# Development of an improved drill string safety valve

Thomas Lysgaard

Benjamin Mikkelsen

Sander H. Solbakken

Bachelor's thesis in Subsea Technology

Bergen, Norway 2020

Høgskulen på Vestlandet

# Development of an improved drill string safety valve

Thomas Lysgaard

Benjamin Mikkelsen

Sander H. Solbakken

Department of Mechanical- and Marine Engineering

Western Norway University of Applied Sciences

NO-5063 Bergen, Norway

IMM 2020-M92

Høgskulen på Vestlandet

Fakultet for Ingeniør- og Naturvitskap

Institutt for maskin- og marinfag

Inndalsveien 28

NO-5063 Bergen, Norge

Cover and backside images © Norbert Lümmen

| | |
|---|---|
| *Norsk tittel:* | Utvikling av en forbedret boresikringsventil |
| Authors, student number: | Thomas Lysgaard, h571714 |
| | Benjamin Mikkelsen, h180330 |
| | Sander H. Solbakken, h571708 |
| Study program: | Subsea Technology, Operation and Maintenance |
| Date: | May 2020 |
| Report number: | IMM 2020-M92 |
| Supervisor at HVL: | Tone Helene Bergset Røkenes |
| | Bjarte Sivertsen |
| Assigned by: | Moonshine Solutions |
| Contact person: | Helge Hope |
| Number of files submitted: | *2/bachelor's report and animation video* |

# Preface

This project is written as part of the completion to attain our bachelor's degree in Subsea Technology - Operation & Maintenance at the Department of Mechanical and Marine Engineering (IMM) at the Western Norway University of Applied Sciences (HVL). The report is written during the spring 2020 as part of the further development of SmartCock, a newly granted patent, in collaboration with Moonshine Solutions AS.

We would like to give a special thanks to *CEO* Helge Hope and *Head of Engineering* Jan Georg Tveiterås at Moonshine Solutions for giving us this opportunity to contribute in the development of SmartCock. We are grateful for all help and guidance that Helge and Jan Georg have provided us during the project.

We would also like to thank our supervisors at HVL, *Assistant Professor* Tone Røkenes and *University College Teacher* Bjarte Sivertsen. This project has deeply benefited from their constructive feedbacks and prompt responses to solve our everyday questions. We also appreciate all the technical advice and help regarding structure of the report.

Finally, we would like to thank *Senior Manager* Tommy Bjørklund at MHWirth for providing relevant roughneck data and to *Senior Project Engineers* Gunnar Polden and Jan-Otto Natland at MRC Global for carrying out an appraisal meeting during hectic and abnormal times.

MOONSHINE
SOLUTIONS

# Abstract

How can a small entrepreneurial company assert itself among the big corporations in the oil & gas industry? By creating innovative and cost-efficient products, Moonshine Solutions challenges several existing technologies in this business. To acquire full ownership of its own inventions, Moonshine has prioritized to patent all its technologies in advance of development.

One of Moonshine's latest technologies granted with a patent is "An improved drill string safety valve device", which further is commercialized as SmartCock. This patent combines existing field proven solutions to function together as new technology. During drilling operations a kick can occur, this must be handled by using approved equipment for such incidents. "Kick" is the terminology the oil & gas industry uses when there is an unintended influx of formational fluids into the wellbore. Kicks have the potential to escalate into serious situations which the Snorre A incident in 2004 is an example of. Kelly Cock and Inside Blow Out Preventer (IBOP) are two valves with different functions, that in some scenarios are used to handle and control a kick situation. The Kelly Cock is used to stop the influx of fluids into the drill string and the IBOP allows to safely circulate the well with kill mud. The newly invented SmartCock combine those functions into the same assembly and is intended to be light enough for one person to quickly install it onto the drill string to avoid a kick situation to escalate.

Even though the patent is granted as new technology there are still uncertainties whether it is suitable to use as a drill string safety valve. This project will commence a design process to further develop the SmartCock technology. As a part of this process a digital prototype of the SmartCock is modelled with use of a CAD software. Criteria from Moonshine and relevant standards is used as framework, but creative solutions are also implemented to make a successful design. One of them is inventive weight reduction measures to strive for the lowest possible weight of the valve body.

All components have carefully been given a dedicated material and a thorough review on different titanium alloys is conducted to find a suitable strength-to-weight ratio material for the valve body and other crucial components. Further is a weight analysis performed to check whether a combined weight below 25 kg is achievable.

Simulations are performed to test the SmartCock with forces it must cope with as a drill string safety valve. Structural analyses considering forces from pressure, clamp force, tensile- and torsion capacity are conducted to verify structural integrity with design criteria.

A technical appraisal meeting with MRC Global has been held to discuss feasibility of the ball valve assembly. In collaboration with their engineers' have several aspects been accounted for.

To help Moonshine commercialize the SmartCock, it is beneficial for them to be in possession of advertising objects to show their customers. A downscaled 3D-printed model was planned, but due to COVID-19 the access to 3D-printing facilities were not available. As a substitute, it is made an animation video that covers all aspects of SmartCock from assembly to function and surrounding environment.

Ultimately this report will figure out if any drawbacks are to be found that would put an end to the continuation of the development.

# Table of contents

# 1. Introduction

This chapter will cover information about the project. This includes background information, aim, objectives and limitations of the project.

## 1.1 Background

Moonshine Solutions AS is an entrepreneurial company that was founded in 2013 by a group of three people who combined have more than 60 years of offshore experience. Between them they have had job positions like Driller, Rig Manager, Rig Safety, Drilling Engineer and more. The company creates innovative new solutions targeting the offshore oil & gas sector. They are currently in possession of 4 patents, where the first one is developed and commercialized by the name Smartalizer. This is an innovative centralizer that reduces costs during cementing operations of casing or liner in an offshore well [1]. Another patent that is granted, was approved in June 2019 and is described as “An improved drill string safety valve device” with the Norwegian patent number 343784. This invention is only briefly developed to ensure that assessors have enough information to either approve or decline this invention as new technology. The valve is intended to be used during drilling operations as a barrier if a kick occurs. The reason for this invention is that Moonshine sees a potential in streamlining today’s solution which uses two separate valves with different functions to meet the barrier requirements during drilling. The two different functions of those valves are known as a ball valve and a check valve. In their respective use offshore they are referred to as Kelly Cock (ball valve) and Inside Blow Out Preventer (check valve). The newly invented valve from Moonshine combine those functions into the same assembly to reduce the number of valves needed. The valve generally consists of a body which houses a stem-operated ball valve assembly with a spring load function. Another objective for this invention is for the valve to be light enough for one man to install it onto the drill string. Since Moonshine commercializes all their products with a name that contains the word “Smart”, the valve is intended to be entitled SmartCock. The valve will therefore be referred to as SmartCock in this report.

Being a small company in the oil & gas industry is challenging, reason is that it demands big budgets for innovation. The cost of developing, certifying and producing are expensive. Usually the bigger established companies will either create a similar solution or just buy of the technology. It is therefore a priority to patent an invention at an early stage to ensure that the owners have an exclusive right on the technology. The World Intellectual Property Organization (WIPO) is a source for global intellectual properties, such as patents, they describe a patent in this way:

> *“A patent is an exclusive right granted for an invention, which is a product or a process that provides, in general, a new way of doing something, or offers a new technical solution to a problem. To get a patent, technical information about the invention must be disclosed to the public in a patent application.”* [2]

To grant a patent is complex process and therefore time consuming. It can often take more than a year before applications is granted or declined. For Moonshine it took approximately 18 months of processing time before the technology regarding SmartCock was granted. The development of SmartCock is only briefly began because Moonshine wanted to grant the patent and be certain of ownership of the technology before investing more time into the idea.

## 1.2 Aim and objectives

The aim of this project is to further develop the design of SmartCock regarding the technology that is patented and achieve a weight below 25 kg.

