thread was to avoid contact between the tip of the guide pillar and the testing machine, so as not to damage either of them. According to the testing machine's data sheet, the height of the actuator where the lower plate will be assembled is 63 mm [23]. In the case of maximum preload, the guide pillar thread exceeds the lower part of the lower plate by 31 mm, considering that the punch has a penetration of 20 mm will result in a movement of the guide pillar by 20 mm, which will correspond to a maximum of

51 mm. Check that there will be no contact, however the testing machine still has the functionality to lift the actuator.

Regarding the length of the guide pillars, attention had to be paid to:

- the housing of the bushings;
- the length of the springs;
- the thickness of all tool components;
- the guide pillars should have sufficient length so that they never leave their bushings during operation. This is a safety feature to prevent possible crushing of fingers accidentally introduced between posts and bushings as the die is operated [29];
- sufficient opening of the tool to be able to change the active components;
- prevent the guide pillars from touching the testing machine, so as not to damage both.

Regarding the diameter and material to be used in the guide pillars, it was necessary to carry out verifications both on the thread and on the notch where the key will be housed. It should be noted that the process of determining the diameter was an iterative process between checking the thread, the notch for the key and the key itself.

The *Equinotec* company was contacted to manufacture the guide pillars. The material used was heat treated Cf53 (1.1213), with the mechanical properties shown on Figure 38 [33]:

Mechanical Properties in Hardened+Tempered Condition

Diameter (mm)	0.2 % proof stress (N/mm ²)	Tensile strength (N/mm ²)	Elongation (%)	Reduction (%)
up to 16	510	740-880	12	25
17-40	430	690-830	14	35
41-100	400	640-780	15	40

Figure 38- Mechanical properties of the guide pillar's material [19].

Starting with the thread verification, the process used was identical to the one previously presented for the lower plate and upper drive plate. The thread was considered to be subject to compressive stresses, so it is possible to calculate the resistance of the thread through Expression 13 [31].

Since the connection is made through an M30x2 thread, by consulting *Morais et.al.* [31], can be obtained the following value for the resistant section area:

$$A_{sf} = 621 \text{ mm}^2 \quad (22)$$

The guide pillars are manufactured in heat treated Cf53 (1.1213), so its nominal yield point value is [33]:

$$R_{p0,2} = 430 \text{ MPa} \quad (23)$$

Thus, the maximum force possible to apply can be obtained:

$$F_{max} = 267.030 \text{ kN} \quad (24)$$

The maximum force applied to the guide pillars is due to the preload, and after the contact of the dies with the sheet, force is no longer exerted on this component. It was previously mentioned that the maximum preload value would be 60 kN, however this force is distributed by the two guide pillars, considering a uniform distribution, the maximum applied force would be 30 kN. Thus, the thread of the guide pillar has a factor of safety in the order of 9.

As for the necessary checks for the notch of the shaft, it started by checking its traction stress [26]:

$$\sigma = \frac{F}{A_s} \quad (25)$$

Where F is the value of the applied force, as seen earlier, it has a value in the order of 30 kN. And A_s is the value of the area of the resistant section which can be given by:

$$A_s = \frac{\pi * D^2}{4} \quad (26)$$

Knowing that the shaft notch diameter, D, is 25 mm, it was obtained the following resistant section area:

$$A_s = 490.874 \text{ mm}^2 \quad (27)$$

With these values, it is now possible to obtain the value of the tension existing in the keyway of the guide pillar:

$$\sigma = 61 \text{ MPa} \quad (28)$$

As seen earlier, the guide pillar has a yield point stress in the order of 430 MPa, which means that has a factor of safety in the order of 7.

However, for the verification of the shaft notch, it is still necessary to analyze the stress concentration. Stress concentration is a highly localized effect that in some instances, it may be due to a surface scratch. If the material is ductile and the load static, the design load may cause yielding in the critical location in the notch. This yielding can involve strain strengthening of the material and an increase in yield strength at the small critical notch location [26].

In the development of the previous stress equations for tension and compression, it was assumed that no geometric irregularities occurred in the member under consideration, but it is quite difficult to design a machine without permitting some changes in the cross sections of the members. For example, rotating shafts must have shoulders designed on them so that the bearings can be properly seated and so that they will take thrust loads. Other parts require holes, oil grooves, and notches of various kinds. Any discontinuity in a machine part alters the stress distribution in the neighborhood of the discontinuity so that the elementary stress equations no longer describe the state of stress in the part at these locations. Such discontinuities are called stress raisers, and the regions in which they occur are called areas of stress concentration. Stress concentrations can also arise from some irregularity not inherent in the member, such as tool marks, holes, notches, grooves, or threads [26].

A theoretical, or geometric, stress-concentration factor k_t or k_{ts} is used to relate the actual maximum stress at the discontinuity to the nominal stress. The factors are defined by the equations:

$$k_t = \frac{\sigma_{max}}{\sigma_0} \quad (29)$$

$$k_{ts} = \frac{\tau_{max}}{\tau_0} \quad (30)$$

Where k_t is used for normal stresses and k_{ts} for shear stresses. The nominal stress σ_0 or τ_0 is the stress calculated by using the elementary stress equations and the net area, or net cross section. Sometimes the gross cross section is used instead, and so it is always wise to double check the source of k_t or k_{ts} before calculating the maximum stress. The stress-concentration factor depends for its value only on the geometry of the part. That is, the particular material used has no effect on the value of k_t . This is why it is called a theoretical stress-concentration factor. The analysis of geometric shapes to determine stress-concentration factors is a difficult

problem, and not many solutions can be found. Most stress-concentration factors are found by using experimental techniques. Though the finite-element method has been used, the fact that the elements are indeed finite prevents finding the exact maximum stress. Experimental approaches generally used include photoelasticity, grid methods, brittle-coating methods, and electrical strain-gauge methods. Of course, the grid and strain-gauge methods both suffer from the same drawback as the finite element method [26].

As such, the verification of the stress concentration in the shaft notch was performed through simulations in the *SOLIDWORKS2021*'s software. To carry out these simulations, the static mode of the software was considered. The force was applied to the notch of the guide pillar, with a value of 60 kN, presenting a safety factor of 2. For fixing the component, the entire length of the thread was considered. In order to obtain the best possible results, the mesh in the notch area was refined to the lowest possible value, as shown in Figure 39.

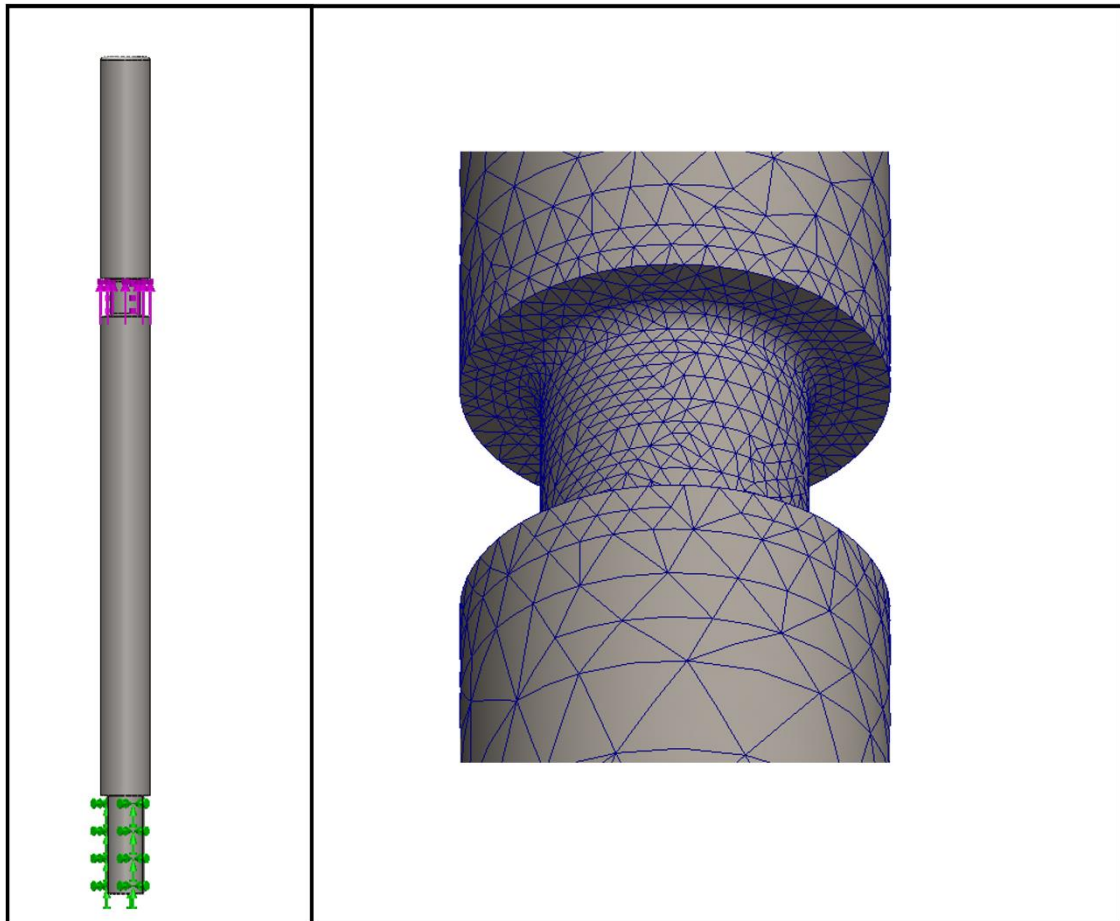


Figure 39- Mesh of the guide pillar's notch.

The results obtained from simulations are presented in Figure 40:

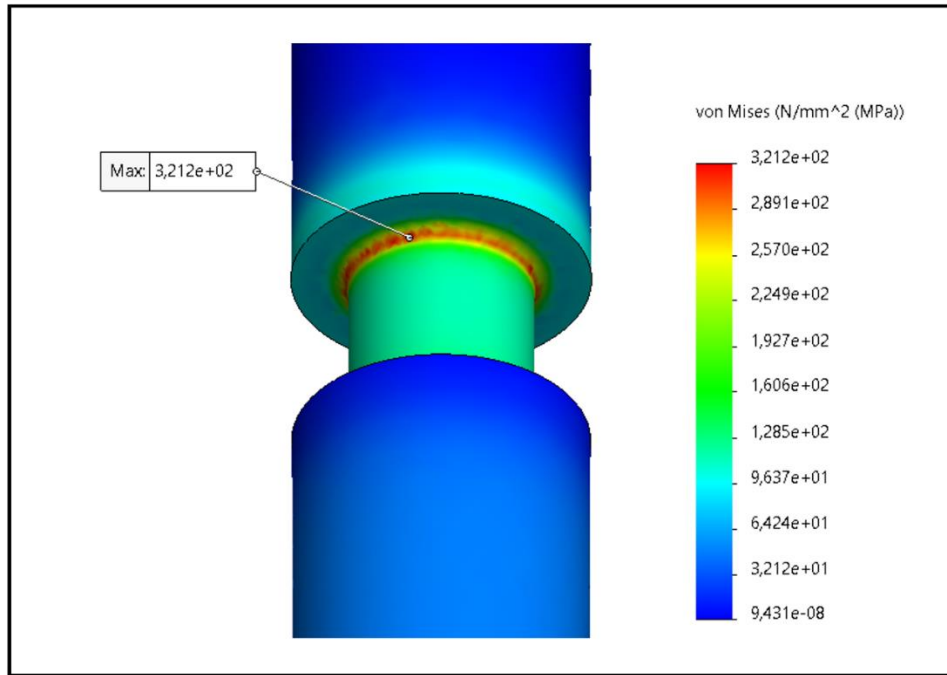


Figure 40- Von Mises stress results off the guide pillars.

The maximum value obtained through simulations in the SOLIDWORKS2021 software for the maximum von Mises stress was 321.2 MPa, so the notch of the guide column is safe.

3.5.8. Keys

The keys used for our tool have the function of preventing the ascending movement of the middle plate, through its compression against the guide pillars, these being prevented from the upward movement through the nuts, Figure 41 and 42.

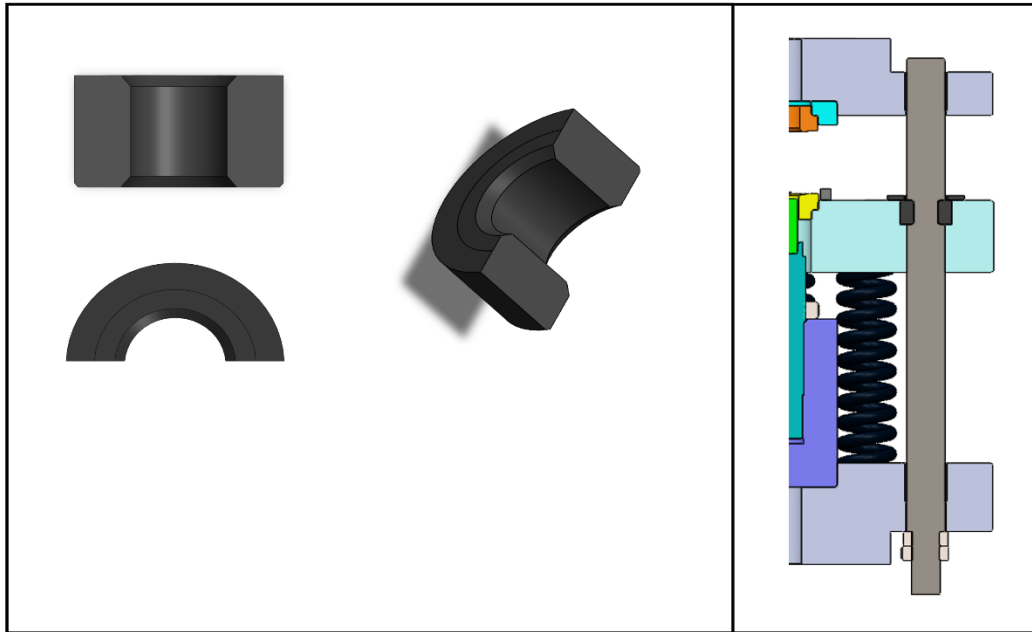


Figure 41- Schematic representation of the keys and their position in the hole hemming tool.

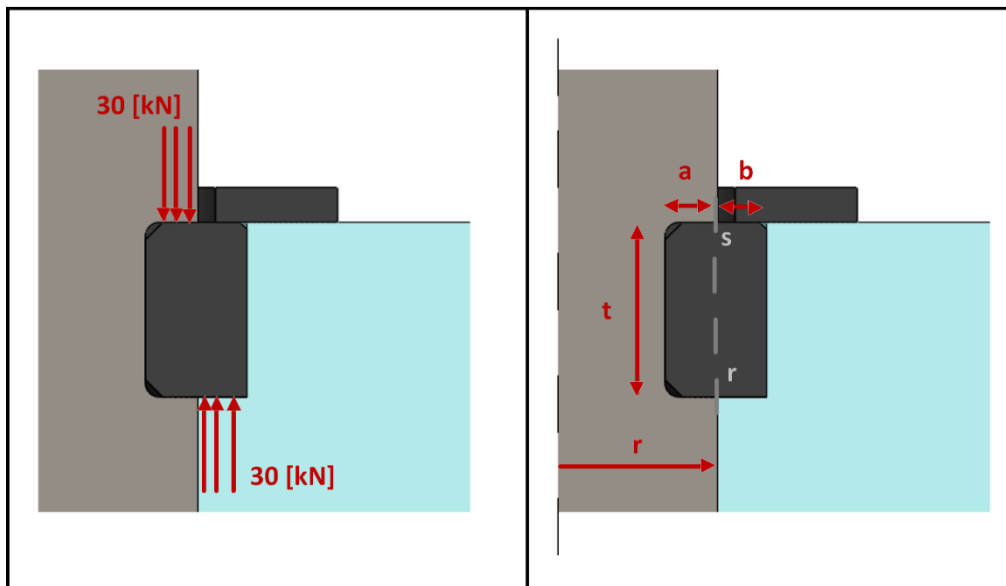


Figure 42- Schematic representation of the forces applied in the keys and key dimensions.

A set of 2 keys will be used for each guide pillar. As previously mentioned, the force exerted on each guide pillar will be a maximum of 30 kN, however, as two keys will be used for each guide column, the force considered for each one, F , will be 15 kN. A safety factor, n , of 4 was considered.

By the distortion-energy theory, the shear strength is [26]:

$$S_{Sy} = 0.577 * S_y \quad (31)$$

Where S_y is the yield stress. The material used for the production of the keys was the F10 from *Ramada Aços*, which has a yield stress in the order of 330 MPa [32]. Therefore, the value of shear strength is:

$$S_{Sy} = 190.41 \text{ MPa} \quad (32)$$

Failure by shear across the area $s - r$ will create a stress of [26]:

$$\tau = \frac{F}{Area} = \frac{F}{t * \pi * r} \quad (33)$$

Substituting the strength divided by the design factor, n , for t gives:

$$\frac{S_{Sy}}{n} = \frac{F}{t * \pi * r} \quad (34)$$

To resist crushing, the area of one-half the face of the key is used [26]:

$$\frac{S_{Sy}}{n} = \frac{F}{\frac{t * \pi * r}{2}} \quad (35)$$

Since the radius of the shaft, r , is 20 mm, it was obtained:

$$t = \frac{F * n * 2}{S_{Sy} * \pi * r} = 10 \text{ mm} \quad (36)$$

In order to obtain the values of a and b , *Morais et.al.* [31] was consulted, from which the typical values for the value of t obtained were taken.

$$a = 5 \text{ mm} \quad (37)$$

$$b = 4 \text{ mm} \quad (38)$$

Then it was necessary to check the compressive stress existing in each of the half faces:

$$\sigma_i = \frac{F}{A_i} \quad (39)$$

$$\sigma_o = \frac{F}{A_o} \quad (40)$$

Where A_i is the value of the area of the inner half face and A_o the value of the area of the outer half face.

$$A_i = \frac{\pi * r^2 - \pi * r_i^2}{2} \quad (41)$$

$$A_o = \frac{\pi * r_o^2 - \pi * r^2}{2} \quad (42)$$

From which the following values were obtained for the compressive stresses:

$$\sigma_i = 54 \text{ MPa} \quad (43)$$

$$\sigma_o = 54 \text{ MPa} \quad (44)$$

So, the key is secure.

However, these first values obtained only served as a first indication. This is because the keys are a component of great responsibility and safety in the tool. As such, it was decided to carry out the verification of the keys with the help of the SOLIDWORKS2021 software. To carry out these simulations, the static mode of the software was considered. A force of 15 kN was considered applied to the outer half face of the lower part of the keyway and the inner half face of the fixed upper part was considered.

After several simulations and different values for dimensions a , b and t , a compromise of values was reached. The final values for the dimensions were:

$$t = 25 \text{ mm} \quad (45)$$

$$a = 7.5 \text{ mm} \quad (46)$$

$$b = 7 \text{ mm} \quad (47)$$

Using these values and the best possible mesh available by the software, the results obtained for the von Mises stress distribution are presented in Figure 43.

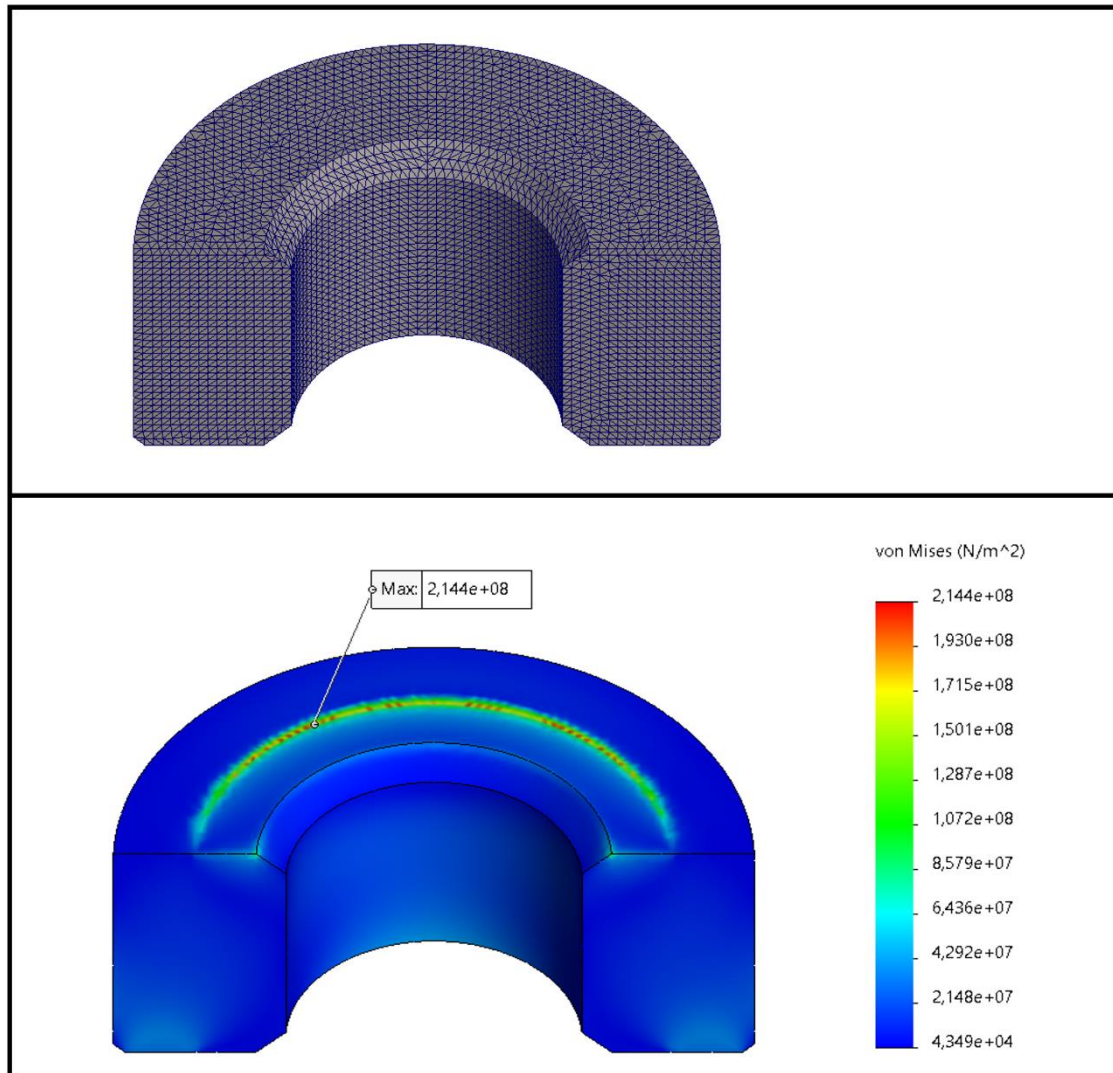


Figure 43- Mesh and von Mises stress results of the keys.

The value of the maximum von Mises stress was 214.4 MPa, presenting a safety factor in the order of 1.5.

3.5.9. Nuts

The nuts used for the tool, complement the work of restricting the movement of the middle plate carried out by the keys. The functionality of the nuts is to prevent only the upward movement of the pillars, through their compression against the lower plate. However, the downward movement of the guide pillars is still possible in order to perform the hole hemming process.

Two nuts are used for each guide pillar, the second nut working as a lock nut, so that the position of the nuts is not changed during the hole hemming process. Another feature of the nuts is the variation of the preload. As previously explained, for the variation of the preload it is necessary

to screw or unscrew 27 mm, allowing a variation of the preload in the order of 20 kN. Therefore, it is convenient to use the smallest possible thickness for the nuts.

The nuts used were Hexagon thin nut MF DIN 439 2 Steel Zinc plated 04 M30X2.00 purchased from *Fabory*, figure 44. These nuts have a thickness of 15 mm each and a material class of 04 [34].



Figure 44- Technical parameters of the nuts. [34]

According to the *Moraes et.al.* [31], the nuts are subject to compression forces, so it is sufficient to know the value of the nominal tensile strength, R_m . The material class of the nuts is expressed only by the first material class number of the mating bolt. In low nuts, as is in this case, the class is preceded by 0, to signify small load capacity [31].

Being the class of our nut 04, we get:

$$R_m = 4 * 100 \text{ MPa} = 400 \text{ MPa} \quad (48)$$

The compression stress is given by Expression 25. Consulting *Moraes et.al.* [31], can be obtained the area of the resistant section:

$$A_s = 621 \text{ mm}^2 \quad (49)$$

Manipulating the Expression 25, it can be obtained the maximum force that the thread can withstand:

$$F = \sigma * A_s = R_m * A_s = 248.4 \text{ kN} \quad (50)$$

The maximum force that the thread can withstand is 248.4 kN, however the maximum preload that can be applied is 60 kN, which is distributed over the two guide pillars. Considering this uniform distribution, the maximum force applied to the threads will be 30 kN. So, the nuts have a safety factor in the order of 8.

3.5.10. Bushings

The bushings mounted on the tool were 4 EGB4040-E40 bushings from *Schaeffler*, the most important specifications are shown in Figure 45 [35]. Two bushings were assembled on each guide pillar, one of them housed in the lower plate and the other in the upper drive plate.

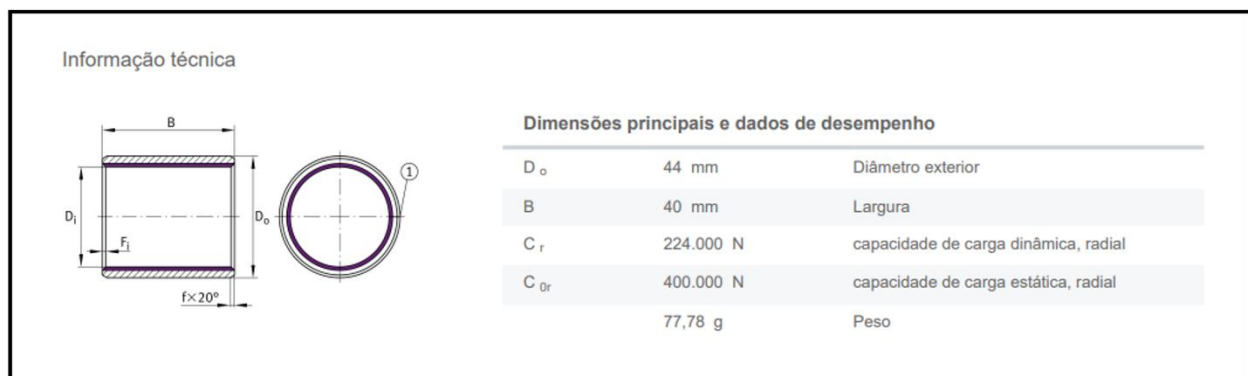


Figure 45- Technical parameters of the bushings. [35]

To assemble the bushings correctly on the upper drive plate and on the lower plate, the dimensions shown were followed. For the bushings to settle and be completely inserted in the upper drive plate and in the lower plate, in these components a small projection was created that allows the seating of the bushings so that they are completely inserted, Figure 46.

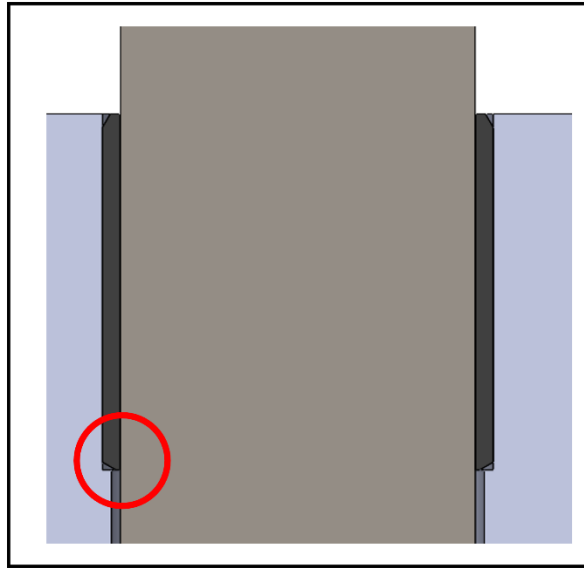


Figure 46- Schematic representation of the assembly of the bushings.

This protrusion, in addition to guaranteeing the seating of the bush, also makes it possible to dismantle it.

3.5.11. Plates

The 'plates', Figure 47, were designed with the aim of covering the keys and preventing them from being dismantled for some reason.

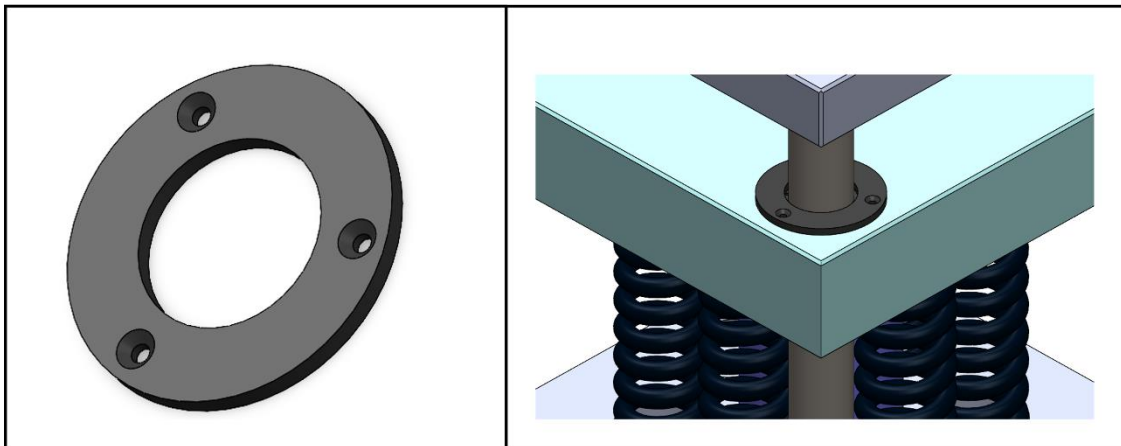


Figure 47- Schematic representation of the keys and its assembly in the hole hemming tool.

Given their functionality, these components do not require any verification. These components were made of F10 steel by *Ramada Aços*.

3.5.12. Pins

The pins were used to ensure alignment and centering between the center of the hole in the sheets and the center of the hole in the dies.

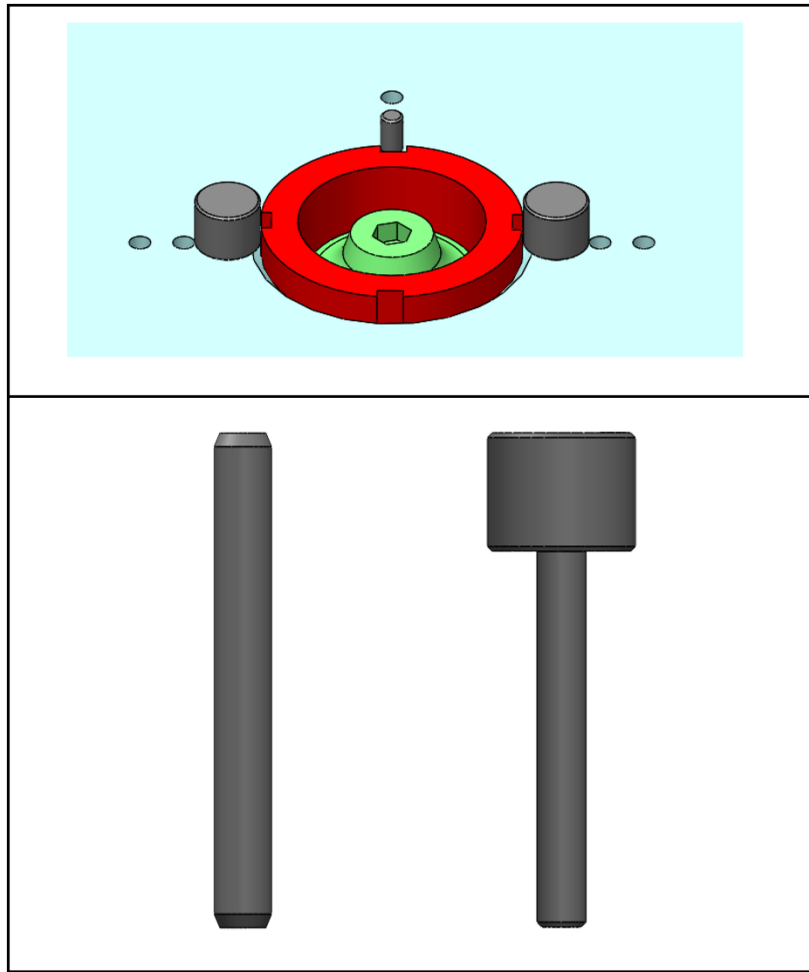


Figure 48- Schematic representation of the pins and its assembly on the hole hemming tool.

It was considered necessary to use 3 pins with the arrangement shown in the Figure 48, in order to guarantee the easy positioning of the sheets. Two types of pins were designed, one without a head and the other with a head to allow a greater variation in the dimensions of the sheet to be tested.

The headless pins were purchased from the *Fabory* company, being DIN 6325 Tolerance H6 steel pins, 5 mm in diameter and 50 mm in length. As can be seen in the attached technical drawings, this component will be placed in the middle plate, which has a 38 mm cavity, so the pins are 12 mm out of this component.

As for the headed pins, these were produced by *FEUP mechanical engineering department workshop*. The material used was F10 from *Ramada Aços* [32]. Like the headless pins, these are outside the middle plate 12 mm, and the head of these new pins must have a height of 12 mm. A head diameter of 15 mm with h6 tolerance was considered, in order to have the same precision as headless pins. As for the diameter of the body of the pin, it was considered equal to that of the previous pins, 5 mm with tolerance h6, to be possible to mount it in the same cavities.

For these components, it wasn't necessary to carry out any verification, since its objective is only to align and center the sheets, however, it was necessary to pay attention to the height of the pins in the dimensioning of the dies to avoid contact between the pins and the die holder.

3.5.13. Screws

For the connection of some structural components to the passive components that they couple, screws were used. Below is presented Table 10 with a list of the screws used, all of which are obtained from *Fabory*.

Table 10- List of screws utilized for the structural components.

Qt.	Norm	Component
4	ISO4762 M5x0.8x20-8.8	Lower plate
4	ISO4762 M5x0.8x20-8.8	Upper drive plate
6	ISO10642 M4x0.7x10-8.8	Plates

As for the screws that connect the plates to the middle plate, it was not necessary to carry out any verification.

For the screws that connect the upper drive plate to the die holder, a check was carried out to ensure that they support the weight of the die holder and die. Knowing that the bolt class is 8.8, there is a nominal tensile strength of [31]:

$$R_m = 8 * 100 [MPa] = 800 [MPa] \quad (51)$$

And a nominal elastic limit of [31]:

$$R_{p0.2} = 0.8 * R_m = 640 [MPa] \quad (52)$$

The tensile stress of a bolt is given by Expression 13 [26]. According to *Morais et.al.* [31], are obtained the following value for the resistant section area:

$$A_s = 14.2 \text{ mm}^2 \quad (53)$$

As for the applied force, is known that the die holder has a mass of 2 kg, and the dies approximately 1 kg. Considering the gravity acceleration value as 9.81 m/s^2 , we will have a force of 29.43 N. However, this force is distributed by 4 screws, so the force value to be considered will be:

$$F = 7.35 \text{ N} \quad (54)$$

It is now possible to obtain the value of the tractive effort:

$$\sigma = 518 \text{ Pa} \quad (55)$$

It is then concluded that the screws can support the die holder and its matrix with a very significant margin.

For the screws that connect the lower plate to the height adjuster, these only ensure that the component does not come apart. Since the assembly is performed using an H7/g6 adjustment, the height adjuster is located on the lower plate and not suspended as in the case of the die holder [30].

To manufacture the structural components, after all the mechanical drawings were reviewed, the order of all the materials was needed. So, all the steel components were bought from *Ramada Aços*, except for the guide pillars, whose material was purchased and produced under the responsibility of *Equinotec*. The springs were bought from *Unceta*, and the bushings were bought from *Schaeffler*. The keys, pins and plates were able to be manufactured by *FEUP mechanical engineering department workshop*. The upper drive plate and lower plate were sent to *Tecnogial*.

3.6. Passive tool components

This section describes the design and validation process for building the passive tool components of the hole hemming tool, shown in Figure 49. The passive tool components of the hole hemming tool were designed based on the following requirements:

- Accommodate the active tool components;
- Provide guidance and centering;
- Allow adjustment of the position of the active tool components.

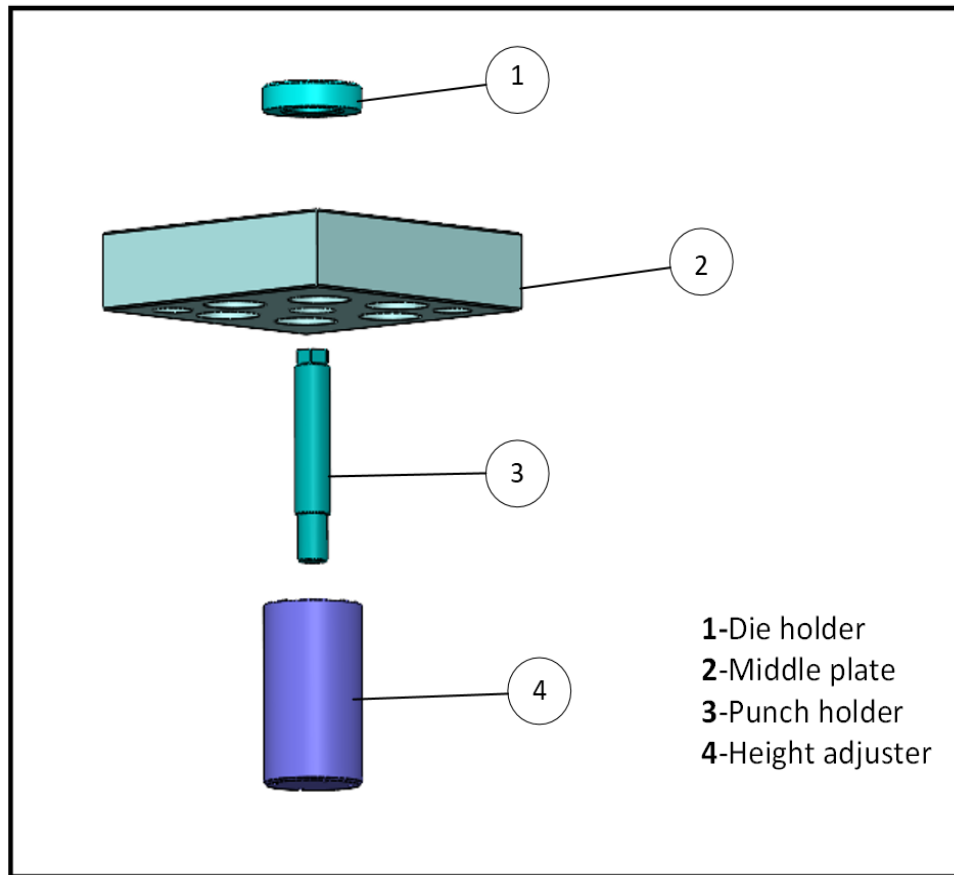


Figure 49- Schematic representation of the passive tool components of the hole hemming tool.

3.6.1. Adjustment of the active tool components position

The need to adjust the position of the active components, more specifically the punches, arises from the possibility of varying the preload of the hole hemming tool. As previously explained, the preload variation is achieved through the compression or expansion of the springs, therefore, when the working length of the springs varies, the distance between the middle plate and the lower plate also varies, so the relative position between the middle plate and the punch is changed. The need arises to adjust the position between these two components with relative ease. The possibility of adjusting this position also allows the variation of some process conditions, namely the penetration distance of the punch.

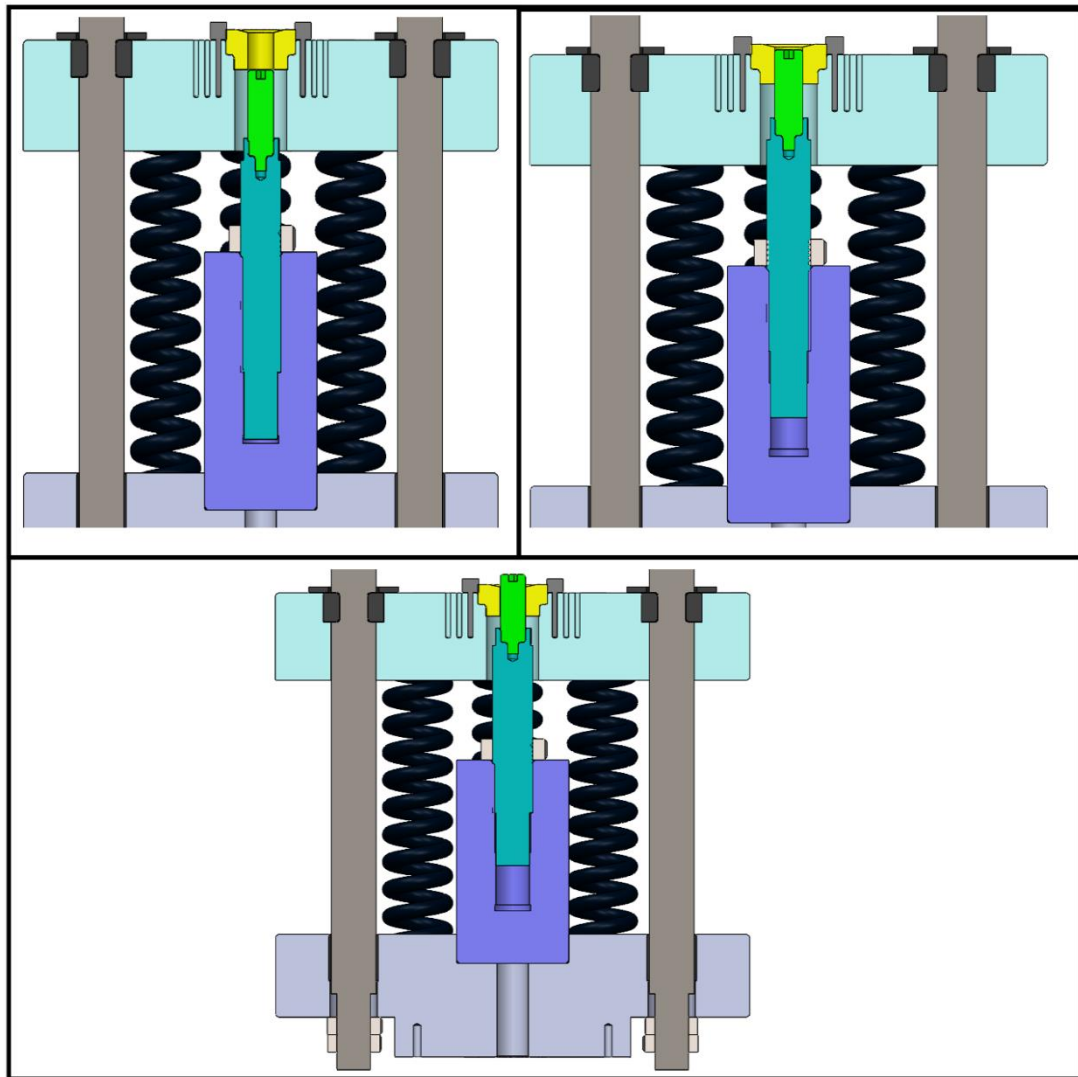


Figure 50- Schematic representation of the variation of the punch position.

The solution found for this adjustment involves dividing the typical punch holder into two components: the punch holder and the height adjuster. These components have, respectively, an external thread and an internal thread, so that it is possible to screw or unscrew the punch holder on the height adjuster, causing the punch to rise or fall. To guarantee that the defined position does not change during the process, it is necessary to introduce a nut. The adjustment process of these components is illustrated in Figure 50.

3.6.2. Height adjuster

The height adjuster is the component responsible for supporting, centering and varying the position of the punch holder and consequently of the punches, Figure 51.

Starting with the centering, this component is assembled on the upper drive plate through an H7/g6 adjustment, which guarantees precision guidance. The punch holder is mounted on the height adjuster through a thread, so the thread does not guarantee any centering or precision.

To guarantee the centering, a $\varnothing 30$ H7 hole was drilled before the thread, in this way the centering of the punch holder is guaranteed by an adjustment H7/g6 in this hole and not by the thread [30].

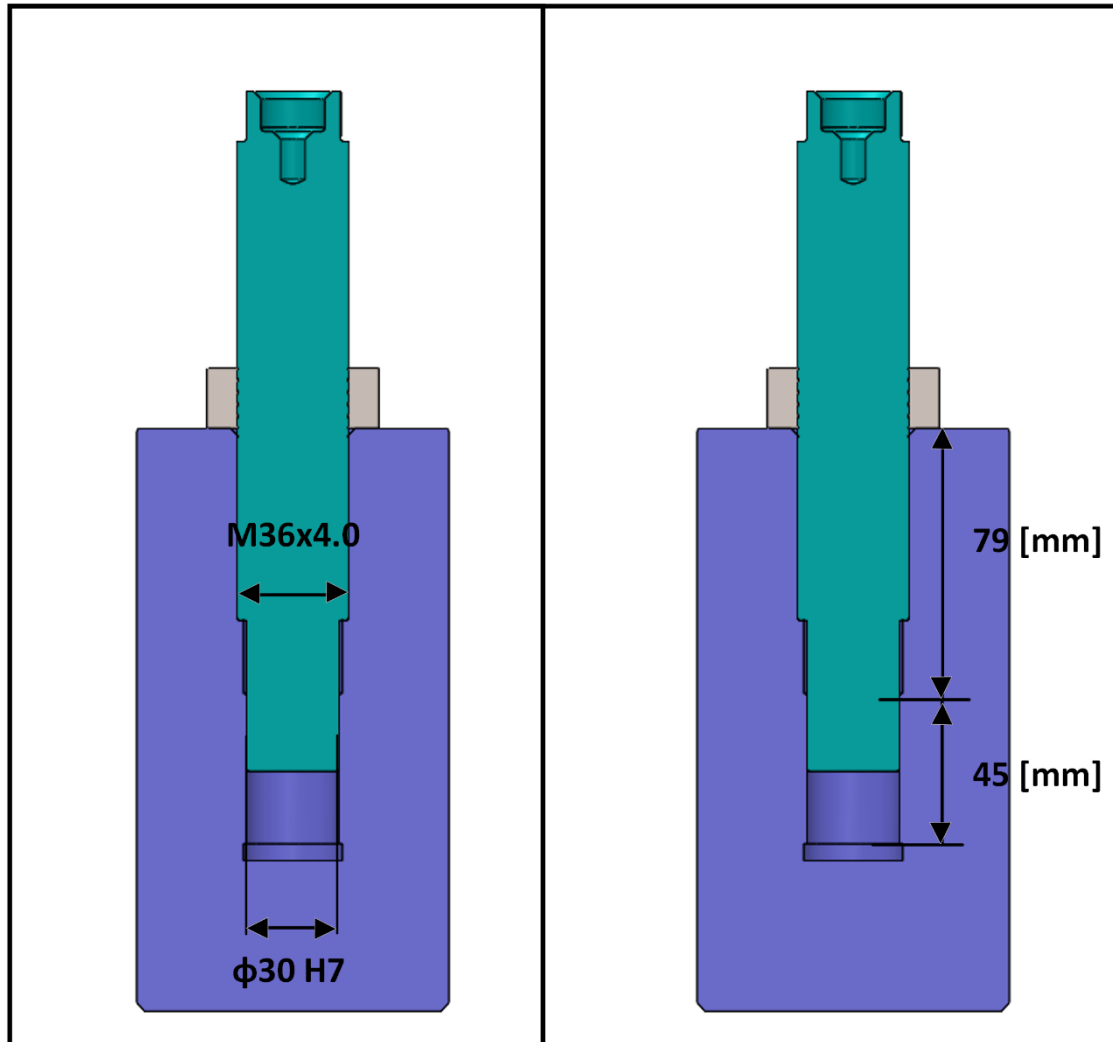


Figure 51- Dimensions of the height adjuster.