Even though the patent is granted as new technology, it does not necessarily mean that the technology will be suitable for the intended working environment. This project will go in depth to clarify if the technology is suitable for use as a drill string safety valve with the given specifications from Moonshine and relevant standards. During the development will force, dimension, weight and function be the factors mainly considered.

**Objectives:**

1. Design the valve by use of a CAD software in accordance with relevant standards and criteria.
2. Choose a material for each item of the valve.
3. Simulate forces that are applicable to ensure structural integrity of body and stem.
4. Collaborate with a third party to achieve a professional assessment on ball valve function and sealing.
5. Create a 3D-printed model for advertising and functional testing purposes.

*Note: Due to extraordinary measures made by the Norwegian government in March 2020 regarding the COVID-19 outbreak will objective 4 be limited, and objective 5 be neglected from the project.*

## 1.3 Limitations

There are many factors to consider when designing a valve that is intended to be used as a barrier during a drilling operation. This project is limited to solve technical issues regarding the design based on available information and calculations. Some limitations are:

1. Design specifications and standards are interpreted in best ability but still limited because the newly invented patent is not directly considered in them.
2. NACE MR0175/ISO-15156 is not considered regarding material choice.
3. Thread sections for connection to drill pipe will only be of cosmetic character.
4. Assumptions are made to simplify the flow through closed ball valve (pump through capacity).
5. Gaskets, sealings and sealing surfaces will only be considered in such a way that the design allows space for them to appear on a later basis.
6. Engineering tolerances are not considered.

## 1.4 Structure of report

This report will cover the design process carried out regarding the SmartCock patent. Chapter 1 covers background, aim, objectives and limitations of the project. In chapter 2 is background theory presented. It will cover some basics regarding drilling and where SmartCock will apply. Chapter 3 will describe what design process and formulas that are used to solve the task. Chapter 4 clarifies standards, criteria and objectives which the valve must fulfil. Chapter 5 describes the modelling, calculations and material choices regarding the design. In chapter 6 is simulation results presented and evaluated. In chapter 7 is the conclusion of the project disclosed and will also include recommendation for further work.

# 2. Background theory

This chapter will give a brief description of drilling a well with some main equipment, barrier elements and kick situations. It will then highlight some equipment used to handle a kick situation and where the new improved drill string safety valve will apply.

## 2.1 Drilling

Drilling wells have various purposes, one of them is to access reservoirs containing hydrocarbons in form of oil and/or gas. Independent of drilling into subterranean reservoir onshore or offshore is the goal usually the same; To achieve a safe access by use of a well system. When drilling the well, a broad range of equipment is used to ensure safe and effective operation. The main part that physically enters the formation when drilling is the drill string with the drill bit in front doing the hard work with crushing the formation into cuttings and debris. The mud will carry the cuttings out of the well by use of a circulation system topside. The drill string is composed by different pipes and assemblies which usually are screwed together with high torque performed by a roughneck. A top drive rotates and keeps tension in the drill string in addition to facilitate safety features and mud. A very important safety equipment during the drilling operation is the Blow Out Preventer (BOP) which act as a safety valve if the well must be shut in.

**Barriers during drilling**

According to NORSOK D-010 it is applicable to have two independent well barriers when drilling in formations that might contain hydrocarbons. D-010 describes a "well barrier" like this:

> "*Envelope of one or several well barrier elements preventing fluids from flowing unintentionally from the formation into the wellbore, into another formation or to the external environment.*" [3]

The intension of the well barriers is to achieve safe access to the reservoir. The pore pressure that is build up in the reservoir due to the overlaying rock formation and other conditions will try to "push out" the medium (normally water, gas or oil) if an access occur. This access is the well that is drilled to enter the reservoir. During drilling operations, the two barriers are normally referred to as primary barrier and secondary barrier.

**The primary barrier** is normally the drilling mud that is used. The weight of this fluid is calculated by the information gathered during seismological research. The geologist will use all the information that is available for the actual well and calculate the required mud properties. The weight of the fluid generates a pressure in the well due to the height column in the system and is balanced to slightly exceed the formation pressure.

**The secondary barrier** is normally of a mechanical character. The main secondary barrier during drilling is the BOP. This safety valve consists of an annual preventer, pipe rams and shear seal rams. For the BOP to function as a barrier for the inside of the drill string, it would need to cut the string and close the whole cross section, this is not desirable. The BOP should preferably shut in the well in the annular space which is the outside of the drill string. Inside the drill string there will be "fluid connection" all the way from the drill bit to the top drive on the rig. The inside of the top drive is equipped with valves referred to as manual and automatic Kelly Cock. These valves will prevent bore fluid to escape the drill string if the top drive is connected to drill string. If the top drive is not connected to the drill string, then a Kelly Cock located on the drill floor must be placed on the drill string to achieve the secondary barrier.

**Kick situation**

According to API 53 is a “kick” referred to as: “*Unintended influx of formation liquids or gas into the wellbore”* [4]. This means that an unbalance in the weight of the mud and the pressure in the formation leads to influx of fluids into the wellbore. On the drilling rig is the wellbore always monitored to detect an influx of fluids. If a kick occurs, the well is closed and actions to establish well control is conducted. If the top drive is connected to the drill string, the automatic Kelly Cock will close the inside, and the BOP will close the outside of the drill string. If the top drive is not connected to the drill string, then the Kelly Cock located on the drill floor is installed to the drill string, and then closed. This should preferably be done as fast as possible since kicks often have the potential to escalate rapidly.

## 2.2 Kelly Cock and Inside Blow Out Preventer

**The Kelly Cock** (shown left in Figure 1) is often referred to as a “Full opening safety valve”. This means when the valve is open, the inside bore diameter is smooth to not restrict the outflow of fluid. The Kelly Cock is mounted to the drill string in open position to allow fluid to pass, otherwise it would be impossible to enter the threads. After it has been mounted to the drill string and tightened, the valve is closed to secure the well. The valve is closed by use of a hex key, as illustrated in Figure 1 [5].

**The Inside Blow Out Preventer (IBOP)** (shown right in Figure 1) is a check valve that can be mounted to the drill string after the Kelly Cock is installed. Sometimes when a kick occur it is beneficial to lower the drill string to the bottom of the well before starting to circulate kill mud. If that is necessary, an IBOP is installed above the Kelly Cock. The Kelly Cock valve is opened to let the well pressure reach the IBOP. This allows the string to be lowered into the well and every time the top drive disconnect the string to install another stand (set of drill pipes), the secondary barrier is achieved with the check valve function in the IBOP. When the drill string is bottomed in the well, the killing operation can begin, and the kill mud will pass the IBOP as illustrated on Figure 1. The spring will try to force the sealing element towards closed position, but when the kill mud in the drill string is pressurised to a certain level above the well pressure, the spring will compress [5].

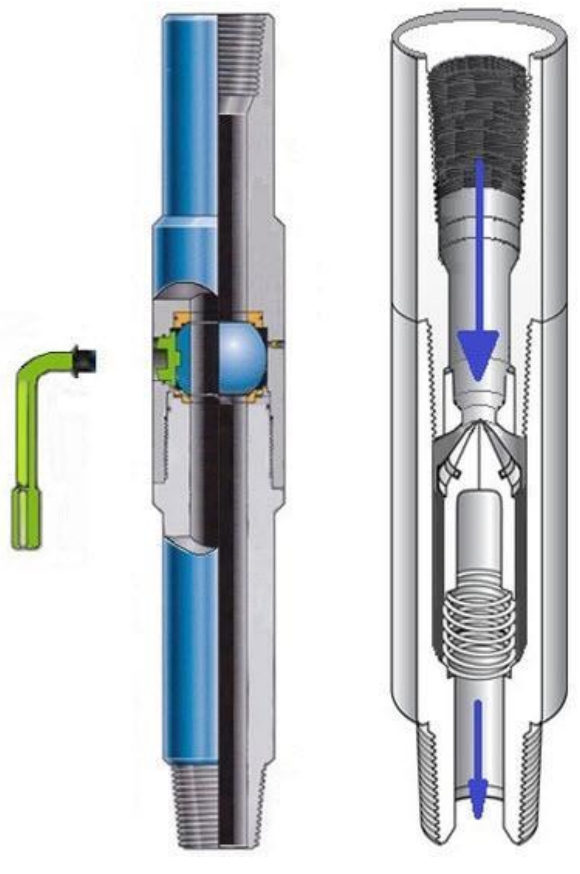


Figure 1: Kelly Cock and IBOP [5]

## 2.3 The improved drill string safety valve (SmartCock)

This sub chapter will cover some of the essential information regarding the patent “An improved drill string safety valve device”. The complete patent application is given in *Attachment 3*. As mentioned before, the name of this valve will also be referred to as SmartCock. Original figures from the patent application are used to illustrate how the patent look like and works. Not all the numbers and highlighting of components and functions will be mentioned in the text below.

The main objective with the patented technology regarding SmartCock is to combine the functions from the Kelly Cock and IBOP. It is also desirable to achieve a weight below 25 kg so it can be lifted by one person. The Figure 2 below shows one of the models from the patent application. Item 2 is the body that interface with the drill string and house all the internal components. Item 5 is the stem (type of bolt or crank that operates a valve) which operates the ball. All the components inside the ring A are internal items that must be placed within the body. Item 10 is a seat that seals towards the ball valve and body. Item 9 is a valve cage that keeps the valve assembly, included the spring, in place. Item 7 is the ball valve that open or shuts fluid to pass through the valve. Item 11 is a seat that fits the ball valve and the spring. Item 8 is the spring that allows the ball valve to be forced from its seat during pump through operation. The last two items most left in the picture is a retainer arrangement to keep all the internal items in place.

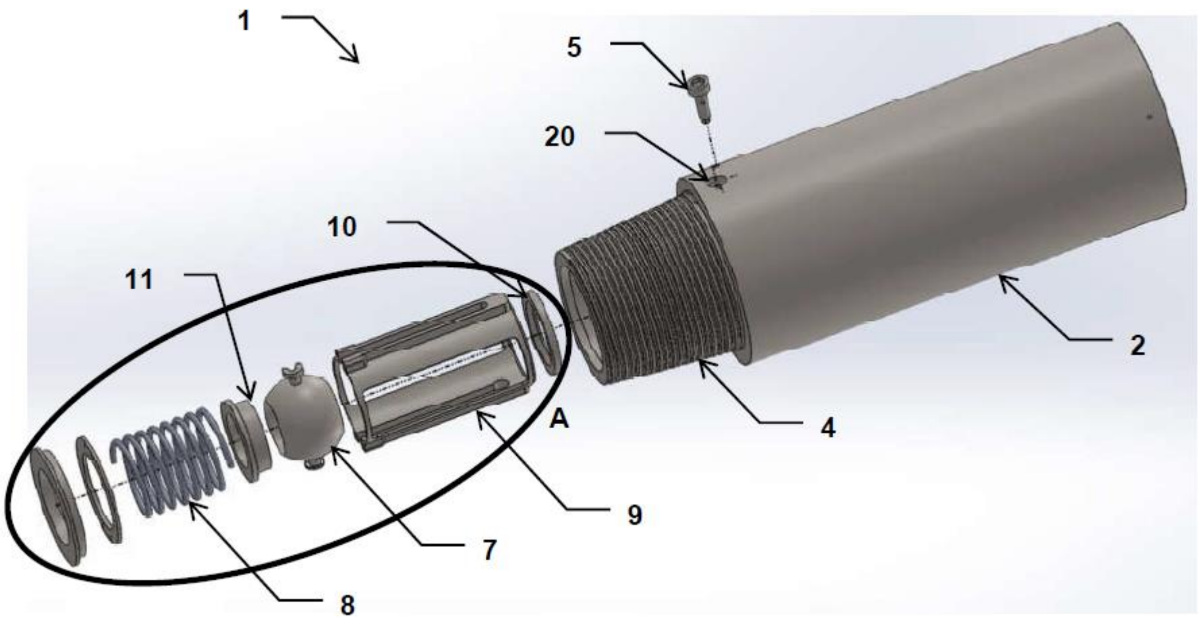



Figure 2: Patent figure 2

Figure 3 shows how SmartCock act as a ball valve (Kelly Cock function). In the left and middle picture is the ball valve in an open state and the arrow F indicates fluid passing inside the valve. This is the valve’s initial state when it is fitted to the drill string during a kick. The fluid will pass through the valve while allowing the pin threads (point 4) to enter the drill string, and then the valve is tightened. The picture to the right shows the ball valve in a closed state and the outflow of fluid is stopped.

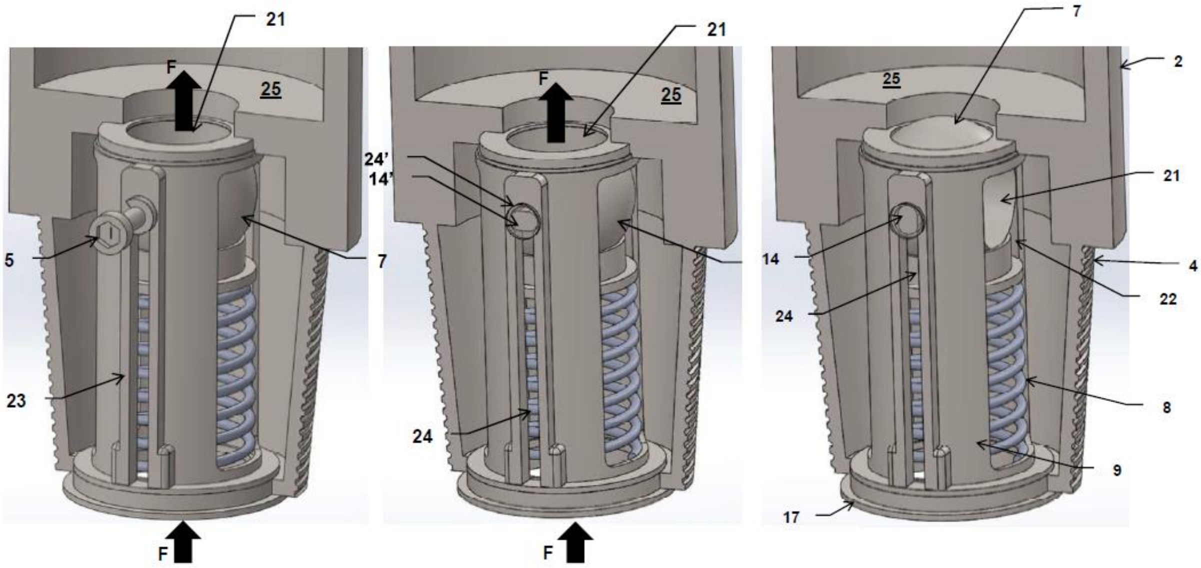



Figure 3: Patent figure 5 & 9

Figure 4 below shows how SmartCock act as a check valve (IBOP function). In closed state it is possible to force the ball valve to leave its upper seat by applying pressure on the top side of the valve as illustrated in the figure below. The arrows indicated with F shows the suggested fluid path through the internal components and into the drill string below. There are some details to highlight during this operation. Point 13 shows the first guide peg on the valve which has a shape and a cut that allows it to leave the Stem (point 5) when in closed state. Point 14 shows the second guide peg that is shaped to fit in the guide slot of the sleeve only in closed state. This is to secure that the valve is not unintentionally rotated to open state during the pump through action. If it did, it would not have fitted into the slot in the stem and it would not have sealed when the spring forced it back up to its seat.