As for the variation of the punch holder position, it was said earlier that for the variation of the preload the springs could be compressed up to 27 mm. Therefore, the height adjuster must at least allow the variation of this height in the punch holder, for this it was necessary to define the thread length and the hole length that allows the punch holder to be centered. According to the *Morais et.al.* [31], the implantation length, j , for hard steels, can be given by:

$$j \geq d \quad (56)$$

As the thread diameter used is 36 mm, it is obtained a j of 36 mm. However, the possibility of a variation of 27 mm is still necessary, so the value of j will be:

$$j = d + 27 \text{ mm} = 63 \text{ mm} \quad (57)$$

It is still necessary to calculate the value of the thread length, B :

$$B \geq j + 4 * P \quad (58)$$

Since the thread used has a pitch of 4 mm, P , the following value was obtained for the thread length:

$$B = 79 \text{ mm} \quad (59)$$

As for the length of the hole that allows the adjustment, it was necessary to ensure that when the punch holder is completely threaded into the height adjuster, it does not touch the bottom, and to ensure that for the smallest implantation used, the tip of the punch holder is already inside the hole.

Finally, the punch holder support. This support is guaranteed through an M36x4.0 thread. The tensile stress of a thread, as seen before, can be given by the Expression 13 [12],[31].

The resistant section area can be given by:

$$A_s = \left(\frac{\pi}{4}\right) * (d - 0,935P)^2 \quad (60)$$

$$A_s = 817 \text{ mm}^2 \quad (61)$$

Where d is the diameter (larger diameter of the thread) and P is the pitch [31]. The material used for the height adjuster was F10 from Ramada Aços, which presents a nominal yield limit of [32]:

$$R_{p0,2} = 330 \text{ MPa} \quad (62)$$

By manipulating the previous expressions, it was obtained the following expression for the maximum force possible to apply to the height adjuster thread:

$$F = R_{p0,2} * A_s = 269.610 \text{ kN} \quad (63)$$

The force applied to the height adjuster thread will be the force applied by the sheet to the punch, which in turn will apply the same force to the punch holder, which in turn will apply the same force to the height adjuster thread. As seen earlier, through the simulations carried out by *Pereira et.al.* [22], the maximum value applied to one of the punches (hemming punch) is approximately 60 kN. Therefore, it is verified that the height adjuster thread has a safety factor of 4.5.

3.6.3. Punch holder

Looking now at the punch holder, Figure 52, this component is responsible for supporting, centering, and varying the position of the punches.

This component requires attention in a few more aspects than the height adjuster, in addition to its centering and force applied to the thread, namely, the compression exerted by the punch and buckling. As for the centering and force applied to the thread, the same procedure as the height adjuster was followed.

Starting with the length of the threaded part, since the implantation length and preload variation criteria were analyzed when defining the height adjuster thread length, for the punch holder thread length, the criterion used was to ensure that this was long enough to fully thread the height adjuster and still fit an M36x4 low nut.

As seen earlier, the height adjuster has a thread length of 79 mm. The nut used in this set is an M36x4 low nut, which has a thickness of 18 mm [31].

$$B \geq 79 + 18 \geq 97 \text{ mm} \quad (64)$$

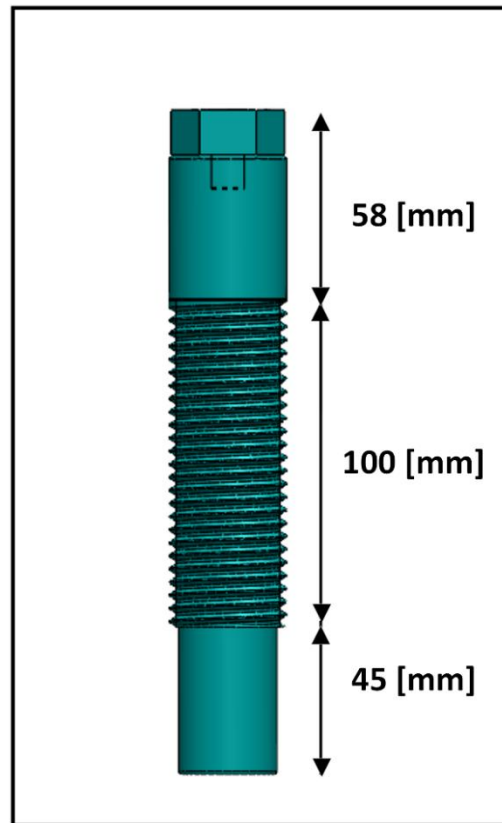


Figure 52- Dimensions of the punch holder.

As for the shaft length responsible for the adjustment between the punch holder and height adjuster, it was considered that this length should be equal to the length of the hole in the height adjuster. This is justified because, even if the punch holder is threaded to the end of the height adjuster thread, there is still 5 mm below this hole, which ensures that the punch holder tip and the height adjuster bottom never made contact.

Regarding the total length of the punch holder, this was defined together with the length of the punches, the thickness of the middle plate and its height. This is explained because it is necessary to ensure that the punches are positioned at a specific distance from the dies. However, this process will be explained in more detail later.

To facilitate the screwing and unscrewing of the punch holder when assembled in the height adjuster, a hexagonal socket was created on the upper end of the punch holder. The designed hexagon followed the dimensions given by *Budynas et.al.* [26], for hexagonal head screws, Figure 53.

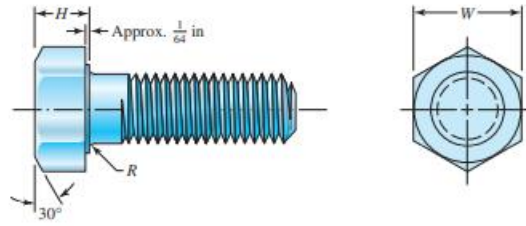


Figure 53- Schematic representation of a hexagonal head screw. [12]

The following dimensions were used:

$$w = 30 \text{ mm} \quad (65)$$

$$H = 15 \text{ mm} \quad (66)$$

As for the necessary verifications on this component, start by checking the thread. As seen earlier, the tensile stress of a thread, can be given by Expression 13 and the resistant section area by Expression 60 [26], [31].

For the punch holder, the steel used was PM300 from *Ramada Aços*, this choice is explained by the greater demands of this component, as will be seen later. For this material, *Ramada Aços* only presents the value of the minimum and maximum hardness supplied [32]:

$$\text{Min} = 290 \text{ HB} \quad (67)$$

$$\text{Max} = 330 \text{ HB} \quad (68)$$

However, it is possible to convert hardness into tensile strength.

Table 11- Conversion table of tensile strength to hardness [36].

T.S. Rm 820-2180 [MPa]	Hardness Vickers [HV]	Hardness Brinell [HB]	Hardness Rockwell C [HCR]
965	300	285	29,8
995	310	295	31
1030	320	304	32,2
1060	330	314	33,3
1095	340	323	34,3
1125	350	333	35,5
1155	360	342	36,6

By consulting Table 11, is expected a tensile strength value of [36]:

$$R_m \geq 995 \text{ MPa} \quad (69)$$

However, to verify the thread, it is necessary to know the value of the yield stress. It is not possible to obtain this from tensile strength. Through the literature review it was possible to find studies where the yield stress of this material was determined.

The study found is presented in Figure 54 [37]:

TENSILE STRENGTH Approx. values. Samples were taken from a flat bar, 90 x 300 mm (3.5" x 11.8"). Hardness: 325 HB.		
Testing temperature	20°C (68°F)	200°C (390°F)
Ultimate tensile strength $R_m \text{ N/mm}^2$	1020	930
Yield strength $R_{p0.2} \text{ N/mm}^2$	900	800

Figure 54- Mechanical properties of PM300. [37]

Is expected a value for the yield point of PM300 of approximately [37]:

$$R_{p0.2} \approx 800 \text{ MPa} \quad (70)$$

As for the area of the resistant section, since the thread is the same as the height adjuster, the area of the resistant section will also be [31]:

$$A_s = 817 \text{ mm}^2 \quad (71)$$

By manipulating the Expression 13, the following expression for the maximum force possible to apply to the punch holder thread is obtained:

$$F = R_{p0.2} * A_s = 653.600 \text{ kN} \quad (72)$$

As previously seen, the maximum force applied to this component is 60 kN, so the thread of this component has a safety factor of 10.8.

Regarding the contact of the punch with the punch holder, as will be seen later, the punch only exerts force on the red area, Figure 55. This is because the thread used will be a loose thread so that alignment and centering are guaranteed by the adjustment and not by the thread. Therefore, no verification is required for this thread.

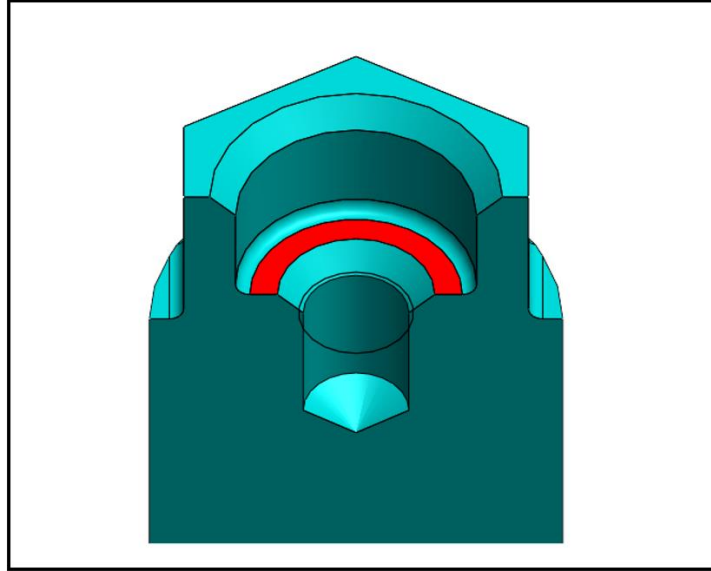


Figure 55- Schematic representation of the contact area of the punch holder.

Therefore, the verification of the compression exerted by the punch on the punch holder can be calculated using the Expression 25 [26]. However, it was also decided to carry out simulations using the SOLIDWORKS software, in order to be able to identify stress concentrations more easily. The SOLIDWORKS simulation was used in Static mode, a force of 100 kN was considered and the mesh was refined in the areas prone to stress concentrations, as shown in Figure 56.

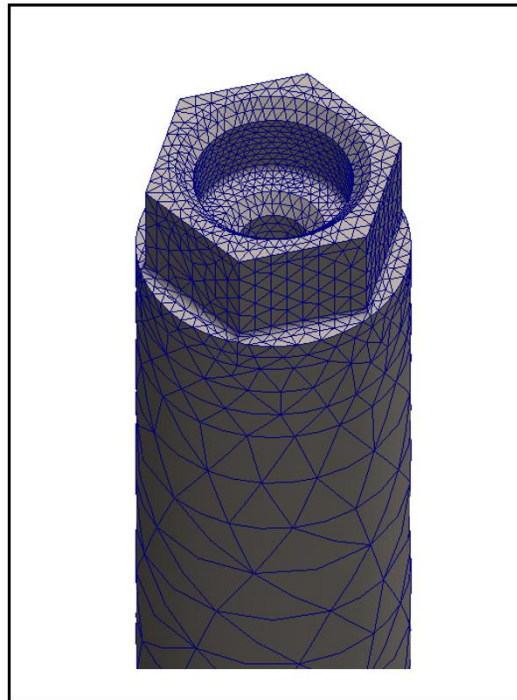


Figure 56- Mesh of the punch holder.

A force of 100 kN was applied to the contact area and the engagement of the entire punch holder was considered, as shown in Figure 57. This is because for the stress analysis in question, the length of the punch holder does not interfere.

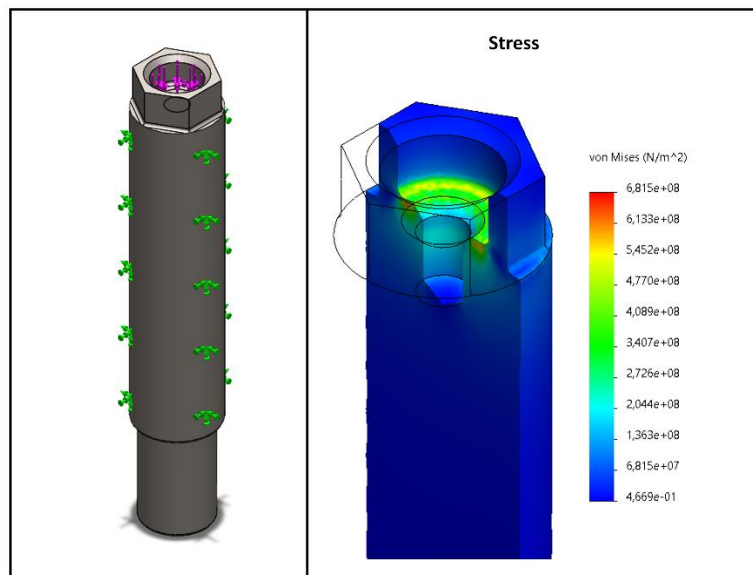


Figure 57- Fixtures, forces and von Mises stress of the punch holder.

Table 12- Results obtained for the punch holder

Punch holder	
Max von Mises stress	680 [MPa]

A von Mises stress value of 680 MPa was obtained, as shown in table 12, applying a force 1.67 times greater than the real force. This guarantees safety due to the compressive forces.

Lastly, buckling. According to *Nielsen et.al.* [17], the buckling case can be studied the expression:

$$\sigma_{cr} = \frac{F_{cr}}{S} \rightarrow \sigma_{cr} < \alpha C \frac{\pi^2 EI}{L^2 S} \quad (73)$$

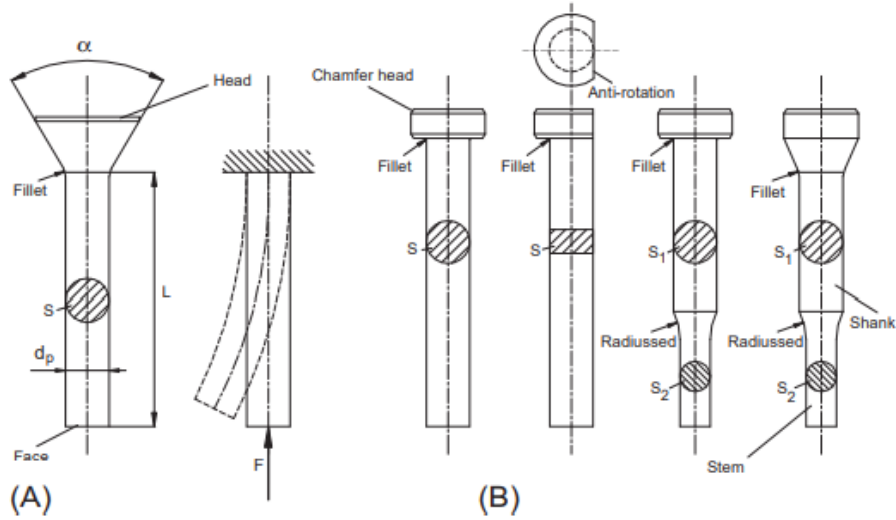


Figure 58-Punches. (A) Notation and illustration of Euler buckling (Euler case I) and (B) schematic details of different types of straight punches [17].

Where σ_{cr} is the critical instability stress, E is the elasticity modulus, I is the area moment of inertia, L is the length of the punch holder and C is a coefficient related to the punch holder support condition. Typical values of $C = 0.25$ are used in cases like that shown in the Figure 58, where the punch head is clamped, and the face is free [17].

Taking now into consideration that the area moment of inertia:

$$I = \frac{\pi}{64} * d^4 \quad (74)$$

Considering the punch holder end to be free and neglecting the safety factor, we can rewrite the expression:

$$L \leq \sqrt{\frac{c \cdot \pi^2 \cdot E \cdot d^2 \cdot 4}{64 \cdot \sigma_{cr}}} \quad (75)$$

Since the material used is PM300, it has a modulus of elasticity of [32]:

$$E = 205 \text{ GPa} \quad (76)$$

As previously mentioned, for this material the critical instability stress is [37]:

$$\sigma_{cr} = 800 \text{ MPa} \quad (77)$$

As for the punch holder diameter, d , it is 36 mm.

Then, the following value can be obtained:

$$L \leq 226 \text{ mm} \quad (78)$$

Corresponding this value to the maximum length that the punch holder can be found outside the height adjuster.

However, it was considered necessary to carry out simulations in the SOLIDWORKS2021 software in order to verify this result. To carry out these simulations, the buckling mode of the software was considered. A force of 100 kN was considered applied to the aforementioned area. As for fixing the material, the critical case was considered, that is, when only 36 mm of the punch holder is threaded in the height adjuster. Thus, for the buckling simulations, only the length:

$$L = 158 - 36 = 122 \text{ mm} \quad (79)$$

Was considered. And the lower end of the punch holder was considered fixed.

To carry out the simulations, the previously specified mechanical properties of the material were considered. The results obtained are presented in Figure 59 and Table 13:

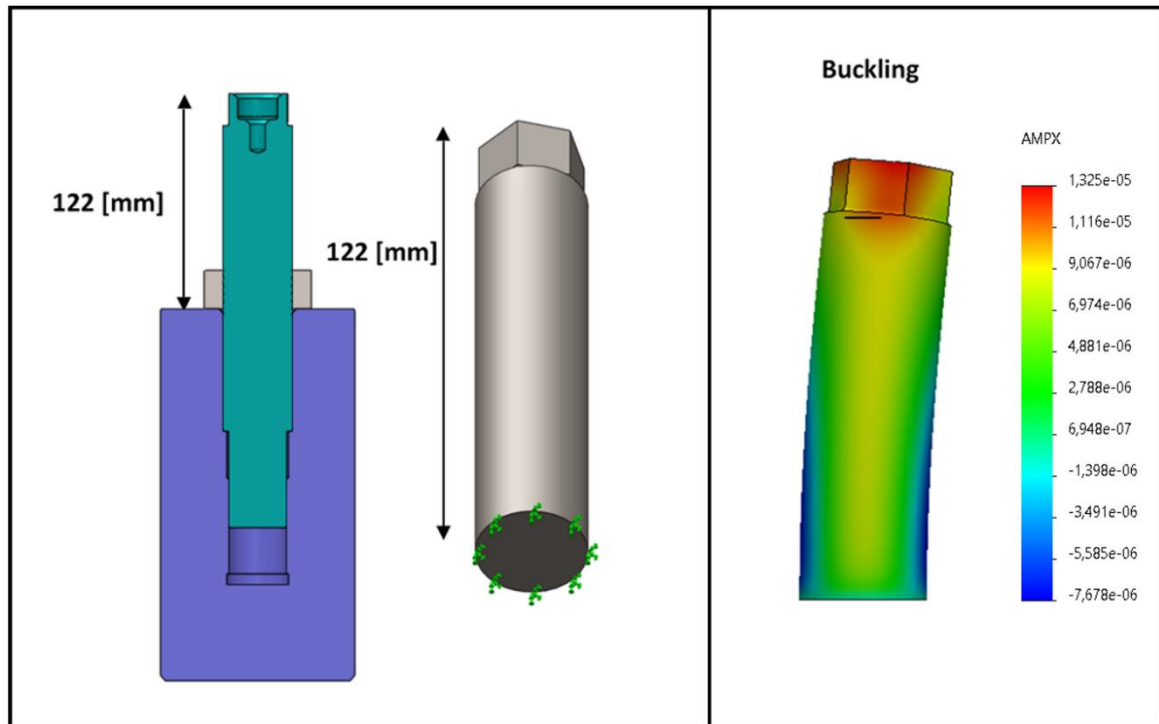


Figure 59- Buckling simulation of the punch holder.

Table 13- Results obtain from buckling simulation of the punch holder.

Punch holder	
Load factor	32

In buckling mode simulation, the value to extract is the Load Factor:

$$Load\ factor = \frac{estimated\ critical\ load}{applied\ load} \quad (80)$$

Therefore, the higher the value of this parameter, the more safety our component presents and the closer to the unit, the closer to the critical case.

As it is possible to see, this component is more demanding than the previous one, thus justifying the selection of a superior quality steel.

3.6.4. Die holder

The die holder is the component responsible for supporting and centering the dies.

The centering of the dies is achieved through an adjustment H7/g6, and the support of the dies is obtained through a thread [30]. However, this will be a loose thread so that all force is applied to the contact surface just before the thread and not to the thread.

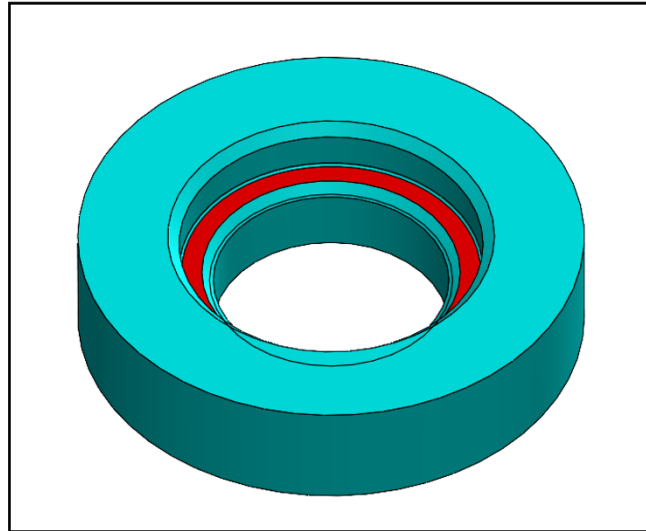


Figure 60- Schematic representation of the die holder.

Therefore, for this component it is only necessary to check the compressive forces exerted on the red area, Figure 60. The verification of the compression exerted by the die on the die holder can be calculated using the Expression 25 [26].

However, it was also decided to carry out simulations using the SOLIDWORKS software, in order to be able to identify stress concentrations more easily. The software was used in Static mode, a force of 120 kN was considered and the mesh was refined in the areas prone to stress concentrations, Figure 61. For this component, the maximum force applied corresponds to the maximum force applied in one of the dies, this value being in the order of 60 kN, however for this component a safety factor of 2 was considered. This is due to this component can be used for other processes, and even to be able to be used in the assembly of the machine. Thus, with a coefficient in the order of 2, the force will be 120 kN, which exceeds the force limit of the machine. Thus, the functioning of this component is guaranteed for any process that is used in this machine.

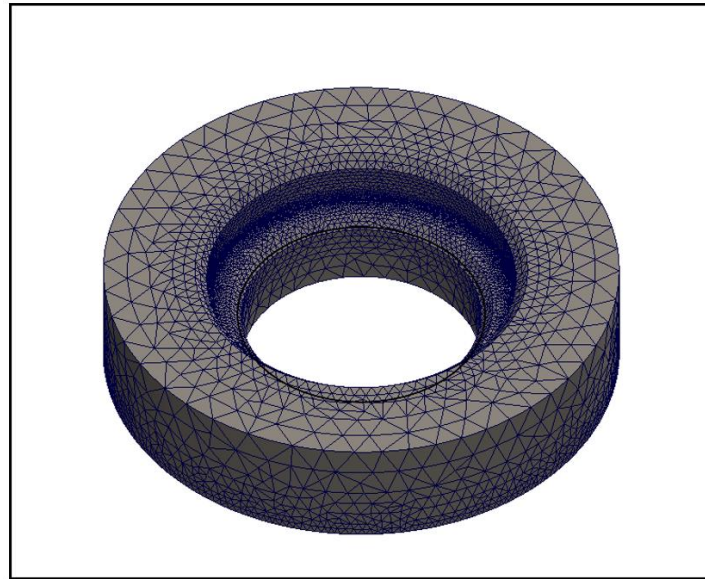


Figure 61- Mesh of the die holder.

In the simulations carried out, the force applied to the previously mentioned area was considered and the matrix was fixed by its lower part (surface that rests on the upper drive plate). The results obtained are presented in Figure 62 and Table 14.

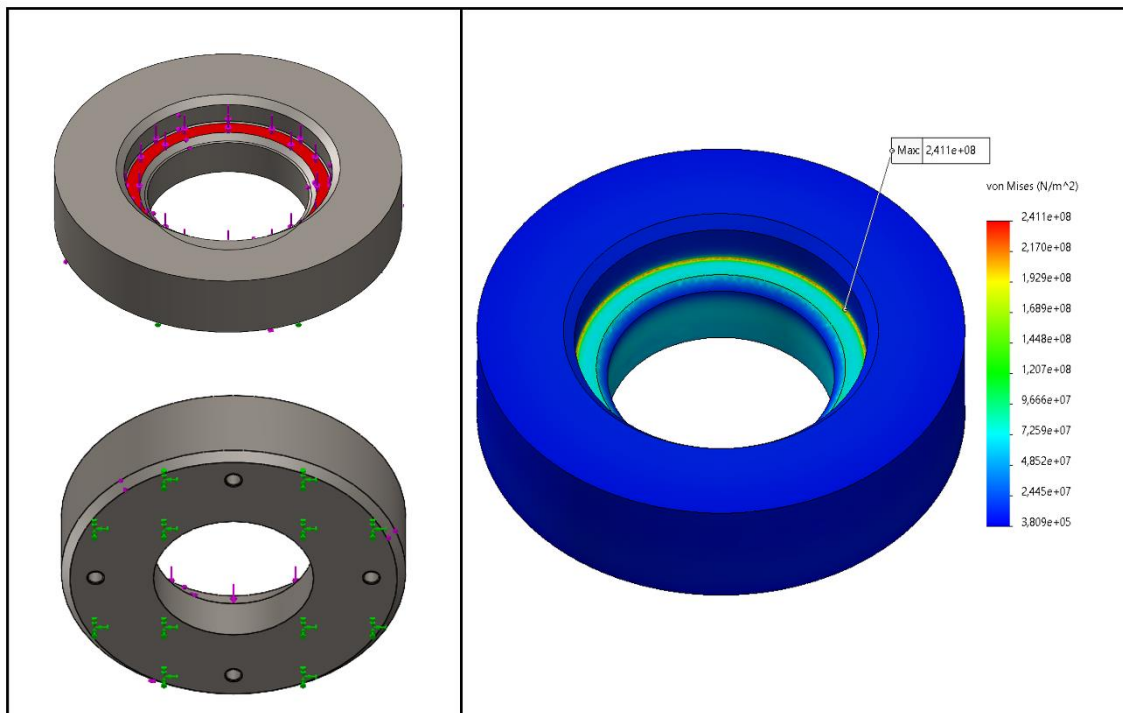


Figure 62-Fixtures, forces applied and von Mises stress of the die holder.

Table 14- Results obtained for the die holder.

Die holder	
Max von Mises stress	241.1 [MPa]

The die holder was manufactured in F10 from *Ramada Aços*, which has a yield stress in the order of 330 MPa [32]. As it turns out, this component is safe.

3.6.5. Middle plate

Regarding the passive tool components of the hole hemming tool, another component that deserved special attention was the middle plate, Figure 63, as this component is responsible for assembling the dies, guide pillars, keys and springs, it is one of the components that guarantees the alignment and centering between the punches and the dies and is a component that exhibits displacement during the process. So, any misalignment or deformation in this component can cause misalignment between the punches and the dies, poorly distributed forces or even jamming of the tool's operation.

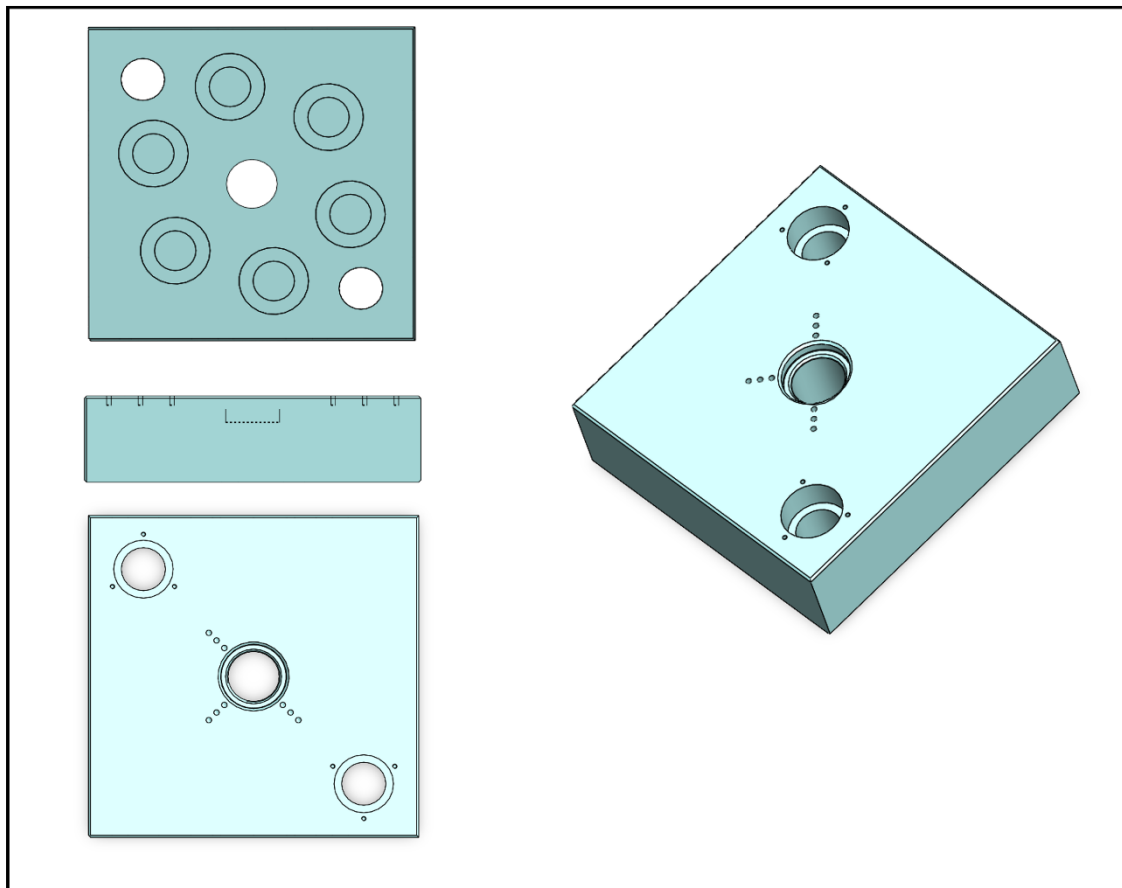


Figure 63- Schematic representation of the middle plate.

To avoid these problems, special care was needed in the quotation, geometric tolerance, and dimensional tolerance of this component. It was also necessary to carry out several simulations using the SOLIDWORKS software to be able to evaluate the deformations and stresses existing in this component in a critical case of operation.

Starting with the centering of the dies, this was guaranteed, as in the previous cases, through an H7/g6 adjustment, Figure 64, in order to obtain precision guidance [30]. As for the method of fixing these, the same procedure was also followed, namely the use of a loose thread, which guarantees the fixation of the component, but allows the entire force to be exerted on the contact surface immediately before.

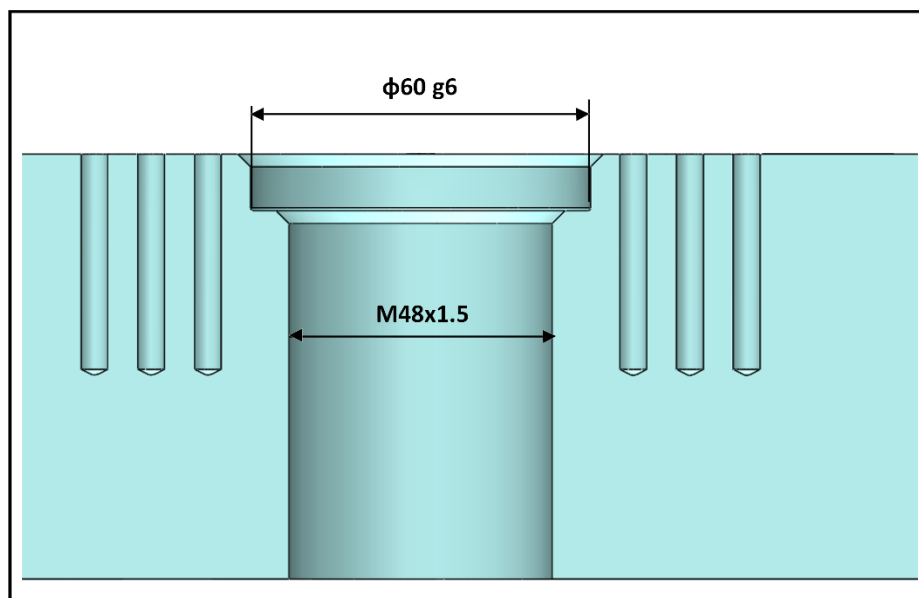


Figure 64- Dimensions of the adjustment between middle plate and dies.

Therefore, for this component it is necessary to check the compressive forces exerted on this area. The verification of the compression exerted by the die on the middle plate can be calculated using the Expression 25 [26].

However, it was decided to carry out simulations using the SOLIDWORKS software, in order to be able to identify stress concentrations more easily.

As for the assembly of the guide pillars in this component, they must be fixed, after the assembly of these there must be no relative movement between these two components. This fixation, as seen before, is achieved through the use of keys, the force of springs and plates. As for the adjustment of the guide pillars with the holes in the middle plate, this was achieved through a G6/h7 adjustment, Figure 65, which allows us to accurately guide [30].

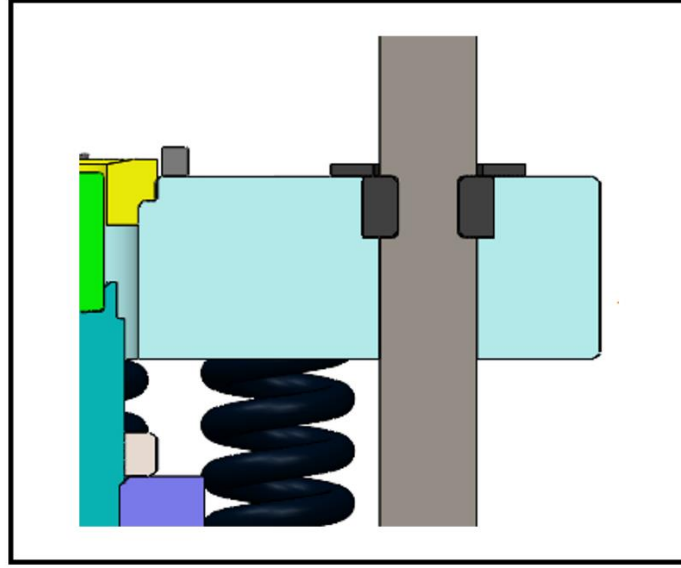


Figure 65- Schematic representation of the assembly of guide pillars and keys in the middle plate.

In this component, as in the upper drive plate and lower plate, the forces applied during the process provide buckling. To guarantee alignment and centering, it was also necessary to ensure that the maximum displacement of the guide pillar holes was less than 5 μm . Due to the distribution of the springs, we know that the maximum displacement due to bending in the holes of the guide pillars will be in the diagonal direction.

The admissible value for the displacement of the holes of the guide pillars, δ_{adm} , must be less than the value of the minimum clearance over two. The minimum clearance is obtained by the following equation [31]:

$$F_{min} = D_{min} - d_{max} \quad (81)$$

Where D_{min} is the value of the minimum diameter of the hole of middle plate and d_{max} is the value of the maximum diameter of the guide pillar. The guide pillar, as previously mentioned, has a diameter of $\varnothing 40\text{h}7$, which indicates [31]:

$$e_s = 0 \mu\text{m} \quad (82)$$

$$e_i = -25 \mu\text{m} \quad (83)$$

Where e_i is the lower deviation value and e_s is the upper deviation value. So, there is a maximum diameter for our guide pillar of:

$$d_{max} = 40.000 \text{ mm} \quad (84)$$

As for the holes in the guide pillars, they have a diameter of $\varnothing 40G6$, which indicates [31]:

$$E_s = 25 \mu m \quad (85)$$

$$E_i = 9 \mu m \quad (86)$$

So, the minimum diameter for the hole of the guide pillar is:

$$D_{min} = 40.009 mm \quad (87)$$

Then having a minimum clearance of:

$$F_{min} = 0.009 mm \quad (88)$$

And the admissible value for the displacement of the holes of the guide pillars is:

$$\delta_{adm} = \frac{F_{min}}{2} = 4.5 \mu m \quad (89)$$

Thus, for this component it is necessary to verify three aspects, namely, the compression force exerted by the middle plate against the keys, the compression force exerted by the dies in the middle and the displacement of the holes of the guide pillars.

Starting with the verification of the compression force exerted by the middle plate against the Keys, the SOLIDWORKS software was used in Static Simulation mode. The component was manufactured in material F10 (ISO 1045) from *Ramada Aços*, so the properties of this material were assigned to the simulations. It was considered a force of 60 kN applied in the positions of the springs, this force corresponds to the maximum force exerted in the preload. As for fixing the middle plate, it was fixed by the surface where the keys sit.

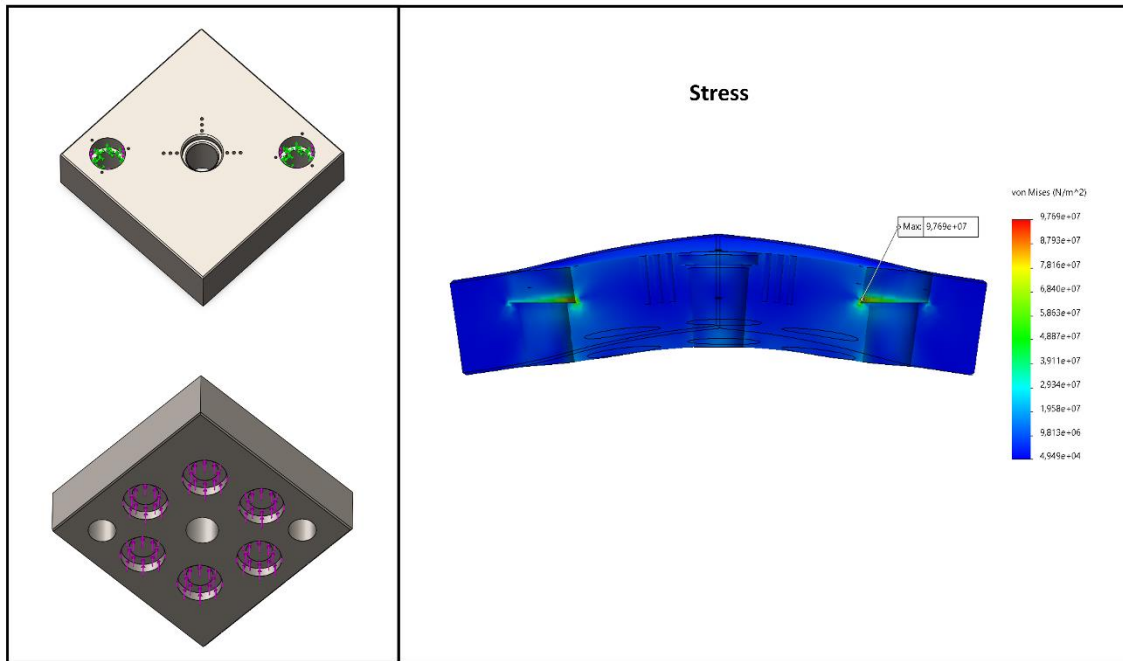


Figure 66-Fixtures, forces applied and von Mises stress of the middle plate.

Table 15- Results obtained from simulations regarding the keys assembly.

Middle plate	
Max von Mises stress	97.6 [MPa]

Through software simulations, a value of 97.6 MPa was obtained for the maximum von Mises stress, as shown in Figure 66 and Table 15. As the middle plate material is an F10 steel, it has a yield strength of 330 MPa, so there is a safety factor in the order of 3.38 [32].

As for the verification of the compression forces exerted by the dies on the middle plate and the verification of the displacement of the holes of the guide pillars, to carry out these simulations, the SOLIDWORKS software was used in Static Simulation mode. The properties of this material were assigned to the simulations. Regarding the applied forces and the fixing method, this simulation was more demanding than the others we have seen so far. This is since this component, during the process, is not fixed, it is only supported by the springs and guided by the guide pillars.

To obtain results consistent with reality, simulations were carried out on the middle plate and guide pillars set. To be able to apply the forces on the middle plate without applying fixtures to this component, it was decided to apply the fixings to the guide pillars, which is more in line with reality and provided results consistent with the physics of the problem. As for the applied forces, the forces applied by the dies and by the springs in the middle plate were considered, each one presenting a magnitude of 70kN. In this case, the maximum value allowed by the springs was

considered, not presenting a safety factor as high as in the lower plate and upper drive cases, because in the other cases we had uncertainty about the value of the minimum clearance, so it was necessary to consider a greater safety factor to check the displacement of the holes of the guide pillars. However, for this case, it was possible to determine with certainty the value of the minimum clearance. So, a value closer to the actual force can be considered, which allows obtain the displacements of the holes of the guide pillars closer to reality and not being necessary over dimension the thickness of the middle plate.

The results obtained from SOLIDWORKS simulations are presented in Figure 67 and Table 16.

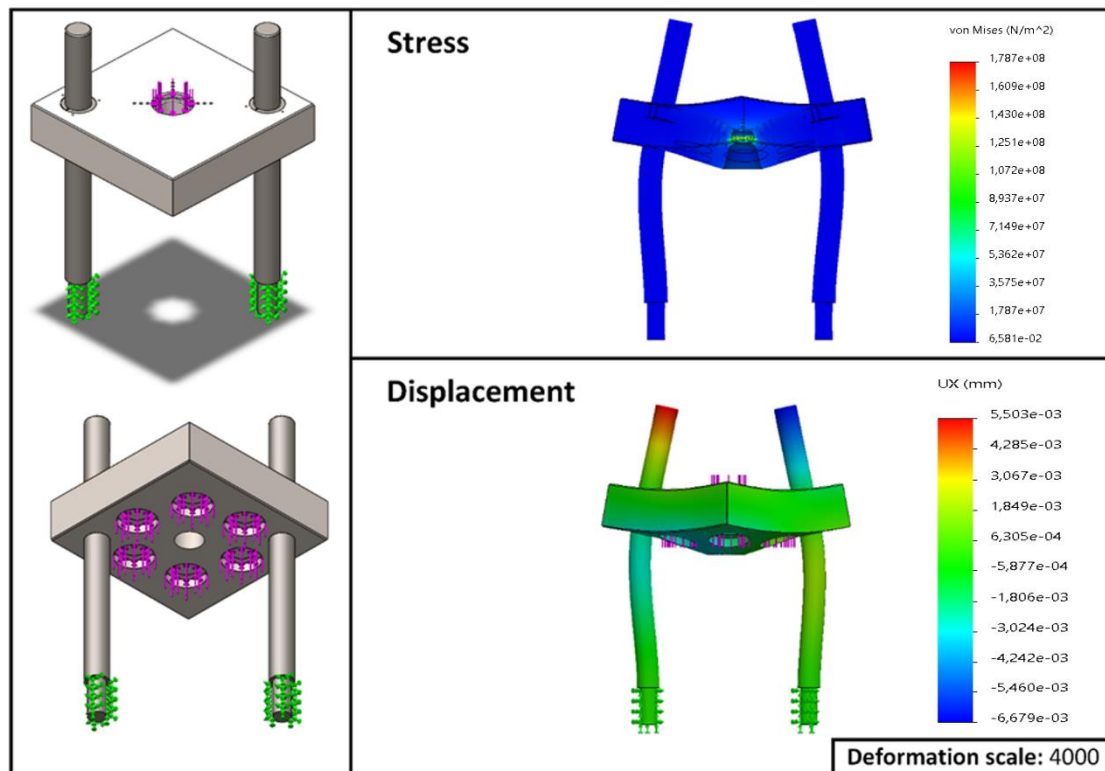


Figure 67- Fixtures, forces applied, von Mises stress and displacement of the assembly: middle plate and guide pillars.

Table 16- Results obtain for the assembly: guide pillars and middle plate.

Middle plate	
Max von Mises stress	170 [MPa]
Max displacement in holes	< 4.5 [μm]

To ensure that the displacement of the holes in the guide pillars did not exceed 4.5 μm , it was necessary to increase the thickness of the plate beyond the values observed in the initial catalogues. The final dimensions of this component are shown in Table 17:

Table 17- Final dimensions of the middle plate.

Middle plate	
Width	300 [mm]
Length	300 [mm]
Thickness	105 [mm]

The value obtained for the maximum von Mises stress was 170 MPa, knowing that the middle plate material has a yield strength of 330 Mpa, there is a safety factor of approximately 2 in compression due to the dies [32].

As previously considered the possibility of using 4 springs instead of 6 springs, it was considered necessary to carry out simulations to validate its functionality. As in the case of the lower plate, the same parameters of the simulation presented above were used, with the exception of the spring forces. For the spring force, first it was considered applied in the distribution shown in Figure 68, secondly, a total value for the spring force in the order of 50 kN was considered. The use of this value and not the 80 kN is justified by the same reasons presented for the previous simulation.

The results obtained from the simulations are presented in Figure 68 and Table 18.

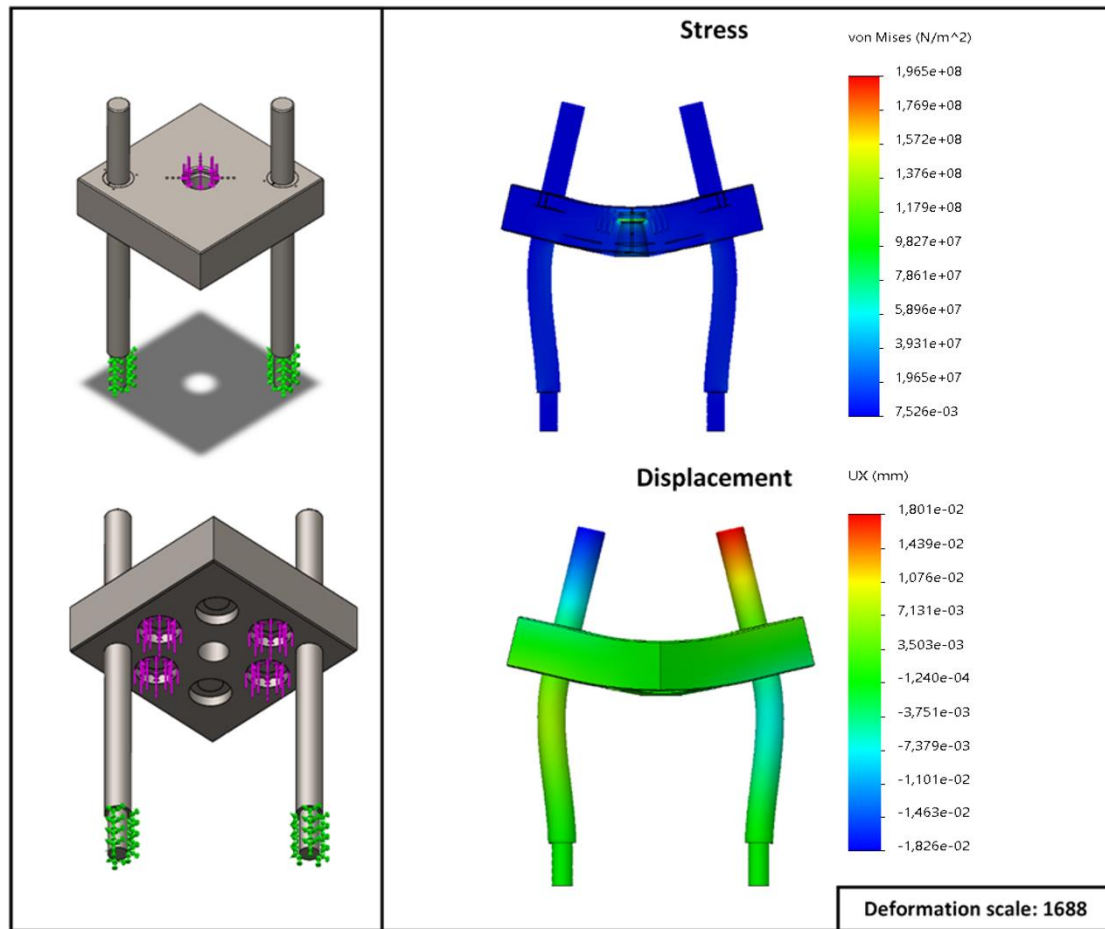


Figure 68-Fixtures, forces applied, von Mises stress and displacement of the assembly: middle plate and guide pillars. Considering the use of 4 springs.