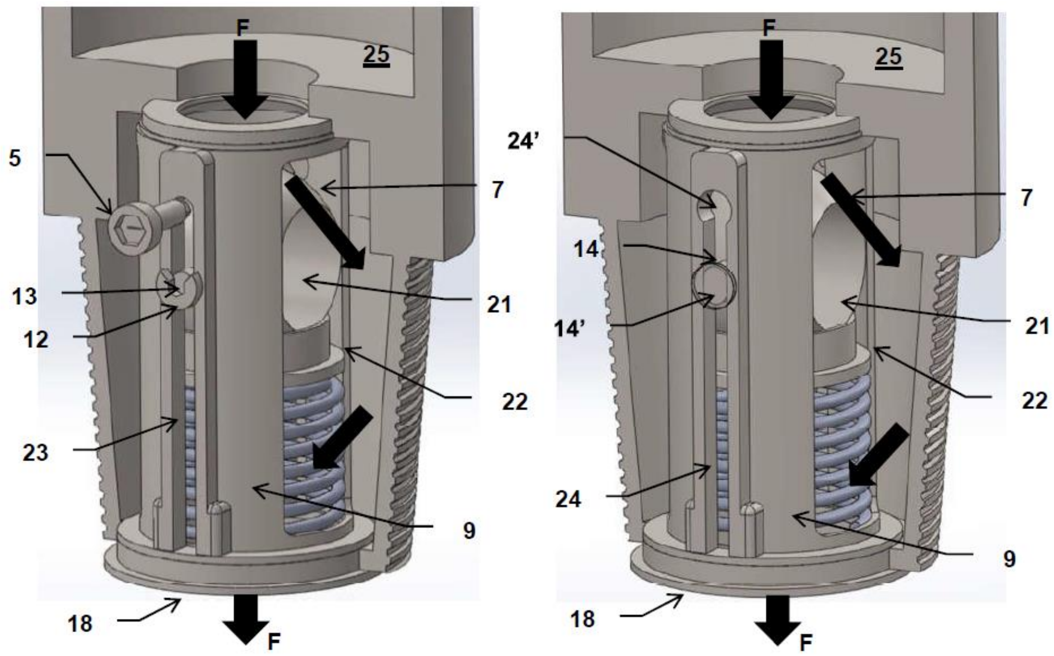



Figure 4: Patent figure 7

# 3. Theoretical approach

This chapter will give an overview of the initial phase of the project on elements that needs to be declared before approaching the design phase. These are elements such as a defined design method, software and formulas.

## 3.1 Design process

The aim and objectives for this project relies mostly of design of the valve, it will therefore be beneficial to define a design process that the project will follow. Shigley's design process covers a six-step process on how one should design a product seen in Figure 5. This design process is a broad and general way of working towards a design, it can be used for both improvements of already existing parts and for brand new designs. The whole process starts with *Recognition of need* and is often done by an individual or a group that sees the possibility to either improve, innovate or fix a design. For this specific project, Moonshine has already covered the recognition of need.

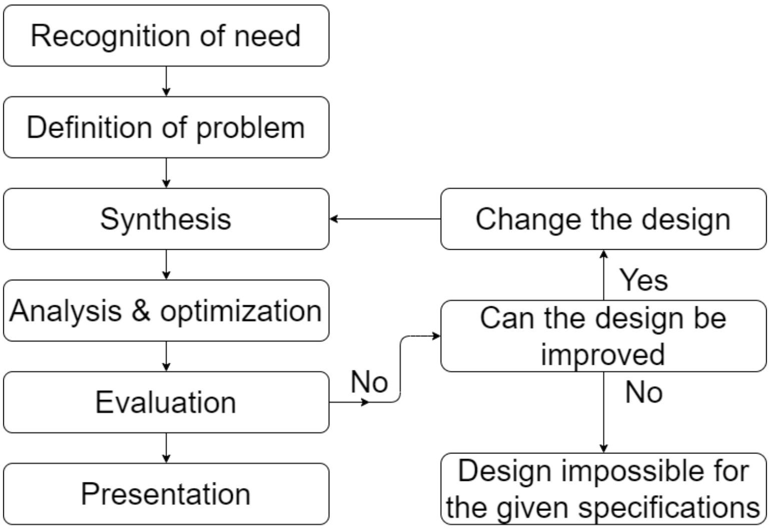



Figure 5: Shigley's design process [6]

Shigley's next phase goes by the name of *Definition of problem*, and its where one shall cover the dominating specifications for the product to function as intended and to abide its working environment. Regarding the SmartCock design its dominating criteria are set by both Moonshine and standards that are related to rotary drill stem elements. Most of the standards used in offshore operations are usually not specific but covers more broad and general definitions. The reasoning for this is that all wells are different in pressure, size, formation, and there are also different methods that do the same task. Though it has defined three commonly used pressure classes (5k, 10k, 15k psi systems). These classes are typically higher than the maximum well pressure as another safety factor. To summarize, the design criteria will define the specifications of SmartCock such as quality, performance, functional- and physical characteristics, which will be further elaborated in chapter *4 Design criteria*.

The *Synthesis* phase is intended to gather as much inspiration as possible towards a digital prototype. For this project, it will be done by combining all sources of information like the design criteria, pre-existing designs and meet with experts/companies that specializes in valve configuration and design.  It is beneficial to make an early digital prototype as no materials manufacturing are required, and changes on the design can be done swiftly without losing much time. Now a days it is common to use Computer-Aided Design (CAD) software to create the digital prototype of the product. There are many different CAD software programs to choose from, and they all have its advantages and disadvantages depending on what the product requires. This project will use Autodesk Inventor, due to that it is user-friendly and easy access. However, there is an issue with designing a digital prototype, there are uncertainties in how the prototype would behave in a real-life scenario.

The solution is simulation; a way to mimic the physical properties of the real world onto a digital one. After a prototype is made it will go through designated tests and analysis, this is what Shigley's design process defines as the *Analysis & optimization* phase. Typical test used are structural analysis, heat transfer, manufacturability, materials, and fluid flow to name a few. For SmartCock the material and structural analysis are the most relevant due to the large forces involved, and the desire to make the part as light as possible. The material analysis will be covering material properties such as yield strengths, Charpy-V notch impact, tensile capacity and hardness, to see its capability of resisting plastic deformation. The density of the materials also needs to be accounted for, due to weight criteria. To solve the structural analysis a simulation software uses mathematical algorithms to approximate how an object would react to the real-life load and constraints. One of the most common methods of solving the simulations is the Finite Element Method (FEM). This report will not go into detail about the math involved in the model, but the concept in FEM is to divide a bigger object to smaller sized objects, known as elements, and using partial differential equations to approximate the stress and displacement around these elements. A collection of all these elements are what makes up the complete 3D-model. But FEM is not without limitations, one of them is the appearance of singularities sometimes referred to as hotspots and are often found on sharp edges and can provide misleading results. As there are many different CAD software, there are also many simulation software to choose from. For simplicity reasons Autodesk Nastran will be used for simulating the SmartCock, the Autodesk suite have an excellent collaborative capability between its software programs.

After the calculations are done the results can be used to *Evaluate* if the object would break, cease to fulfil its task, or successfully withstand the loads. If the design does not fulfil its design criteria regarding both functional and physical characteristics the prototype will need further modification or redesign and will therefore repeat the process and go back to the synthesis. This process is often the most time-consuming part of a design process, but it is essential to produce a valuable product.

If the results are as anticipated, and there are no more room for improvement the design will be ready for *Presentation*. All documentation regarding the product will be made, often the documentation consists of technical drawings, bill of material, design development, product brochure, etc. [6].

## 3.2 Formulas

A Summary of all formulas relevant to the development will now be presented. The outer shell will mostly be calculated with use of pressure and tensile capacity formulas, while others are related to flow configuration for the push through function.