Table 18-Results obtain for the assembly: guide pillars and middle plate. Considering the use of 4 springs.

Middle plate	
Max von Mises stress	190 [MPa]
Max displacement in holes	< 4.5 [μm]

Regarding the design of this component, it was necessary to pay special attention to the location of the wheelbases, so that the upper drive plate, middle plate and lower plate are perfectly aligned. Attention was also needed in the location of the cavities for the springs, so that this position corresponds with the position previously defined in the lower plate. In the design of this component, 9 holes were also added to accommodate the pins.

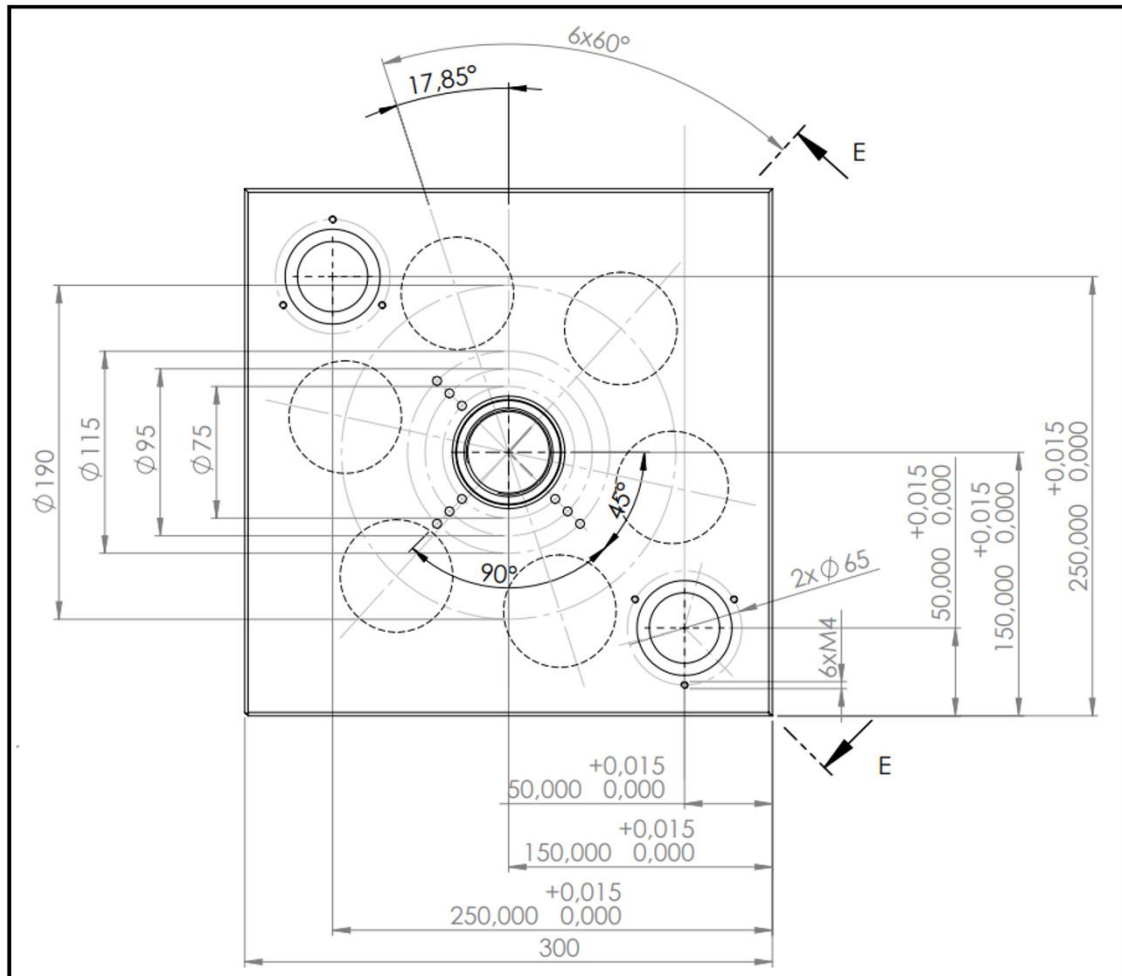


Figure 69- Upper view of the middle plate's technical drawing.

Three holes were placed with a diameter of 75, 95 and 115 mm centered in the center of the middle plate and distributed as shown in Figure 69. As seen earlier, the pins serve to facilitate the positioning of the plates for the hole hemming process. For the tool under development, there are pins with two different diameters, some with a diameter of $\varnothing 15$ mm and others with a diameter of 5 mm. In this way, it was possible to test sheets with the variety of dimensions shown in Figure 70 and Table 19:

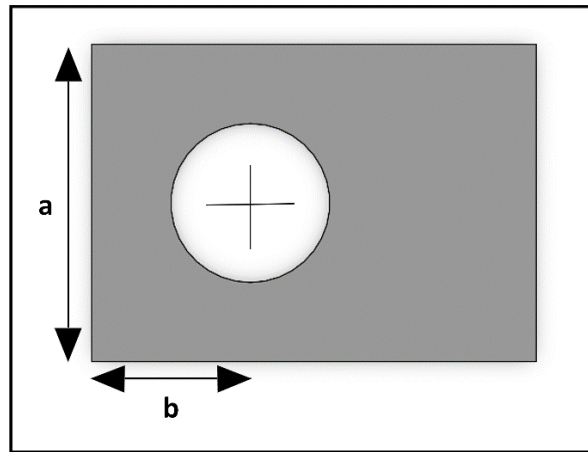


Figure 70- Schematic representation of the specimens.

Table 19- List of the available dimensions for the specimens.

a	b	pinos [a]	pino [b]
60 [mm]	30 [mm]	15 [mm]	15 [mm]
60 [mm]	35 [mm]	15 [mm]	5 [mm]
70 [mm]	30 [mm]	5 [mm]	15 [mm]
70 [mm]	35 [mm]	5 [mm]	5 [mm]
80 [mm]	40 [mm]	15 [mm]	15 [mm]
80 [mm]	45 [mm]	15 [mm]	5 [mm]
90 [mm]	40 [mm]	5 [mm]	15 [mm]
90 [mm]	45 [mm]	5 [mm]	5 [mm]
100 [mm]	50 [mm]	15 [mm]	15 [mm]
100 [mm]	55 [mm]	15 [mm]	5 [mm]
110 [mm]	50 [mm]	5 [mm]	15 [mm]
110 [mm]	55 [mm]	5 [mm]	5 [mm]

Thus, with a combination of 6 pins (3 with a diameter of 15 mm and 3 with a diameter of 5 mm) and 9 holes, sheets of 12 different dimensions can be centered.

To manufacture the passive tool components, after all the mechanical drawings were reviewed, the order of all the materials was needed. So, all the steel components were bought from *Ramada Aços*. The nut was bought from *Fabory*, Table 20.

Table 20- List of the normalized components used in the assembly of the passive tool components.

Qt.	Norm	Component
1	ISO4035 M36x4.0x18-04	Punch holder

The middle plate, die holder, punch holder and height adjuster were able to be manufactured by the workshop of FEUP.

3.7. Active tool components

This section describes the design and validation process for building the active tool components of the hole hemming tool. The active tool components of the hole hemming tool, Figure 71, were designed based on the following requirements:

- Create the desired deformations;
- Be easily interchanged;
- Provide guidance and centering;
- Withstand the applied forces.

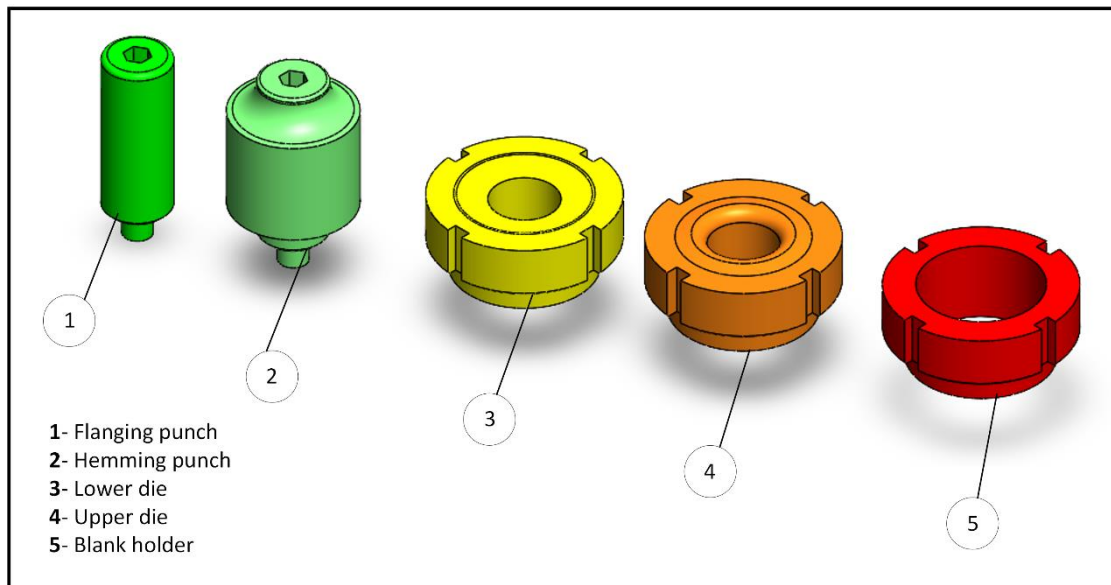


Figure 71- Schematic representation of the active tool components of the hole hemming tool.

Starting with the first requirement, creating the desired deformations, this was approached by José Pereira, who through several simulations in the ABAQUS software obtained the ideal geometries and dimensions of these components for the testing of the materials under study.

The following requirement, easy interchange of active components, is a requirement of great relevance to the process. As explained above, the hole hemming process is divided into two

stages, being necessary to change the active components from one stage to the other, it is therefore essential that this replacement is easy and fast. This requirement also opens the door to testing different materials that require different active components. The solution found to satisfy this requirement involves the use of a thread both in the dies and in the punches, thus making it easy to assemble and disassemble these components. However, the use of a thread does not guarantee centering or alignment of the components, and the components can fracture in the thread due to the high forces applied. To solve this problem, the thread used will be a loose thread, so the forces will be applied to the surface immediately above the thread and not to it. As for the alignment and centering of these components, this is guaranteed by the side surfaces, as shown in Figure 72, which will fit in the die holder and middle plate, in the case of dies, and in the punch holder, in the case of punches.

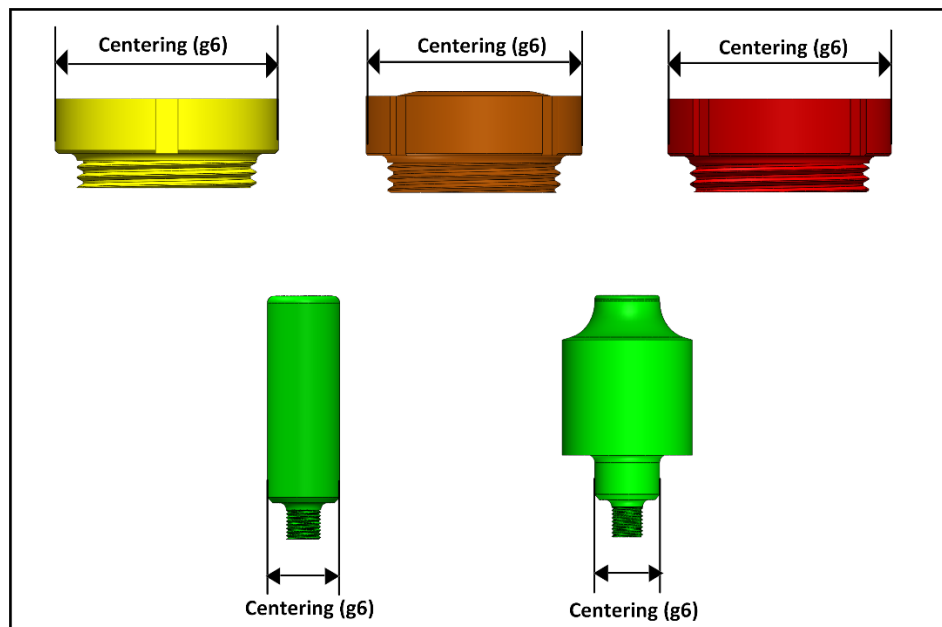


Figure 72- Adjustment used for the dies and punches.

3.7.1. Dies

Regarding the design of the dies, the dimensions that provide the contact between the die and the plate were defined by José. This dissertation is focused on the definition of the dimensions that provide the housing of the dies in the tool, namely, height of the dies, size and thread length, the design of a mechanism to easily change the dies and the selection of the material.

As explained above, the solution for fixing the dies involves the use of a loose thread, with the dies being guided by the lateral surface. For guidance through the lateral surface, it was considered that at least 8 mm was necessary. However, both in the die holder and in the middle

plate the housing location has a 2 mm chamfer. Therefore, the depth that the dies will be inserted into these components will be 10 mm, Figure 73.

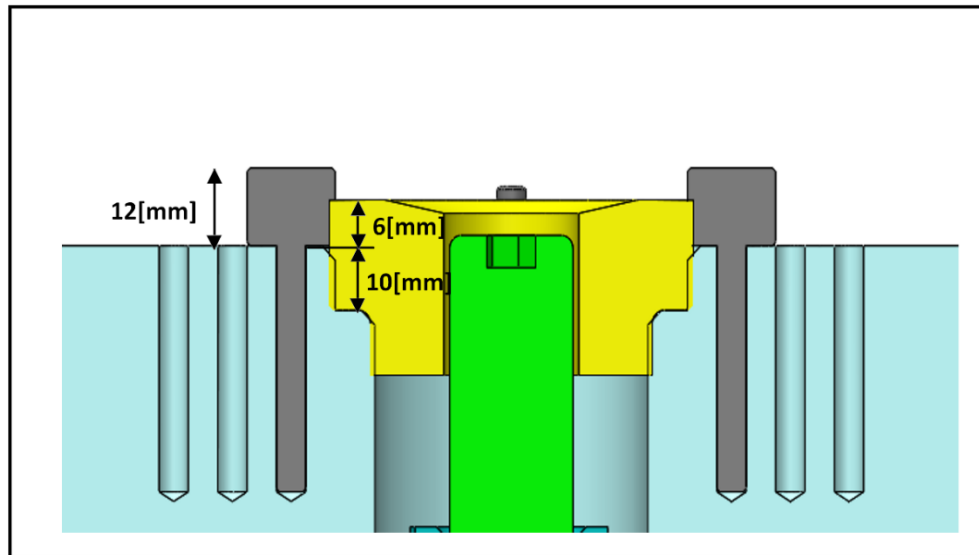


Figure 73- Schematic representation of the assembly of the dies in the middle plate.

The pins, previously presented, have a height of 12 mm, in order to be able to use a wider range of plate thicknesses, it was considered that the height at which the dies should come out of the die holder or middle plate, should be half the height of the pins. With this height of the dies, when they are closed (with no sheet between them) the pins come into contact with the die holder. Thus, when a plate is placed between the dies, it is ensured that there is no contact between the pins and the die holder, with no restriction on the minimum thickness of the plate. As for the maximum thickness of the sheet, in the case of using only one sheet, there is also no restriction. However, when two sheets are used, the sheet at the bottom must be less than 6 mm thick, so that the sheet at the top can also be centered by the pins.

Although the thread used to assemble the dies is a loose thread, it was considered necessary to implement a mechanism that allows us, if necessary, to use a tool to thread or unscrew the dies. However, this mechanism must not interfere with the contact areas between the dies and the sheet. The solution used was based on slotted nuts, this solution consists of opening grooves on the lateral surface of the dies, allowing the use of a hook wrench to thread or unscrew the dies. The dimensions used for the slots were taken from the dimensions of a slotted nut from the company *Fag*, which has an external diameter identical to the external diameter of the dies [38]. These dimensions are presented in Figure 74 and Table 21.

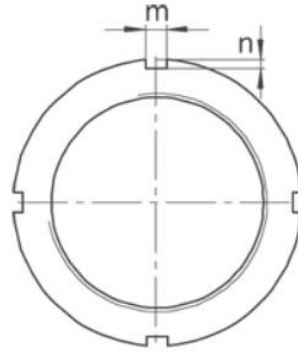


Figure 74- Schematic representation of slotted nut FAG KM8 [38].

Table 21- Dimensions of the slotted nut FAG KM8 [38].

m	n
6 [mm]	2.5 [mm]

The choice of the thread size of the dies was conditioned by two aspects, the first was the inner diameter of the dies, since the thread diameter must be greater than this diameter, in order to have enough material to create the thread. The second was the surface area that is subjected to compression, since the larger the diameter of the thread, the smaller the value of this area, which causes an increase in tensions due to compression. Bearing in mind these two aspects, the thread with the smallest possible diameter was selected, allowing the existence of sufficient material for the creation of the same. The value of the largest inner diameter is 43 mm, corresponding to the inner diameter of the blank holder. The M48x1.5 thread was then selected. According to *Moraes et.al.* [31], this thread is known to have a smaller diameter of 46,376 mm, so there is enough material to create the thread.

As for the length of the thread, since it is loose and all the force is exerted on the surface immediately before the thread, no verification is necessary for this value. However, it was considered that there should be at least 5 turns of thread, in order to have a good support of the matrices. Since the thread pitch is 1.5 mm, approximately 8 mm is needed to obtain 5 turns.

For the dies, simulations using the software SOLIDWORKS2021 were performed only with the aim of studying the effect of compressive forces on component stresses. In carrying out these simulations, the previously explained fixing method was considered, dies fixed on the surface immediately after the thread and not on the thread. As for the applied forces, the same magnitude value was considered for all, 120 kN, a safety factor of approximately 2. These forces were applied to the contact surfaces. Care was also taken to refine the mesh for the zones conducive to stress concentration.

The results obtained through the simulations are presented in Figures 75, 76, 77 and Table 22:

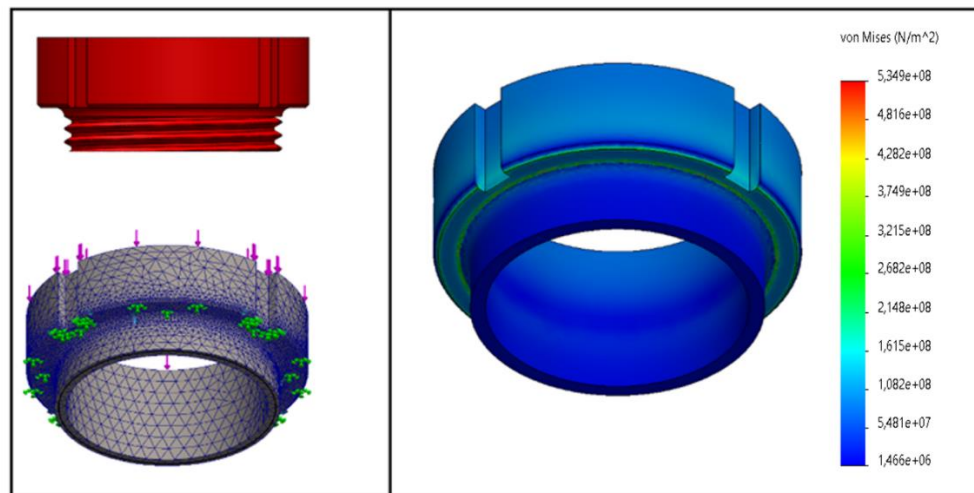


Figure 75-SOLIDWORKS static simulation of the blank holder.

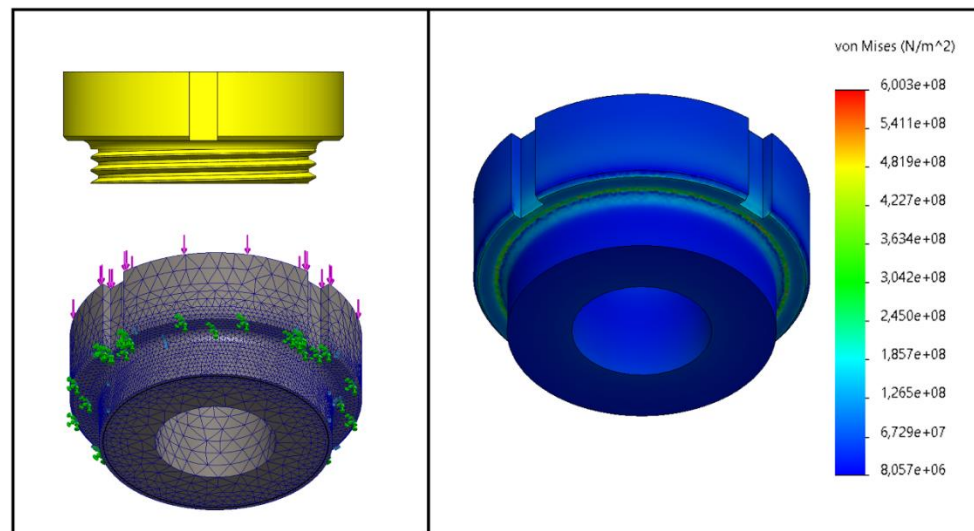


Figure 76-SOLIDWORKS static simulation of the lower die.

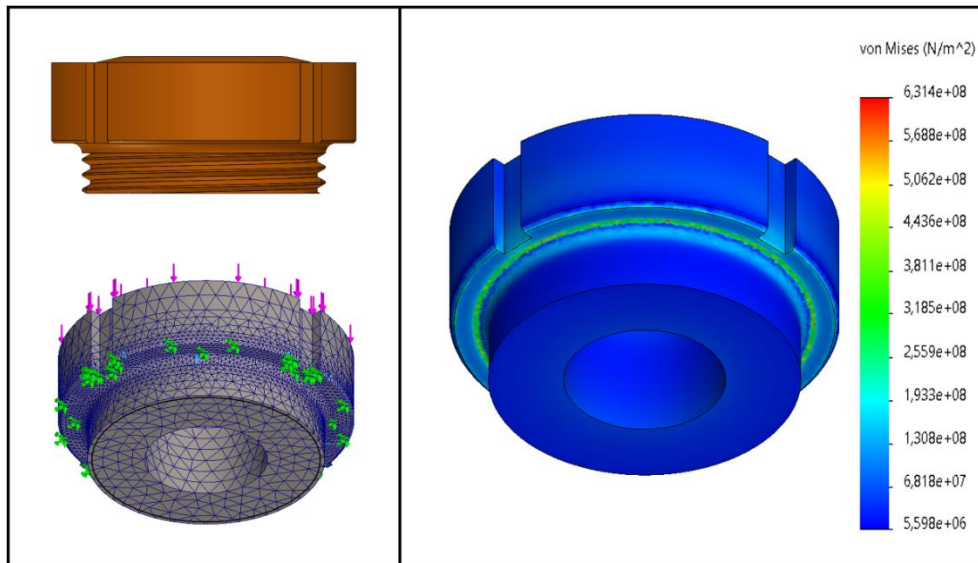


Figure 77-SOLIDWORKS static simulation of the upper die.

Table 22- Results obtained from the SOLIDWORKS simulations of the dies.

Component	Load applied	Max von mises stress
Blank holder	120 [kN]	530 [MPa]
Lower die	120 [kN]	600 [MPa]
Upper die	120 [kN]	650 [MPa]

Due to the high tensions existing in these materials, a more careful selection of the material to be used was necessary. We chose to use the material C265, a material from *Ramada Aços* recommended for the following applications: dies and punches; blades for guillotines; razors; woodworking tools. This material is supplied with a minimum hardness of 255 HB, which corresponds to a tensile strength of 850 MPa, due to the existence of some uncertainty in the yield stress for this material, it was considered necessary to carry out a heat treatment, tempering, in order to obtain adequate hardness for the process under study [32], [37] [39].

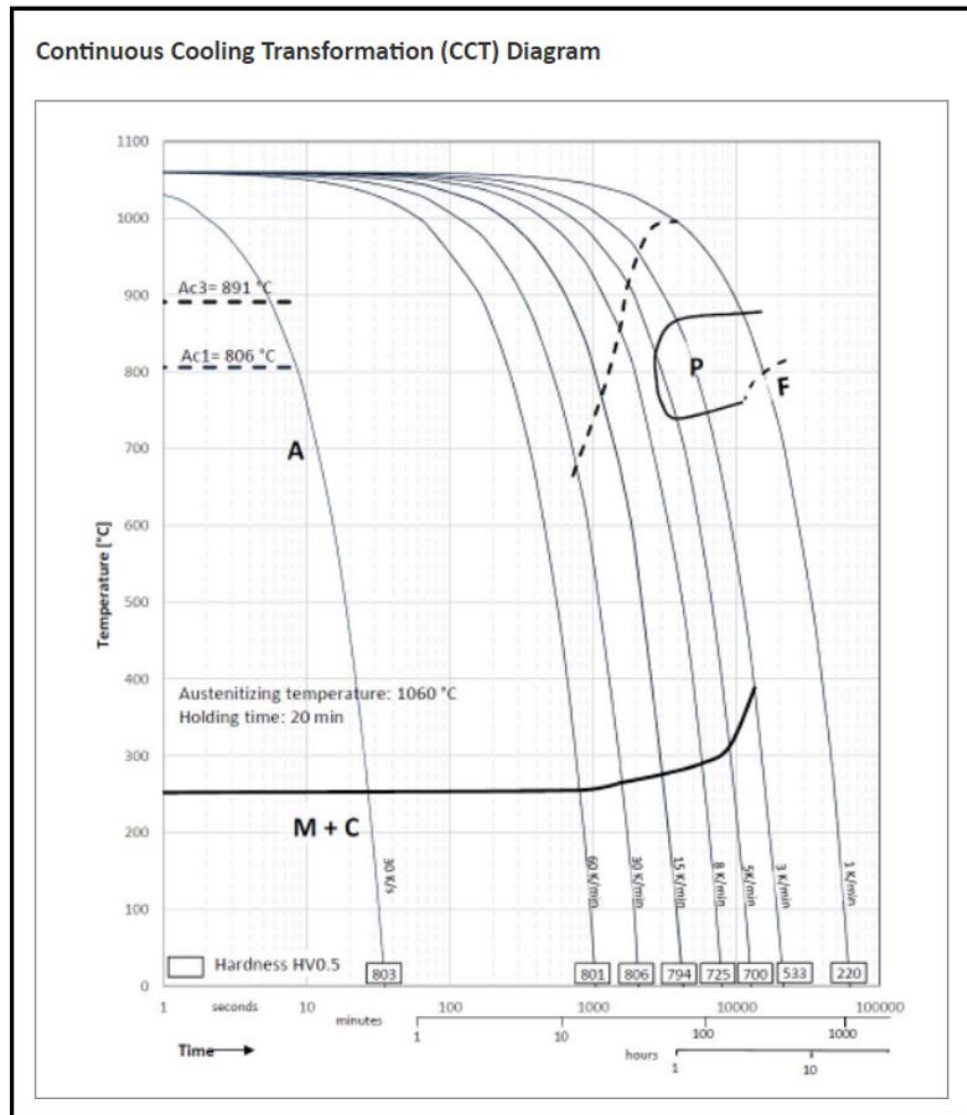


Figure 78- Continuous cooling transformation diagram of C265 [39].

Figure 78 shows the continuous cooling transformation diagram of C265. After analyzing the diagram, it was considered adequate to use a hardness in the order of 43-46 HRC, which provides a tensile strength of approximately 1350-1500 MPa [36] [39].

The thermal treatment of these components was carried out by *Ramada Aços*, where it was recommended to use an overlay of 2 mm and the use of radius in the order of 2-3 mm. The treatment was carried out in a vacuum oven, which guarantees a smaller variation of the initial dimensions.

After heat treatment it was necessary to rectify the components. This correction was performed by the workshop of FEUP. However, during the rectification process one of the dies, blank holder, fractured, Figure 79.



Figure 79- Fracture of the blank holder.

To circumvent this problem, it was decided to use a different material for the dies, which does not require a heat treatment, in order to make their machining easier. Considering the von Mises stresses presented in Table 22, the material PM300 from *Ramada Aços* was selected. As previously seen, this material has a tensile strength of approximately 1000 MPa and a yield strength in the order of 800 MPa [32], [37]. These components were able to be manufactured by *FEUP mechanical engineering department workshop*.

In the end, were obtained a total of 5 matrices, as shown in Table 23 and Figure 80.

Table 23- List of manufactured dies.

Component	Material
Blank holder	PM300
Upper die	PM300
Lower die	PM300
Upper die	C265
Lower die	C265

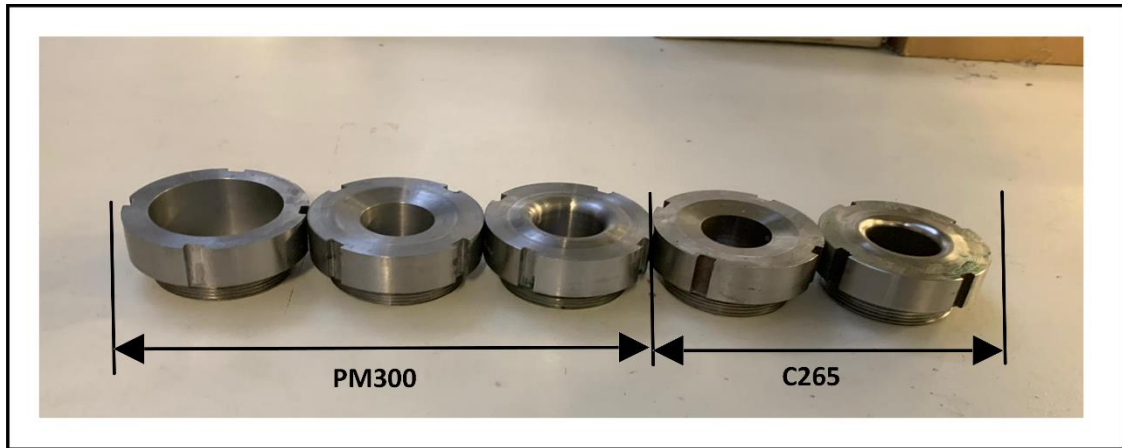


Figure 80- Manufactured dies.

3.7.2. Punches

As for the design of the punches, the situation is the same as for the design of the dies. The dimensions that provide the contact between the punch and the sheet were defined by José. This dissertation focuses on the definition of the dimensions that provide the housing of the punch in the tool, namely, height of the punches, size and thread length, the design of a mechanism to easily change the punch and the selection of the material.

As for the gap between the dies and the punches, this was also defined by José. For the first stage, hole flanging, this clearance is 2 mm, while for the second stage, hemming, it is only 1 mm. The proper clearance to apply depends upon the material, its degree of hardness, and its thickness [29].

To guarantee the clearance defined by José, special attention was paid to the dimensioning and tolerances of the technical drawings of the components. Regarding the dimensioning of the lower plate, middle plate and upper plate, the dimensions and tolerances used were the same.

For these components, the position of the axes was made using Cartesian coordinates with a tolerance of 15 μm in both directions, as shown in Figure 69. This results in a maximum clearance from the component's center axis position of:

$$\sqrt{2} * 15 = 21.21 \mu\text{m} \quad (90)$$

As for the perpendicularity tolerance used for these components, its value was 10 μm , which guarantees that the axis of these components is inside a cylinder with a diameter of 10 μm and the axis perpendicular to the reference plane used [31].

For the die holder, punch holder, punches and dies, coaxiality tolerances in the order of 10 μm and perpendicularity tolerances in the order of 10 μm were considered.

To obtain the maximum clearance between the axis of the dies and punches, its necessary to start by determining the maximum clearance between the punch axis housed in the lower plate and the die axis housed in the middle plate. First, the maximum clearance of the position of the central axis of the plates was added, with the coaxiality tolerance of the respective component that is housed in them, and the sum of the coaxiality tolerances of these components was divided by two. For the lower plate the component to house will be the height adjuster and for the middle plate it will be an die.

$$21.21 + 5 + 5 = 31.21 \mu m \quad (91)$$

On the lower plate, in addition to the height adjuster, it is still necessary to add the punch holder and the punch itself. Thus, a maximum clearance between the axis between the lower plate punch and the middle plate die is obtained:

$$31.21 + 5 + 5 = 41.21 \mu m \quad (92)$$

Then, the maximum clearance between the punch axis housed in the lower plate and the die axis housed in the upper drive plate was determined. Again, the maximum clearance of the position of the central axis of the plates was added, with the tolerance of coaxiality of the respective component that is housed in them. For the lower plate the component to house will be the height adjuster and for the upper drive plate it will be the die holder.

$$21.21 + 5 + 5 = 31.21 \mu m \quad (93)$$

As seen earlier, on the lower plate it is still necessary to house the height adjuster and the punch, while on the upper drive plate it is still necessary to house the die. Thus, a maximum clearance between the axis between the lower plate punch and the upper drive plate die is obtained:

$$31.21 + 5 + 5 + 5 = 46.21 \mu m \quad (94)$$

Finally, the maximum clearance between the die axis housed in the middle plate and the die axis housed in the upper drive plate was determined. Again, the maximum clearance of the position of the central axis of the plates was added, with the tolerance of coaxiality of the respective

component that is housed in them. For the middle plate, the component to house will be the die and for the upper drive plate it will be the die holder.

$$21.21 + 5 + 5 = 31.21 \mu m \quad (95)$$

On the upper drive plate, it is still necessary to house the die. Thus, a maximum clearance between the axes between the middle plate die and the upper drive plate die is obtained:

$$31.21 + 5 = 36.21 \mu m \quad (96)$$

It can be seen that the greatest clearance between axis will be between the punch and the die housed in the upper drive plate.

Finally, for the dimensions of the inner diameter of the dies and for the dimensions of the outer diameter of the punches, only the general tolerance ISO2768-mH was considered. In the worst possible case, the inner diameter of the dies will be -0.3 mm and the outer diameter of the punches will be +0.3 mm [31]. Therefore, the clearance initially of 1.5 mm becomes 1.2 mm. Considering now the value previously determined for the maximum clearance of the axis, 46.21 μm , it's possible to determinate the minimum distance between the die and the punch:

$$1.2 - 0.04621 = 1.15379 mm \quad (97)$$

As for the dimensioning of the punch height, it starts by defining the punch length that would be guided by the lateral surface of the punch holder. As in the case of the dies, it was considered that at least 8 mm was necessary. However, in the punch holder the housing location has a 2 mm chamfer. Therefore, the depth that the dies will be inserted into these components will be 10 mm.

The punch height dimensioning was performed simultaneously with the punch holder height dimensioning, as these components must guarantee that the punch is positioned at a specific distance from the top of the dies housed in the middle plate. Another aspect to bear in mind in this dimensioning is the height of the punch being sufficient to allow its penetration into the dies without the contact of the punch holder with the die or the nut with the middle plate.

To carry out this dimensioning, the case of maximum preload was considered, by guaranteeing the aforementioned aspects for this position, also guaranteeing the other positions due to the possibility of adjusting the position of the punch.

According to the requirements provided by José, in the first stage of the process, hole flanging, the top of the flanging punch must be approximately 2 mm from the edge of the die housed in the middle plate. For the second stage, this value should be 3 mm. In order to avoid having to change the position of the punch holder from one phase to the other, the punches were dimensioned taking into account this small difference, Figure 81.

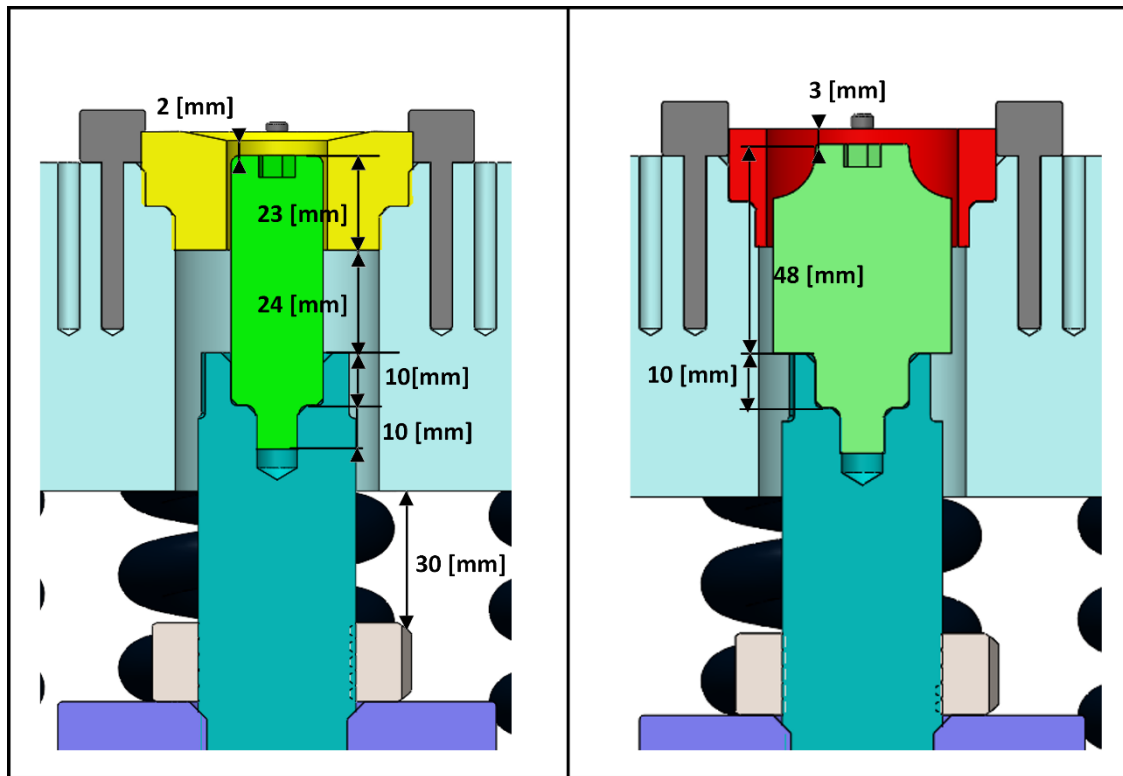


Figure 81- Schematic representation of the assembly of the flanging punch and hemming punch on the hemming tool.

Another of the requirements of the process was a penetration length in the order of 12 mm, so it was necessary to guarantee a distance greater than this value between the top of the punch holder and the bottom of the die housed in the middle plate, and between the top of the nut used to adjust the position of the punch and the bottom of the middle plate.

As in the case of dies, although the thread used to assemble the punches is a loose thread, it was considered necessary to implement a mechanism that allows, if necessary, to use a tool to thread or unscrew the punches. However, this mechanism must not interfere with the contact areas between the punches and the sheet. The solution found was the use of a hexagonal hollow at the top of the punches, since only the edges of the punches come into contact with the plates. For the dimensions of the hexagonal hollow, *Morais et.al.* [31] was consulted, where, by consulting the ISO23429- Hexagonal Hollow screws, are obtained the following dimensions, Figure 82:

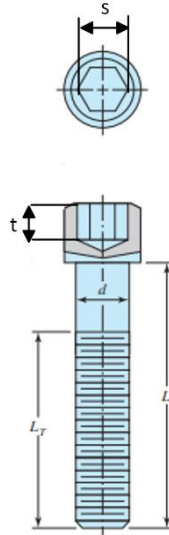


Figure 82- Schematic representation of hexagonal hollow screws [26].

$$S = 8 \text{ mm} \quad (98)$$

$$t = 5 \text{ mm} \quad (99)$$

As for the choice of punch thread size, in this case, it was only necessary to consider the surface area that is subjected to compression, since the larger the diameter of the thread, the smaller the value of this area, which causes an increase in tensions due to compression. An M10x1.0 thread was chosen.

As for the length of the thread, since it is loose and all the force is exerted on the surface immediately before the thread, no verification is necessary for this value. However, it was considered that there should be equal to the length used for the dies.

The punch design process was accompanied by two verifications, the verification of elastic stability (buckling) and the verification of compressive forces. As seen earlier for the punch holder, the verification of the compression strength can be calculated using the Expression 25 [26]. And the case of buckling can be studied using the Expressions 73-75 [5]:

However, it was considered more appropriate to carry out simulations using the SOLIDWORKS2021 software, in order to study the effect of compressive forces on component stresses, as well as their effect on buckling. In carrying out these simulations, the previously explained fixing method was considered, punches were fixed on the surface immediately before the thread and not on the thread. As for the applied forces, different magnitude values were considered for the punches. The values used for these components are based on the values

presented at the beginning of this paper, with the application of a safety factor of 2. These forces were applied to the contact surfaces and care was taken to refine the mesh to the areas conducive to stress concentration, as in the previous cases, Figure 83 and 84.

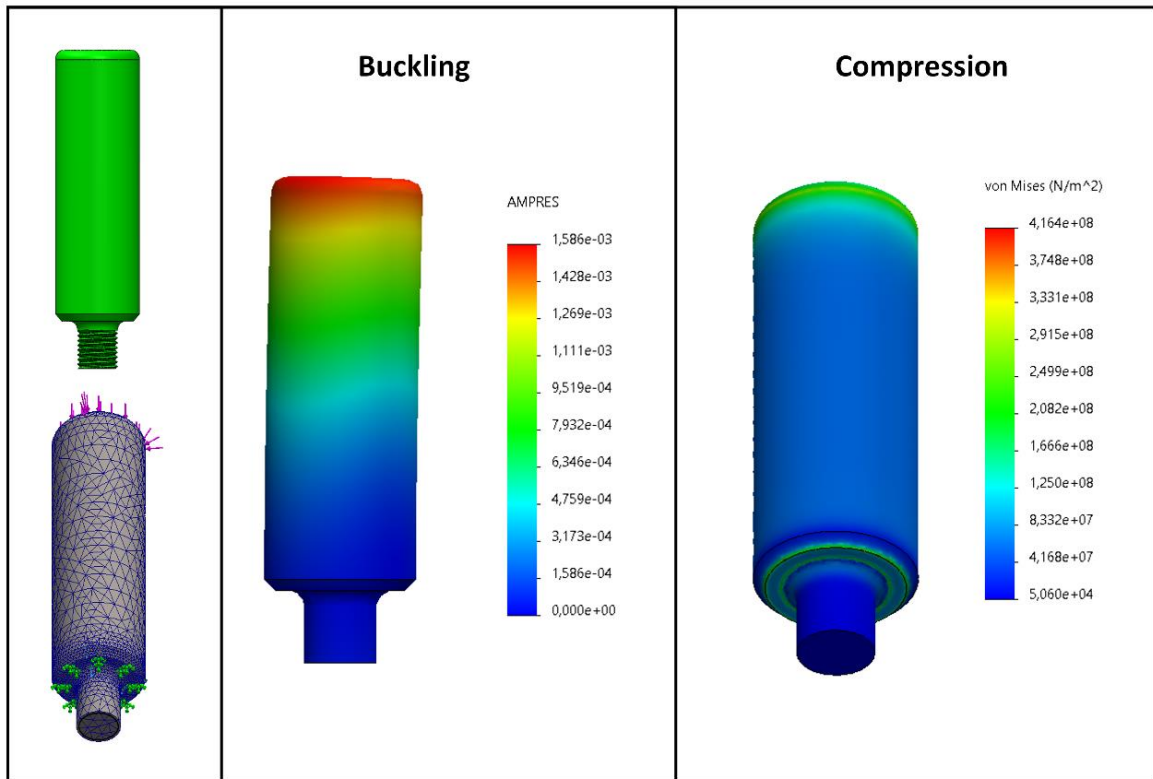


Figure 83- Buckling and static simulations of the flanging punch.

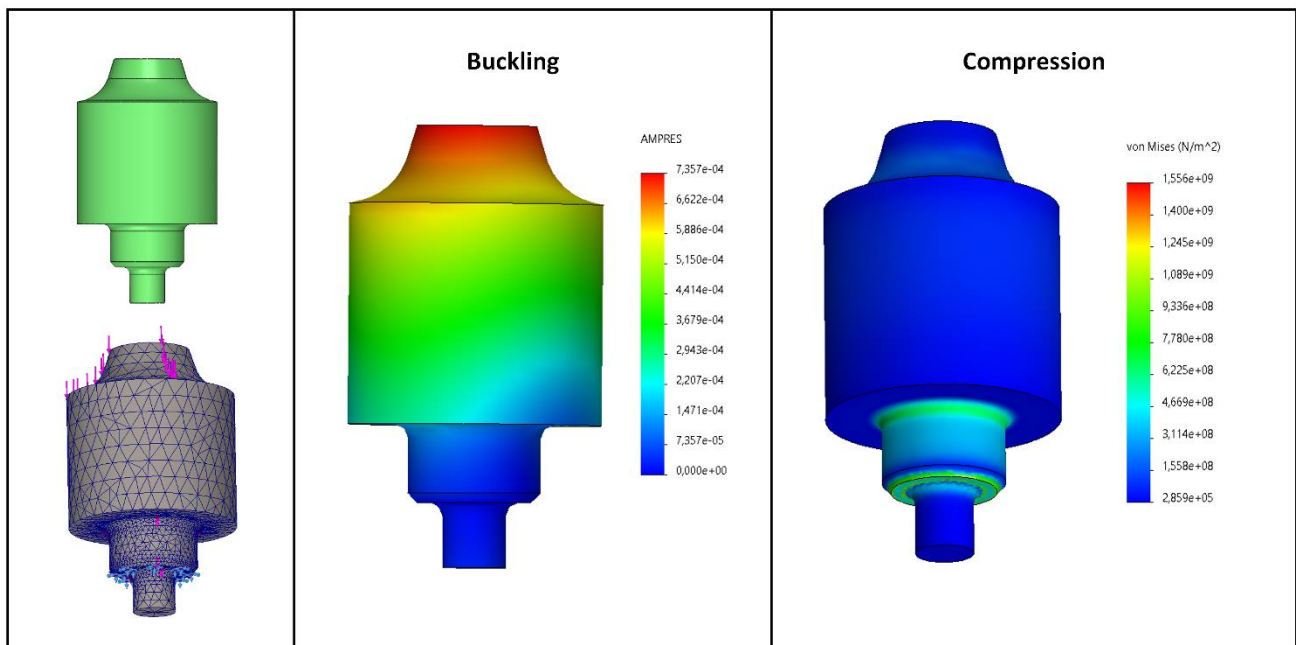


Figure 84- Buckling and static simulations of the hemming punch.

The table 24 presents a summary table of the simulations performed.

Table 24- Results obtained from the buckling and static simulations performed on the punches.

Component	Load applied	Max von mises stress	Buckling (load factor)
Flanging punch	20 [kN]	450 [MPa]	70
Hemming punch	120 [kN]	1600 [MPa]	25

Due to the high tensions existing in these materials, a more careful selection of the material to be used was necessary. The material C265 was chosen that was from *Ramada Aços* recommended for the following applications dies and punches; blades for guillotines; razors; woodworking tools. This material is supplied with a minimum hardness of 255 HB, which corresponds to a tensile strength of 850 MPa [32], [36]. For hemming punch, it is necessary to carry out a heat treatment, tempering, in order to obtain adequate hardness for the process under study. For the hemming punch, the hardness provided by the supplier is sufficient, however, as it is necessary to carry out heat treatment for the hemming punch, it was also decided to carry out treatment on this component too.

Table 25 shows the hardness used for each of the components.

Table 25- Hardness and equivalent tensile stress of the punches.

Component	Hardness [HV]	Tensile stress [MPa]
Flanging punch	430 - 460	1385 – 1485
Hemming punch	600 – 650	1995 – 2180

For the flanging punch it is guaranteed that the yield strength of the material is not reached. In the case of hemming punch, through literature review it was possible to determine the yield strength of the material for the desired hardness.

Table 26- Mechanical properties of C265 [17]..

AISI	UHB-name	Hardness HCR	Elasticity modulus (GPa)	Compression test $\sigma_{0.2}$ (MPa)	Bending test $\sigma_{0.1}$ (MPa)
A2	Rigour	62	206	2150	2350
		60		1950	2550
		58		1650	2450
D2	Sverker 21	62	209	2150	2350
		60		1950	
		58		1650	

The material in question AISI D2 or C265 for a hardness of 58 HRC or 650 HV has a yield strength of 1650 MPa, as shown in Table 26 [17].