The valve will have to withstand pressures and the following formula is used:

$$p = \frac{F}{A} \tag{1}$$

Where p is pressure, F is force, A is area

Reynolds number is a dimensionless ratio to indicate whether a flow is laminar or turbulent, and is calculated as followed:

$$Re = \frac{\rho VL}{\mu} \tag{2}$$

Where ρ is density, V is average velocity, L is diameter of a pipe, μ is viscosity

By flow through a check valve (closed position for the valve in this project) is *flow rate in a turbulent nozzle* used. This formula only applies with turbulent flows and is calculated as followed [7]:

$$Q = \mu A \sqrt{\frac{2}{\rho}(p_1 - p_2)} \tag{3}$$

Where Q is flow rate, μ is outflow coefficient, A is area, ρ is density, $p_1$ - $p_2$ is the differential pressure.

To calculate relation between velocity, flow, and area this formula is used:

$$Q = V \cdot A \tag{4}$$

Where Q is flow rate, V is velocity and A is area

The Tensile Capacity of a drill pipe or drill string equipment is the maximum tension that can be applied to a pipe/equipment before entering the plastic deformation [8]. It can be calculated as followed:

$$TC = CSA \cdot YS \tag{5}$$

Where TC is Tensile Capacity, CSA is Cross Section Area of the pipe, YS is Yield Strength of the material.

# 4. Design criteria

This chapter will cover criteria that are required to successfully design the SmartCock. The first sub chapter describes what Moonshine demands from the design, and the second sub chapter quotes some specifications from relevant standards.

## 4.1 Moonshine criteria

The design must be similar as the original patent description and drawings, as mentioned earlier the granted patent is titled “An improve drill string safety valve device” and can be found in *Attachment 3*.

Moonshine has a goal to limit the weight of the valve to **under 25 kg**. The reason for this is that all rigs that are operating on the Norwegian continental shelf, follows The Norwegian labour Inspection Authority guidelines which recommend that single lift by one person shall not exceed 25 kg [9].

The SmartCock should be designed for use with 5 7/8” drill pipes. This is a considerable large size, and if a weight below 25 kg is achieved at this dimension it is likely to be achieved for smaller dimensions as well. The end sections of the 5 7/8” drill pipe, referred to as tool joints, have an Outer Diameter (OD) of 7” (177.8 mm). The SmartCock can therefore not exceed the **177.8 mm** diameter.

The connection is requested to be designed as **MT57** connection. The MT57 thread profile is designed by DP masters, and is required an authorised license to use. Moonshine is not in possession of this license, and because of the thread systems confidentiality the connection profile is now only made for cosmetic reasons. But should not need much configuration when specifications are available.

The maximum working pressure rating for the valve is designed to be **10k psi system**, which is a common working pressure rating on the Norwegian continental shelf.

MHWirth produces iron roughnecks that are used in drilling operation. It is a hydraulic machine that connects/disconnects drill pipes and downhole assemblies. One of MHWirth’s senior managers stated that a commonly used roughneck of theirs are equipped with 2-grip clamp system and a clamp force at **970 kN** [10]**.** A height requirement was also set to ensure that the roughneck would have enough outer shell to grip around, without interfering with the stem. Moonshine required a length of **390 mm** between the centre of the stem to the top of valve.

The ball valve is operated by turning the stem with use of a hex key. To limit the possibility for human error during operation, it is a criterion to only allow the stem to rotate 90 degrees in the design. It is also a criterion to implement a solution that secures the stem in locked position during downhole operation.

## 4.2 Criteria from standards

There are many applicable standards related to equipment used in the oil sector. For the design of SmartCock, two standards are mainly taken into account. *API Standard 53: Well Control Equipment Systems for Drilling Wells* and *API- Spec 7-1: Rotary Drill Stem Elements*. NS-EN ISO 10424-1 is similar to API-Spec 7-1 and is used for this project as a replacement for API-Spec 7-1.

**API Standard 53**

> "*Provides requirements for the installation and testing of blowout prevention equipment".*

The OD of the drill pipe safety valve shall be suitable for running into the hole.

Subsequent operational pressure testing (subsea BOP stacks) Drill pipe safety valve, frequency not to exceed 21 days with a high pressure test at maximum anticipated wellhead pressure [4].

**API- Spec 7-1 / NS-EN ISO 10424-1**

> *"The function of this part of ISO 10424 is to define the design and the mechanical properties of the material required for rotary drill stem elements (...), particularly applicable where there is innovative or developing technology".*

It is important to specify that this standard is requirements for upper and lower Kelly valves, square and hexagonal Kellys, drill stem subs, standard steel, and non-magnetic drill collars. SmartCock is a valve that is not included as any of these elements that the standard cover, however, it will still follow the design specifications.

The standard defines to types of classes of the drill string safety valves. Class 1 is a valve that only can operate on surface, while a class 2 can operate both on surface and downhole. The SmartCock will operate downhole and therefore must follow specifications for class 2.

NS-EN ISO 10424-1 indicates that for 10k psi systems, the hydrostatic shell test for new valves should be performed at 15k psi. For the test to be approved it should have no visually detectable leakage during the test.

- Working pressure test from below is conducted with a closed valve and pressurised from bottom with the hydrostatic shell test pressure (15k psi).
- Working pressure test from above, this test only applies to class 2 type valves. It is stated in ISO10424-1 5.4.3.3 *"This testing applies to valves with ball-type closure mechanisms only"*. The new improved drill string safety valve has a ball type closure with a check-valve function and will therefore not seal with pressure above.
- External Stem-seal pressure test shall be conducted with minimum 2000 psi.
- The average impact value of the three specimens (in accordance with ISO 148) shall not be less than 42 J, with no single value below 32 J when tested at -20 °C [11].

# 5. Modelling phase

This chapter will cover the modelling and engineering of SmartCock. For an easier handling of the design specifications, all criteria are grouped into four factors. They are featured as dimension, force, functionality, and weight. The specification statements are placed in their respectively factors, for example OD of 177.8 mm is a dimensioning factor. According to the Shigley’s design process, the modelling of SmartCock is the *synthesis* phase where all information is systemized and combined to create a prototype. The end of this chapter will cover what materials that are selected and a summary of the total weight. Technical drawings created during this chapter are found in *Attachment 1*.

## 5.1 SmartCock design

The SmartCock is designed part by part. All parts interact with other parts which makes the design process complex. A small change in one part leads to corrections in several other parts, for the components to still cooperate with each other. To develop the complete assembly, it is necessary to start with essential parts which dominates the rest of the configuration. According to the design criteria, most of the crucial development objectives are related to the Body. This is the leading component which sets the initial parameters. The other components are designed to fit the Body while still attaining the design specifications.