The thermal treatment of these components was carried out by Ramada Aços, where it was recommended to use an overlay of 2 mm and the use of radius in the order of 2-3 mm. The treatment was carried out in a vacuum oven, which guarantees a smaller variation of the initial dimensions.

After heat treatment it was necessary to rectify the components. This correction was performed in the workshop of FEUP.

3.8. Tool fabrication

In this section, the fabrication of the hole hemming tool is summarized, presenting the manufactured components, their material, manufacturing process and the corresponding mechanical drawing number. This section makes it easier to manufacture tool components for future works.

Table 27 presents the material, manufacturing process, company that performs the process and the mechanical drawing's number of structural components of the hole hemming tool.

Table 27- Fabrication of structural components of the hole hemming tool.

Components	Material	Manufacturing process	Company	Mechanical drawing's number
UPPER DRIVE PLATE	F10	Machining	Machining CNC Unipessoal Lda	2022050402
LOWER PLATE	F10	Machining	Machining CNC Unipessoal Lda	20220504001
GUIDE PILARS	1.1213	Machining	Equinotec	20220601013
KEYS	F10	Machining	FEUP	20220517008
PLATES	F10	Machining	FEUP	20220524009
PINS	F10	Machining	FEUP	20220529010

Figure 85 shows the structural components after the fabrication process.

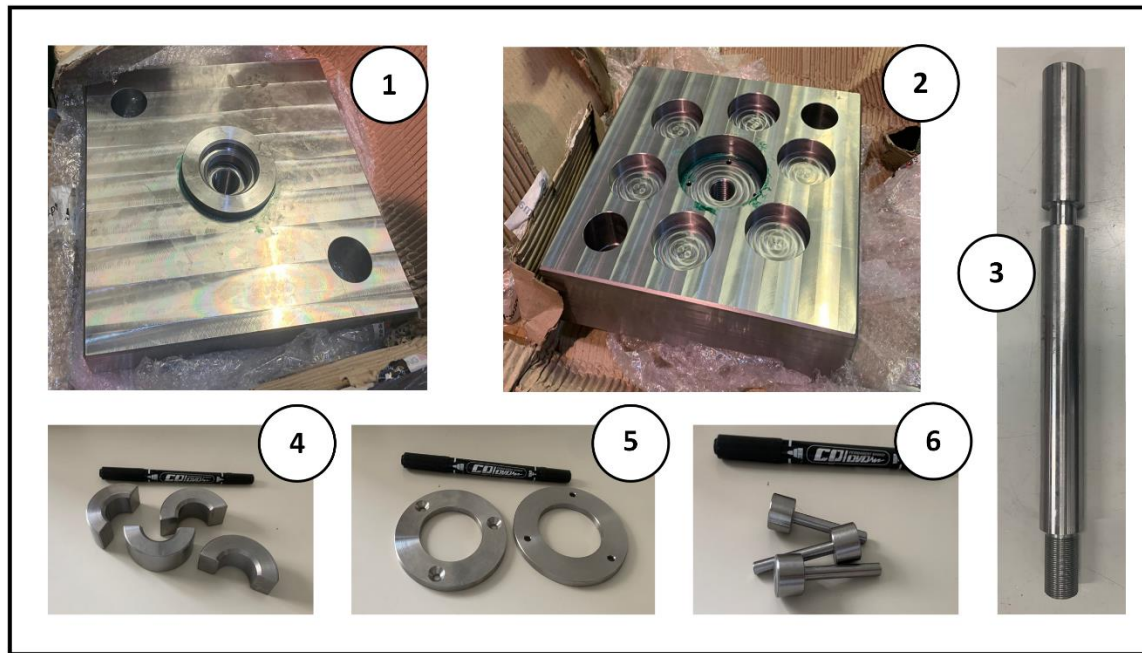


Figure 85- Structural components of the hole hemming tool.

Table 28 presents the material, manufacturing process, company that performs the process and the mechanical drawing's number of passive components of the hole hemming tool.

Table 28- Fabrication of passive components of the hole hemming tool.

Components	Material	Manufacturing process	Company	Mechanical drawing's number
HEIGHT ADJUSTER	F10	Machining	FEUP	20220522006
PUNCH HOLDER	PM300	Machining	FEUP	20220524007
DIE HOLDER	F10	Machining	FEUP	20220522005
MIDDLE PLATE	F10	Machining	FEUP	20220517004

Figure 86 shows the passive tool components after the fabrication process.

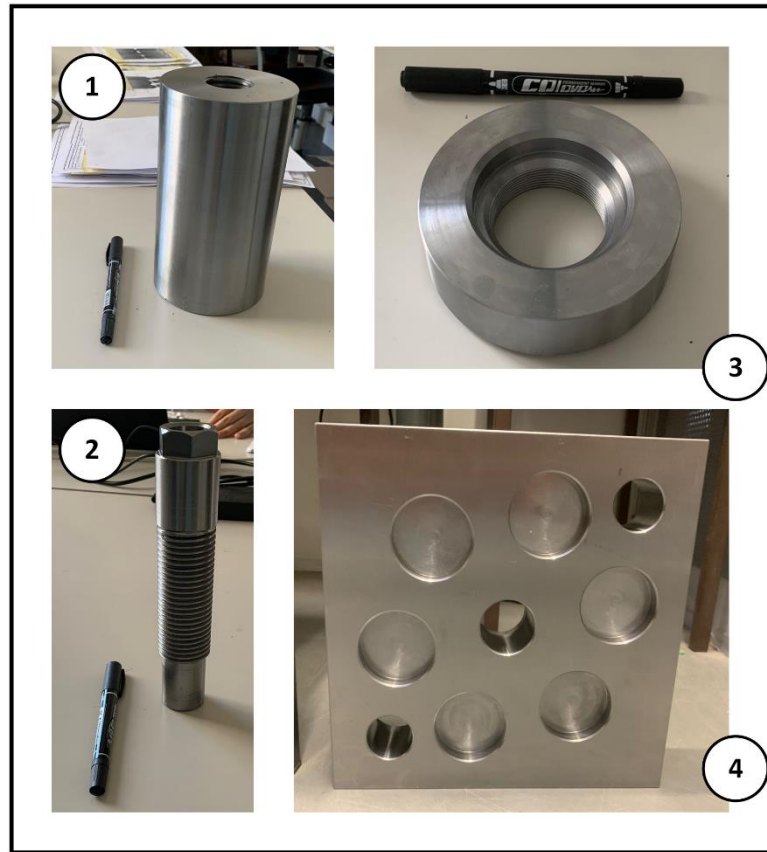


Figure 86- Passive components of the hole hemming tool.

Table 29 presents the material, manufacturing process, company that performs the process and the mechanical drawing's number of active components of the hole hemming tool.

Table 29- Fabrication of atctive components of the hole hemming tool.

Components	Material	Manufacturing process	Company	Mechanical drawing's number
UPPER DIE	PM300	Machining	FEUP	20220531011
LOWER DIE	PM300	Machining	FEUP	20220602013
BLANK HOLDER	PM300	Machining	FEUP	20220602014
FLANGING PUNCH	C265	Machining and heat treatment	FEUP and Ramada Aços	20220601012
HEMMING PUNCH	C265	Machining and heat treatment	FEUP and Ramada Aços	20220602015

Figure 87 shows the active tool components after the fabrication process.

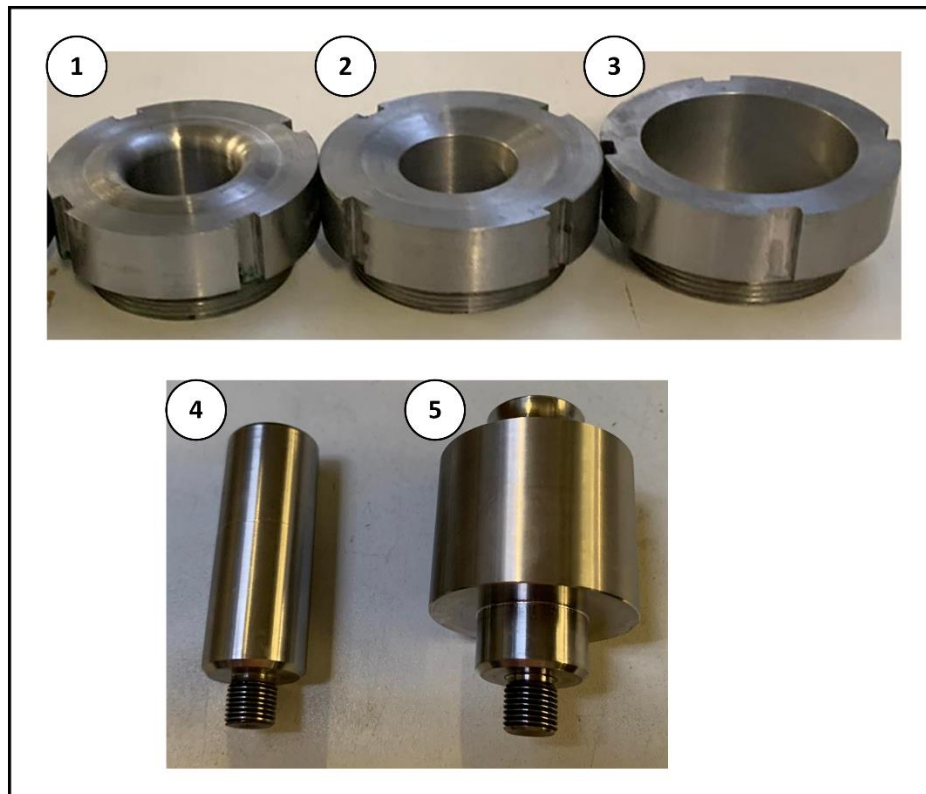


Figure 87- Active components of the hole hemming tool.

Table 30 presents the normalized components of the hole hemming tool.

Table 30- Normalized components of the hole hemming tool.

Components	QT.	Company	Norm
SPRINGS	6	Unceta	ISO10243 – XM 198
BUSHINGS	4	Schaeffler	EGB4040 – E40
NUTS	4	Fabory	ISO8675- M30x2
NUTS	1	Fabory	ISO4035 – M36x4
SCREWS	8	Fabory	ISO4762 – M5x0.8
SCREWS	6	Fabory	ISO10642 – M4x0.7
PINS	3	Fabory	Ø5 -H6

3.9. Assembly process

In this section, the assembly process of the hole hemming tool, Figure 88 and 89, is described.

This process makes it easier to assemble the tool for future works.

The tool assembly process is as follows:

1. Mount the die holder on the upper drive plate and tighten the respective screws - 4 ISO4762 M5x0.8x20-8.8 screws;
2. Next, the shank is assembled on the lower surface of the upper drive plate;
3. The upper drive plate is mounted on the actuator of the testing machine;

4. Another shank is now mounted on the upper surface of the upper drive plate;
5. The actuator of the testing machine is raised so that it's possible to screw the shank on the upper part of the testing machine, and at the same time unscrew the other shank of the actuator of the testing machine and consecutively from the upper drive plate;
6. Mounting the shank on the lower part of the lower plate;
7. Assembly of the lower plate on the actuator of the testing machine;
8. Mounting the guide pillars on the lower plate;
9. Align the upper drive plate and the lower plate, using the guide pillars, and fix their position using the nuts on the shank;
10. Mounting the height adjuster on the lower plate and tightening the corresponding screws - 4 ISO4762 M5x0.8x20-8.8 screws;
11. Assembly of the middle plate on the guide pillars, already assembled on the lower plate. A wooden shim must be used over the height adjuster in order to allow the placement of the middle plate;
12. Placement of the 6 springs between the lower plate and the middle plate, removing the previously placed wooden shim;
13. Assembly of the keys on the guide pillars and on the middle plate, using the plates and the respective screws – 6 ISO10642 M4x0.7x10-8.8 screws;
14. Through the movement of the upper part of the testing machine, carefully assemble the upper drive plate on the guide pillars;
15. Place the upper drive plate against the middle plate, using a wooden shim between the two;
16. Through actuator movement, compress the springs until reaching the desired preload;
17. After reaching the desired preload, mount the nut and lock-nut on the guide pillars. This step must be performed without the actuator relieving the load;
18. After correctly fixing the nut and the lock-nut, the load must be relieved by moving the middle plate and upper drive plate apart;
19. Next, the punch holder and nut are assembled;
20. Assembly of the respective punch is carried out;
21. Assembly of the dies, both in the upper drive plate and in the middle plate;
22. Adjustment of punch position through punch holder and screw thread.

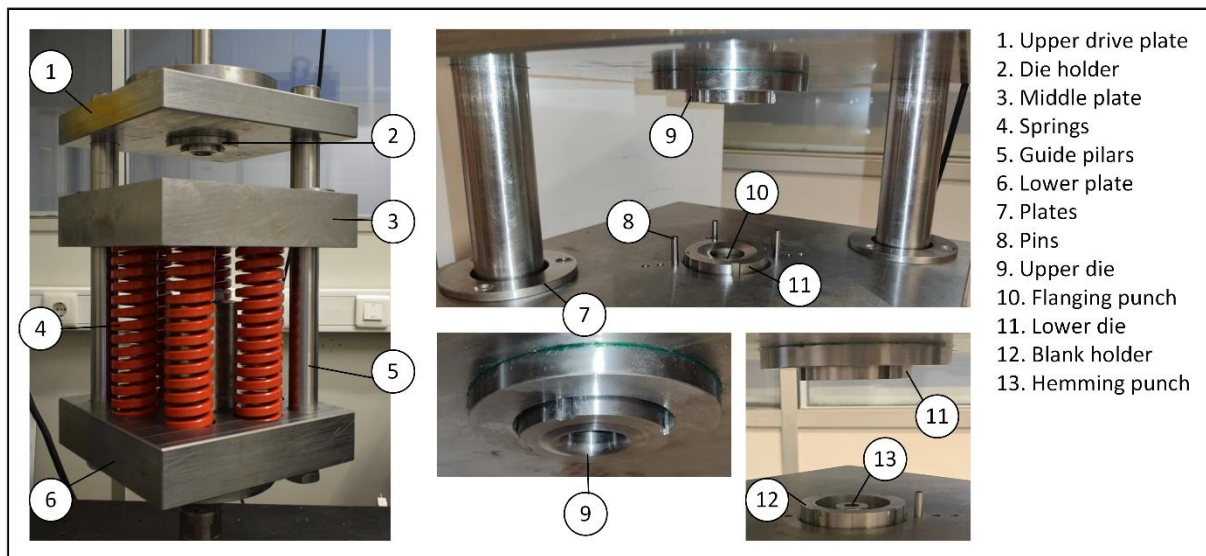


Figure 89- Hole hemming tool.

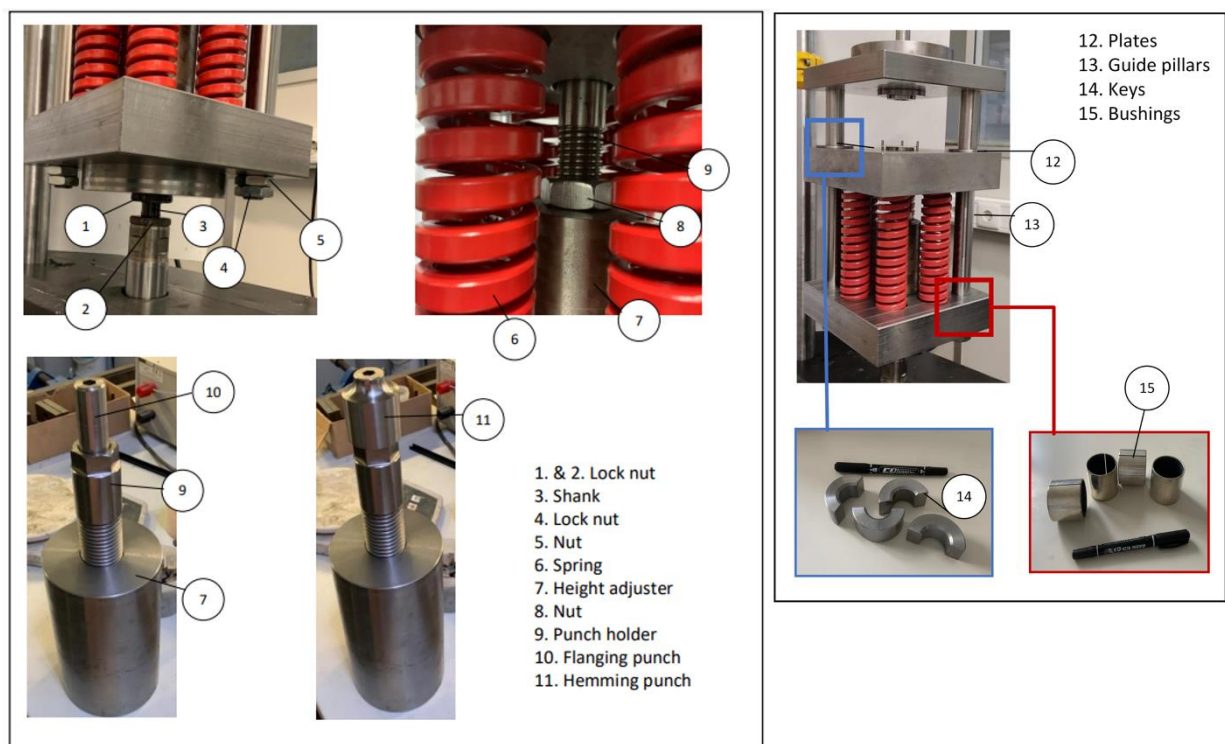


Figure 88- Hole hemming tool components

4. Evaluation of hole hemming tool

In this chapter the validation of the hole hemming tool is done by performing first stage and second stage on specimens with different dimensions. First, the operating conditions are defined, then, the results obtained in the ADFEUP's hole hemming tool are presented. Additionally, these results are also compared with numerical data results obtained from simulations made on SOLIDWORKS2021.

4.1. Test parameters

For the first stage of the process, hole flanging, the AA 6082-T4 with a thickness of 2 mm was deformed, Figure 90, (i.e., the material AA 6082-T4 was the “outer sheet”), presenting a greater ductility. For this material, several specimens were tested, all with the same thickness, but with different hole diameter, including 16 mm, 17 mm and 18 mm.

However, as the purpose of this dissertations is the development of the hole hemming tool and not the development of the hole hemming process, only the tests performed for the specimens with the hole diameter of 17mm will be presented.

The test parameters used for the first stage of the process were:

- Hole diameter: 16, 17 and 18 mm
- Preload: 40 kN
- Displacement: 17 mm
- Velocity: 6 mm/min

For the second stage of the process, hemming, the material AZ31 with a thickness of 0.9 mm was first placed on the outer sheet. In other words, the material AZ31 was the “inner sheet”. For this material, all the specimens had a hole diameter of 29mm, Figure 90.

For this stage the test parameters used were slightly different:

- Hole diameter: 29 mm
- Preload: 26 kN
- Displacement: 12 mm
- Velocity: 4 mm/min



Figure 90-Specimens of outer sheet and inner sheet.

4.2. Hole hemming experiments

The experimental tests, as mentioned in chapter 3.2., were performed on the Instron 8801 servo hydraulic testing machine with a capacity of 100 KN, which provided the evolution of force vs displacement during the experimental tests.

4.2.1. Hole flanging

The AA 6082-T4 specimen was first placed on the die. Three pins were used to fix the specimen in the right position. Then, the test was run, and the upper plate started moving down. As shown in Figure 91, the desired deformations were obtained, creating a flange that allows the assembly of the inner sheet and the creation of the mechanical interlock in the second stage.

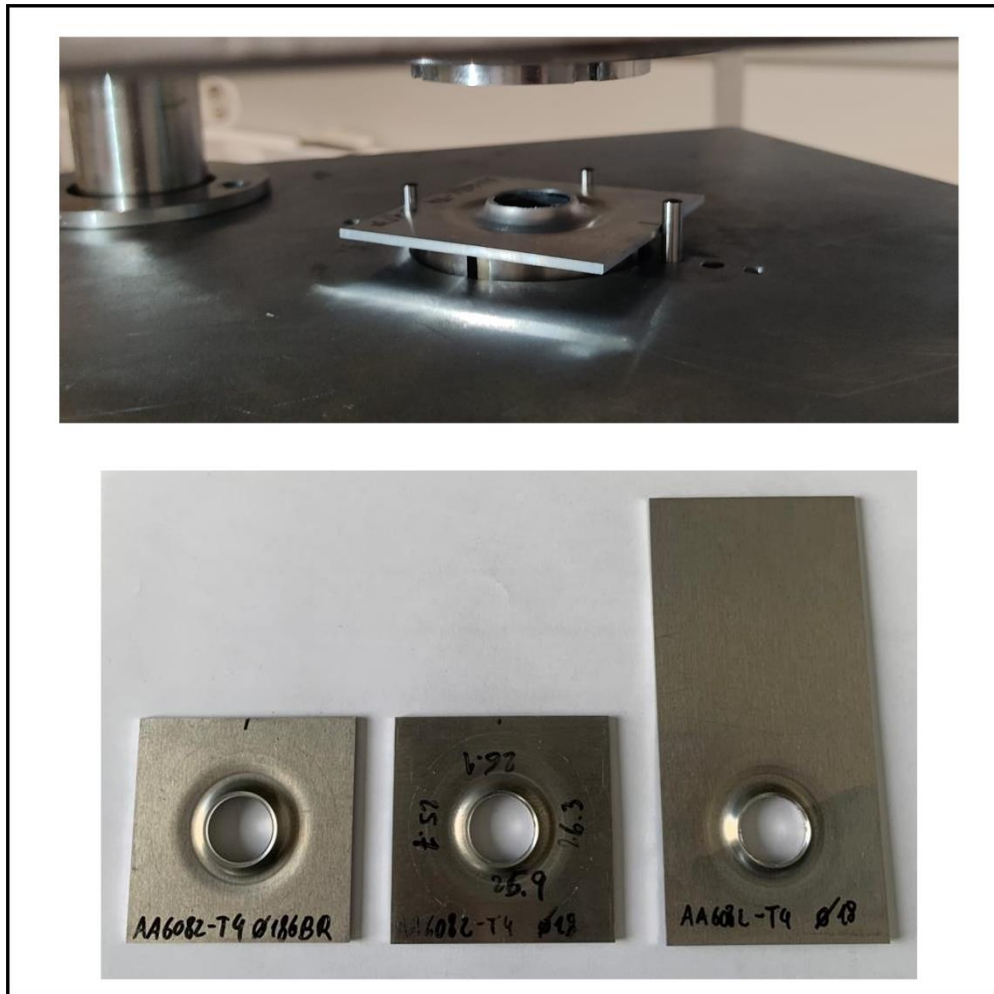


Figure 91-Specimens of the outer sheet after the first stage.

The force vs displacement curve was plotted in Figure 92. Note that this curve contains all the forces involved in the process.

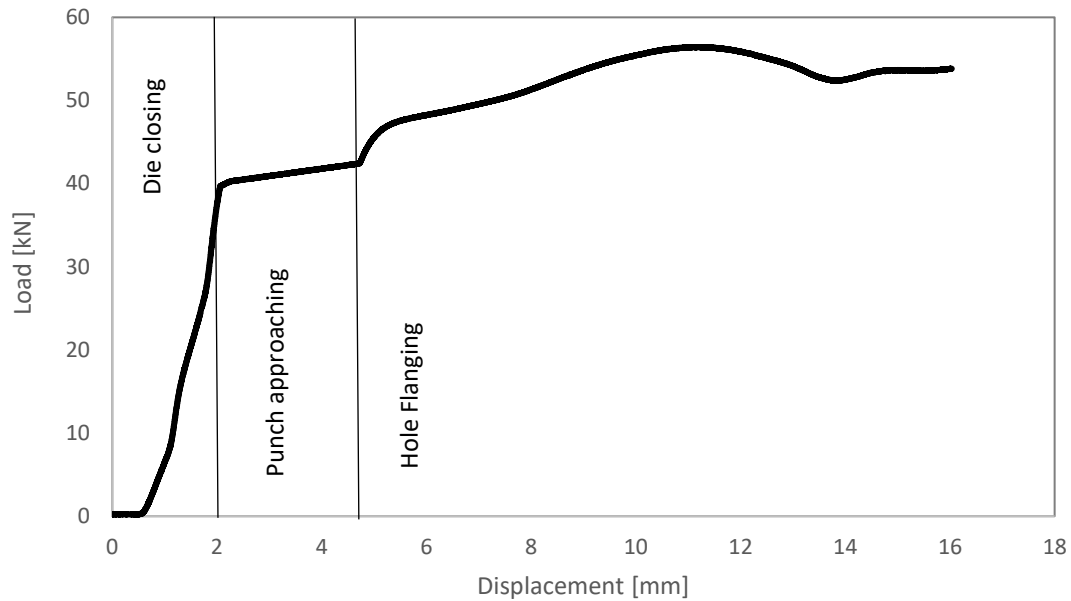


Figure 92-Force vs displacement curve of the hole flanging stage.

As it is possible to see in Figure 92, the force vs displacement curve can be divided into 3 parts: die closing, punch approaching and flanging.

The first part of the force vs displacement curve corresponds to the die closing. At this stage, there is only contact between the dies and the outer sheet, and the deformation of the sheet is only the result of the work of the dies. The graph is approximately a straight line.

The second part of the force vs displacement curve corresponds to punch approaching. This phase is defined by the preload of the springs, because for the punch to start contacting the outer sheet, this force must be overcome. That is, by setting a certain value for the preload, it is ensured that the outer sheet is compressed by the dies to that value before the punch comes into contact with the sheet. For the case under analysis, a preload in the order of 40 kN was defined, and as it is possible to observe when this value is reached, the slope of the graph is completely altered, this slope corresponding to the stiffness coefficient of the set of springs.

The last part of the force vs displacement curve corresponds to the performance of the flanging punch. The start of contact between the punch and the sheet can be identified through the variation in the slope of the graph.

Since the springs were not modelled in the simulation, the approximate value of the force applied to the flanging punch was obtained by removing the force applied by the springs from the total force at each moment and compared with the value obtained from the numerical simulation [22]. As can be seen, in Figure 93, the forces obtained in the flanging punch are in

very good agreement with those predicted by the numerical simulation. Thus, the tool works very close to the ideal design and can create a flange with the desired quality in the first stage.

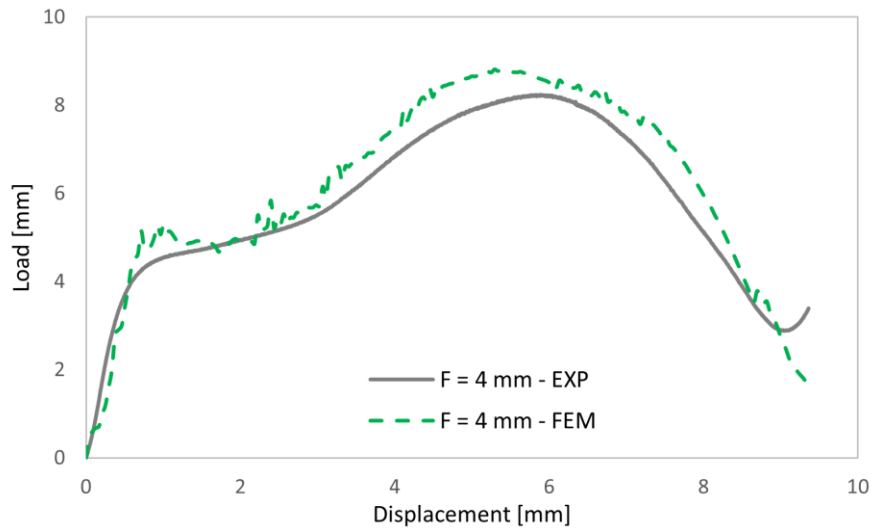


Figure 93-Force vs displacement curve of flanging punch [22].

4.2.2. Hemming

To start the second stage of the process, it was first necessary to change the active tool components. Namely, replace the lower die with the blank holder, the upper die with the lower die and the flanging punch with the hemming punch. Then the AZ31 specimen is assembled under the 6082-T4 specimen. Next both specimens were placed on the blank holder. Three pins were used to fix the specimen in the right position. Then, the test was run, and the upper plate started moving down. As shown in Figure 94, the desired deformations were obtained, the flange is formed over the hole edge of the inner sheet, producing a mechanical interlock. The hole edge of the inner sheet is also slightly deformed and placed inside the indent.

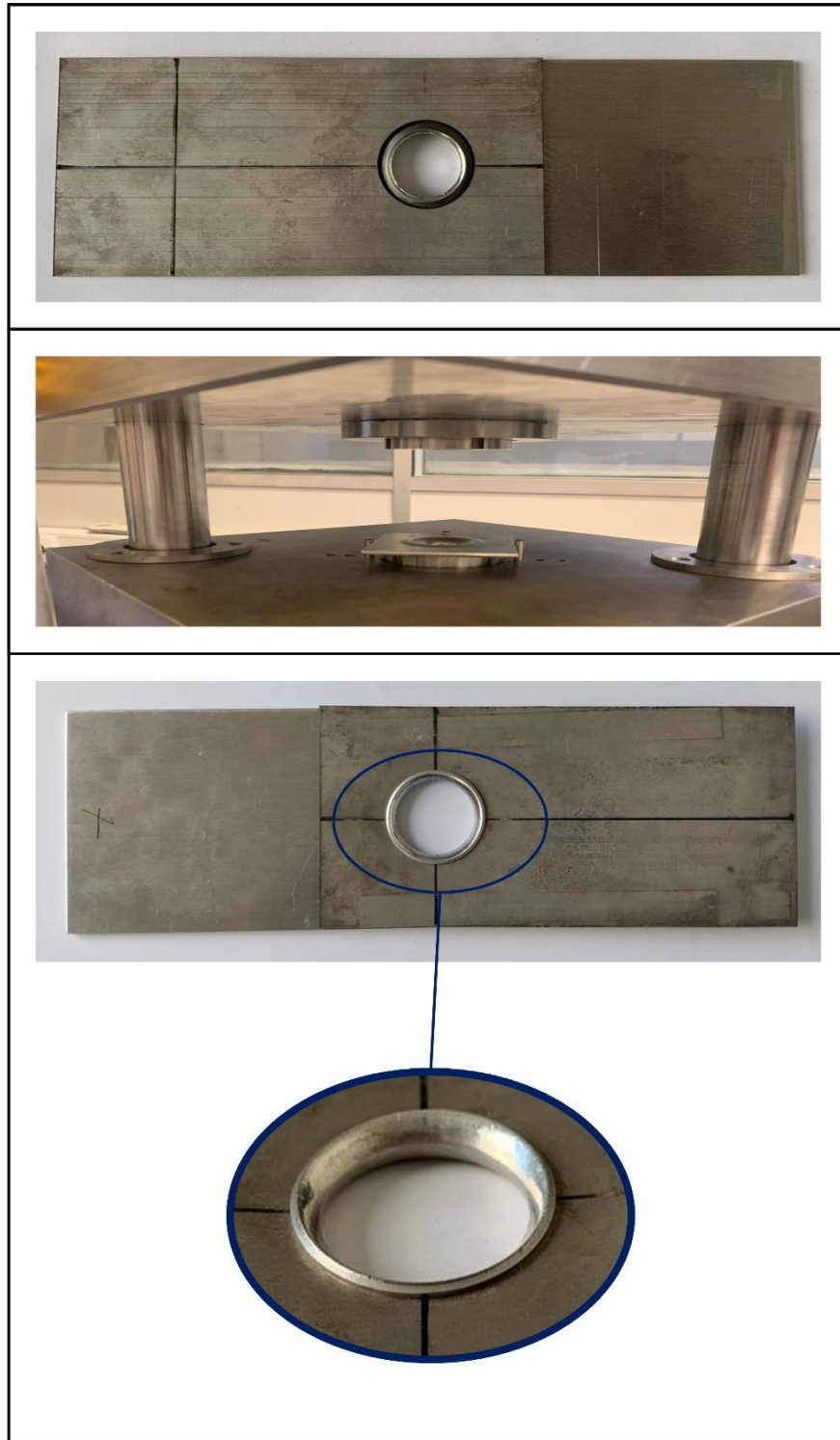


Figure 94-Specimens at the hemming stage.

The force vs displacement curve was plotted in Figure 95. Note that this curve contains all the forces involved in the process.

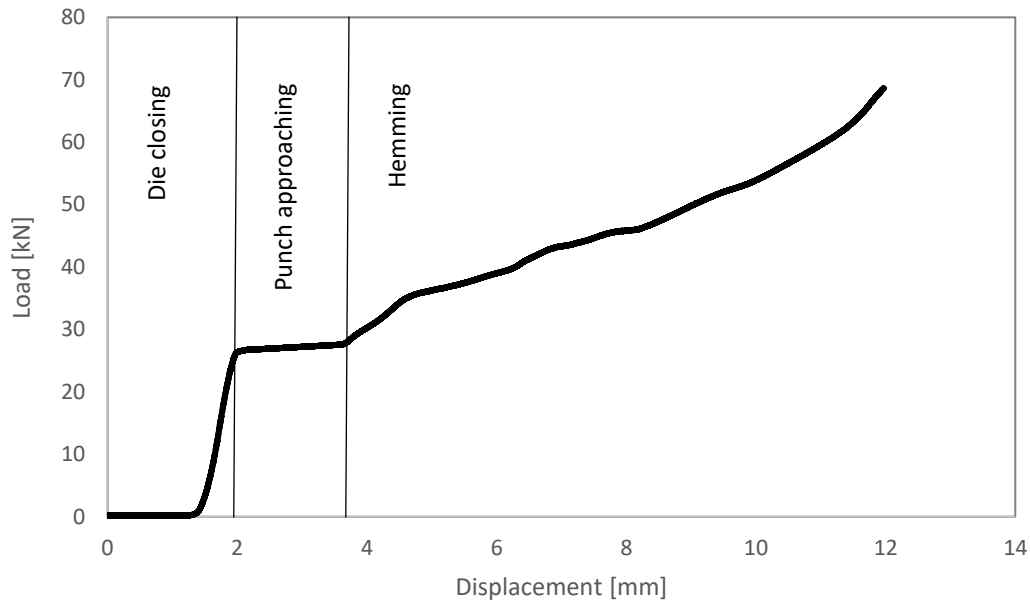


Figure 95-Force vs displacement curve of the hemming stage.

As it is possible to see in Figure 95, the force vs displacement curve can be divided into 3 parts: die closing, punch approaching and hemming.

As in the first stage, the first part of the force vs displacement curve corresponds to the die closing. The beginning of the crushing of the plates by the dies is identified through the beginning of the increase in force. It is verified that this part of the graph corresponds to an approximately linear line. As in the hole flanging stage, this first part of the force vs displacement curve is the one with the greatest slope (largest increase in force for a given displacement).

The second part of the force vs displacement curve, the punch approaching, is identified by a sudden change in the slope of the graph. In this stage, the required preload was 26.7 kN, and as can be seen, the slope of the graph changes precisely at this value, with the slope of the graph being the stiffness coefficient corresponding to the set of springs used.

The last part of the force vs displacement curve begins when the hemming punch start contacting the specimens. This moment can be identified through the variation in the slope of the plot.

As in the previous stage, the approximate value of the force applied to the hemming punch was obtained by removing the force applied by the springs from the total force at each moment and compared with the value obtained from the numerical simulation, Figure 96 [22].

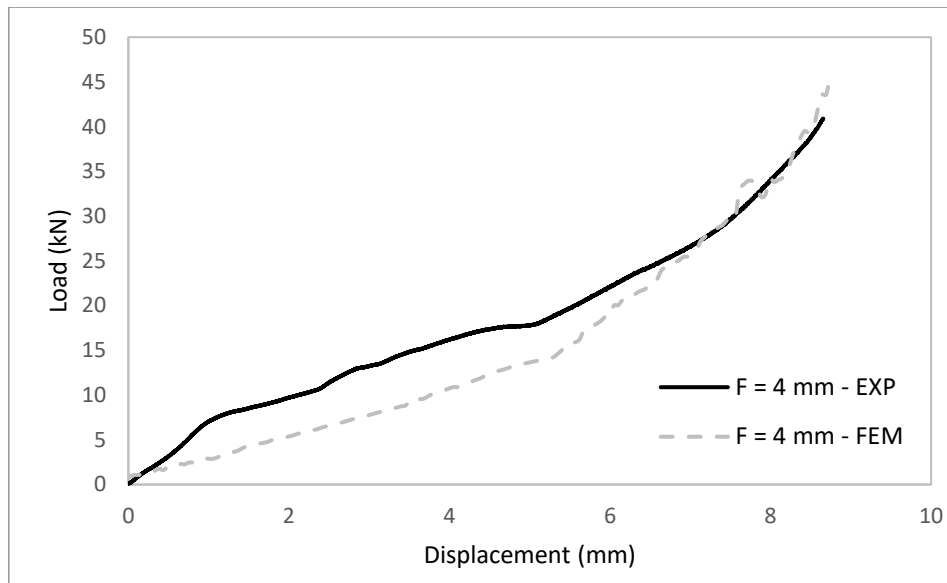


Figure 96-Force vs displacement curve of hemming punch [22].

It is possible to observe that the forces obtained in the hemming punch are practically identical to those obtained through numerical simulations.

4.2.3. Hole hemming tool

As seen in the previous sections, the developed tool allows performing both stages of the hole hemming process, reaching the strength and quality requirements initially proposed.

Concerning the preload, it was possible to see that the values of preload defined were both achieved during the process.

Although initially a minimum preload of 40kN was proposed for both the stages, during the experimental tests, it was found that with the use of this preload for the hemming stage, the required force was approaching the capacity of the testing machine. As a result, four springs instead of 6 springs were used, which allows reducing the preload from 40kN to 26 kN, thus achieving the desired force for the hemming punch. This change in the preload was easily done, which is another evidence of the flexibility of this developed tool.

However, it was not possible to verify the variation of the preload up to its maximum value, being necessary to verify it in the future.

The adjustment of the punch position in relation to the dies was also found to be functional, allowing the use of different punch positions in the tests, especially at the beginning of each stage to achieve the best position. The replacement of punches and dies from one stage to the other was easily carried out. Thus, it can be concluded that this tool meets all the considered requirements well.

However, problems arose regarding the alignment and centering of the sheets, Figure 97. This was due to the difficult machining of the specimen holes, making it difficult to obtain specimens with equal dimensions.

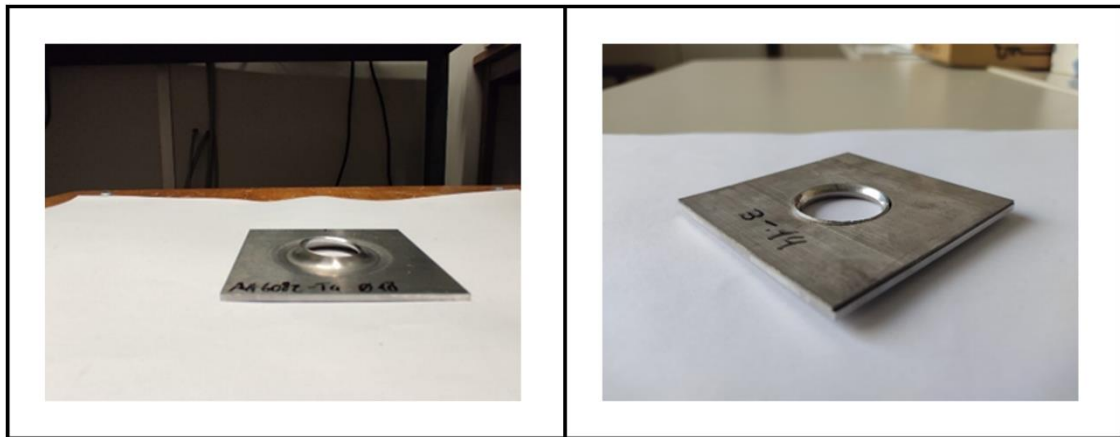


Figure 97- Specimens with problems in the centering

5. Conclusions and future developments

This chapter summarizes the main conclusion of this dissertation, reporting the objectives achieved and addressing the possible future improvements to be implemented in the hole hemming tool.

5.1. Conclusions

In this dissertation, a flexible tool was developed that allows to perform the hole hemming process for dissimilar materials, with different thicknesses and geometries under laboratory conditions.

The initial step of this dissertation was to obtain the concept that provides the best conditions for the process under study. Bearing in mind the characteristics of the testing machine where the tool would be implemented and the purpose of developing the hole hemming tool, the first concept presented in this dissertation was used, allowing an easy variation of the parameters to be tested and an easy replacement of the active components of the tool.

The structural components were designed in a way to allow an easy variation of the preload, an easy handling of the active and passive components of the tool, and that this tool could be used for other joining processes by plastic deformation. The structural components were then dimensioned in order to withstand the working conditions of the testing machine used. For these components, the use of 4 springs instead of 6 was also analyzed and verified.

Regarding the passive components of the tool, they were designed with the aim of accommodating the active tool comments and adjusting the position of the punch. Similar to the structural components, these components were dimensioned in order to withstand the working conditions of the testing machine, thus allowing the use of these components in other processes.

The active components of the tool have been designed with the modular concept in mind, in order to provide easy replacement of these components. These components were dimensioned to support the conditions mentioned for the hole hemming process.

All the structural, passive and active components of the tool were successfully manufactured and implemented. In order to validate the tool, experimental tests of the hole hemming process were performed. The experimental tests verified all the previously mentioned characteristics; however, it was not possible to verify the total variation of the preload, being only verified the preloads of 40 and 26 kN. Regarding the guidance and alignment of the entire tool, the design was proved to be enough for the correct performance of both stages of the process.

However, there was some difficulty in correctly positioning the specimens on the tool and it was verified some aluminum stuck in the punches. Concerning the aluminum in the punches, this is a common problem when using aluminum specimens.

To complement the information given in this dissertation's report, a paper entitled "Flexible tool for joining dissimilar materials by the novel hole hemming process" was written that is supposed to submit to the "Mechanics Based Design of Structures and Machines".

Despite the objectives were successfully achieved and the performance of hole hemming tool was verified, there is still some aspects for more development. Thus, some suggestions for future developments are presented in the next section.

5.2. Future developments

Although the hole hemming tool is ready to be used for performing the hole hemming process, there are still aspects that can be improved, and additional components can be implemented so that the tool can reach its final state.

First, a physical barrier should be implemented, so that the safety of the operators is guaranteed. This is necessary as the springs are subject to a high preload, and in the event of a disaster they can seriously injure the user.

The last validations of the tool must be done, namely the possibility of varying the preload by about 20 kN. It is also recommended that the tool is examined for joining different material combinations that require less or more force than the studied case.

The positioning of the sheets in the tool is the biggest challenge, If the holes are not carefully made in the sheets. To solve this problem, the centering of the sheets must be done through their holes and not through their edges by pins. Another solution to this problem is to improve the hole hemming tool, so that the holes are made in the sheets before the hole hemming process.

It is also considered interesting to study and carry out treatments for the punches produced, due to the high aluminum stuck on the contact surface, with special emphasis on the flanging punch.

Finally, although the tool was designed only for experimental tests under laboratory conditions, required checks must be carried out regarding the fatigue of their component. It is also advisable to carry out periodic checks on the conditions of the tool components to detect any defects that may affect the performance of the tool.

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Appendices

A flexible tool for joining dissimilar materials by the novel hole hemming process

A.G.C. Conceição¹, M.M. Kasaei², R.J.C. Carbas², E.A.S. Marques², L.F.M. da Silva¹

1 Departamento de Engenharia Mecânica, Faculdade de Engenharia (FEUP), Universidade do Porto, Porto, Portugal

2 Instituto de Ciência e Inovação em Engenharia Mecânica e Engenharia Industrial (INEGI), Porto, Portugal

Abstract

This research work presents the development of a flexible tool for performing the new joining process, called hole hemming, between dissimilar materials with very different mechanical properties. This work includes the design, fabrication, and assembly of the structural, active and passive components of the hole hemming tool and the evaluation of their performance. The tool is designed to have adjustable structural and passive tool components as well as easily interchangeable active tool components. The tool is evaluated for joining the AA6082-T4 aluminium sheet to the AZ31 magnesium sheet by experimental tests. The experimental data is also compared to the numerical results obtained from the finite element analysis of an ideal design. The results in both stages allow concluding the developed tool meets the considered requirements and can be efficiently used for creating hole-hemmed joints between dissimilar materials.

Keywords

Deformation assisted joining, Hole hemming, Tool design, Flexible tool, Lightweight materials

1. Introduction

Due to the increasing pollution of the atmosphere, the automotive industry has been pressured to reduce the weight of automotive bodies for the reduction of the consumption of fuels. However, it is necessary to ensure that the decrease in automotive bodies is not associated with a decrease in collision safety. As a solution, new lightweight materials such as advanced high strength steels, magnesium alloys, aluminum alloys and composites have been created and suggested for the manufacturing of automotive body parts. However, the development of lightweight automotive bodies still requires new joining processes that have the ability to connect such dissimilar materials [1].

Plastic deformation, although is generally used in forming processes, can also be employed for joining materials, especially in cases of dissimilar materials where conventional joining methods are ineffective. In mechanical joining by plastic deformation a mechanical interlock is produced between sheets by creating material plastic flow, which connects them without the need for melting and metallurgical bonding. The mechanical joining by plastic deformation offers the clearest opportunities to create versatile joining processes due to their low heat input, robustness to metallurgical incompatibilities and various process variants [2].

There are various joining processes by plastic deformation[3]. In these processes, to attain a proper interlock, either both materials must be adequately ductile, or an auxiliary element must be used which increases the joining cost. Among all the processes, clinching and self-pierce riveting stand out due to their high productivity, low cost, and ability to join different materials. However, it is still difficult to join materials with low ductility or high strength, especially when joining aluminum alloys to advanced high strength steels or to magnesium alloys that are often required for the manufacturing of automotive bodies. The materials with low ductility tend to fracture due to plastic deformation [4] while it is difficult to pierce high strength steel sheets in self-pierce riveting [5].

To expand the scope of application of joining by plastic deformation, *Kasaei and da Silva* [1] proposed a new hemming-based joining process, called 'Hole Hemming'. The 'Hole Hemming' process enables to join dissimilar materials with different mechanical properties, and without limitation in the joint position. In this process, the hole edge of an outer sheet is bent over an inner sheet to join two sheets together. The process is carried out in a two-stage operation including Hole Flanging and Hemming so that the inner sheet is not deformed, and no additional joining elements are required. Thus, only by controlling the plastic deformation of a sheet, the mechanical interlock can be formed. In this process, unlike the hole clinching process, the high

strength steel sheet is formed to create a mechanical interlock [1]. However, the capability of the process should be thoroughly examined by performing experiments under different conditions. The first step for this purpose is to design a suitable tool for this process.

The tool design, more properly die design in this case, is a complex, fascinating subject. It is one of the most exacting of all the areas of the general field of tool design [6]. The design and construction costs can be significant and may easily reach several tens of thousands of Euros, therefore, a lot of time and attention should be spent on this section, in order to avoid errors that can significantly increase the final costs [7].

Wiesenmayer et.al. [8] used a tool set to test the joinability of the high-strength alloy and AA7075 in shear-clinching processes. The die-sided tool consists of a base plate to which the die is mounted. The inner and the outer punches are installed to individual plates, which are connected via springs with a pre-load of 29 kN. Both plates are guided relative to the base plate by pillars, allowing the exact positioning of the tools. The blank holder is also mounted to an individual plate, which is connected via springs with the outer punch plate. The pre-load is 1 kN.

Reichel et.al. [9] design a flexible tool demonstrator capable of replicating material flow in the lancing, bending and compression stages of a new joining by forming process without auxiliary elements. The flexible tool demonstrator is defined by a modular concept that allows the active tool components to be easily interchanged for testing and exploring different materials and thickness combinations, surface conditions and cross-section areas of the strips under laboratory conditions.

Lambiase et.al. [10] developed a numerical model describing the evolution of ductile damage to predict the onset of fracture during the clinch joining of thin aluminum AA6082-T6 sheets. The damage model was calibrated and validated using instrumented punch-out tests to increase the reliability and robustness of the damage parameter. The authors used a prototypal punching frame, which was clamped on an MTS model 322.31 universal testing machine. In the tests performed, dies with three different diameters were used to produce different punch-die clearances. K340 hardened steel material was used for the manufacture of punches and dies in order to prevent smoothing and fractures on the cutting edges. The specimens are centered in the die using an external ring and the clamp of the specimen was secured by the blank holder (with a preload of 6 kN and spring stiffness of 1.2 kN/mm).

There are several parameters that influence the hole hemming process. Thus, to be able to test different parameters and conditions of the process, a tool should be designed in such a way that it has the following features:

- Adjustable structural and passive tool components
- Easily interchangeable active tool components

Under these circumstances, the main objective of this paper is to present a flexible tool for the hole hemming process that allow the joining dissimilar materials with different thicknesses and cross-section areas under laboratory conditions. The presentation of the flexible tool is supported by experiments.

2. Hole Hemming Concept

In the hole hemming process, unlike hemming process, there is no limit to the joint position. This is because the sheets are connected through pre-drilled holes. These holes can be drilled anywhere on the sheets as long as they are concentric. The holes must have different radius so that the sheets with the smaller and larger holes are called 'outer sheet' and 'inner sheet', respectively. The 'outer sheet' must have sufficient ductility to prevent fracture, while the inner sheet has no restrictions [1].

The process is divided in two stages, as shown in Figure 1. The first stage is called 'Hole Flanging', as the name implies this stage is a flanging operation. At this stage, the components that are in contact with the sheet are the flanging punch, the lower die and the upper die. The 'Hole Flanging' begins with the descent of the upper die, so that the dies close and initially forms the 'outer sheet' (for future purposes, this movement is named as 'step1 of stage 1'). Then the edge of the hole is bent to a 90°angle by moving up the flanging punch. The dies are designed to make an indent next to the flange. This causes the hole edge of the inner sheet can be placed at the closest distance to the flange in the next stage. At the hole flanging stage, the parameters that need more care and attention are the hole radius of the outer sheet, the die radius, the fillet radius of the upper die, the indent depth and the nominal length of the flange[1].

The second stage is called 'Hemming' where the hemming punch, the lower die and the blank holder are used for making a mechanical interlock. The blank holder is employed to fix the position of the sheets during the plastic deformation. It is necessary to change or relocate the active components from one stage to the other. First, the 'inner sheet' is assembled under the 'outer sheet', then the flange is formed over the hole edge of the inner sheet by means of a hemming punch to produce a mechanical interlock. In order for the flange to fit inside the hole and the inner sheet to be completely under the outer sheet, the hole radius of the inner sheet must be equal to or slightly larger than the radius of the die hole. The hole edge of the inner

sheet is also slightly deformed and placed inside the indent. This indent has a depth equal to the thickness of the inner sheet, ensuring that no protrusion is created. Better mechanical locking can be obtained by increasing the contact surface between the hole edges of the outer and inner sheets. At the hemming stage, the most important parameter is the fillet radius of the hemming punch [1].

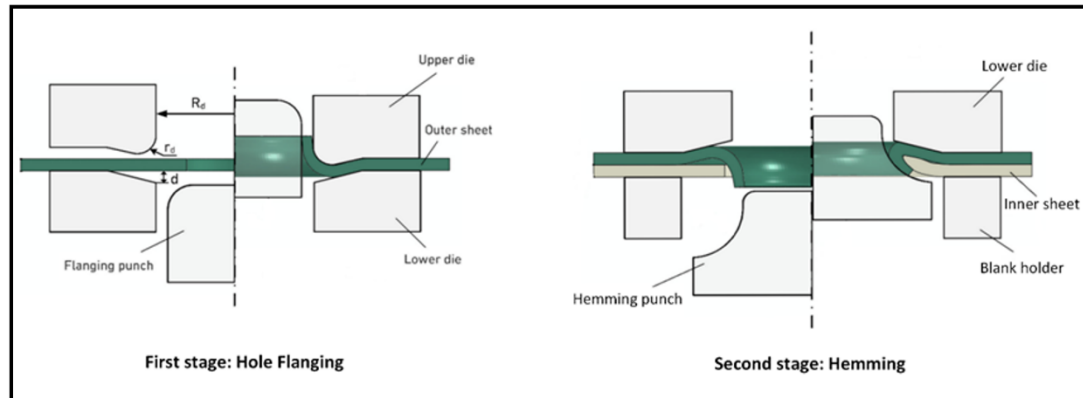


Figure 1- Schematic representation of the hole hemming process.

3. Mechanical properties of the materials

The material AA 6082-T4 is the 'outer sheet', presenting a greater ductility so that fracture does not occur in the first stage, 'Hole flanging'. While the AZ31 material corresponds to the inner sheet', which does not require such high ductility, because this material is only assembled in the second phase, 'Hemming', Table 1 and Figure 2.

Table 1- Mechanical properties of inner sheet and outer sheet.

Material	Thickness [mm]	Young modulus [GPa]	Yield stress [MPa]	Ultimate tensile stress [MPa]	Elongation [%]
AA 6082-T4	2	70	173	278	24.5
AZ31	0.9	44	142	259	16.6

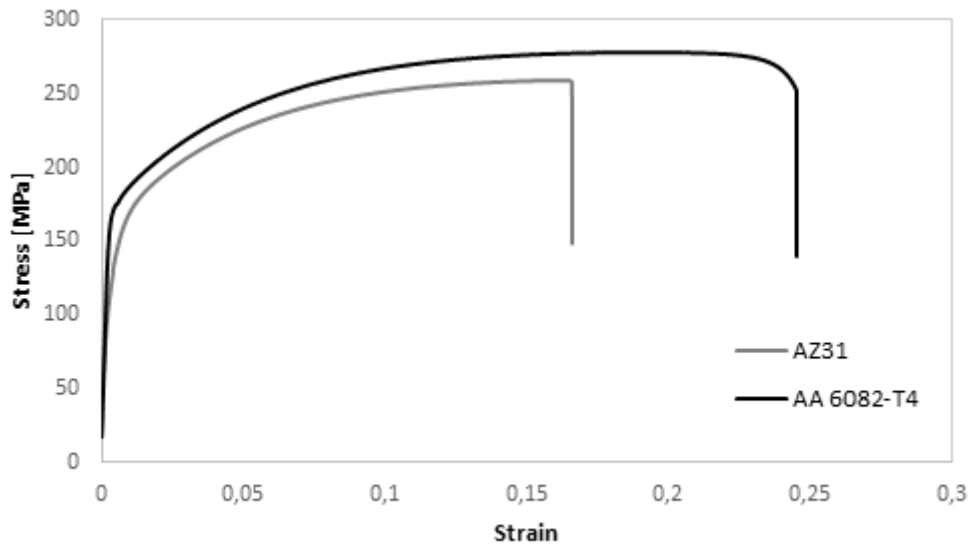


Figure 2- Stress vs strain of inner sheet and outer sheet.

4. Finite element modelling

The numerical simulations of the hole hemming process were also performed in the Abaqus software. Like the formability tests, the sheets were modeled using 3D solid elements C3D8R, considering 8 and 5 elements through thickness for the AA6082-T4 (the outer sheet) and AZ31 (the inner sheet), respectively. A smaller element size of 0.2 mm was applied in the hole sheet edges (Figure 3) where the highest deformations are located. The mechanical properties of each material were defined using the material models obtained in section 4. A dynamic explicit solver with a time scaling approach was used. The calibrated ductile fracture criteria for the AA6082-T4 and AZ31 sheets were separately defined in the user subroutine VUSDFLD to analyze the damage evolution and, finally, predict the fracture during the hole hemming process. The element deletion technique was activated to model the crack propagation when the damage indicator (D) reaches 1.

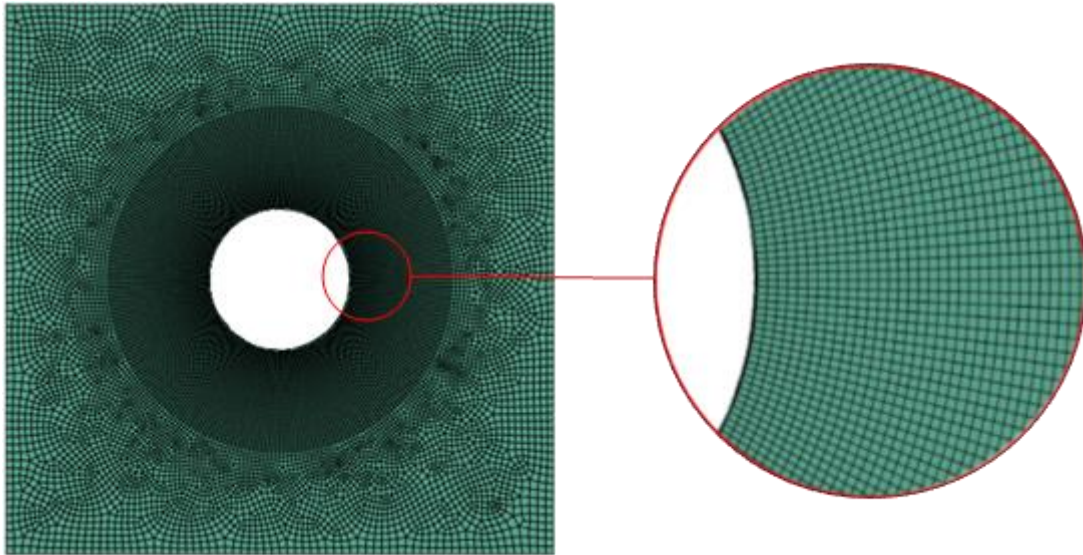


Figure 3- Hole hemming FE model mesh details of a AA6082-T4 sheet.

The punches (flanging and hemming punch) and matrix (bottom and upper die and blank holder) were modeled as rigid bodies and the FEM apparatus can be seen in Figure 4. The simulation is performed in three different steps by applying a displacement to the upper die, flanging punch and hemming punch, in that order. The frictional contact conditions between tools and sheets were defined by a penalty-based contact model using the Coulomb friction law for a friction coefficient of 0.08.

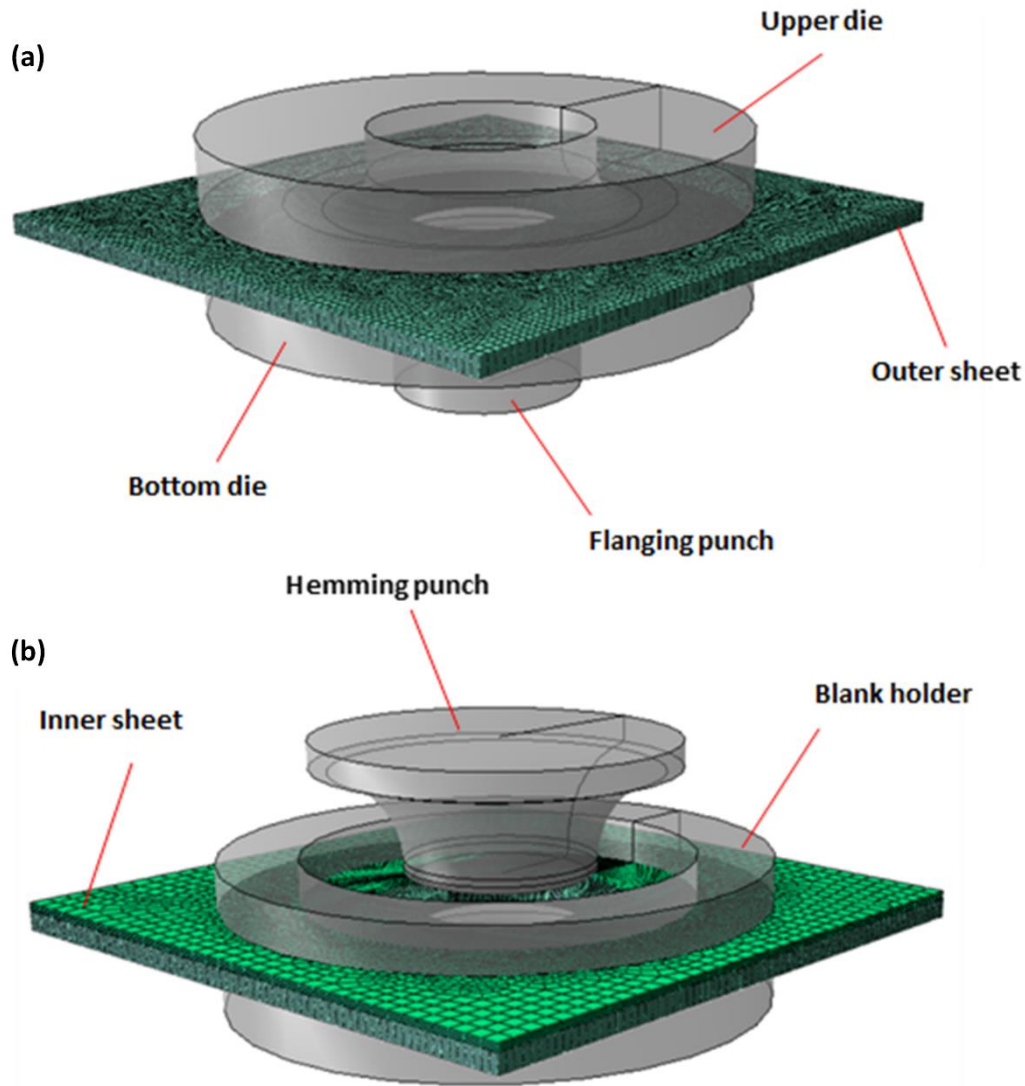


Figure 4- Finite element modelling of the hole hemming process: (a) hole flanging (b) hemming.

5. Tool design

a. Required loads

The forces exerted to the tool components were obtained from the finite element simulation during the process and, in this way, the critical components in each stage were identified [11]. Of all the dies, the upper die is the most critical one, which is used to create the indentation and fix the outer sheet in the first stage. The force applied to the upper die reaches a maximum value of about 55 kN when the dies are closed (Figure 5) and remains constant during the deformation of the hole edge by the flanging punch.

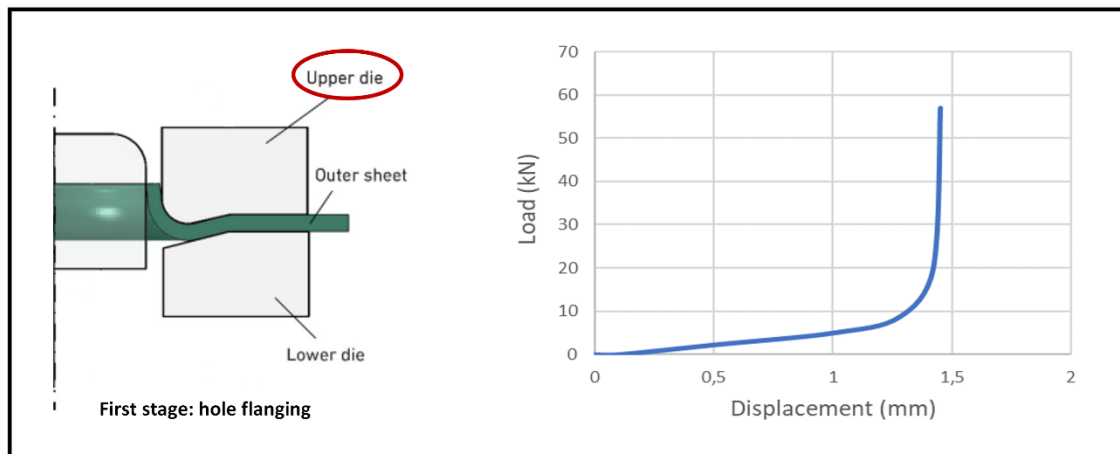


Figure 5- Load vs displacement of the upper die [11].

The punches have main role in forming the flange and the mechanical interlock, respectively. However, the flanging punch experiences a much lower force than the hemming punch due to the bending deformation that is dominant in the first stage (Figure 6 and 7).

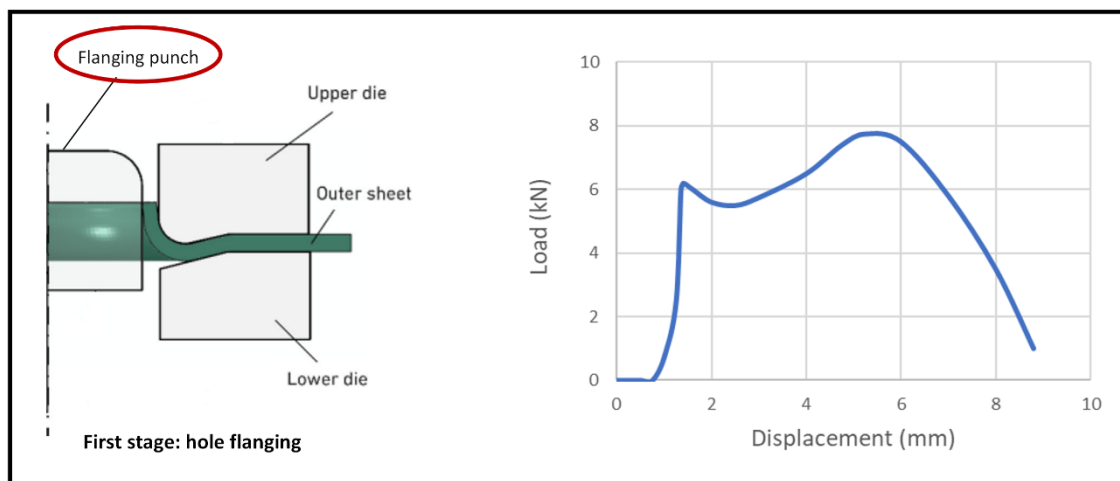


Figure 6- Load vs displacement of the flanging punch [11].

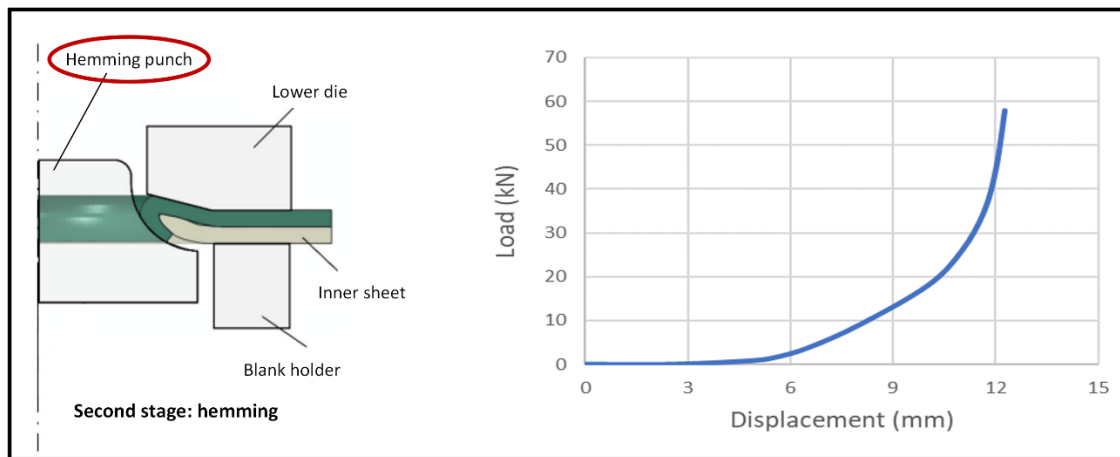


Figure 7- Load vs displacement of the hemming punch [11].

b. Testing machine

In order to start designing and dimensioning the tool, it is necessary to pay attention to the machine where the tool will be mounted. In this work, the machine is an Instron 8801 with a load cell of 100 kN. This machine presents: a double-acting servo hydraulic actuator with force capacity up to ± 100 kN; and high-stiffness, precision-aligned load frame with twin columns and actuator in lower base [12]. Regarding the dimensions, the width and height of the tool should not exceed 562 mm and 1098 mm, respectively. The tool should be fixed through a M30x2 right hand central thread [13].

In order to be able to correctly assemble the tool on the machine, it was necessary to check the mechanical interfaces and dimensions of the machine, in order to ensure that it is properly fixed to the machine and that it is possible to accommodate it.

c. Hole hemming tool

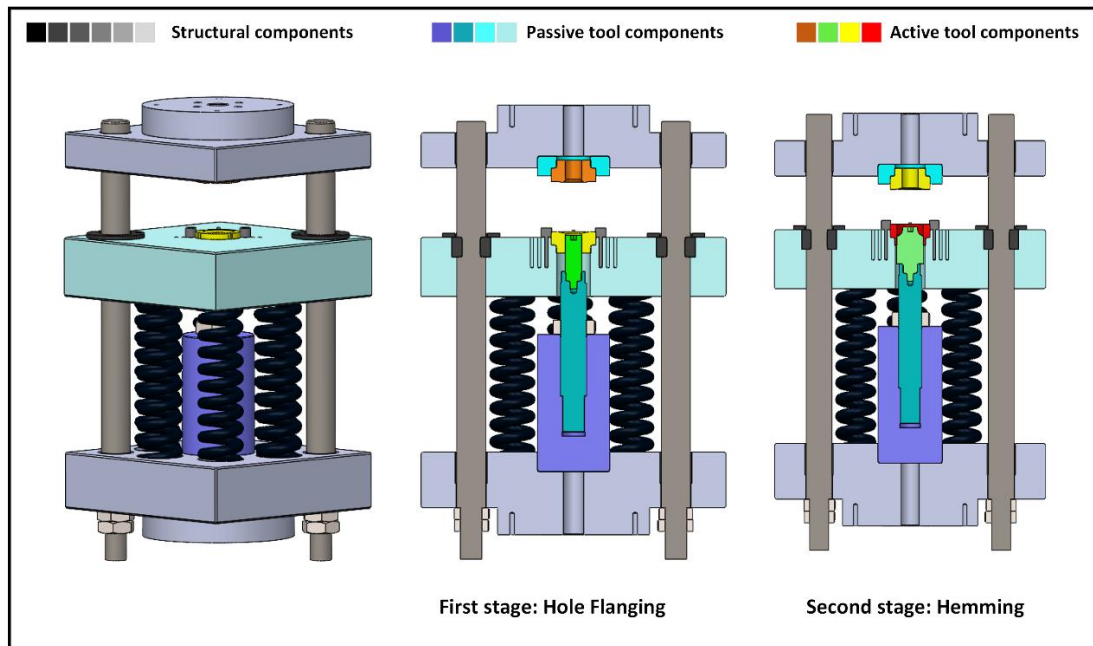


Figure 8- Schematic representation of the hole hemming tool.

The quality and reliability of the process is highly dependent on the hole hemming parameters, as previously seen in section 2. It is necessary to build a flexible tool to be able to explore and test the different combinations of materials, so that it is possible to easily change the parameters of this process and easily change active tool components. Thus, the tool components can be classified into three main categories (Figure 8): active tool components, passive tool components and structural components [9].

The active components of the hole hemming tool are those that come into direct contact with the sheets during joining by plastic deformation, that is, the flanging punch (shown in green), the hemming punch (shown in light green), the upper die (shown in brown), the lower die (shown in yellow), and the blank holder (shown in red). These components only serve to manufacture hole-hemmed joints with specific geometries and sizes and need to be replaced from one stage to the other stage as they are the most stressed components. However, it is possible to easily replace these components with others with different geometries and sizes to produce a wide range of hole-hemmed joints due to the modular concept of the hole hemming tool [9].

As for the passive components of the hole hemming tool, these surround the active components of the tool and support them directly or indirectly. These components include the punch holder (shown in navy blue) that fixes the different types of punches, the height adjuster (shown in dark

blue) that allows to change the height of the punches, the die holder (shown in light blue) and the middle plate (shown in turquoise) that accommodates the different dies. Distribute the forces through the sheets and the tool structure without damaging the hole-hemmed joints and overloading the active tool components, accommodate the active tool components, provide the guidance for the positioning of the sheets, and allow the adjustment of the position of the active tool components are the main functions of the passive tool components [9].

Finally, the structural components of the hole hemming tool are those that accommodate the passive components of the tool and are independent of the process. These components have as main functions to provide orientation and centering between the upper, middle and lower plates, to enable the installation of the tool in the testing machine and to allow the variation of the preload to test different process conditions [9]. Furthermore, as structural components are process independent, they can be used to accommodate active and passive tool components from other processes [9]. These components include the upper drive plate, the lower plate, the guide pillars, the bushings, the pins, the keys, the plates, the nuts, and the springs.

d. Structural components

The structural components of the hole hemming tool, Figure 9, were designed based on the following requirements:

- Accommodate the passive tool components;
- Provide guidance and centering among the upper, middle, and lower plates;
- Enable the installation in the testing machine;
- Allow a pre-load variation for the springs.

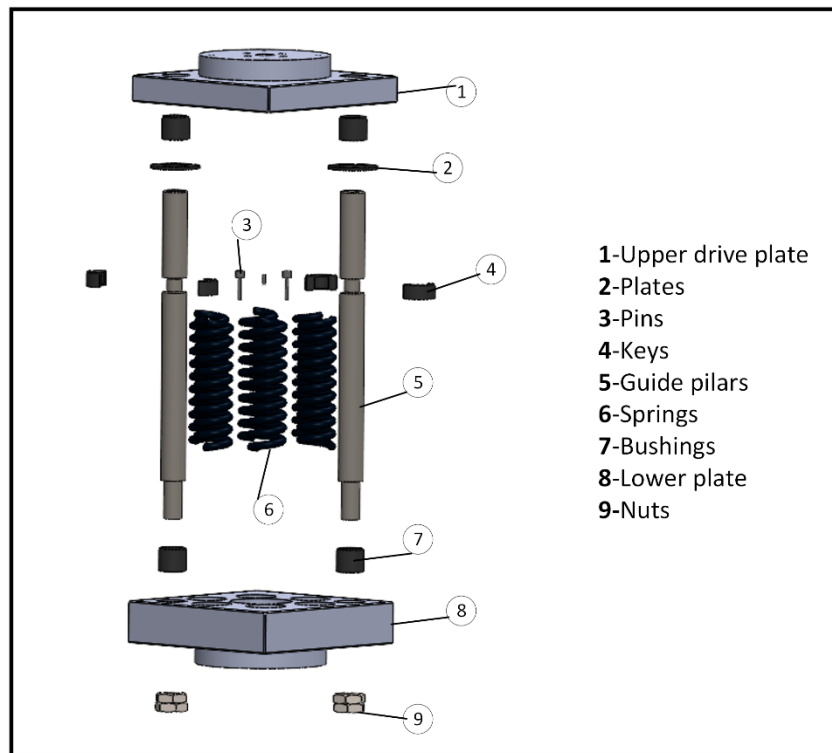


Figure 9- Schematic representation of the structural components of the hole hemming tool.

As shown in Figure 5, there is a peak in the force applied to the upper die, which occurs at the end of the movement of the upper die, before the flanging punch starts contacting the upper sheet (step 1 of stage 1). This force, called preload, is due to the compression of the upper die against the sheet.

The preload exerted in the studied case has a maximum value of approximately 55kN. In order to provide the possibility of testing other material combinations, the preload was considered to vary between 40 and 60kN.

The solution found to allow the existence and variation of the preload is shown in Figure 10.

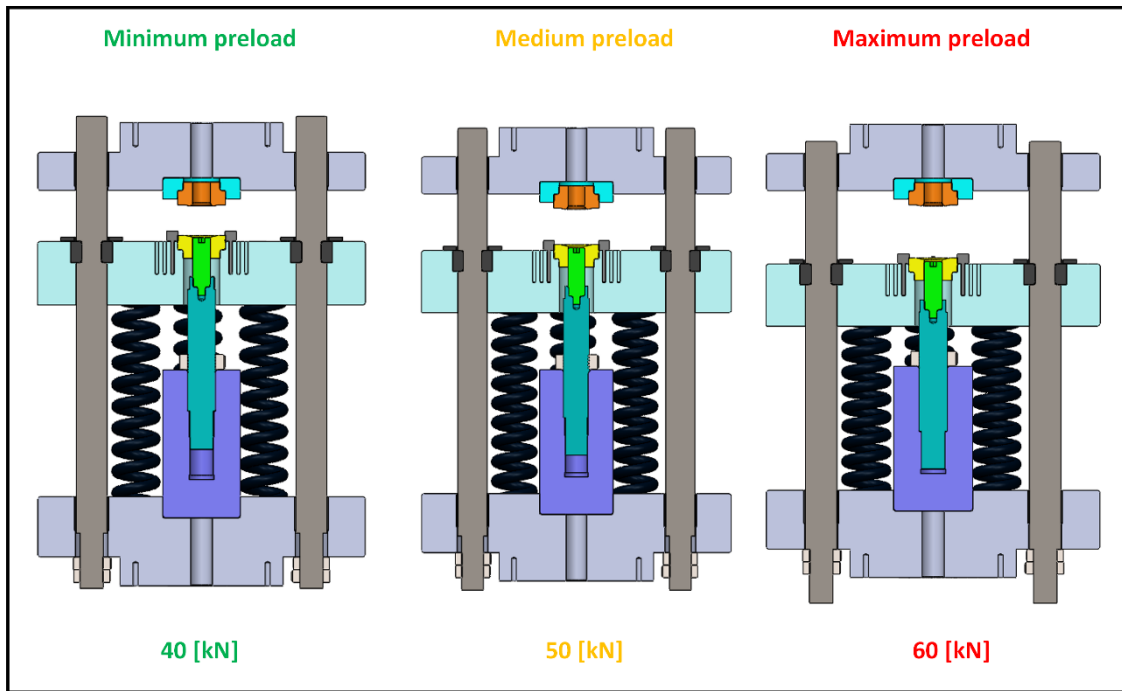


Figure 10- Schematic representation of the preload variation in the hole hemming tool.

This solution involves the use of springs, guide pillars with thread and keys. The preload is ensured by the springs, because after the upper die comes into contact with the sheet, the punch can only move upwards when the force exerted by the springs is overcome. In this way, by controlling the preload of the springs, the preload required in the process is controlled. When screwing or unscrewing the nuts on the respective guide pillars, the middle plate will approach or move away from the lower plate (since middle plate is fixed to the guide pillars through the keys), which causes the springs to be compressed or expanded, allowing to adjust the pre-load.

Springs play a very important role in preload adjustment; therefore they should be carefully selected. For the selection of springs, the following requirements were considered:

- Total preload should vary between 40 and 60kN;
- Operation deflection should be at least 10mm (according to the distance that the punches move);
- Stiffness should be selected so that the increase in force per 10 mm deflection, which is required for flanging the hole of an outer sheet and bending the flange over the hole of an inner sheet, is not significant.

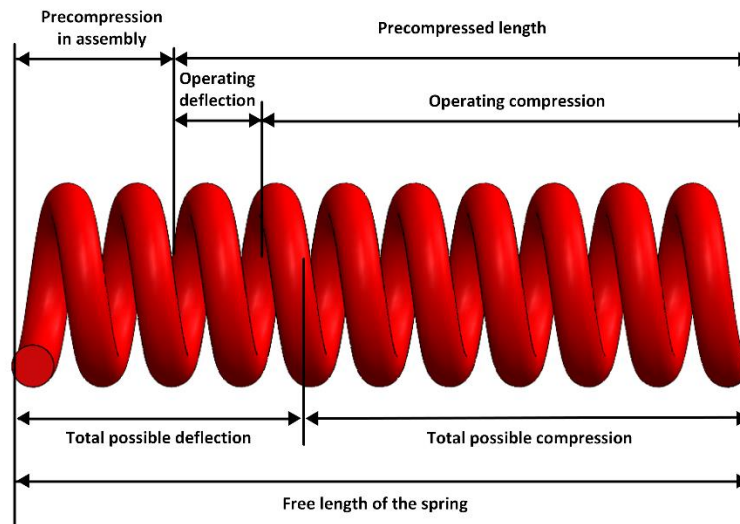


Figure 11- Compression spring and its properties.

According to these requirements, the six ISO 10243 XM198 springs were selected from *UNCETA*. The specifications of the springs are listed in Table 1. The spring life is approximately 1,500,000 cycles [14]. Figure 11 shows the most important properties of a compression spring and Table 2 the mechanical properties of the selected springs [14].

Table 2- Properties of the selected springs [13].

Selected spring	ISO 10243 XM198
Stiffness	128 [N/m]
Number of springs	6
Max deflection [l0 x 30%]	91.5 [mm]

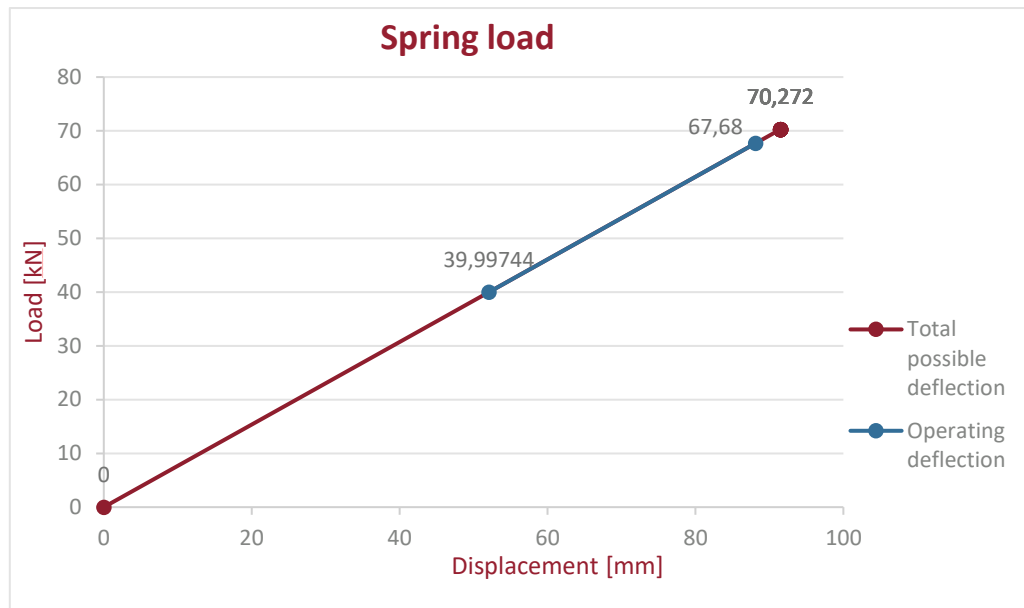


Figure 12- Load vs displacement of the set of springs.

With the use of this set of the springs, it is possible to satisfy all the above-mentioned requirements including the preload between 40 and 60kN, and the operating deflection of at least 10mm for each selected preload, and the small increase in force with 10mm deflection, Figure 12. It should be noted that, in the maximum case, where the preload is 60kN, the increase in force due to a deflection of 10 mm is only 7kN.

Concerning accuracy, alignment, supporting the passive tool components and fixing the tool to the testing machine, the most important components were the upper drive plate and the lower plate.

These two components guarantee the fixing of the hole hemming tool to the testing machine. This fixation is carried out through a bolt that screws simultaneously in the testing machine and the upper drive plate or lower plate, as shown in Figure 13.

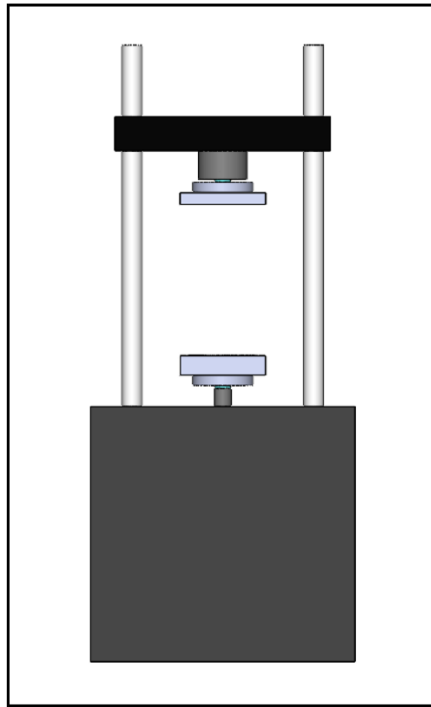


Figure 13- Schematic representation of the assembly of the lower plate and the upper drive plate in the testing machine.

Since these two components are bases for the other components, they play a very important role in the accuracy and alignment of the tool so that any dimensional and geometrical errors in these two components can cause misalignment between the punches and the dies and poorly distributed forces or even jamming of the tool components. In order to avoid these problems, special care was needed in the geometric dimensioning and tolerancing of these two components. It was also necessary to carry out several simulations using the SOLIDWORKS software to be able to evaluate the deflections and stresses applied to the components in a critical case of operation.

Another of the reasons that encouraged to perform the simulations for these two components was the fact that the testing machine used only has a fastening system through a pin (as mentioned above) that fixes the small area in the center of the plates. This fastening system can facilitate the bending of the plates.

The simulations were carried out with the SOLIDWORKS Simulation in the static mode. Both components were manufactured from medium carbon construction steel F10, so its mechanical properties, Table 3, were assigned to the simulations [15].

Table 3- Mechanical properties of F10 from Ramada Aços [14].

Tensile stress	≥ 590 [MPa]
Yield stress	≥ 330 [MPa]
Young modulus	200 [GPa]

The fixing method is the same in both the components (fixed through the M30x2 thread) while the applied forces are different, Figure 14. Starting with the upper drive plate, in this component, the only force applied is from the die holder. For the lower plate, it was assumed that the force applied is only from the springs, to obtain the most critical case of bending (note that in reality the force is applied by the springs and the height adjuster). In both cases, a total force of 100kN was considered, corresponding to the maximum capacity of the testing machine. This value was considered, because in this way the use of these structural components in any other process with this testing machine is guaranteed.

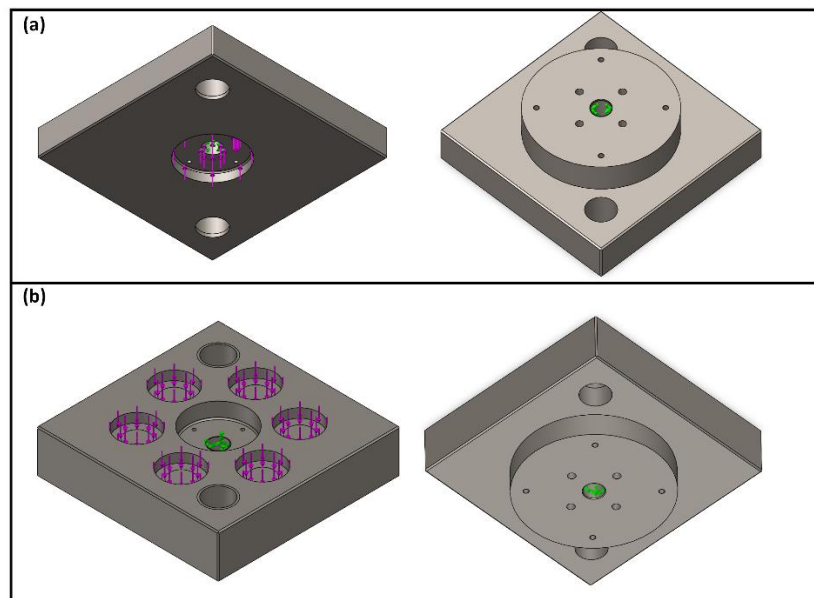


Figure 14- Boundary conditions and forces applied in the static simulations of the lower plate (a) and upper drive plate (b).

In order to guarantee alignment and centering, it was necessary to ensure that the maximum displacement of the guide pillar holes was less than $5 \mu\text{m}$ [6]. Another aspect that had to be ensured was that the stress should not exceed the yield strength of the material. The results obtain from the simulations are presented in Table 4 and Figure 15.

Table 4- Results obtained from the static simulations of the lower plate and the upper drive plate.

	Lower plate	Upper drive plate
Max von Mises stress	100 [MPa]	56 [MPa]
Max displacement in holes	> 5 [μm]	> 5 [μm]

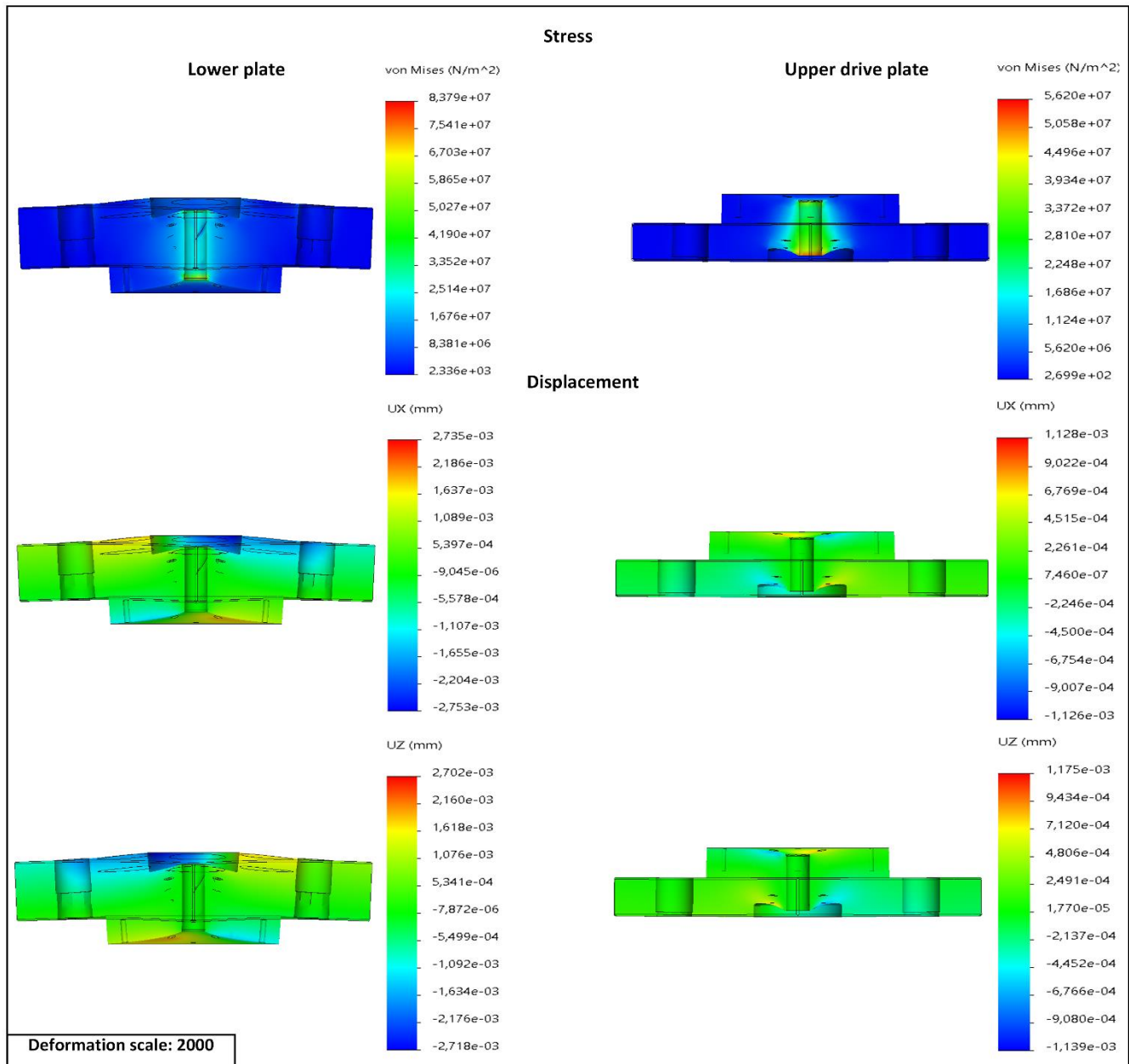


Figure 15- Distribution of Von Mises stress and displacement of the lower plate and the upper drive plate.

e. Passive tool components

The passive tool components of the hole hemming tool (Figure 16) were designed based on the following requirements:

- Accommodate the active tool components;
- Provide guidance and centering;
- Allow the adjustment of the position of the active tool components.

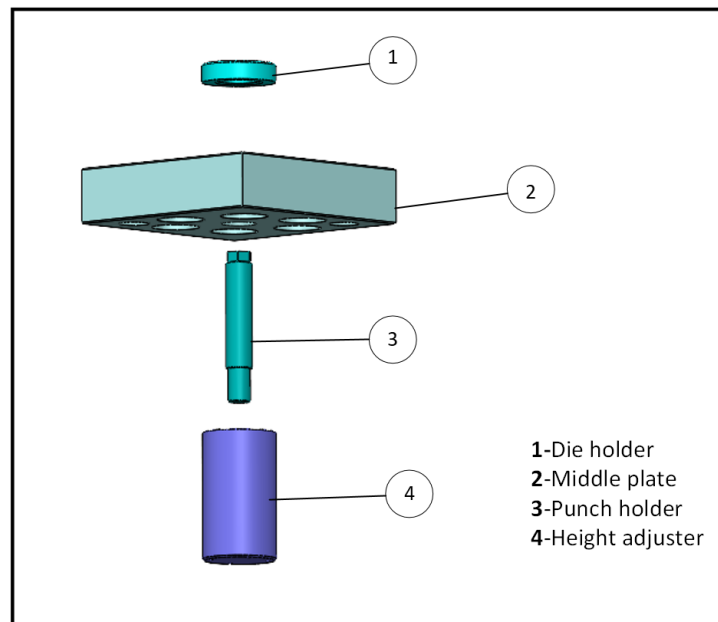


Figure 16- Schematic representation of the passive tool components of the hole hemming tool.

The need to adjust the position of the active components, especially the punches, arises from the possibility of varying the preload of the hole hemming tool. As previously explained, the preload variation is achieved through the precompression of the springs in assembly, leading to the change in the distance between the middle plate and the lower plate and, accordingly, the distance between the middle plate and the punch. Thus, it is necessary to adjust the position of the punch relative to the middle plate. The possibility of adjusting this position also allows the variation of some process conditions, namely the penetration distance of the punch.

The solution found for this adjustment involves dividing the typical punch holder into two components: the punch holder and the height adjuster. These components have, respectively, an external thread and an internal thread, so that it is possible to screw or unscrew the punch holder on the height adjuster, causing the punch to rise or fall. To guarantee that the defined

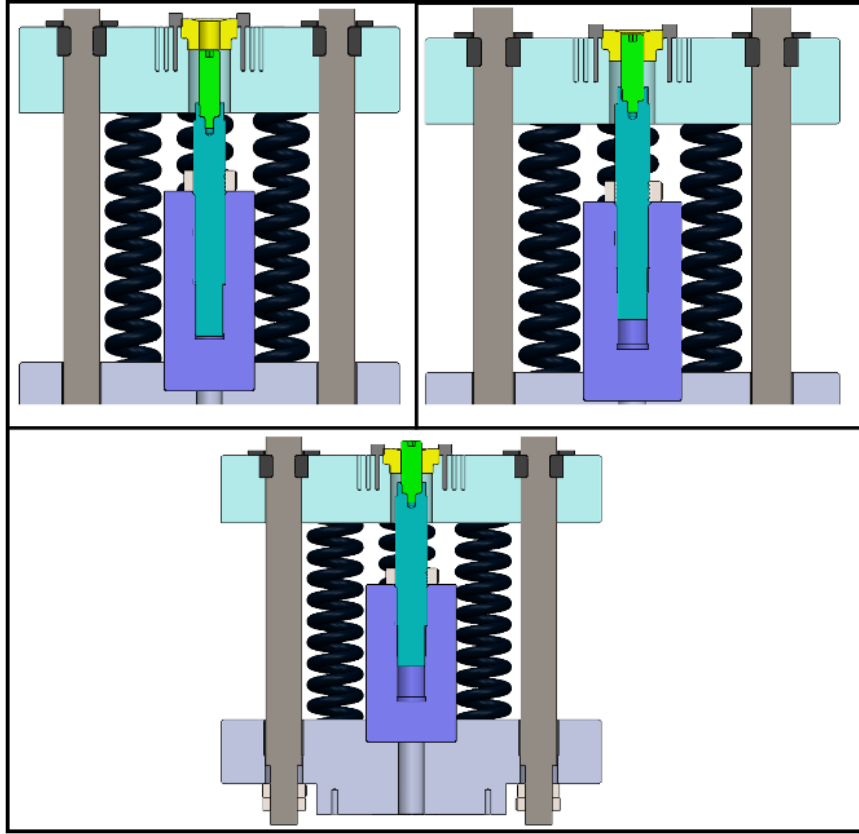


Figure 17- Schematic representation of the adjustment of the punch position.

position does not change during the process, it is necessary to employ a nut. The adjustment process of these components is illustrated in Figure 17.