Six sub chapters are presented in this chapter. All eleven items/parts in the Figure 6 is described in dedicated sub chapters as follows:

- 5.1.1 Body: Item 1
- 5.1.2 Ball Valve Assembly: Item 6, 8 and 9
- 5.1.3 Spring: Item 10
- 5.1.4 Stem and Interlock: Item 2,3,4 and 5
- 5.1.5 Sleeve: Item 7
- 5.1.6 Retainer Cap: Item 11

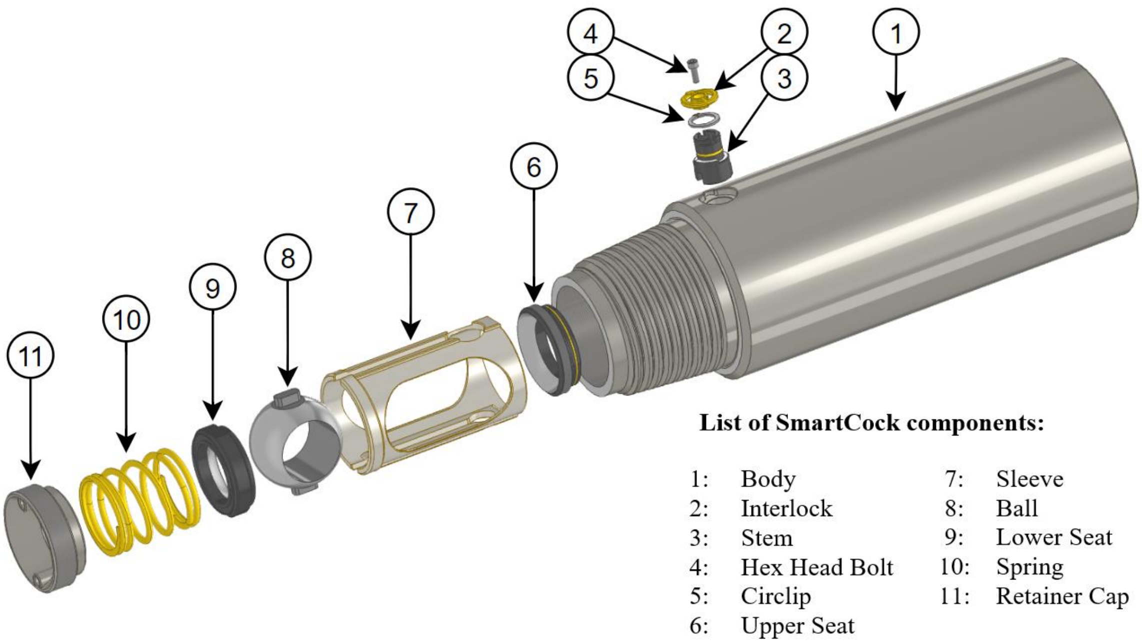



Figure 6: SmartCock components

### 5.1.1 Body

The Body is complex and vast size compared with other components, which means that all the 4 factors are taken into consideration for its design. One of the factors are its weight, the Body covers more than 85 percent of the total mass of the SmartCock. It is also the component that absorbs most forces and therefore also a factor to consider. It is fundamental to find a well figured out balance between the Body being able to resist the forces and still being as light-weight as possible. It also needs to fulfil some extensive design criteria related to dimensions and functions. For a better comprehension of features and later clarifications concerning the Body, an overview picture of the Body will be presented in Figure 7 & Figure 8.

**List of body features:**

- A-1: Pin Threads
- A-2: Pin Thread Seal Shoulder
- A-3: Circlip Slot
- A-4: Stem Seal Face
- A-5: Internal Constrain Slot
- A-6: Pin Thread Makeup Shoulder
- A-7: Retainer Cap Threads
- A-8: Upper Seat Shoulder
- A-9: Inner Diameter
- A-10: Cutout for Flow Optimization
- A-11: Cutout for Valve Pinion
- A-12: Sleeve Anti-Rotation Slot
- A-13: Upper Seat Seal Face
- A-14: Weight Reduction Cutout
- A-15: Outer Diameter (Pin/box diameter)
- A-16: Box Thread Seal Shoulder
- A-17: Wall Thickness
- A-18: Drill bore diameter
- A-19: Box Thread Makeup Shoulder
- A-20: Box Threads

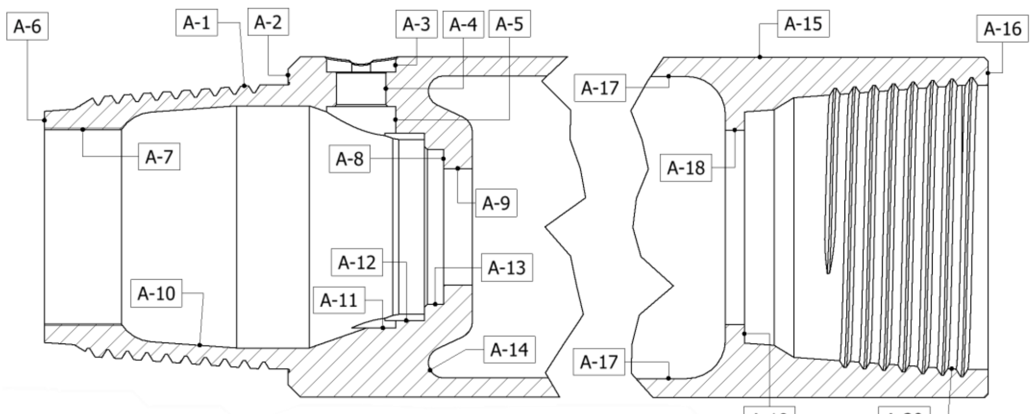



Figure 7: Body features

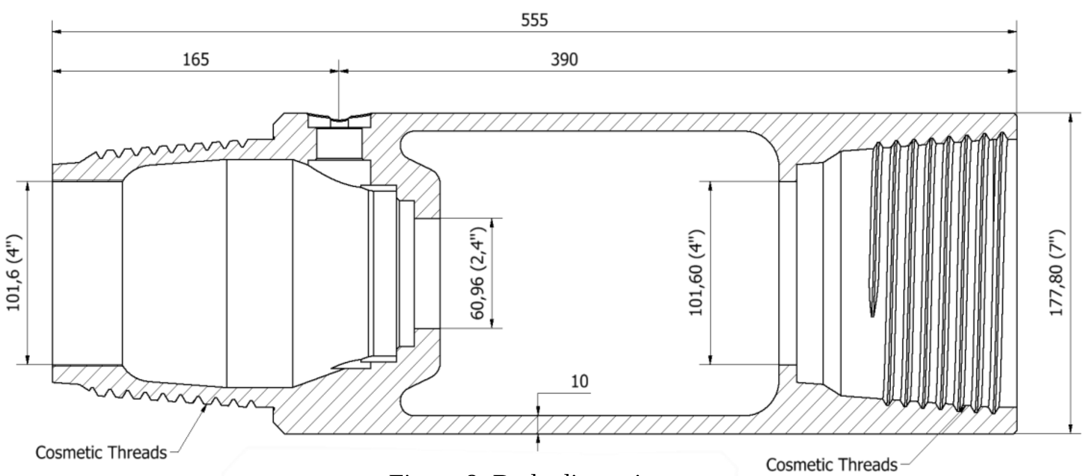



Figure 8: Body dimensions

The Body will have to withstand a large amount of forces, both external and internal. The roughest external force the Body will experience is generated by the clamp force. If the wall thickness holds out against the clamp force it should resist the rest of the forces as well. The gripping of the roughneck has been tested on different places and positions of the Body. It is concluded that the worst outcome came from the grip that was placed in the middle of body where it has no internal support. This leads to either the option to add more internal support or enlarge the wall thickness (A-17) for the specific area. It is then concluded that a wall thickness of 10 mm will withstand the clamp force from a 2-grip clamp system with a clamp force at 970 kN. This will be shown more in depth in the chapter *6.3 Clamp force on Body*.

The SmartCock designated pressure class is 10k psi, but new valves need to conduct a Factory Acceptance Test (FAT) at 15k psi. One of these tests are done with internal pressure towards a closed valve, with some simplified calculations of the pressure from the Upper Seat that would press towards the Upper Seat shoulder (A-8). Where *d* is the diameter of the Upper Seat, and *p* is the pressure class.

$$\mathrm{p} = \frac{\mathrm{F}}{\mathrm{A}} \rightarrow \mathrm{F} = \mathrm{p} \cdot \frac{\pi \cdot \mathrm{d}^2}{4} = 103.42\ \mathrm{MPa} \cdot \frac{\pi \cdot (80mm)^2}{4} \approx 520\ \mathrm{kN} \tag{1}$$

This means that the shoulder needs to withstand 520 kN. As seen in Figure 9, the area that is impacted needs a large amount of support and thickness. The upper part of the shoulder was initially 8 mm, this resulted in a tensile stress at 1500 MPa which will lead to plastic deformation for most materials. Further testing at 13 mm, 15 mm, 25 mm and 30 mm took place. After the analysis, a conclusion was set that the 15 mm thick upper shoulder would be able to withstand the pressure. The upper part of the shoulder is the thinnest part of the shoulder but still has 15 mm thickness. This force is also included in the internal pressure simulation and can be seen in chapter *6.4 Internal pressure on Body*.