For the height adjuster design, the aspects to be taken into account were the centering of the punch holder and the force applied by it to the thread. To guarantee the centering, a $\phi 30$ H7 hole was drilled before the thread, in this way the centering of the punch holder is performed by fitting it in this hole and not by using the thread. For the thread of this component, the instructions provided by *Morais et.al.* [16] were followed. First, the size of the thread and the pitch to be used were defined by the following equations [16] [17]:

$$R_{p0,2} = \frac{F}{A_s} \quad (100)$$

$$A_s = \left(\frac{\pi}{4}\right) * (d - 0,935P)^2 \quad (101)$$

Where $R_{p0,2}$ is the nominal yield strength of the material, A_s is the resistant section area, d is the diameter (larger diameter of the thread), P is the pitch and F is the applied force. Given that the material used for the height adjuster was AISI 1045 and an applied force of 100kN was

considered, the M36x4 thread was selected. In addition, attention was paid to the recommendation made for the implantation length ($j \geq 1.5d$) [16].

In addition to centering and force applied to the thread, the punch holder requires attention in a few more aspects including the compression exerted by the punch and buckling. For the centering and the force applied to the thread, the same procedure as the height adjuster was followed. The compression exerted by the punch on the punch holder can be calculated using the following equation [17]:

$$\sigma = \frac{F}{A} \tag{102}$$

Where σ is the yield stress, F is the applied force and A is the resistant section area. However, it was also decided to carry out simulations using the SOLIDWORKS simulation, in order to be able to more easily identify stress concentrations. The simulation was performed in Static mode with considering a force of 100kN. The mesh was refined in the areas prone to stress concentrations, Figure 18.

Table 5- Results obtained from punch holder's static simulations

Punch holder	
Max von Mises stress	680 [MPa]

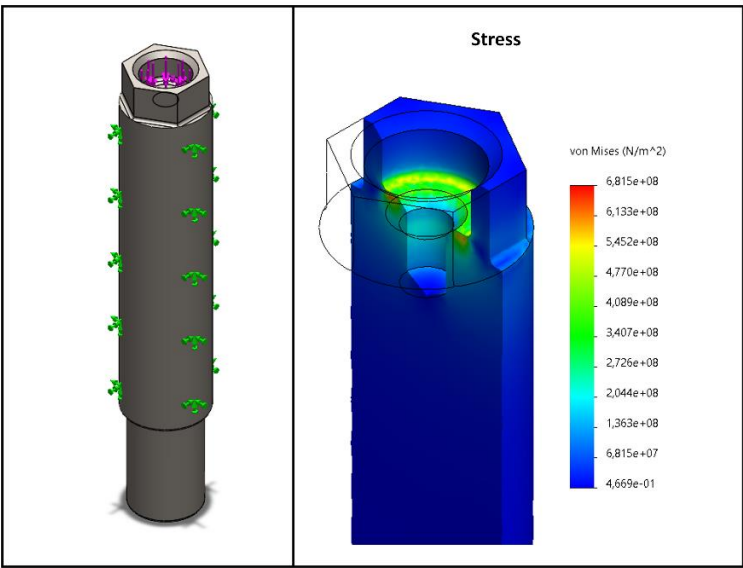


Figure 18- Boundary conditions, forces applied in the static simulation of the punch holder and the resulting von Mises stress.

A von Mises stress value of 680 MPa was obtained, Table 5 and Figure 18, applying a force 1.67 times greater than the real force. This guarantees safety due to the compressive forces.

Buckling was studied according to *Nielsen et.al.* [7] follows:

$$\sigma_{cr} = \frac{F_{cr}}{S} \rightarrow \sigma_{cr} < \alpha C \frac{\pi^2 EI}{L^2 S} \quad (103)$$

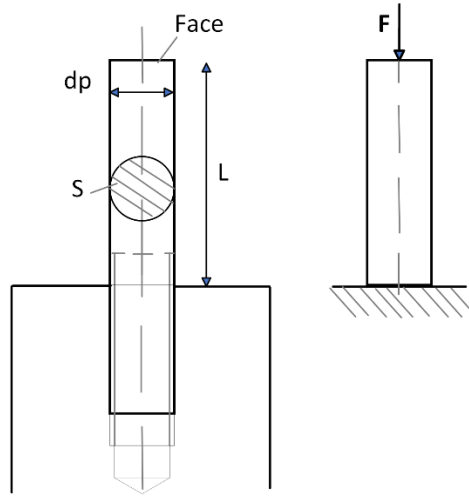


Figure 19- Notation and illustration of Euler buckling for the punch holder (Euler case I).

Where σ_{cr} is the critical instability stress, E is the elasticity modulus, I is the area moment of inertia, L is the length of the punch holder, α is a safety factor and C is a coefficient related to the punch holder support condition. Typical values of $C = 0.25$ are used in cases like that shown in Figure 19 where the punch head is clamped, and the face is free [7].

Taking now into consideration that the area moment of inertia:

$$I = \frac{\pi}{64} * d^4 \quad (104)$$

Considering the punch holder end to be free and neglecting the safety factor, equation can be rewritten as:

$$L \leq \sqrt{\frac{c \cdot \pi^2 \cdot E \cdot d^2 \cdot 4}{64 \cdot \sigma_{cr}}} \quad (105)$$

Since the material used is PM300 from *Ramada Aços*, we have a modulus of elasticity of [15]:

$$E = 205 \text{ GPa} \quad (106)$$

For this material the critical instability stress is [18]:

$$\sigma_{cr} = 800 \text{ MPa} \quad (107)$$

As for the punch holder diameter, d, it is 36 mm.

We can then get the following value:

$$L \leq 226 \text{ mm} \quad (108)$$

Corresponding this value to the maximum length that the punch holder can be found outside the height adjuster.

Buckling simulations were also carried out in the SOLIDWORKS software, in order to validate the results obtained, Table 6 and Figure 20.

In buckling mode simulation, the value to extract is the Load Factor:

$$\text{Load factor} = \frac{\text{estimated critical load}}{\text{applied load}} \quad (109)$$

Therefore, the higher the value of this parameter, the more safety our component presents and the closer to the unit, the closer to the critical case.

Table 6- Results obtained from punch holder's buckling simulations.

Punch holder	
Load factor	31

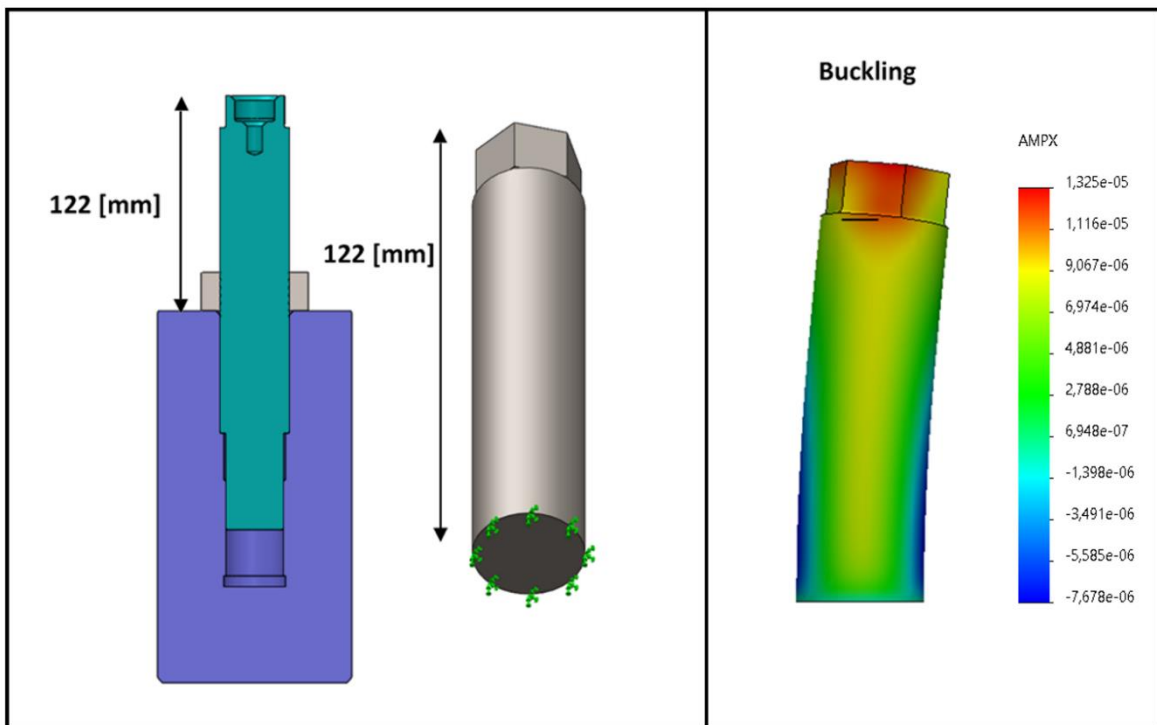


Figure 20- Buckling simulation of the punch holder.

As it is possible to see, this component is more critical than the previous one, so it was justified the selection of a superior quality steel.

Regarding the passive components of the hole hemming tool, another component that needed a special attention was the middle plate, as it is responsible for assembling the dies, it is one of the components that guarantees the alignment and centering between the punches and the dies and is a component that exhibits displacement during the process. Thus, any misalignment or deflection in this component can cause misalignment between the punches and the dies, poorly distributed forces or even jamming of the tool. In order to avoid these problems, special care was needed in the quotation, geometrical and dimensional tolerances of this component. It was also necessary to carry out several simulations in order to be able to evaluate the deflections and stresses existing in this component in a critical case of the operation.

To carry out the simulations, as previously mentioned, the SOLIDWORKS simulation was used in the static mode. The component was manufactured in F10 from *Ramada Aços* [15]. Since this component is not fixed and only supported by the springs and guided by the guide pillars, the definition of the applied forces and the boundary conditions is more default than the components analyzed so far. In order to obtain results consistent with reality, the simulations were carried out on the middle plate and guide pillars set together. In this way, the bottom part

of the guide pillars were fixed and the forces applied by the dies and the springs were considered in the middle plate, presenting a magnitude of 70kN for each one. This is the maximum force allowed for the springs. In this simulation, the misalignment due to the deformation of the holes created in the middle plate for the guide pillars was studied, which can prevent the middle plate from moving on the guide pillars.

Table 7- Results obtained from guide pillars and middle plate simulations.

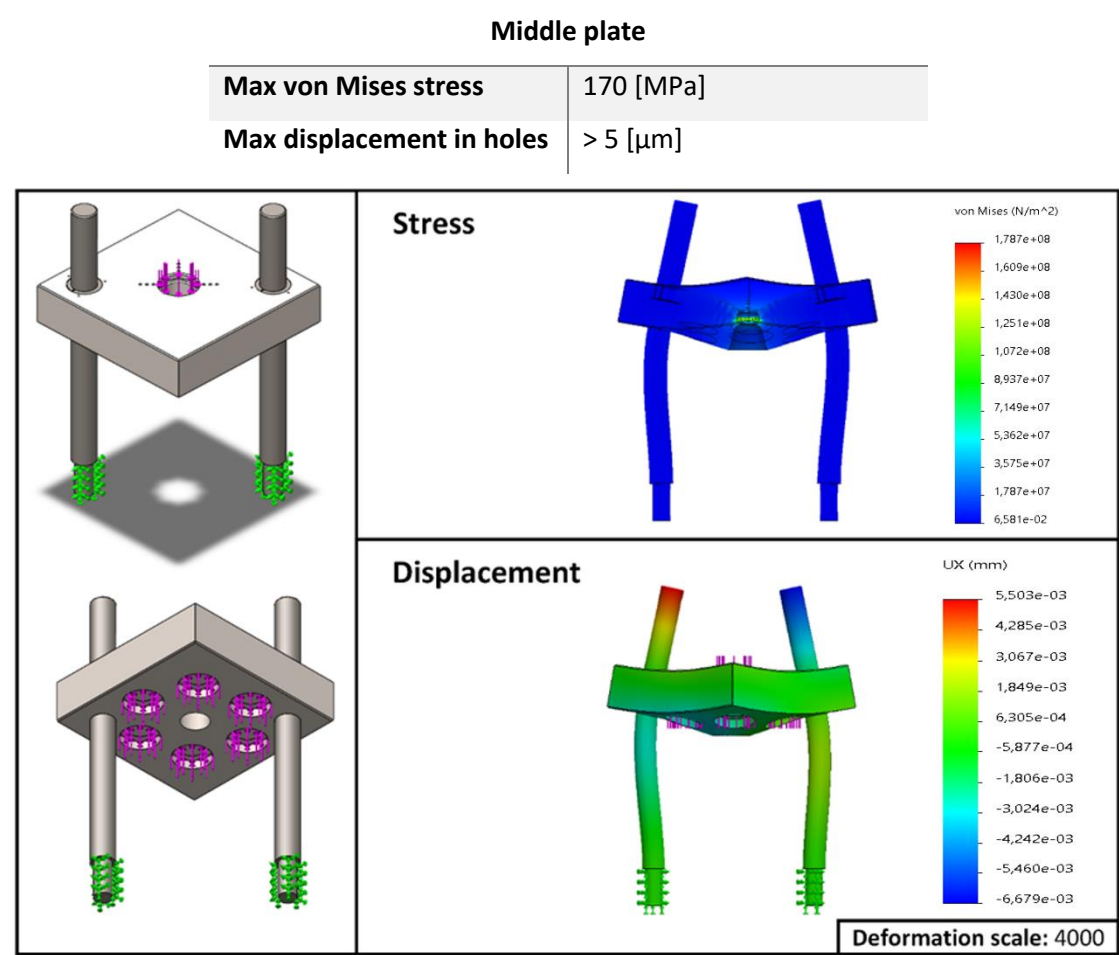


Figure 21- Boundary conditions and forces applied in the static simulation of the assembly of the middle plate and the guide pillars, and the resulting von Mises stress and displacement

Thus, it was necessary to ensure that the maximum displacement of holes was less than 5 μm . Another aspect that had to be ensured was that the yield strength of the material was not exceeded. As it is possible to see in Table 7 and Figure 21, both requirements were accomplished. To ensure that the displacement of the holes in the guide pillars did not exceed 5 μm , it was necessary to increase the thickness of the plate beyond the values observed in the initial catalogues.

f. Active tool components

The active tool components of the hole hemming tool, Figure 22, were designed based on the following requirements:

- Create the desired deformations;
- Easily interchange;
- Provide guidance and centering;
- Withstand the applied forces.

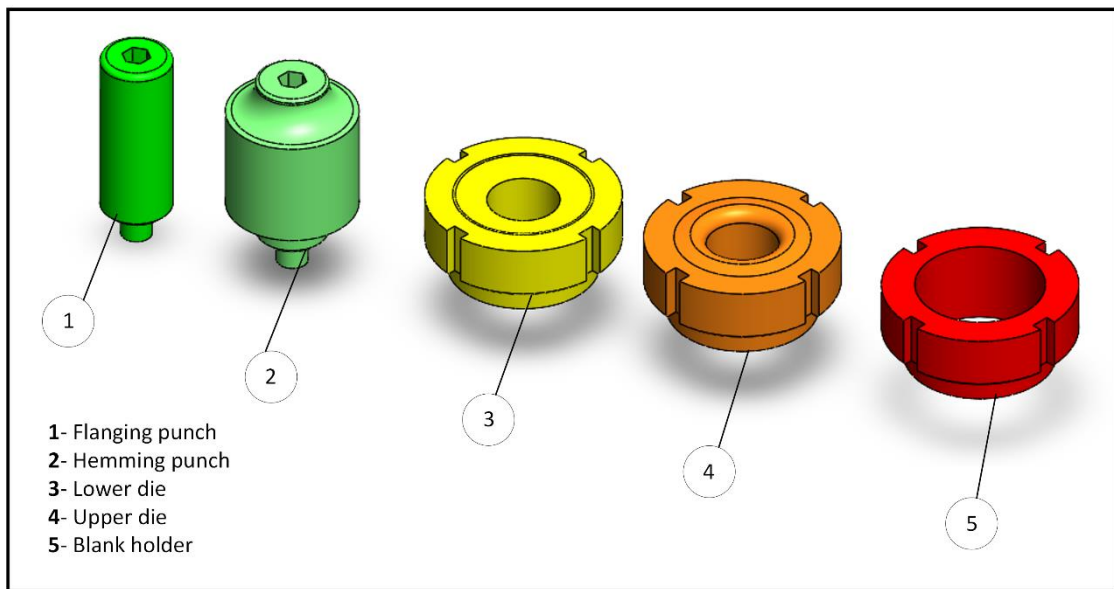


Figure 22- Schematic representation of the active tool components of the hole hemming tool.

Starting with the first requirement, creating the desired deformations, this was approached by *Pereira et al.*[11], who through several simulations in the ABAQUS software obtained the ideal geometries and dimensions of these components for the testing of the materials under study.

The following requirement, easy interchange of active components, is a requirement of great relevance to the process. As explained above, the hole hemming process is divided into two stages, being necessary to change the active components from one stage to the other, it is therefore essential that this replacement is easy and fast. This requirement also opens the door to testing different materials that require different active components. The solution found to satisfy this requirement involves the use of a thread both in the dies and in the punches, thus making it easy to assemble and disassemble these components. However, the use of a thread does not guarantee centering or alignment of the components, and the components can fracture in the thread due to the high forces applied. To solve this problem, the thread used will

be a loose thread, so the forces will be applied to the surface immediately above the thread and not to it. As for the alignment and centering of these components, this is guaranteed by the side surfaces (as can be seen in the Figure 23), which will fit in the die holder and middle plate, in the case of dies, and in the punch holder, in the case of punches.

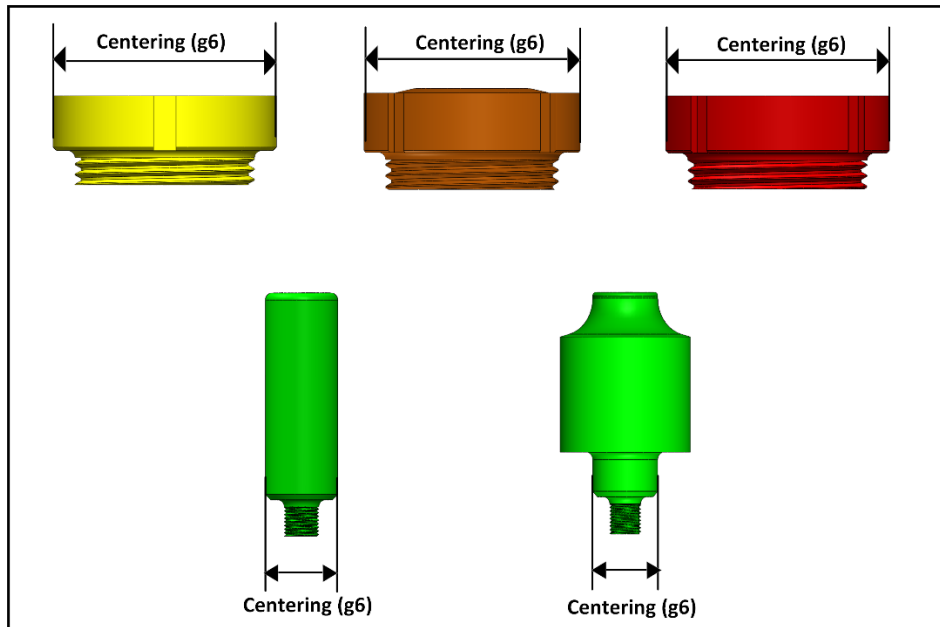


Figure 23- Adjustment used for the dies and punches.

Finally, in order to ensure that both the punches and the dies can withstand the applied forces, several simulations were performed in the SOLIDWORKS software.

For the dies, simulations were performed only with the aim of studying the effect of compressive forces on component stresses. In carrying out these simulations, the previously explained fixing method was considered, dies fixed on the surface immediately after the thread and not on the thread. As for the applied forces, the same magnitude value was considered for all, 100 kN, and these forces were applied to the contact surfaces. Care was also taken to refine the mesh for the zones conducive to stress concentration, Figure 24, 25 and 26.

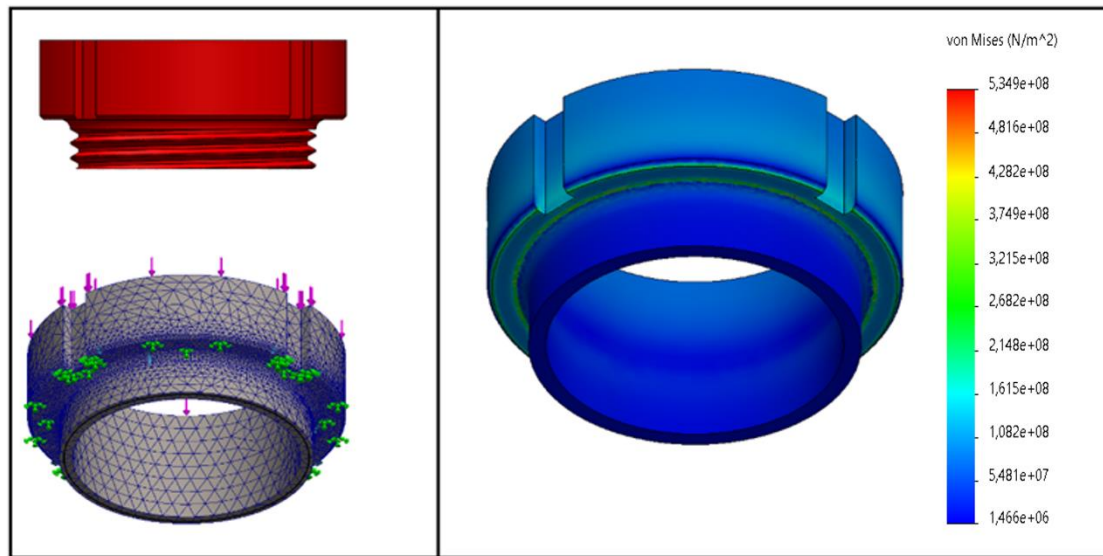


Figure 24- Static simulation of the blank holder.

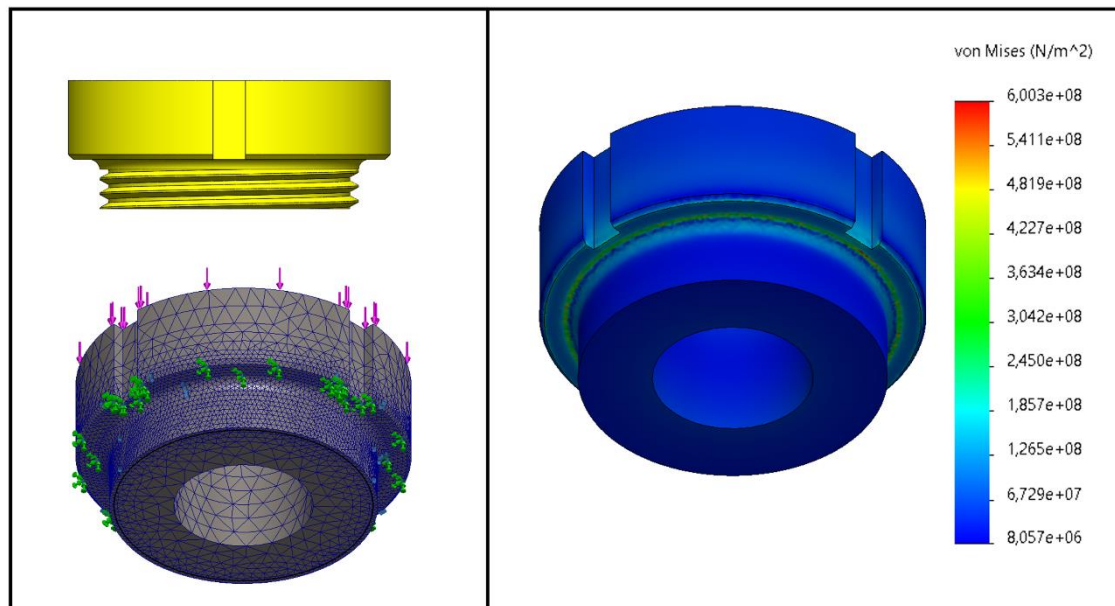


Figure 25- Static simulation of the lower die.

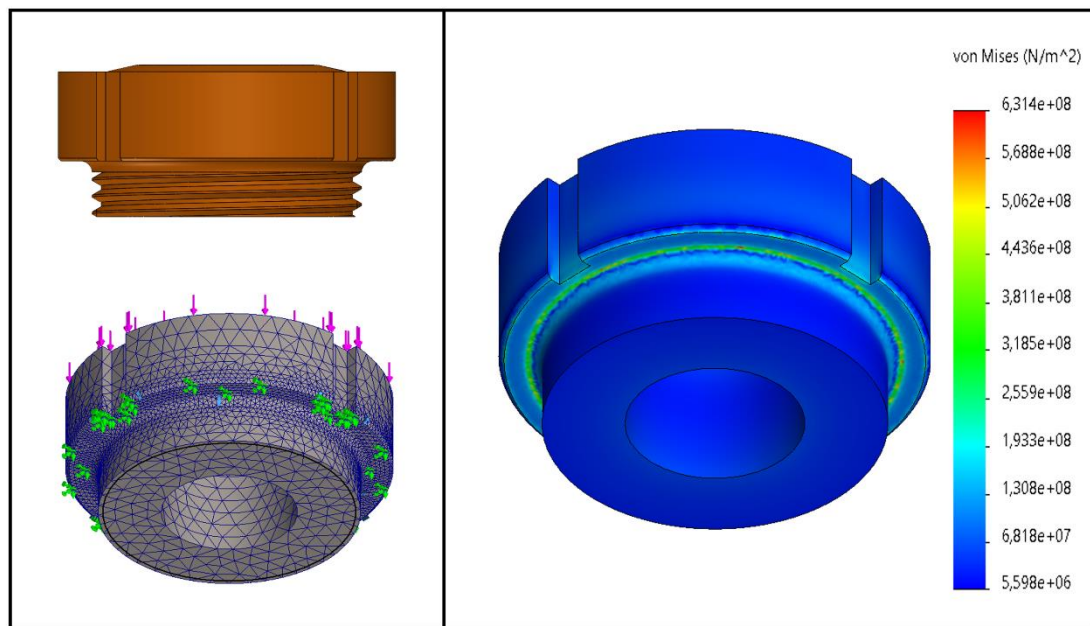


Figure 26- Static simulation of the upper die.

In the case of punches, simulations were carried out to study the effect of compressive forces on component stresses, as well as their effect on buckling. In carrying out these simulations, the previously explained fixing method was considered, punches fixed on the surface immediately after the thread and not on the thread. As for the applied forces, different magnitude values were considered for the punches. The values used for these components are based on the values presented at the beginning of this paper, with the application of a safety factor. These forces were applied to the contact surfaces and care was taken to refine the mesh to the areas conducive to stress concentration, Figure 27 and 28.

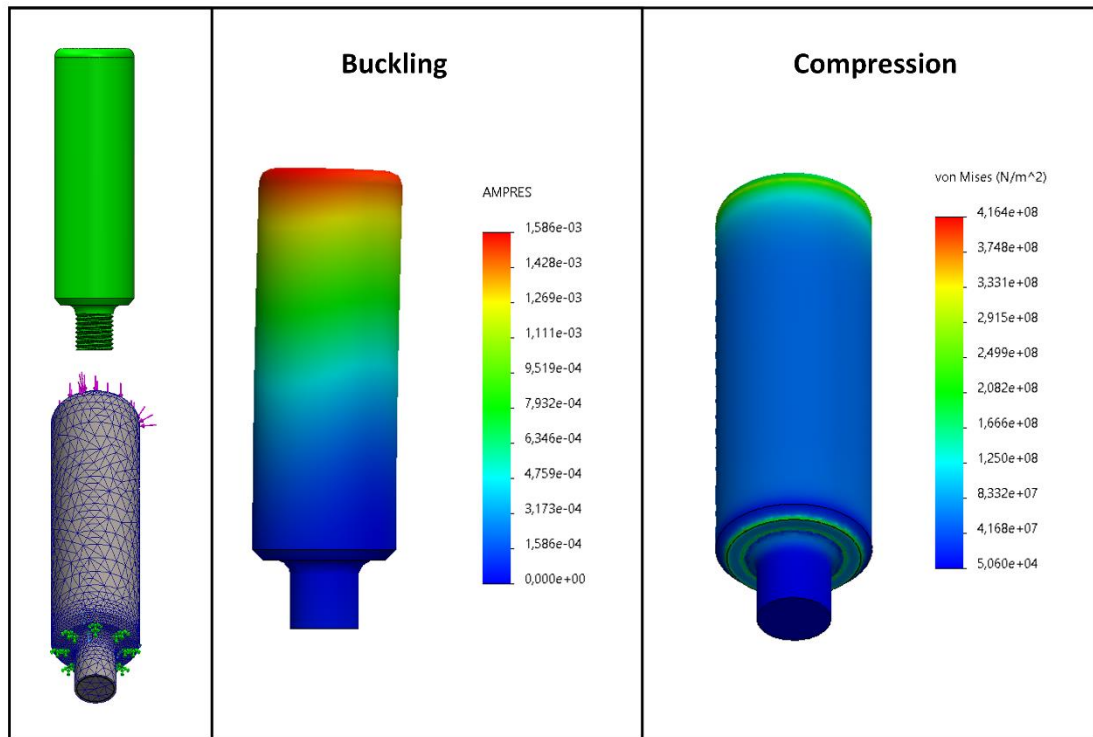


Figure 27-Buckling and static simulations of the flanging punch.

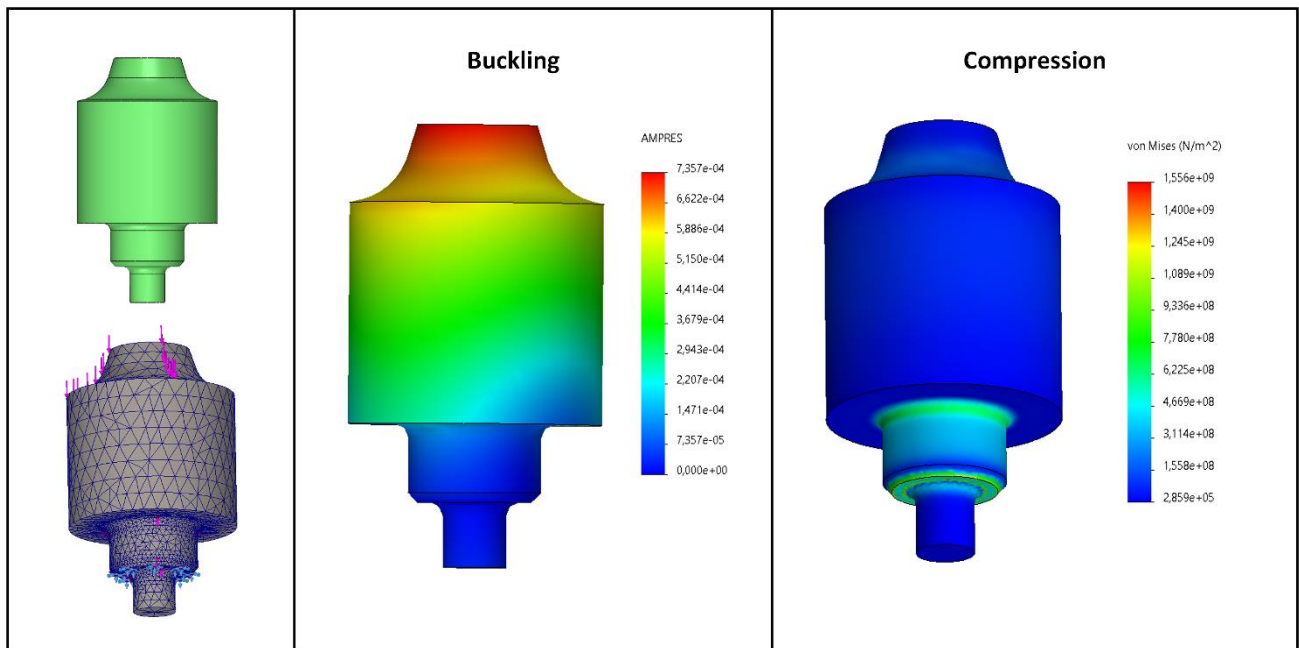


Figure 28-Buckling and static simulations of the hemming punch.

The Table 8 presents a summary table of the simulations performed.

Table 8- Summary of the simulations performed on the active tool components.

Component	Load applied	Max von Mises stress	Buckling (load factor)
Blank holder	120 [kN]	535 [MPa]	-
Lower die	120 [kN]	600 [MPa]	-
Upper die	120 [kN]	631 [MPa]	-
Flanging punch	20 [kN]	416 [MPa]	65
Hemming punch	120 [kN]	1556 [MPa]	21

Due to the high tensions existing in these materials, a more careful selection of the material to be used was necessary. The material C265 was used for the punches, a material from *Ramada Aços* recommended for the following applications: dies and punches; blades for guillotines; razors; woodworking tools [15]. This material is supplied with a minimum hardness of 255 HB, so it was necessary to carry out a heat treatment, tempering, in order to obtain adequate hardness for the process under study. The Table 9 shows the hardness used for each punch [19].

Table 9-Hardness and equivalent tensile stress of the punches.

Component	Hardness [HV]	Tensile stress [MPa]
Flanging punch	430 - 460	1385 – 1485
Hemming punch	600 – 650	1995 – 2180

Regarding the dies, it was decided to use a different material, which does not require a heat treatment, to make their machining easier. Considering the von Mises stresses presented in Table 9, the material PM300 from *Ramada Aços* was selected. As previously seen, this material has a tensile strength of approximately 1000 MPa and a yield strength in the order of 800 MPa [18].

6. Tool fabrication

The tool fabrication of the hole hemming tool was divided into three steps. The first step was the mechanical drawings of each component of the tool. The most complex mechanical drawings were the drawings of the lower plate, middle plate and upper drive plate. In these components, it was considered a special tolerance for the position of the holes, in order to obtain a good

alignment between the three plates. Geometric tolerances of perpendicularity, flatness and coaxility were also considered, not only for these three components but for all the components. It was also necessary to carry out the assembly drawings of the hole hemming tool in first and second stages, Figure 30.

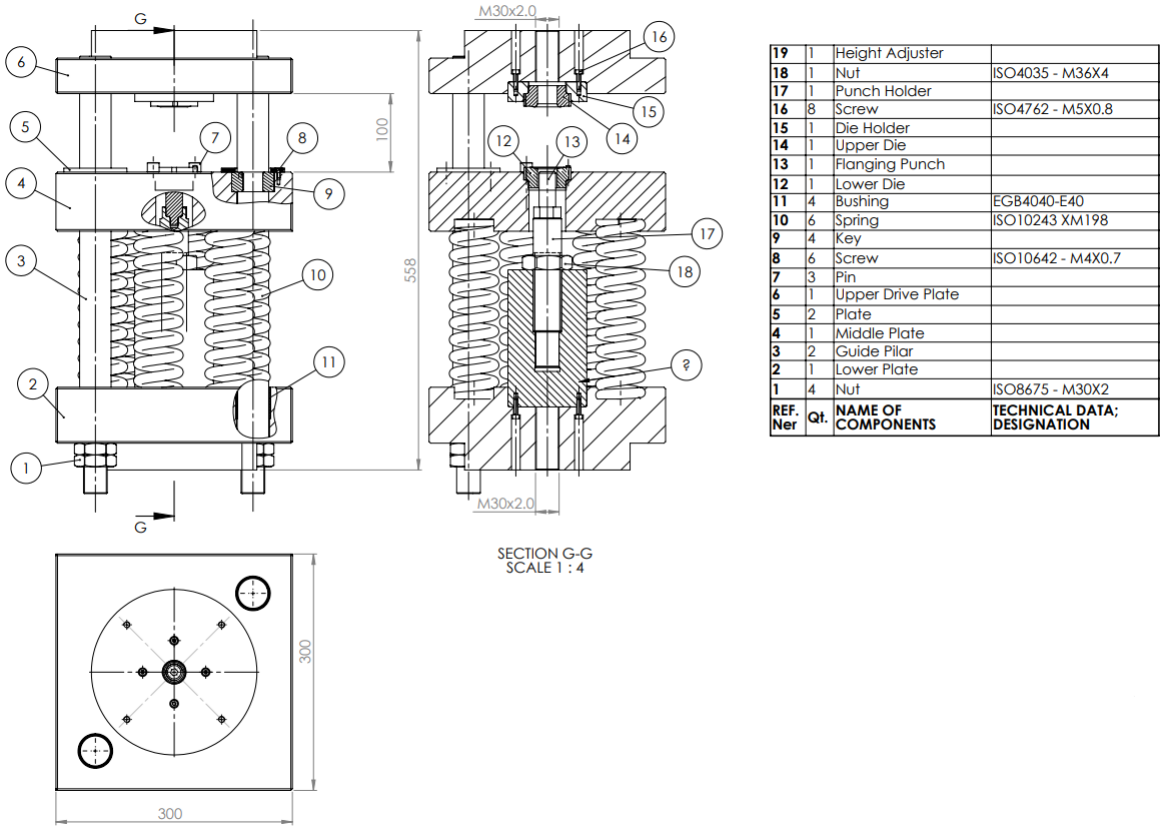


Figure 30- Assembly drawing of the hole hemming tool in the first stage.

The second step was the manufacture of the tool components. In this step, all the components were machined and in the case of the punches, they were also heat treated.

The last step was the assembly process of the hole hemming tool. Due to the weight of some components, namely the upper drive plate, lower plate and middle plate, it was necessary to assemble first on the machine the upper drive plate and the lower plate. And only after this, assembly the remaining components of the hole hemming tool, Figure 31.

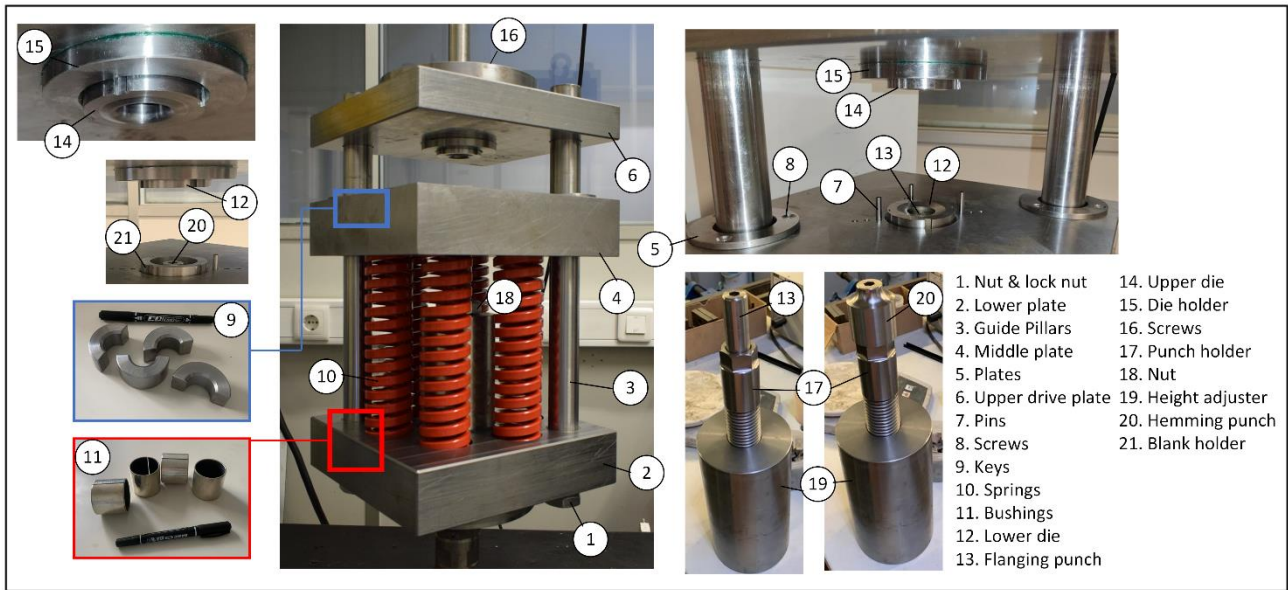


Figure 31- Hole hemming tool.

7. Evaluation of hole hemming tool

In order to validate the developed hole hemming tool, Figure 31, the first and second stages of the hole hemming process were performed on specimens with different dimensions, and the results were compared with numerical data obtained from the finite element analysis.

7.1. Test parameters

For the first stage of the process, hole flanging, the AA 6082-T4 with a thickness of 2 mm was deformed, (i.e., the material AA6082-T4 was the “outer sheet”), presenting a greater ductility. For this material, several specimens were tested, all with the same thickness, but with different hole diameter, including 16 mm, 17 mm and 18 mm.

However, as the purpose of this paper is the development of the hole hemming tool and not the development of the hole hemming process, only the tests performed for the specimens with the hole diameter of 17mm will be presented.

The test parameters used for the first stage of the process were:

- Hole diameter: 16, 17 and 18 mm;
- Preload: 40 kN;
- Displacement: 17 mm;
- Velocity: 6 mm/min.

For the second stage of the process, hemming, the material AZ31 with a thickness of 0.9 mm was first placed on the outer sheet. In other words, the material AZ31 was the “inner sheet”. For this material, all the specimens had a hole diameter of 29mm, Figure 32.

For this stage the test parameters used were slightly different:

- Hole diameter: 29 mm;
- Preload: 26 kN;
- Displacement: 12 mm;
- Velocity: 4 mm/min.

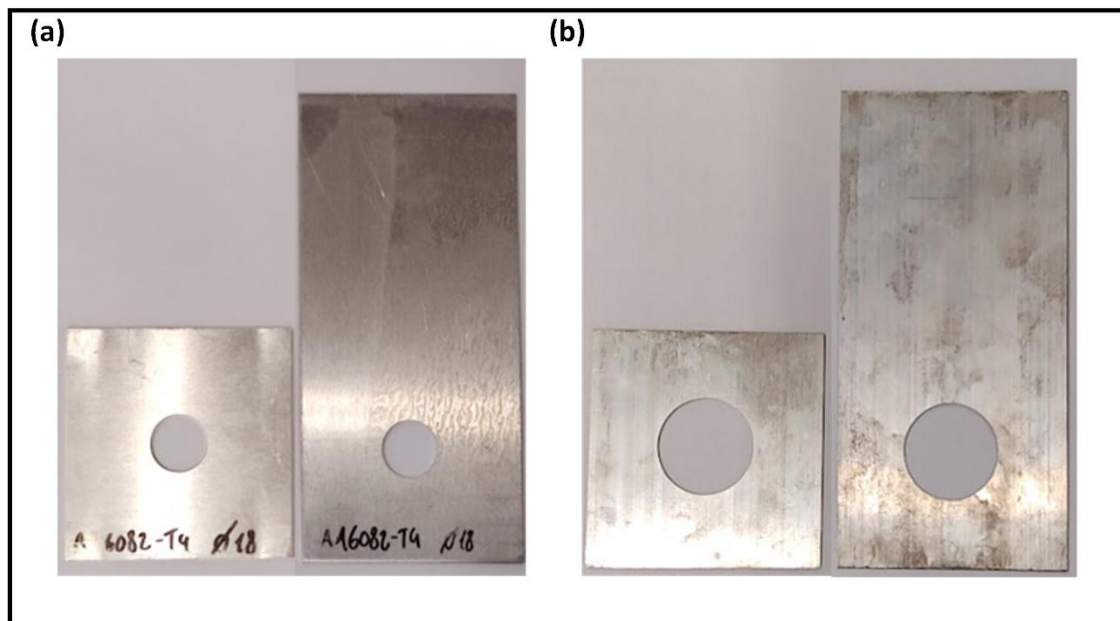


Figure 32- Specimens used for the hole hemming experiments: (a) outer sheet made from AA6082-T4 and (b) inner sheet made from AZ31 (width:70mm, length:70 or 150mm).

7.2. Hole hemming experiments

The experimental tests were performed on the Instron 8801 servo hydraulic testing machine with a capacity of 100 kN, which provided the evolution of force vs displacement during the experimental tests.

7.2.1. Hole flanging

The AA 6082-T4 specimen was first placed on the die. Three pins were used to fix the specimen in the right position. Then, the test was run, and the upper plate started moving down. As shown in Figure 33, the desired deformations were obtained, creating a flange that allows the assembly of the inner sheet and the creation of the mechanical interlock in the second stage.

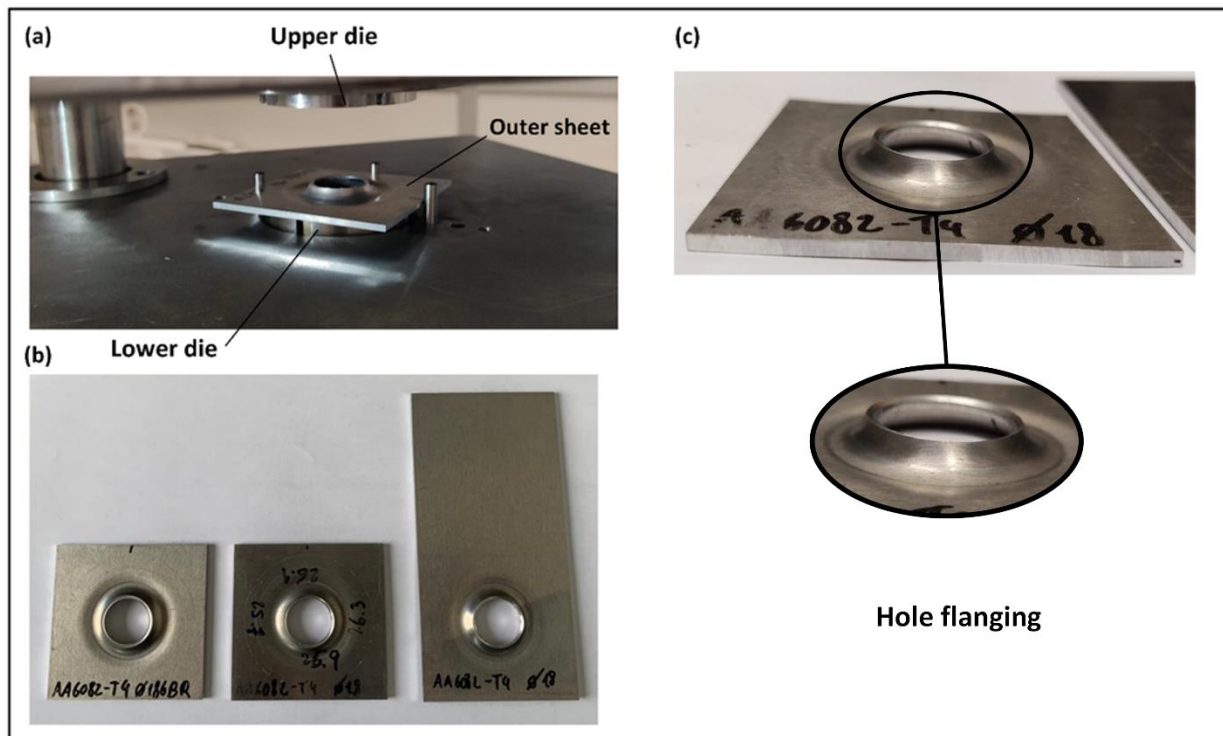


Figure 33- (a) Positioning of the outer sheet by three pins, (b) specimens tested in the hole flanging stage and (c) the resulting hole flange.

The force vs displacement curve was plotted in Figure 34. Note that this curve contains all the forces involved in the process.

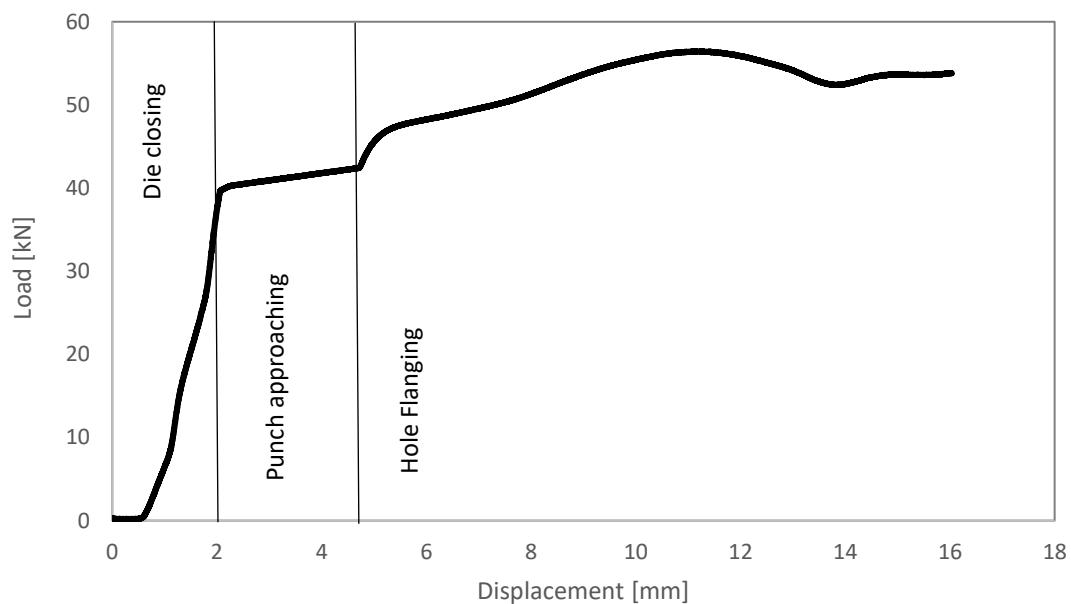


Figure 34- Force vs displacement curve of the hole flanging stage.

As it is possible to see in Figures 34, the force vs displacement curve can be divided into 3 parts: die closing, punch approaching and flanging.

The first part of the force vs displacement curve corresponds to the die closing. At this stage, there is only contact between the dies and the outer sheet, and the deformation of the sheet is only the result of the work of the dies. The graph is approximately a straight line.

The second part of the force vs displacement curve corresponds to punch approaching. This phase is defined by the preload of the springs, because for the punch to start contacting the outer sheet, this force must be overcome. That is, by setting a certain value for the preload, it is ensured that the outer sheet is compressed by the dies to that value before the punch comes into contact with the sheet. For the case under analysis, a preload in the order of 40 kN was defined, and as it is possible to observe when this value is reached, the slope of the graph is completely altered, this slope corresponding to the stiffness coefficient of the set of springs.

The last part of the force vs displacement curve corresponds to the performance of the flanging punch. The start of contact between the punch and the sheet can be identified through the variation in the slope of the graph.

Since the springs were not modelled in the simulation, the approximate value of the force applied to the flanging punch was obtained by removing the force applied by the springs from the total force at each moment and compared with the value obtained from the numerical simulation [11]. As can be seen, in Figure 35, the forces obtained in the flanging punch are in very good agreement with those predicted by the numerical simulation. Thus, the tool works very close to the ideal design and can create a flange with the desired quality in the first stage.

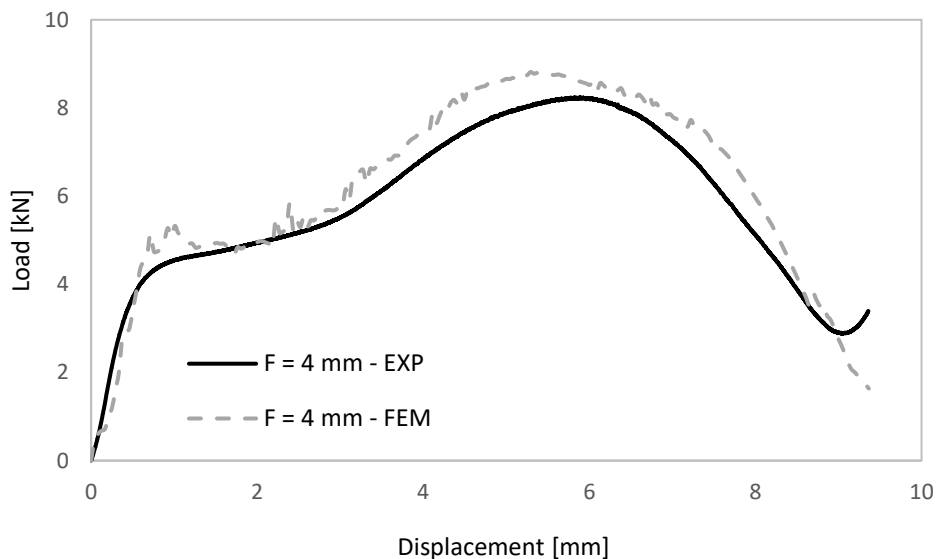


Figure 35- Numerical and experimental force vs displacement curves obtained for the flanging punch [11].

7.2.2. Hemming

To start the second stage of the process, it was first necessary to change the active tool components. Namely, replace the lower die with the blank holder, the upper die with the lower die and the flanging punch with the hemming punch. Then the AZ31 specimen is assembled on the AA6082-T4 specimen. Next both specimens were placed on the blank holder. The pins were used to fix the specimen in the right position. Then, the test was run, and the upper plate started moving down. As shown in Figure 36, the desired deformations were obtained, the flange is formed over the hole edge of the inner sheet, producing a mechanical interlock. The hole edge of the inner sheet is also slightly deformed and placed inside the indent.

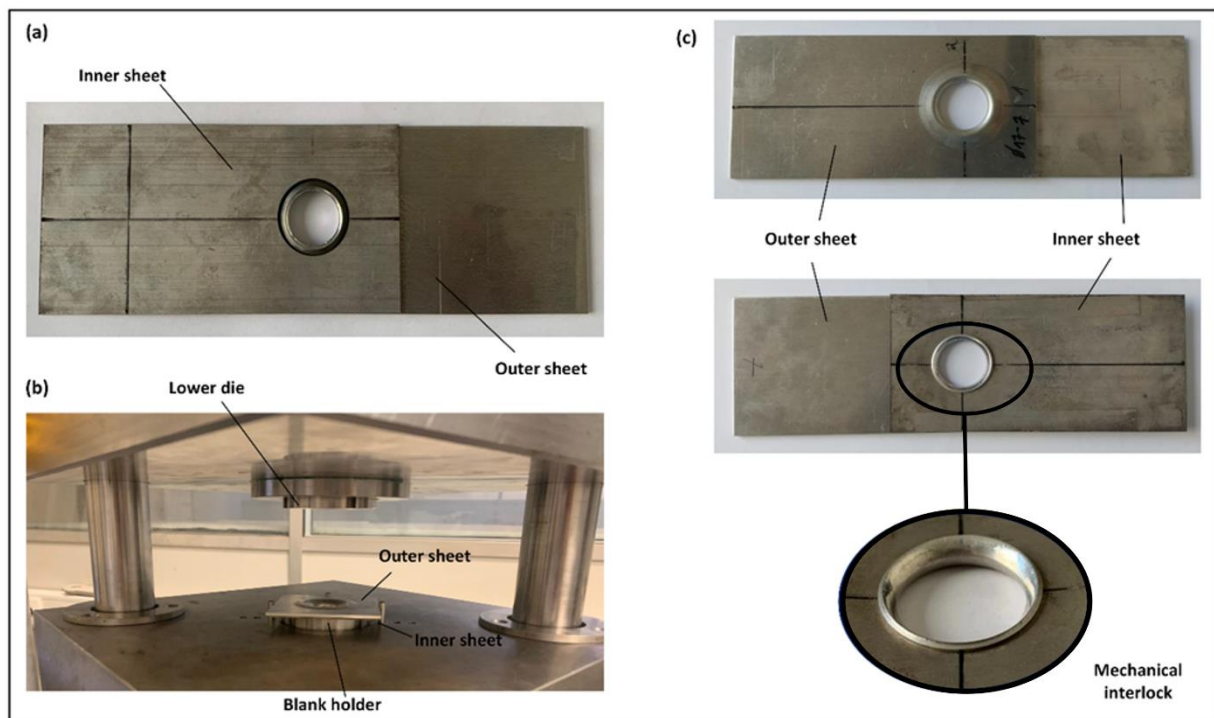


Figure 36-(a) Assembly of the inner sheet on the outer sheet, (b) positioning of the assembly on the blank holder by the pins in the hemming stage and (c) the resulting joint.

The force vs displacement curve was plotted in Figure 37. Note that this curve contains all the forces involved in the process.

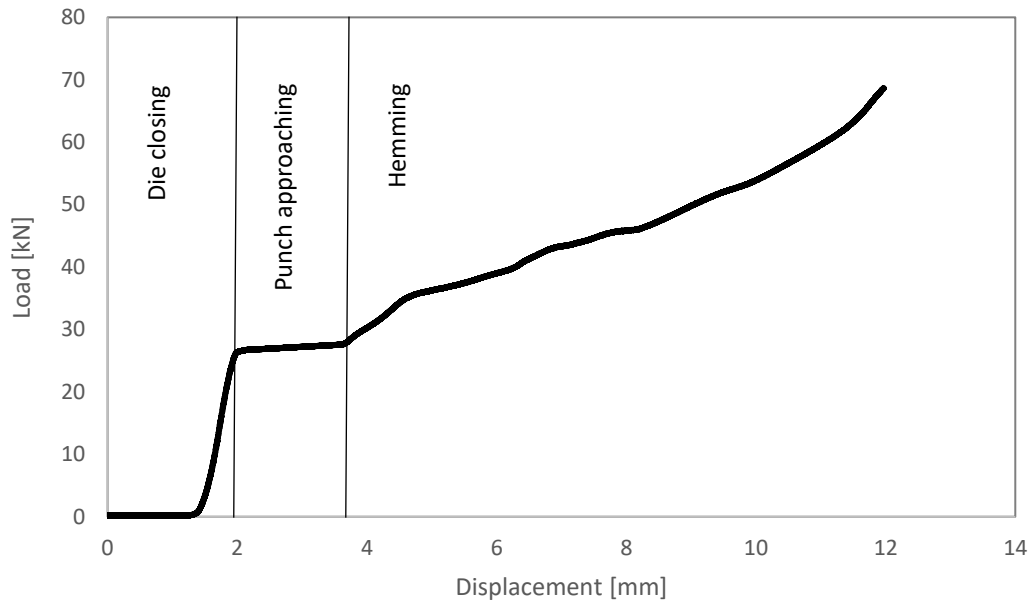


Figure 37- Force vs displacement curve of the hemming stage.

As it is possible to see in Figures 36, the force vs displacement curve can be divided into 3 parts: die closing, punch approaching and hemming.

As in the first stage, the first part of the the force vs displacement curve corresponds to the die closing. The beginning of the crushing of the plates by the dies is identified through the beginning of the increase in force. It is verified that this part of the graph corresponds to an approximately linear line. As in the hole flanging stage, this first part of the force vs displacement curve is the one with the greatest slope (largest increase in force for a given displacement).

The second part of the force vs displacement curve, the punch approaching, is identified by a sudden change in the slope of the graph. In this stage, the required preload was 26.7 kN, and as can be seen, the slope of the graph changes precisely at this value, with the slope of the graph being the stiffness coefficient corresponding to the set of springs used.

The last part of the force vs displacement curve begins when the hemming punch start contacting the specimens. This moment can be identified through the variation in the slope of the plot.

As in the previous stage, the approximate value of the force applied to the hemming punch was obtained by removing the force applied by the springs from the total force at each moment and compared with the value obtained from the numerical simulation, Figure 38 [11].

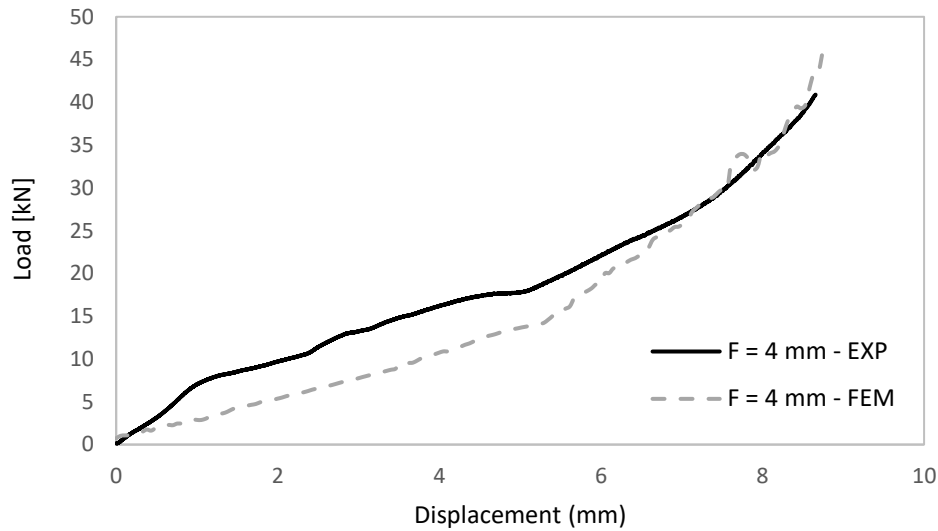


Figure 38- Numerical and experimental force vs displacement curves obtained for the hemming punch [11].

It is possible to observe that the forces obtained in the hemming punch are practically identical to those obtained through numerical simulations.

7.2.3. Hole hemming tool

As seen in the previous sections, the developed tool allows performing both stages of the hole hemming process, reaching the strength and quality requirements initially proposed.

Concerning the preload, it was possible to see that the values of preload defined were both achieved during the process.

Although initially a minimum preload of 40kN was proposed for both the stages, during the experimental tests, it was found that with the use of this preload for the hemming stage, the required force was approaching the capacity of the testing machine. As a result, four springs instead of 6 springs were used, which allows reducing the preload from 40kN to 26 kN, thus achieving the desired force for the hemming punch. This change in the preload was easily done, which is another evidence of the flexibility of this developed tool.

However, it was not possible to verify the variation of the preload up to its maximum value, being necessary to verify it in the future.

The adjustment of the punch position in relation to the dies was also found to be functional, allowing the use of different punch positions in the tests, especially at the beginning of each stage to achieve the best position. The replacement of punches and dies from one stage to the

other was easily carried out. Thus, it can be concluded that this tool meets all the considered requirements well.

8. Conclusions

A flexible tool that allows to carry out the hole hemming process for different materials, with different thicknesses and cross-section areas, under laboratory conditions, was successfully designed, manufactured, and tested.

The structural, passive and active tool components were designed in detail and assisted numerically to determine the loads acting on the different components and identify possible improvements. The structural components were designed in a way to allow an easy variation of the preload, an easy handling of the active and passive components of the tool. The structural components were then dimensioned in order to withstand the working conditions of the testing machine used. For these components, the use of 4 springs instead of 6 was also analyzed and verified. Regarding the passive components of the tool, they were designed with the aim of accommodating the active tool comments and adjusting the position of the punch. The active components of the tool have been designed with the modular concept in mind, in order to provide easy replacement of these components. These components were dimensioned to support the conditions mentioned for the hole hemming process. All the structural, passive and active components of the tool were successfully manufactured and implemented. In order to validate the tool, experimental tests of the hole hemming process were performed. The experimental tests verified all the previously mentioned characteristics

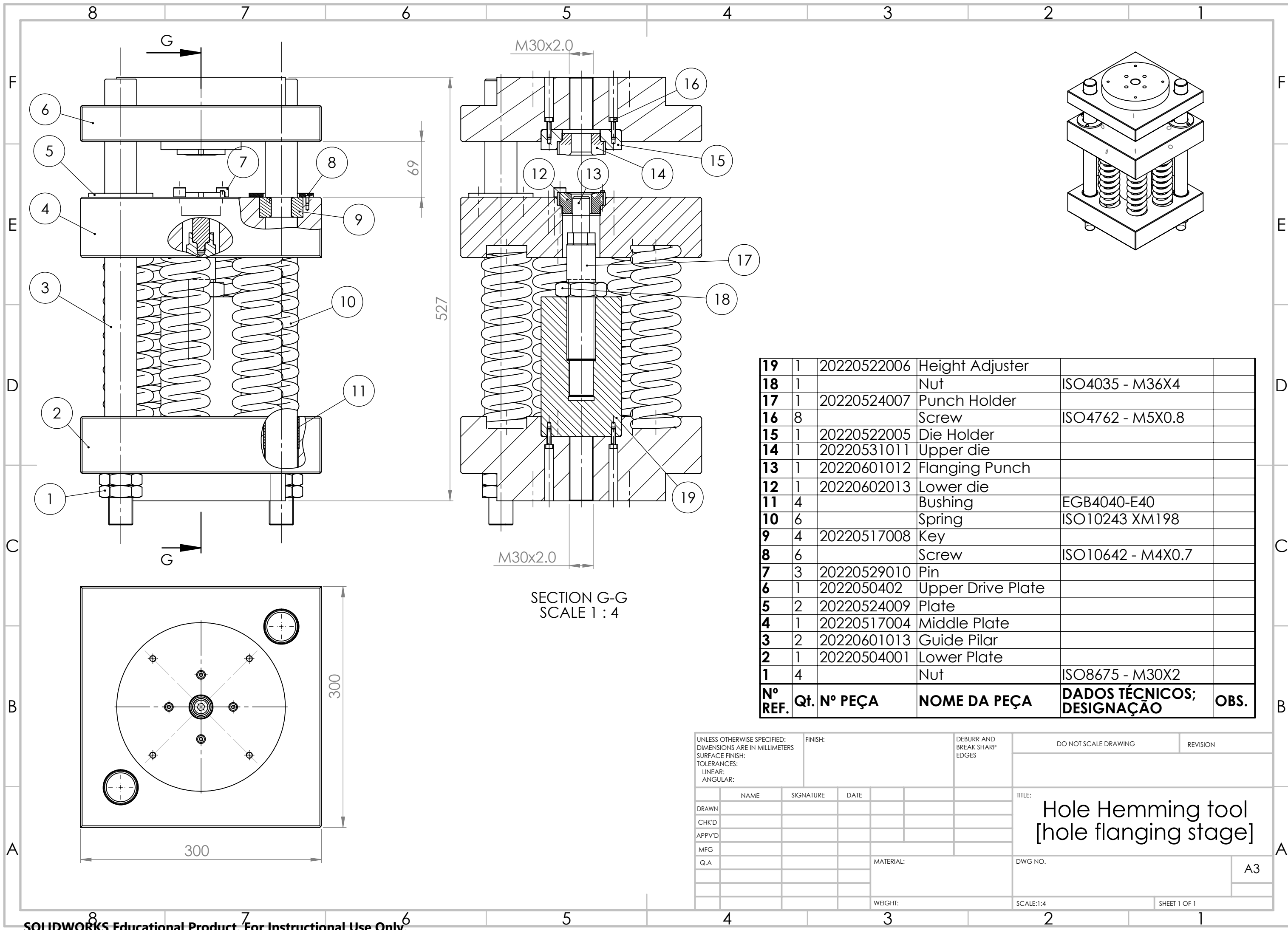
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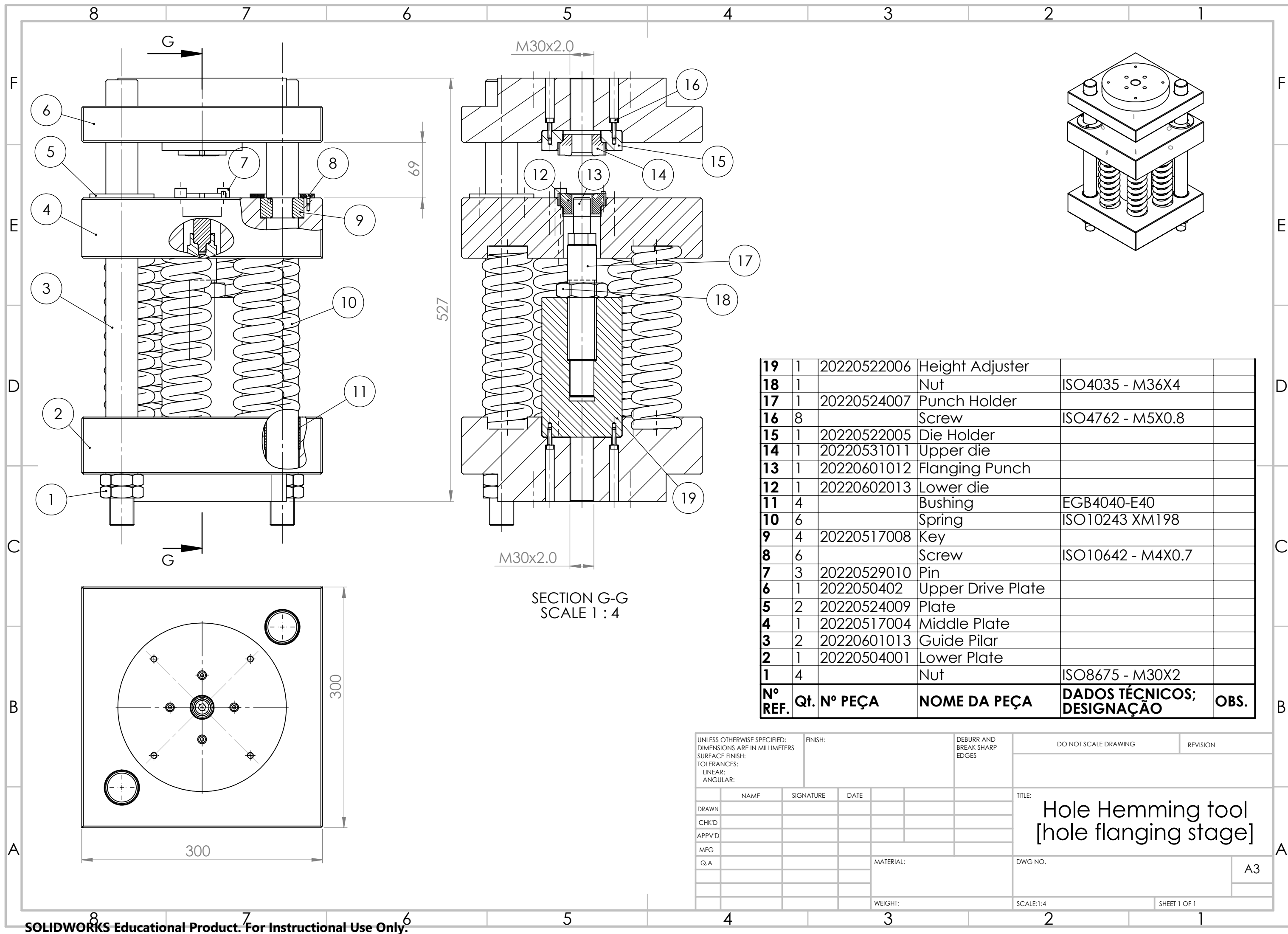
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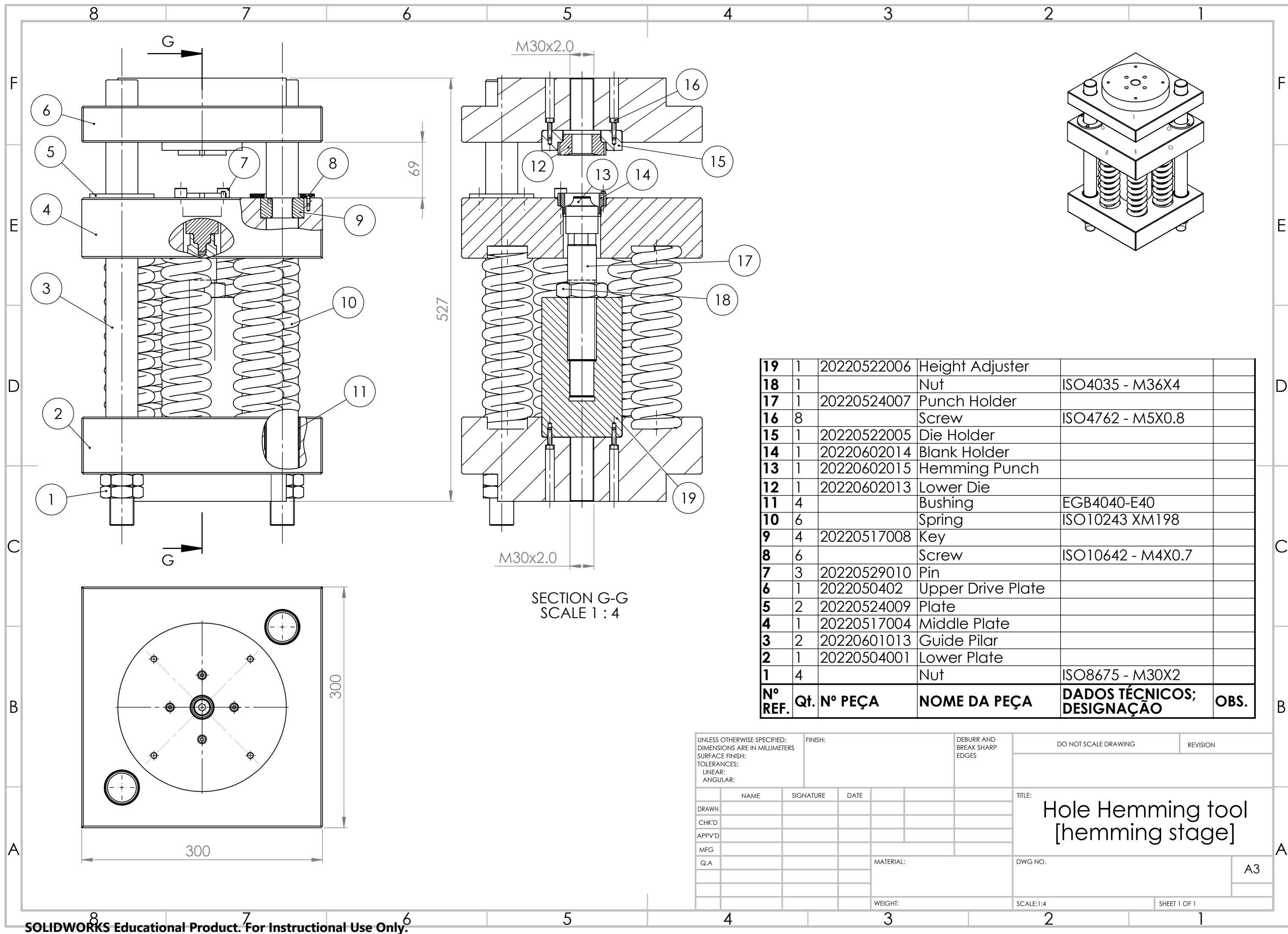
B Mechanical drawings

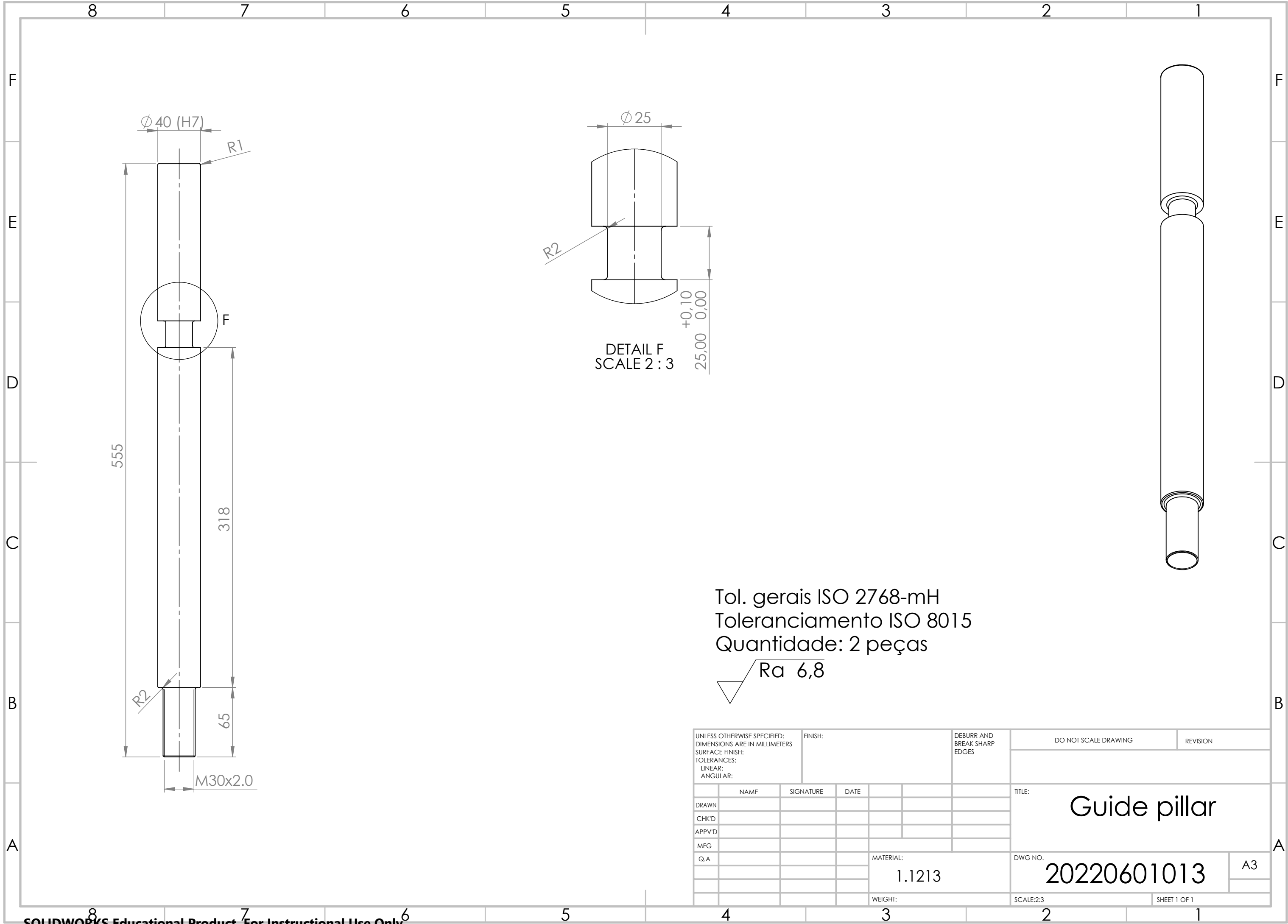


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18	1		Nut	ISO4035 - M36X4	
17	1	20220524007	Punch Holder		
16	8		Screw	ISO4762 - M5X0.8	
15	1	20220522005	Die Holder		
14	1	20220531011	Upper die		
13	1	20220601012	Flanging Punch		
12	1	20220602013	Lower die		
11	4		Bushing	EGB4040-E40	
10	6		Spring	ISO10243 XM198	
9	4	20220517008	Key		
8	6		Screw	ISO10642 - M4X0.7	
7	3	20220529010	Pin		
6	1	2022050402	Upper Drive Plate		
5	2	20220524009	Plate		
4	1	20220517004	Middle Plate		
3	2	20220601013	Guide Pilar		
2	1	20220504001	Lower Plate		
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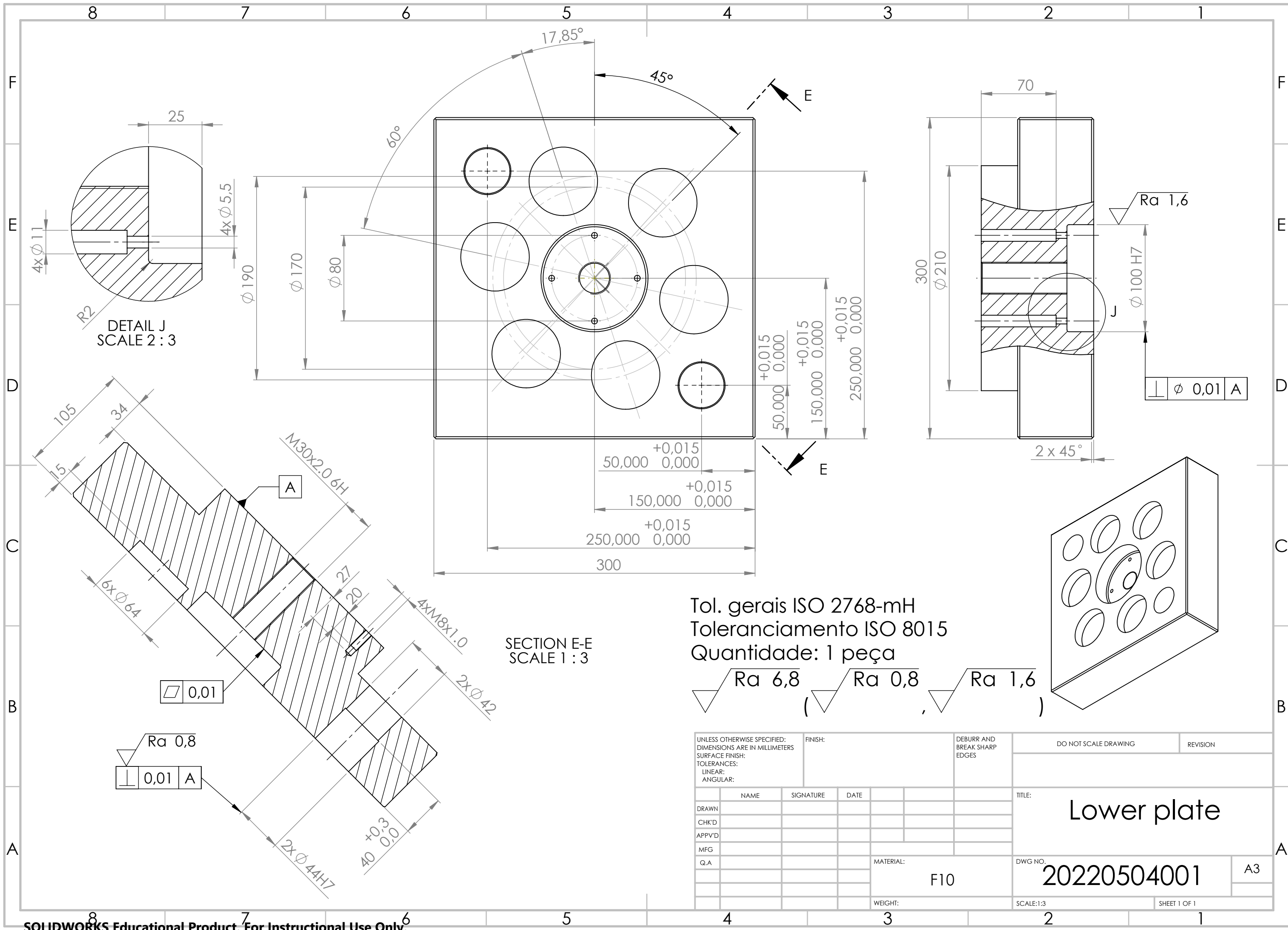
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		NAME		SIGNATURE		DATE						TITLE:	
DRAWN												Hole Hemming tool [hole flanging stage]	
CHK'D													
APPV'D													
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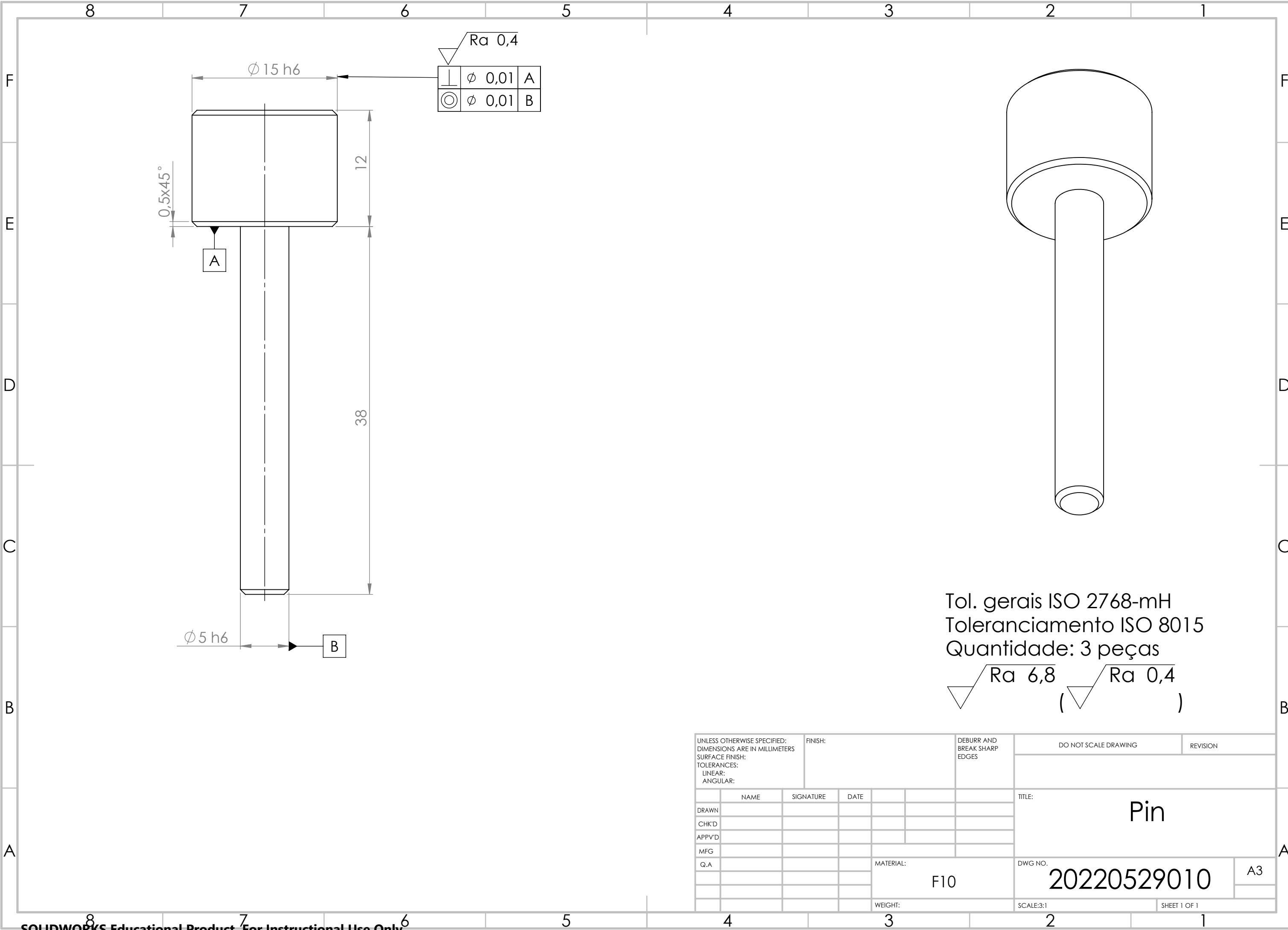




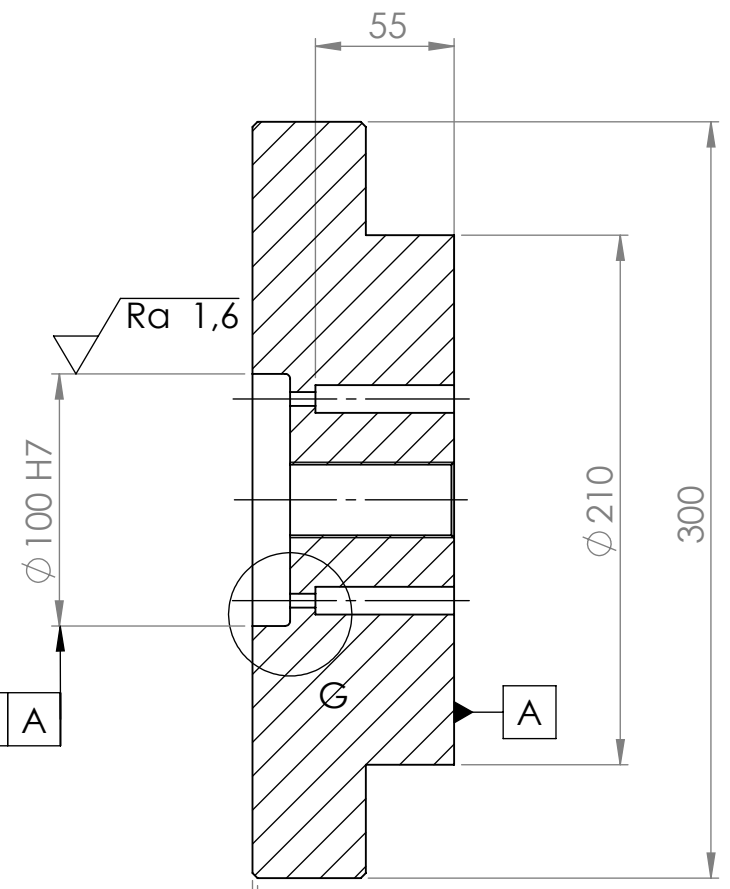
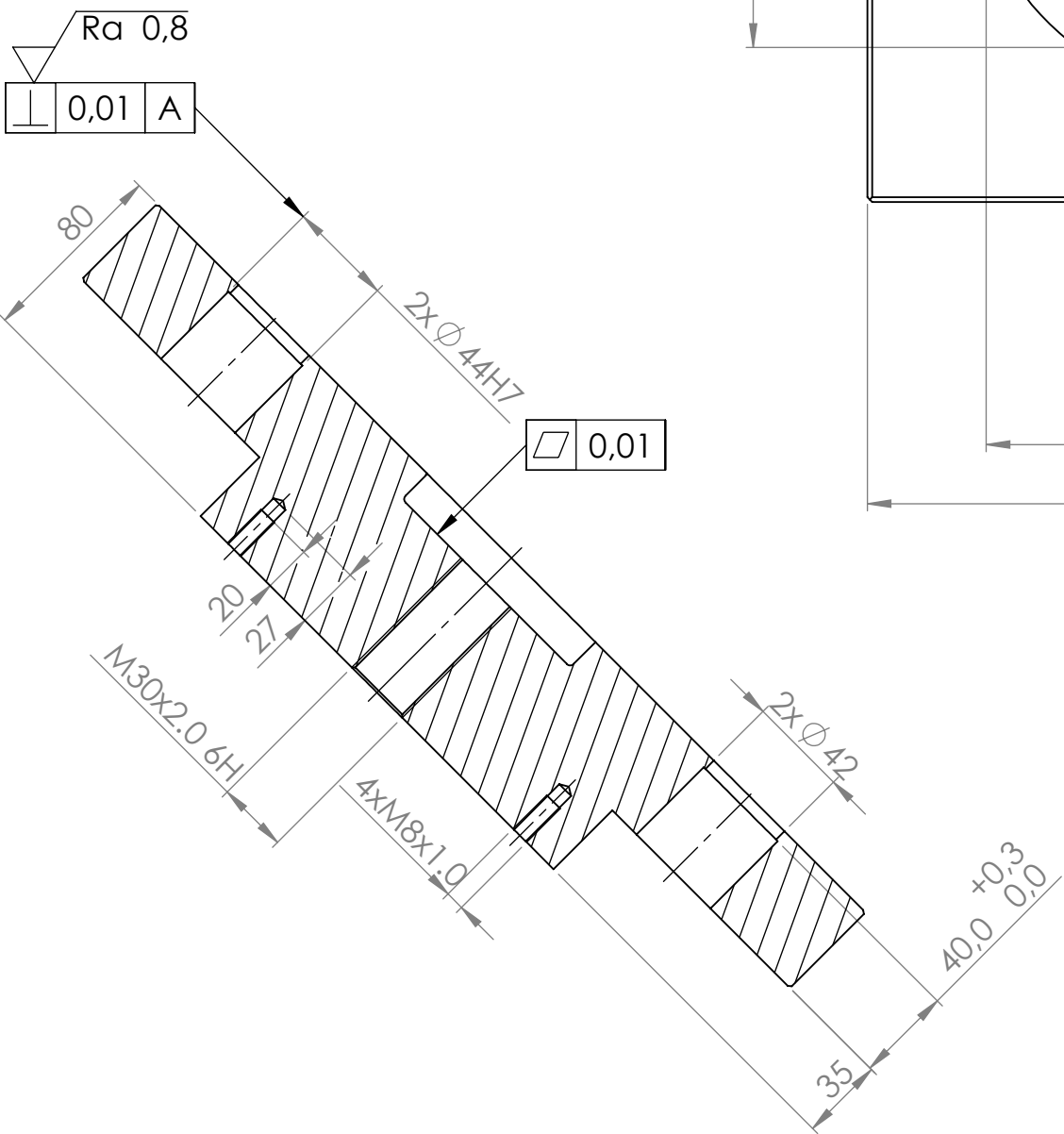
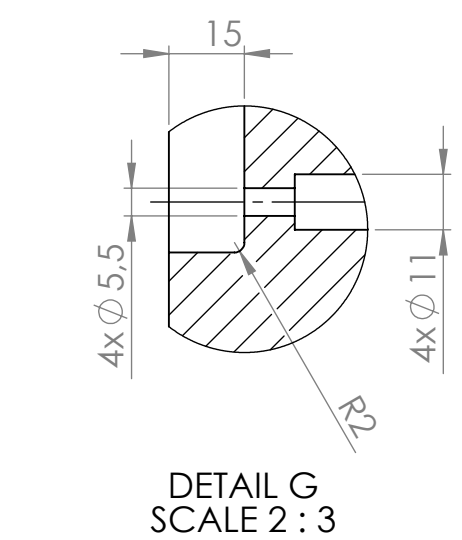
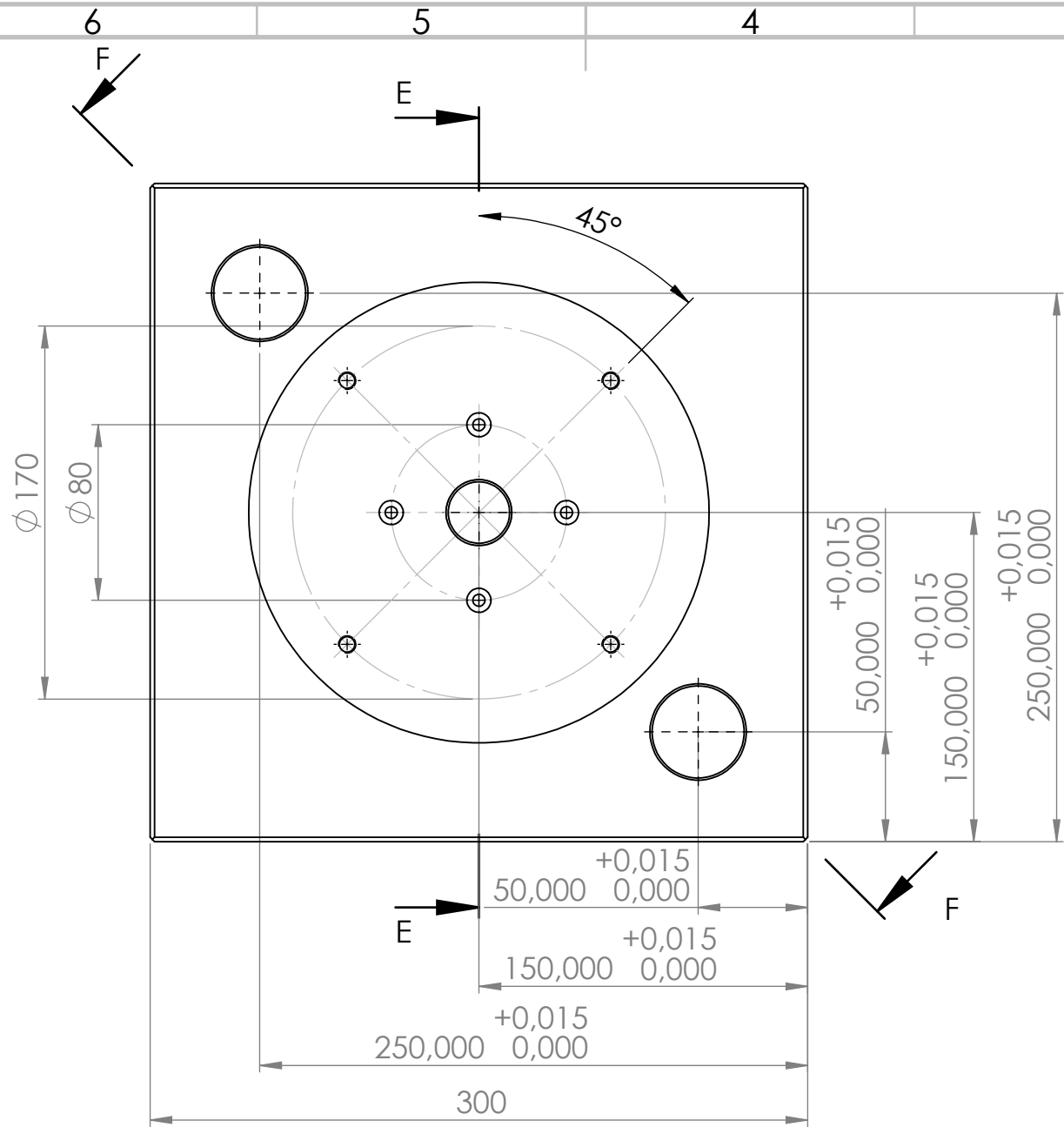