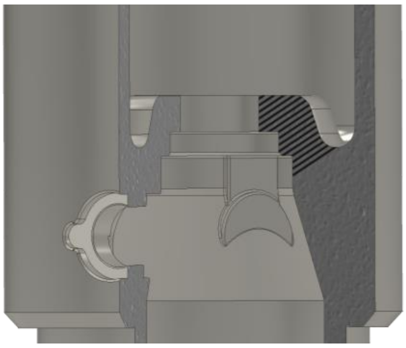


Figure 9: Upper Seat Shoulder

As mentioned, for weight reduction most of the focus has been on the Body, due its high mass percentage over 85% compared to other components. There are many parts of the Body that has a specific form and design that are unchangeable, such as the form of threads and the double make up shoulder, due to their function. To lower the mass, there are two particular areas that can significantly reduce the weight, without interfering with the dominating factors. One of them as mentioned being the wall thickness, initially the wall thickness (A-17) was 12 mm. This number is based on previous experience as a good starting point for the project. After some simulation testing and calculations, the option to reduce the wall thickness to 10 mm was an option up for further testing. All test result for 10 mm walls was approved, and as a result of lowering the thickness on the Body with 2 mm the weight got reduced with 1.1 kg.

Another possibility was to remove excessive mass at A-14. The difference between original design and with the weight reduction cutout can be viewed respectively in Figure 10. This is an odd way to reduce the weight but is required to fulfil the primary weight limit under 25 kg. With this method a more extensive simulation regarding the upper seat shoulder had to be done, which resulted with positive outcome. There are also some uncertainties that may occur during the manufacturing of the Body, flow and fluid residue and will need further investigating. This method will reduce the Body weight with 1.06 kg.

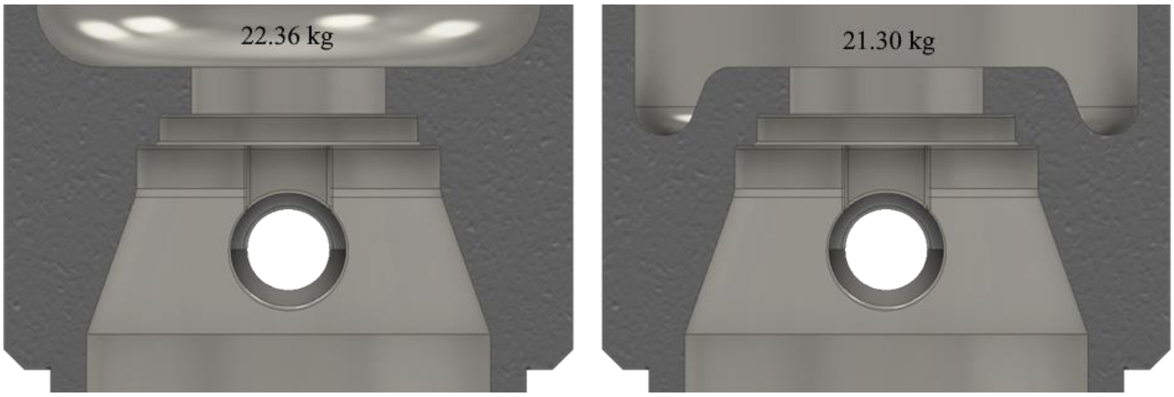



Figure 10: Weight reduction cutout

With the two reductions, the Body was able to cut off 2.16 kg and ends up at a total weight at 21.30 kg. For the Body to tolerate the forces it also needs a material with high yield strength. The material the Body will be using is Titanium Grade 19, further specifications and reasoning for material choice will be described in chapter *5.2 Material choice and assembly* .

Another factor is the dimensions that must be fulfilled, beginning with that there is no part of the Body that can exceed a diameter of 177.8 mm. The height criteria were set by Moonshine with a minimum height of 390 mm between the centre of the Stem and the top of Body. This is done to ensure that the roughneck has enough pipe wall to grip towards, without interfering with the Stem. These dimensions can be further seen in Figure 8.

The modeled pin and box threads on the Body are just for cosmetic purposes and suggests where the threads will be. The technology around the MT57 drill pipe connection, is owned by DP masters and require a certain license to use their connections. The design will be similar, as both the design and MT57 take advantage of a secondary makeup shoulder (A-6 & A-19), the primary sealing shoulder (A-2 & A-16) and the tapered API thread form. The reason for the double shoulder connection is that it will allow additional strength to withstand a higher makeup torque. The tapered threads will also allow faster connection time, this is done by allowing a deeper stabbing of the pin-end into the box connection. The deeper stabbing result in fewer turns to reach full makeup by the roughneck and a shorter time needed to connect the SmartCock to the drill string [12].

The possibility for the Sleeve to rotate or mechanically lock the Stem for rotation, could lead to the valve not sealing as intended. It is therefore made an anti-rotation slot for the Sleeve (A-12). The slot can be seen marked with lines in Figure 11, where it also includes another slot on the opposite side. The Sleeve will be held in place by the walls on each side of the slot. It is assumed that the Sleeve in general will not absorb any great forces that will react as a rotation force; therefore, it does not need much support to keep it in its place. The depth of the walls on each side of the slot is 3.3 mm.

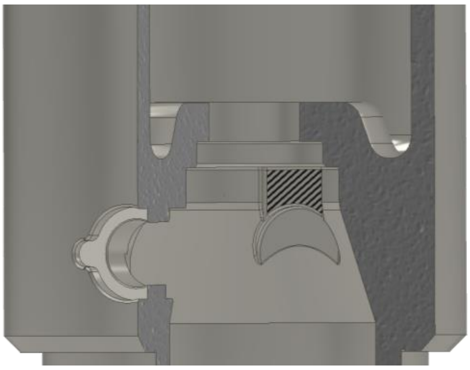


Figure 11: Anti-rotation Sleeve slot

The Stem hole in the Body have been made in such a way that the internal Stem shoulder prevents the Stem from launching out of the Body. Furthermore a 180-degree cut-out have been made to limit the rotation of the Stem. The reasoning for this will be explained later in chapter *5.1.4 Stem and Interlock*. The circular wall inside the Stem hole is intended to function as a seal surface (A-4). The depth of the hole is estimated to be deep enough to achieve a sufficient sealing area. The Stem will seal towards the internal wall by use of a soft seal. Soft seal and surface treatment are not further investigated in this project as this is not part of the scope.