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:				FINISH:		DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING		REVISION
DRAWN	NAME	SIGNATURE	DATE				TITLE: Guide pillar		
CHK'D									
APPV'D									
MFG									
Q.A									
						MATERIAL: 1.1213		DWG NO. 20220601013	
						WEIGHT:		SCALE:2:3	A3
								SHEET 1 OF 1	





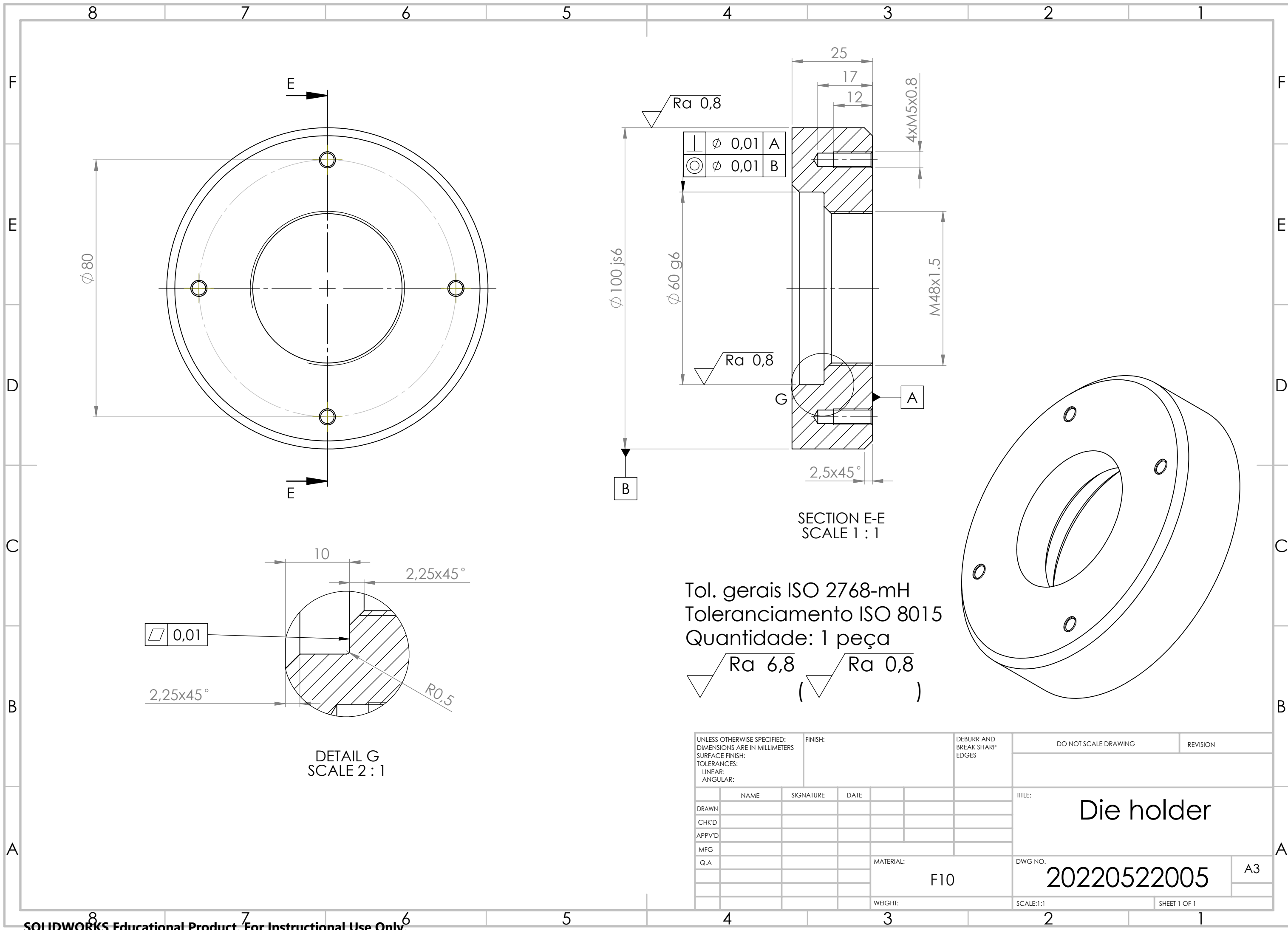
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:				FINISH:		DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING		REVISION		
	NAME		SIGNATURE		DATE				TITLE: Pin		
DRAWN											
CHK'D											
APPV'D											
MFG											
Q.A											
							MATERIAL: F10		DWG NO. 20220529010		A3
							WEIGHT:		SCALE:3:1		SHEET 1 OF 1

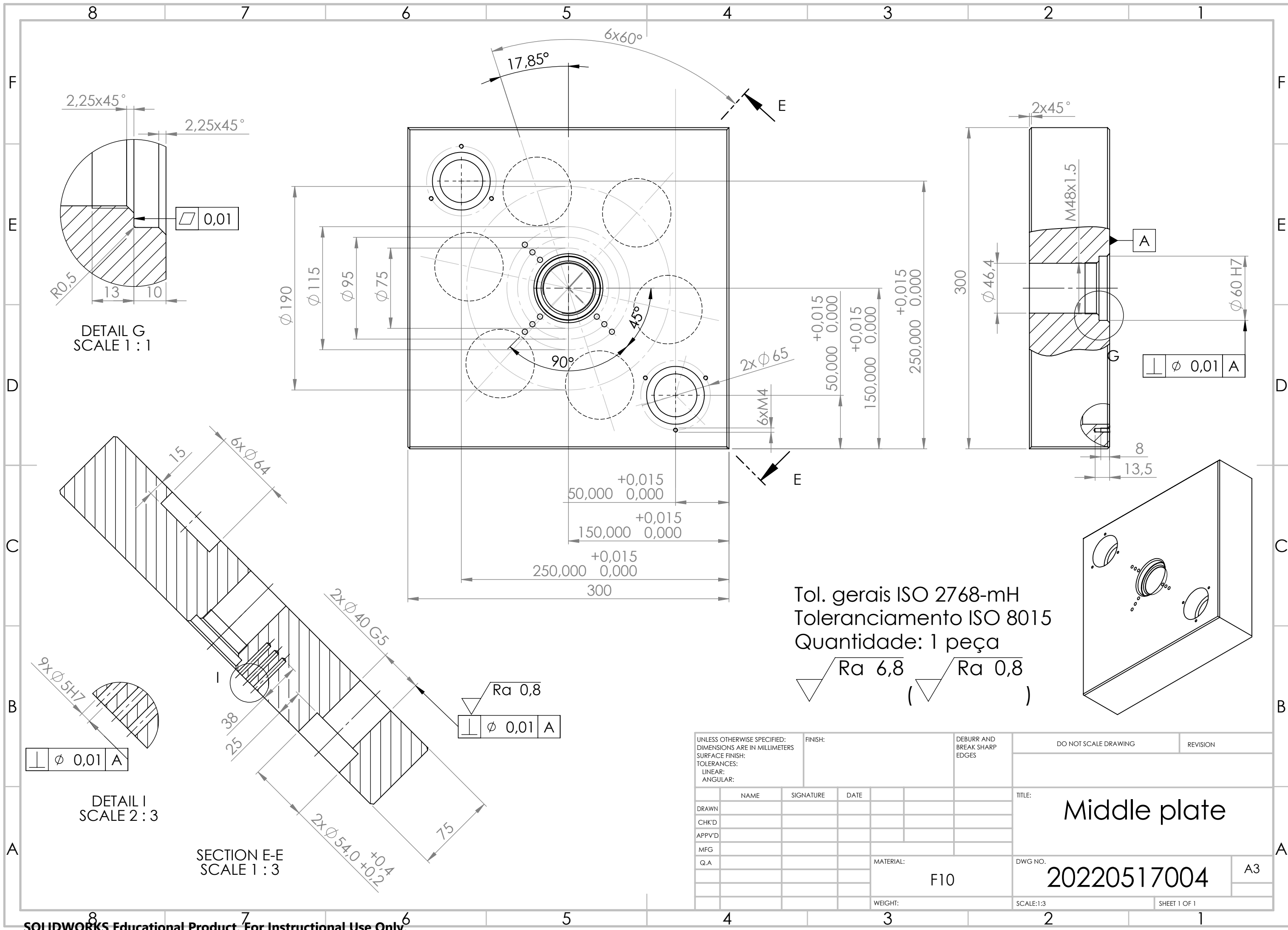


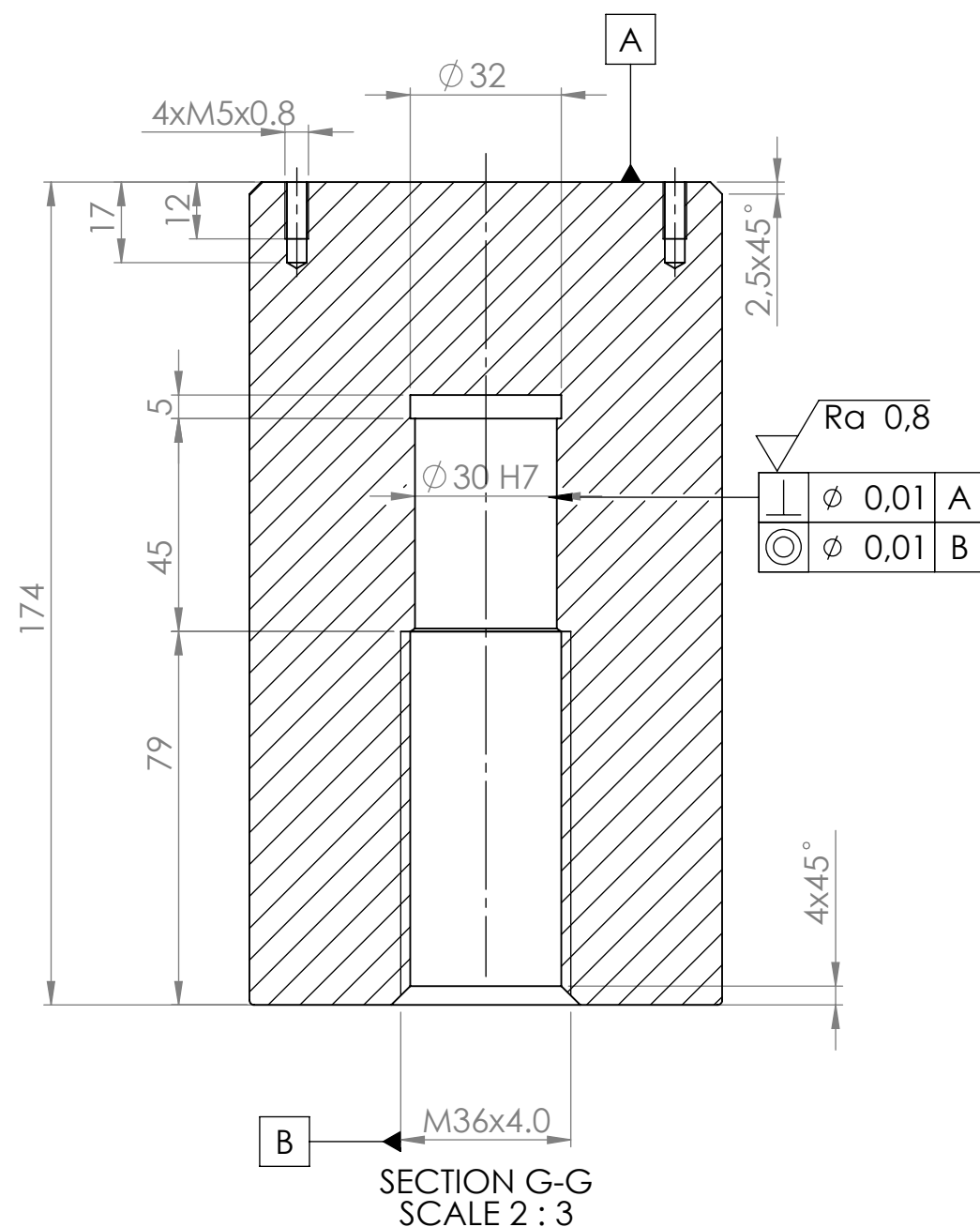
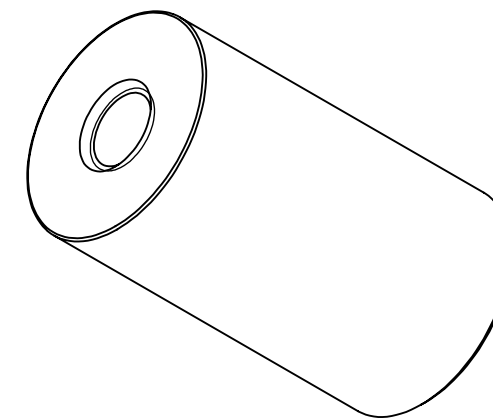
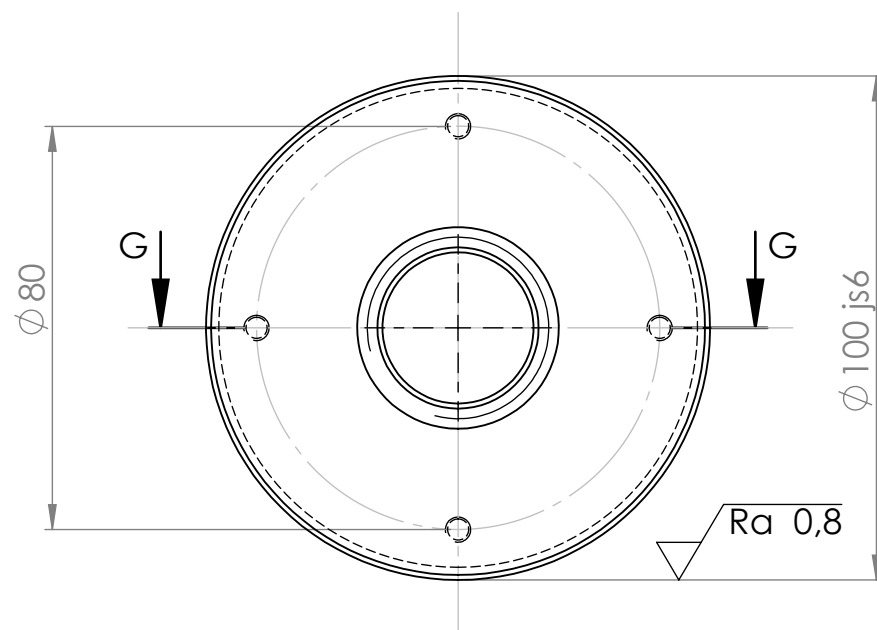
Tol. gerais ISO 2768-mH
Toleranciamento ISO 8015
Quantidade: 1 peça

Ra 6,8 (Ra 0,8 , Ra 1,6)

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:						FINISH:		DEBURR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
		NAME		SIGNATURE		DATE						TITLE: Upper drive plate	
DRAWN													
CHK'D													
APPV'D													
MFG													
Q.A								MATERIAL: F10		DWG NO.		2022050402	
										A3			
								WEIGHT:		SCALE:1:3		SHEET 1 OF 1	

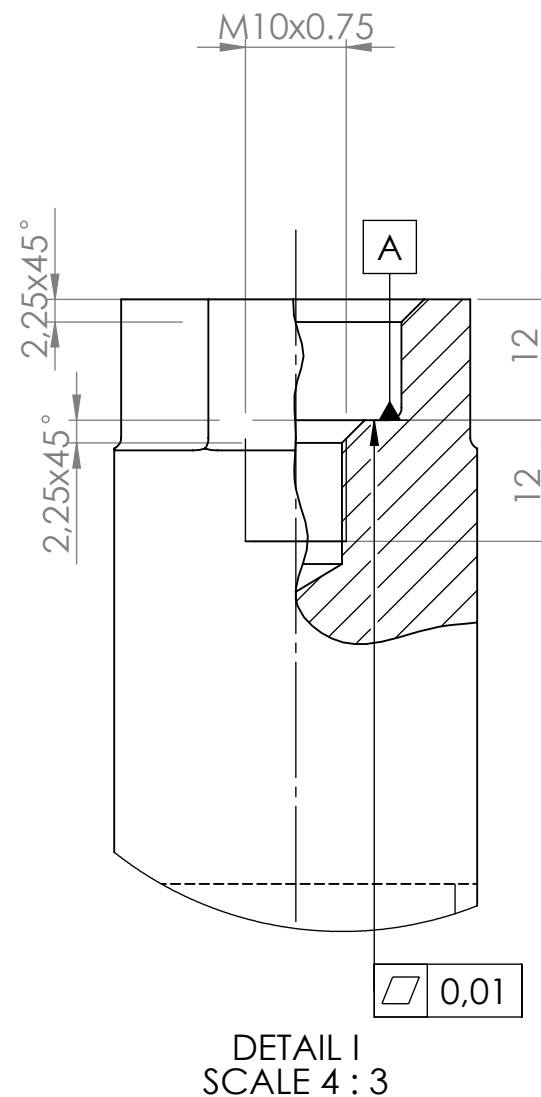
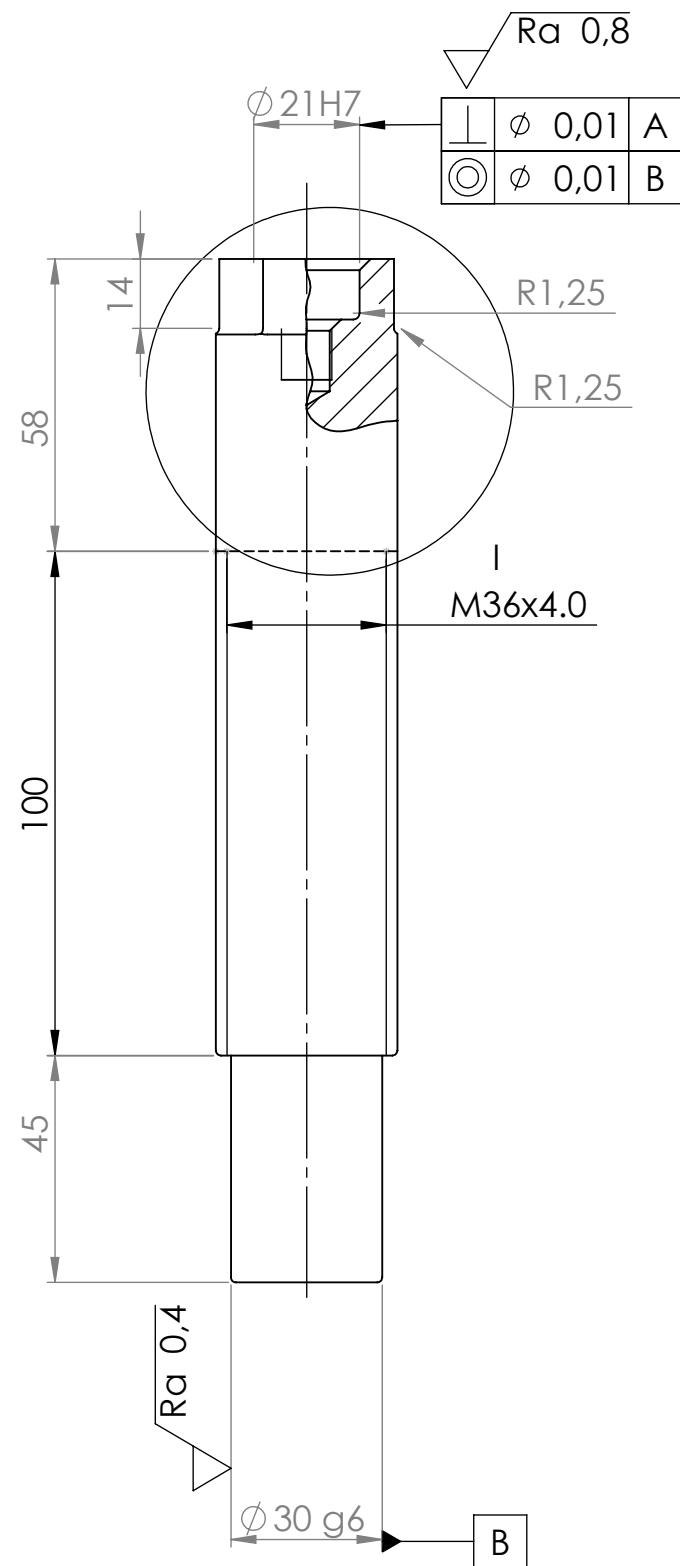






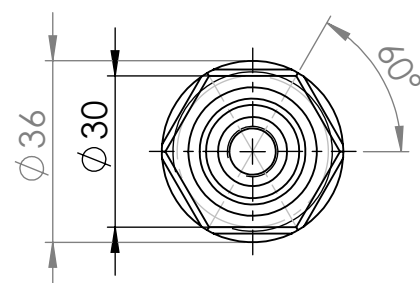
Tol. gerais ISO 2768-mH
Toleranciamento ISO 8015
Quantidade: 1 peça
Ra 6,8 (Ra 0,8)

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:						FINISH:		DEBURR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION		
	NAME		SIGNATURE		DATE						TITLE:			
DRAWN											Height adjuster			
CHK'D														
APPV'D														
MFG														
Q.A														
							MATERIAL:		F10		DWG NO.		A3	
											20220522006			
							WEIGHT:				SCALE:2:3		SHEET 1 OF 1	

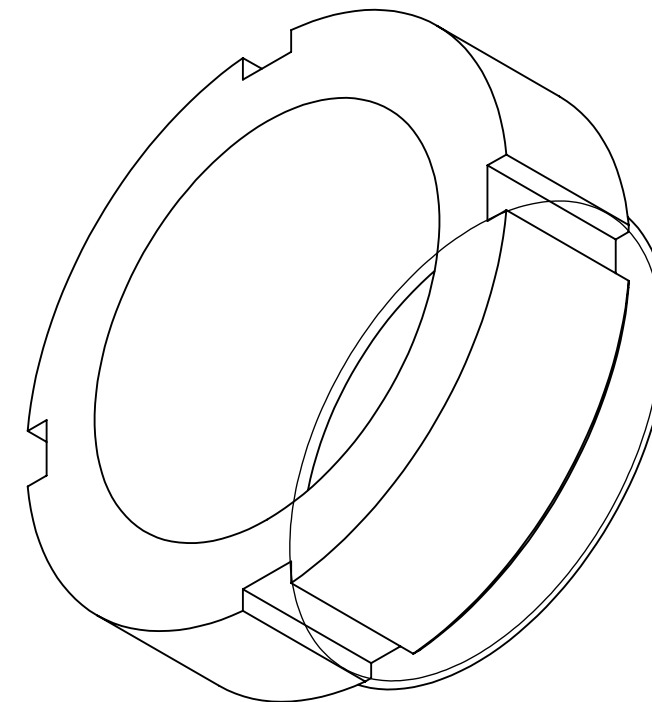
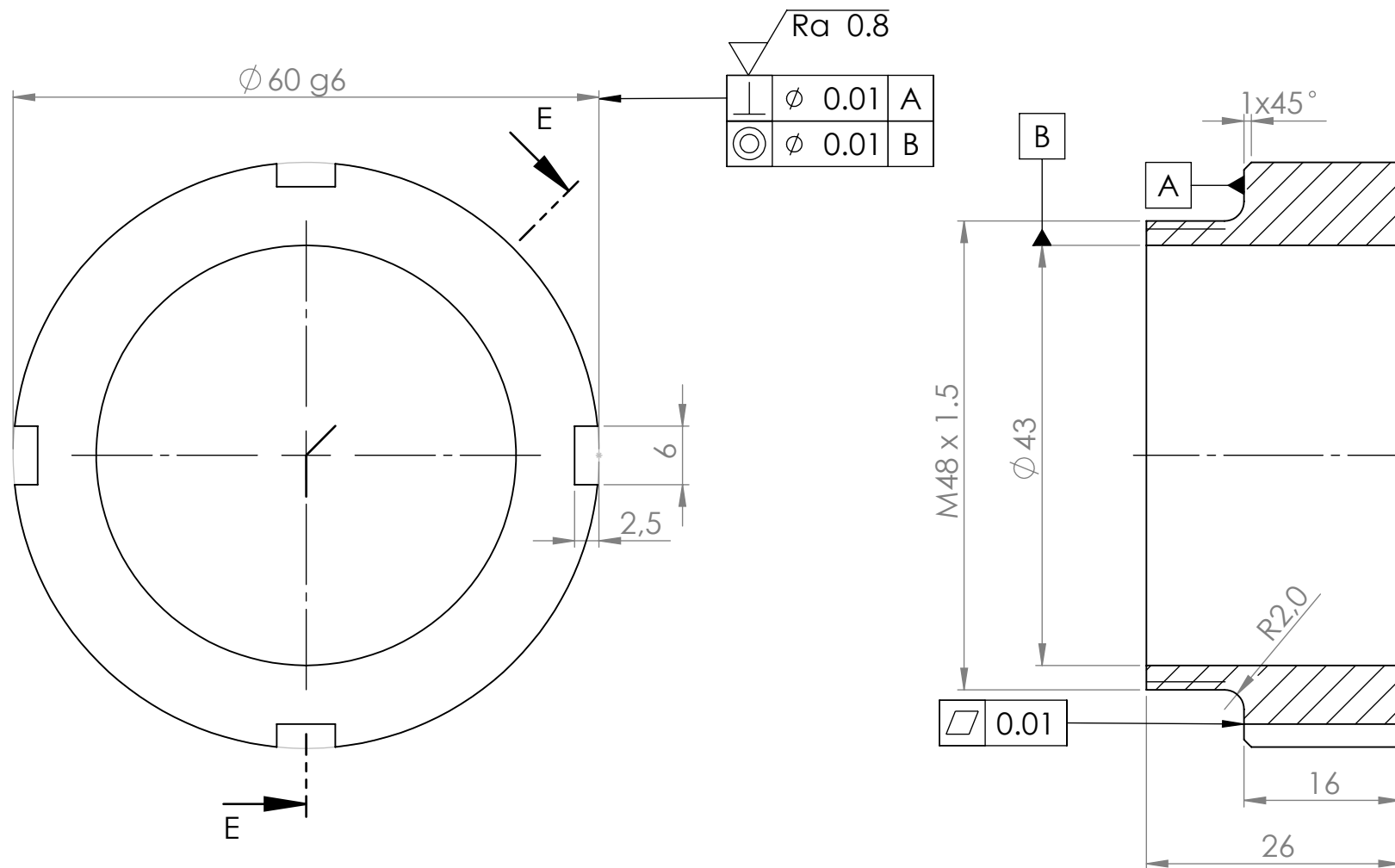


Tol. gerais ISO 2768-mH
Toleranciamento ISO 8015
Quantidade: 1 peça

$Ra 6,8$ ($Ra 0,8$, $Ra 0,4$)



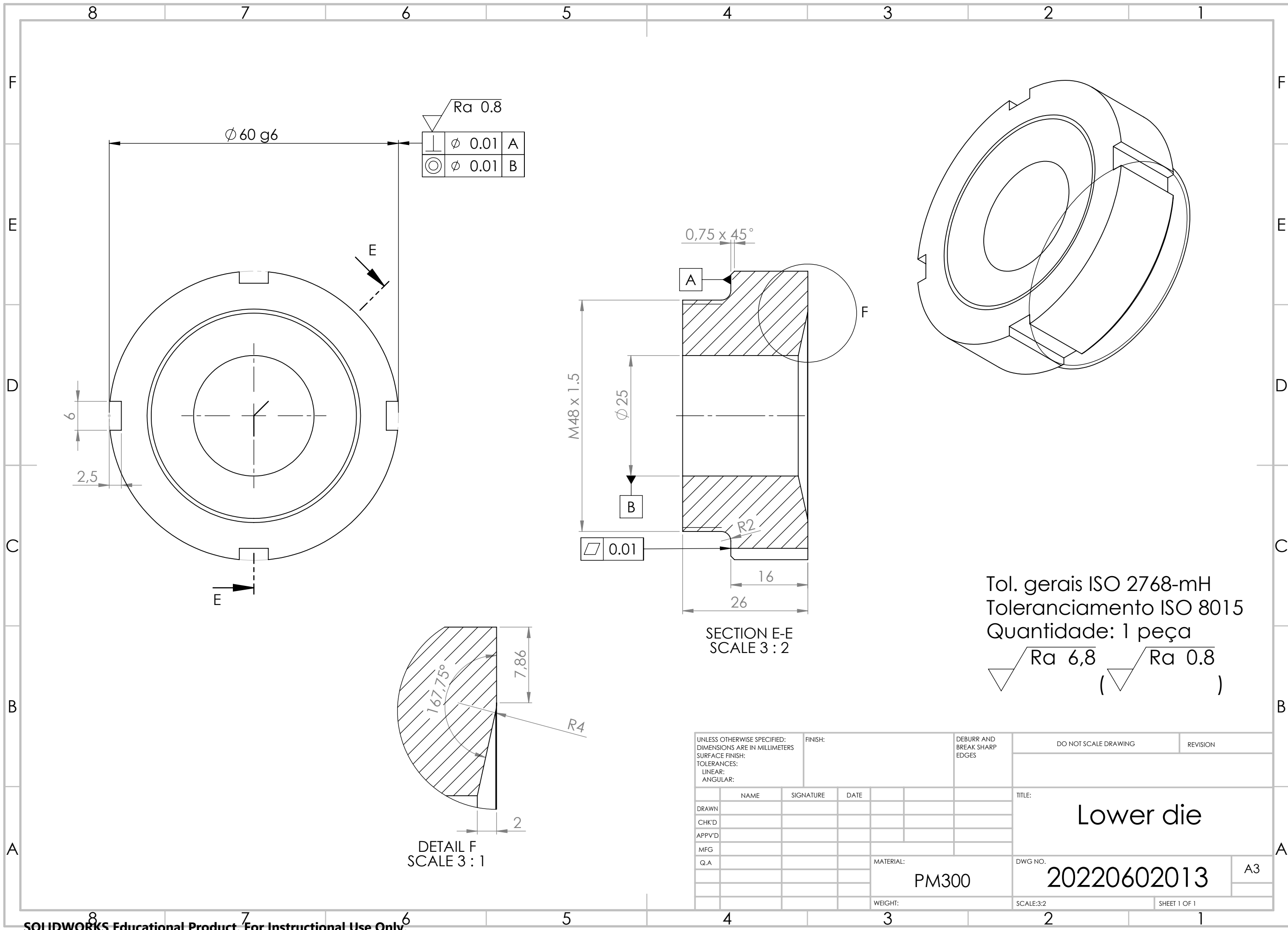
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:						FINISH:			DEBURR AND BREAK SHARP EDGES			DO NOT SCALE DRAWING			REVISION		
	NAME		SIGNATURE		DATE						TITLE: Punch holder						
DRAWN																	
CHK'D																	
APPV'D																	
MFG																	
Q.A							MATERIAL:			DWG NO.			A3				
							PM300			20220524007							
							WEIGHT:			SCALE:2:3			SHEET 1 OF 1				



Tol. gerais ISO 2768-mH
Toleranciamento ISO 8015
Quantidade: 1 peça

$\sqrt{Ra\ 6,8}$ ($\sqrt{Ra\ 0.8}$)

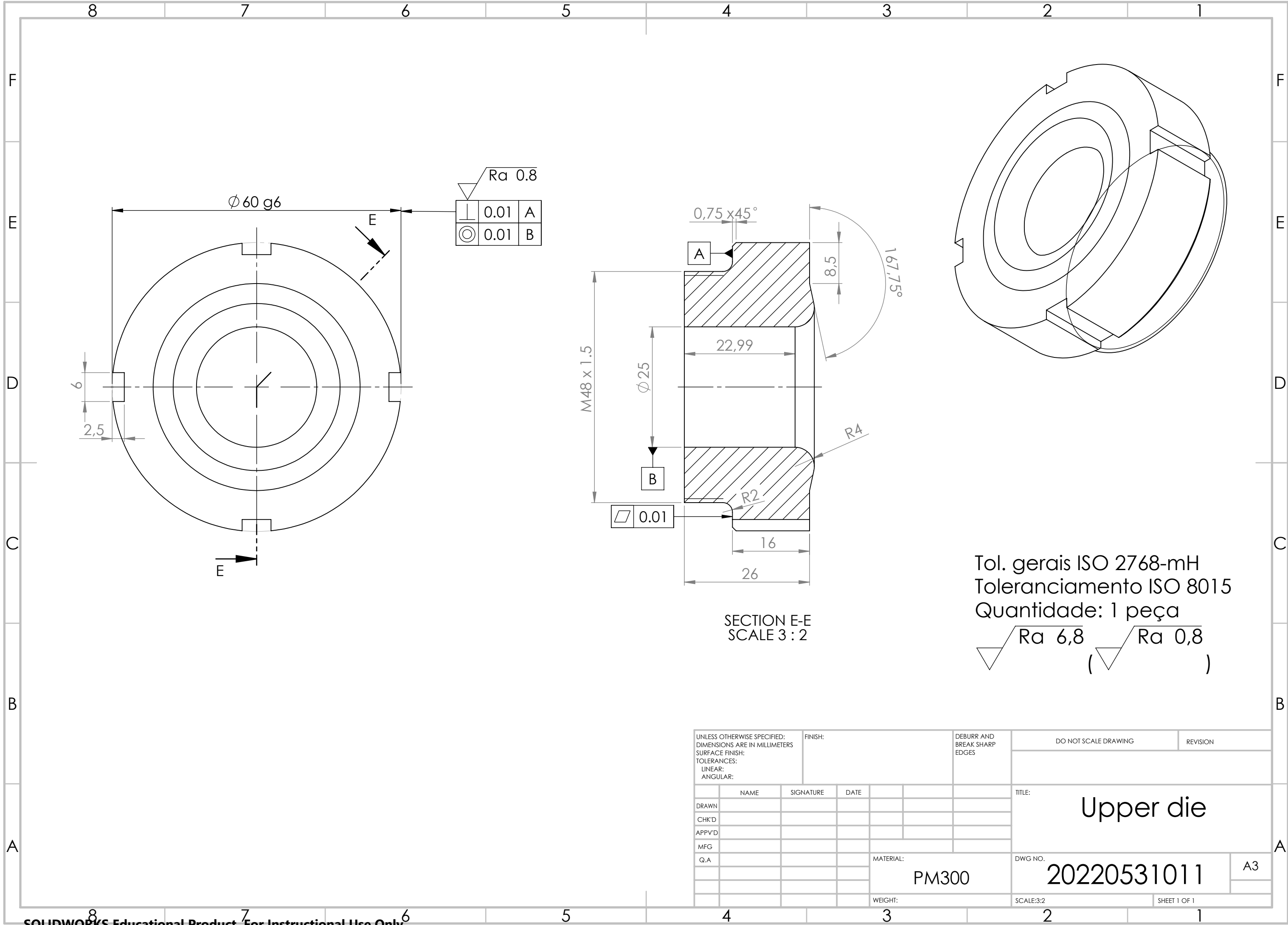
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:				FINISH:		DEBURR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
DRAWN				NAME		SIGNATURE		DATE		TITLE:	
CHK'D										Blank holder	
APPV'D											
MFG											
Q.A											
								MATERIAL:		DWG NO.	
								PM300		20220602014	
								WEIGHT:		SCALE:3:2	
										SHEET 1 OF 1	



Tol. gerais ISO 2768-mH
Toleranciamento ISO 8015
Quantidade: 1 peça

$\sqrt{Ra\ 6,8}$ ($\sqrt{Ra\ 0,8}$)

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:						FINISH:		DEBURR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION		
	NAME		SIGNATURE		DATE				TITLE: Lower die					
DRAWN														
CHK'D														
APP'VD														
MFG														
Q.A						MATERIAL: PM300			DWG NO.				A3	
									20220602013					
						WEIGHT:			SCALE:3:2				SHEET 1 OF 1	



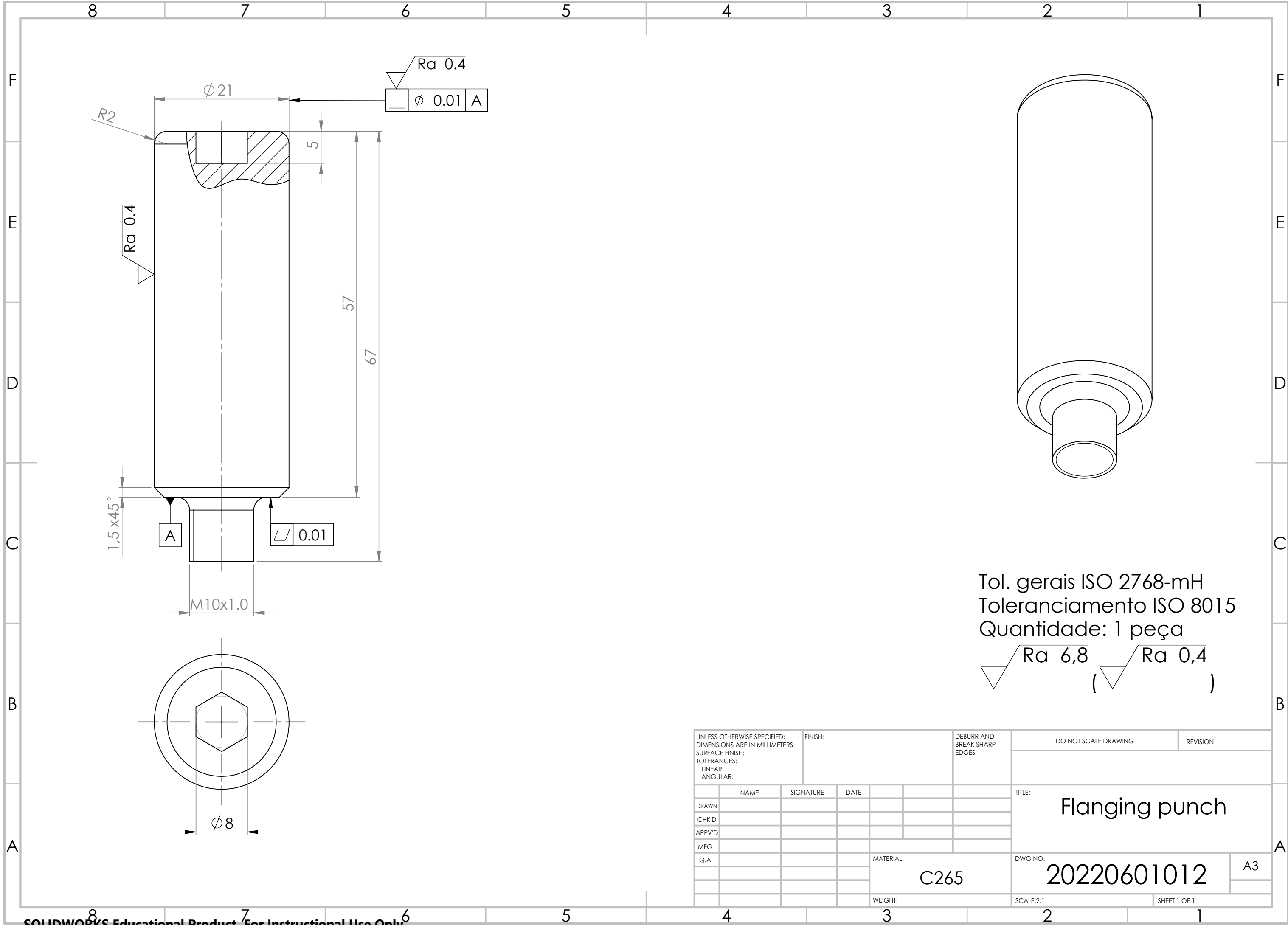
Ra 0.8		
⊥	0.01	A
⊙	0.01	B

▱	0.01
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SECTION E-E
SCALE 3 : 2

Tol. gerais ISO 2768-mH
Toleranciamento ISO 8015
Quantidade: 1 peça
Ra 6,8 (Ra 0,8)

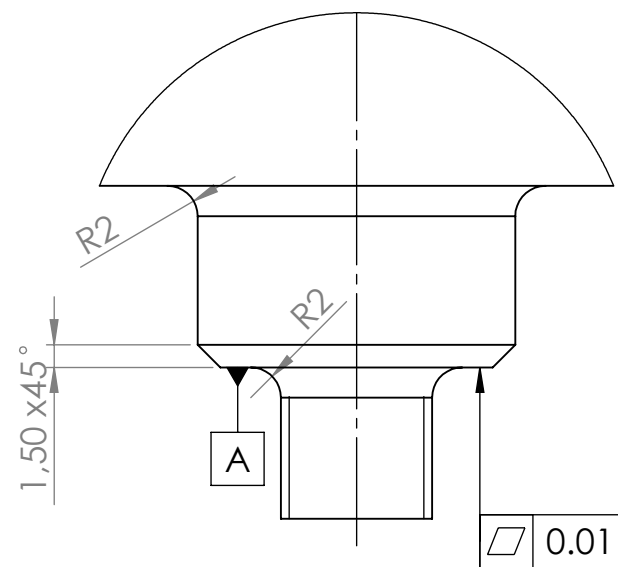
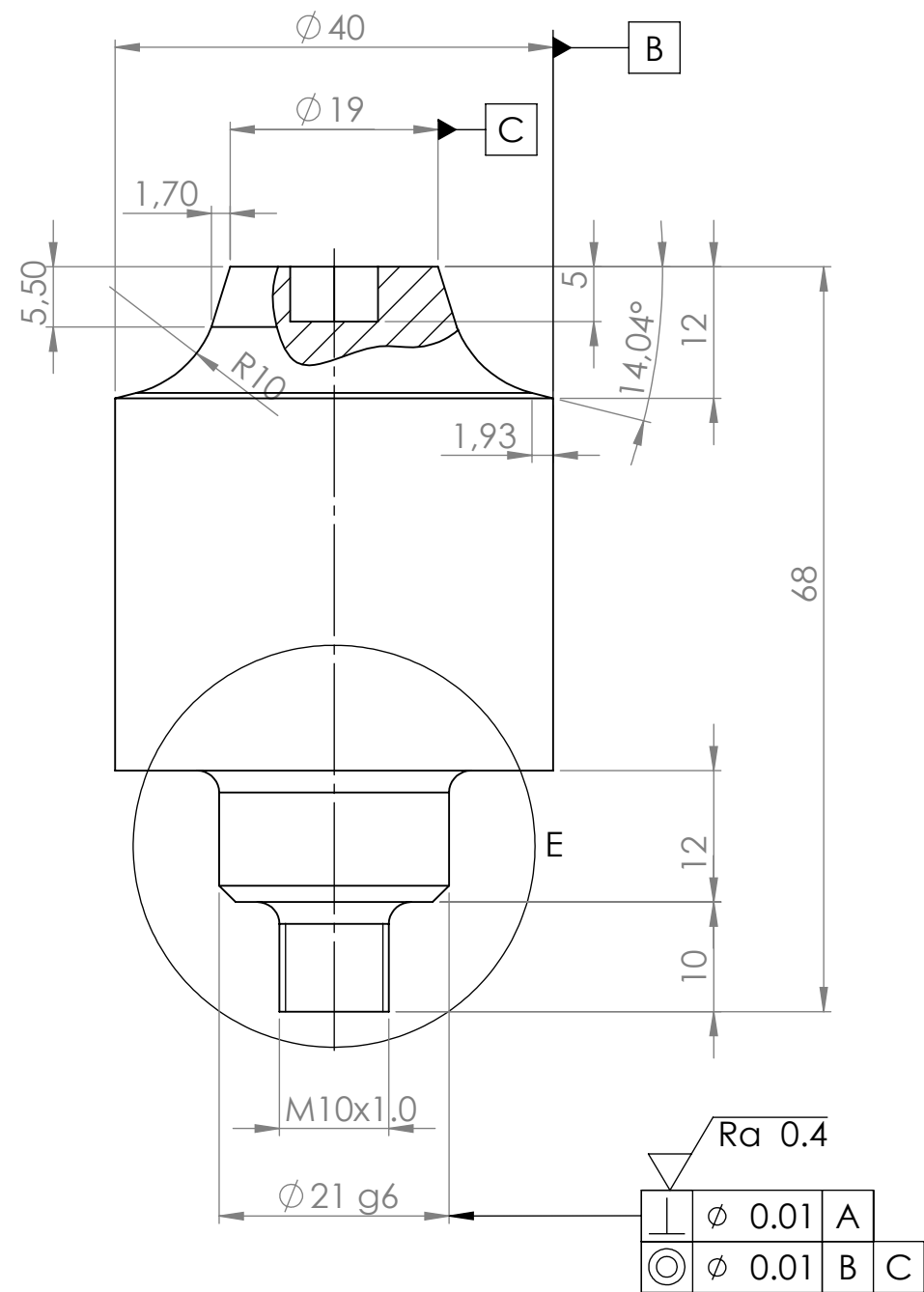
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:						FINISH:		DEBURR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING			REVISION				
	NAME		SIGNATURE		DATE						TITLE: Upper die						
DRAWN																	
CHK'D																	
APPV'D																	
MFG																	
Q.A									MATERIAL: PM300			DWG NO. 20220531011			A3		
									WEIGHT:			SCALE:3:2			SHEET 1 OF 1		



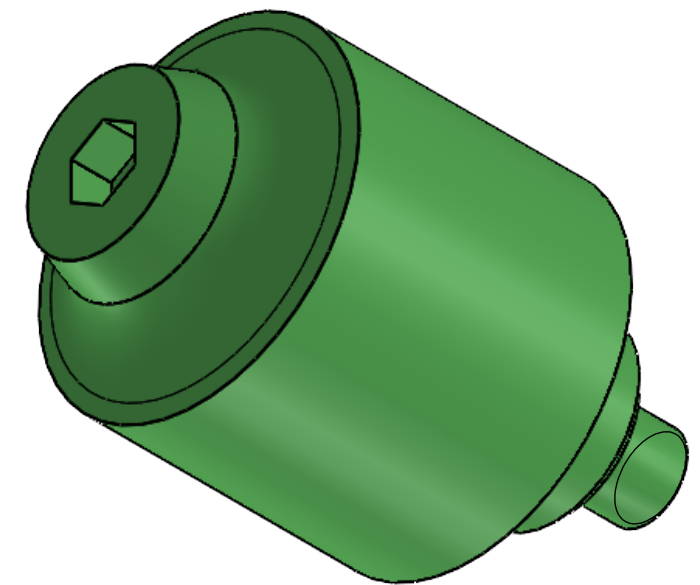
Tol. gerais ISO 2768-mH
Toleranciamento ISO 8015
Quantidade: 1 peça

Ra 6,8 (Ra 0,4)

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:						FINISH:		DEBURR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION		
	NAME		SIGNATURE		DATE						TITLE: Flanging punch			
DRAWN														
CHK'D														
APP'V'D														
MFG														
Q.A									DWG NO. 20220601012				A3	



DETAIL E
SCALE 2 : 1



Tol. gerais ISO 2768-mH
Toleranciamento ISO 8015
Quantidade: 1 peça

$\sqrt{\text{Ra } 6,8}$ ($\sqrt{\text{Ra } 0,4}$)

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:				FINISH:		DEBURR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
DRAWN				NAME		SIGNATURE		DATE		TITLE:	
CHK'D										Hemming punch	
APPV'D										DWG NO.	
MFG										20220602015	
Q.A										A3	
										SCALE:3:2	
										SHEET 1 OF 1	