SmartCock is a valve that will operate in downhole operations, this means that the SmartCock will experience both internal and external pressure. Because the Stem is installed from the inside a solution is required so that the external pressure do not launch the Stem back into the Body, potentially damaging the Ball Valve. It was briefly considered to thread the Stem into place but that could have led to displacement and rotation issues. A much more simplified outcome is to use a circlip. During downhole operation, the Stem must remain in position so that the Ball Valve can return to its sealing position. To lock the stem in place a slot have been added (seen in Figure 12) to allow the interlock to remain stationary, further explanation regard the interlock will be presented in chapter *5.1.4 Stem and Interlock.*

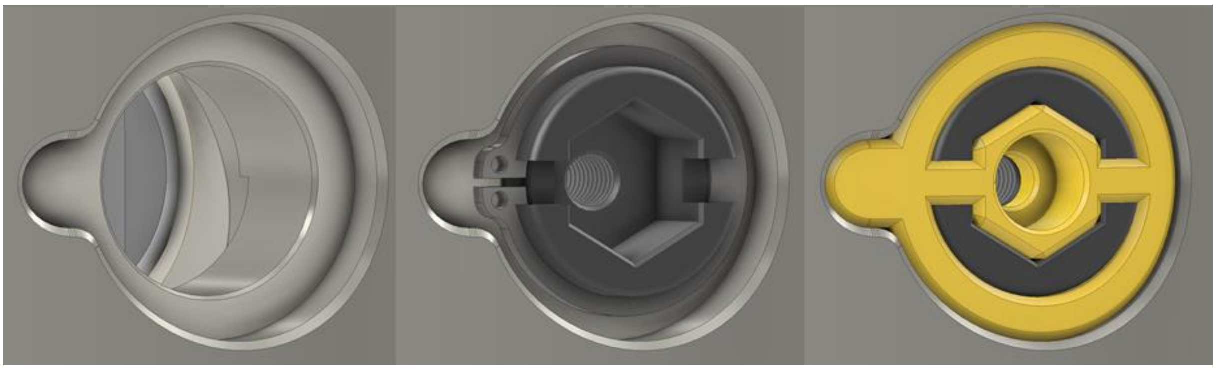


Figure 12: Slot for Interlock

### 5.1.2 Ball Valve Assembly

The Ball Valve Assembly consist of the Ball, Upper- and Lower Seat. The factors mostly considered in this design is the dimensions and functionality. Main dimensions to evaluate are related to the ball's inner and outer diameter. It is desirable with largest possible inner diameter through the bore to achieve low flow restrictions. The Ball Valve with the pegs/guide pins cannot exceed 101.6 mm due to the opening in the Body. The functionality in this design is important because of its specifications in the patent. Factors regarding weight are not prioritized since the integrity of these components cannot be compromised. It is not part of the scope in this project to simulate the forces that the Ball must cope with, but it is assumed a sensible mass for each component in the design.

Figure 13 shows the Lower Seat, Ball Valve and Upper Seat. They are assumed to be in a high alloy stainless steel, typical AISI 316L/AISI 316LN. The main function to the Ball is to allow or disallow flow through the valve. The Upper Seat will seal towards the Ball and Body. The Upper Seat has a groove (marked yellow in the figure below) that is intended for a soft seal. This will seal against the Body to prevent leakage between the Seat and Body. The Lower Seat will be an intermediate part between the Spring and the Ball. This is to transfer force from the Spring to push the Ball towards the Upper Seat. The Ball and Upper Seat are designed to be a metal to metal seal. This will require a surface treatment that will not take damage during operation of the valve. One method to achieve such a though surface is by use of a Tungsten Carbide coating. By thermal spraying components with Tungsten Carbide, they will achieve a hard surface coating. The Upper and Lower Seat will be coated on the surfaces that interacts with the Ball and the Ball itself will be coated on the complete spherical part [13].

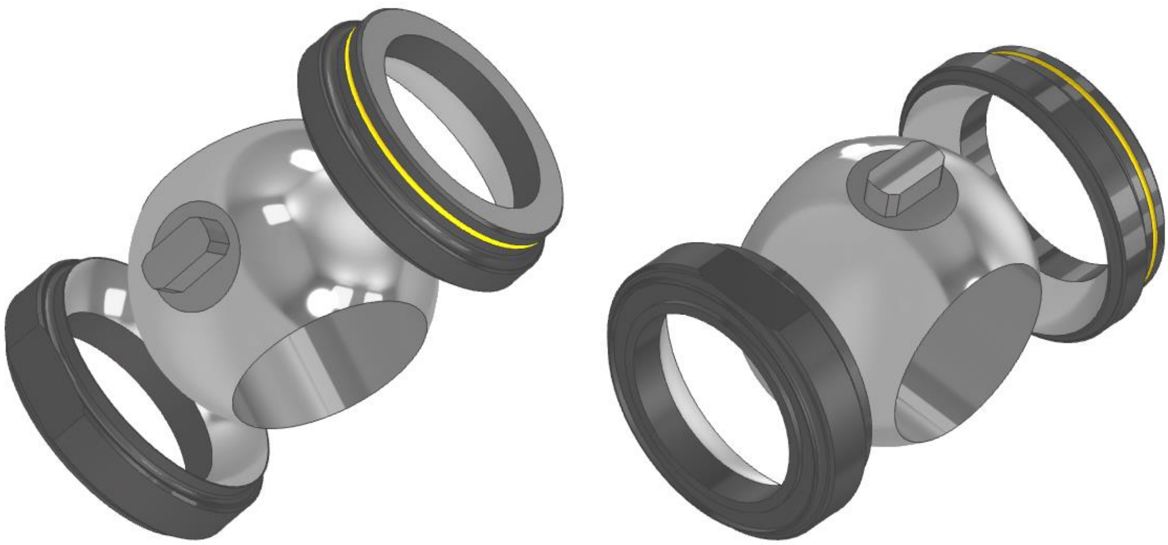


Figure 13: Ball Valve Assembly

The bore opening shown left in Figure 14 shows the smooth transition between the Seats and the Ball. The inner diameter (ID) through the Ball is 60.96 mm (2.4”). This is the largest ID that could be fit to this design. It should be sufficient though to evacuate fluid through SmartCock during installation onto the drill string.

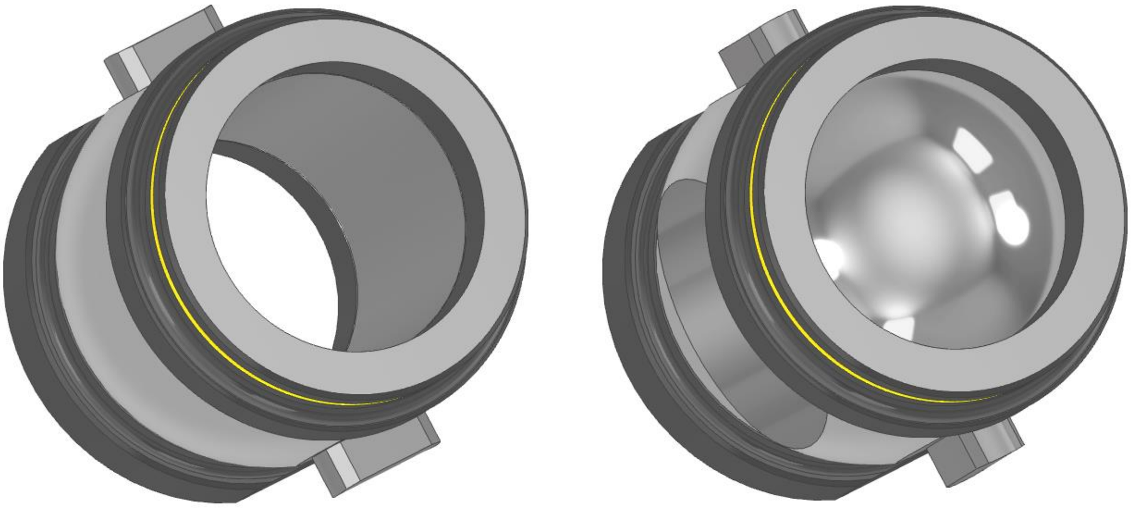


Figure 14: Ball Valve Assembly, open & close

Regarding the guide system there are some design technicalities that has been transferred from the Ball to the Stem. In Figure 15 below is arrow 13 pointing on the *first guide peg* (described in chapter *2.3 The improved drill string safety valve (SmartCock)*). This is a part of the Ball and it is where the Stem interfaces for operating the Ball. In the design carried out in this project is the *first guide peg* function transferred to be a part of the Stem. There are multiple reasons for this. Firstly, it reduces the size of the pegs which further allows for a larger ID. Secondly, the Stem will act as a piston during pressure differentials. To secure the Stem in an adequate way it is decided to install it from the inside. This makes the Stem bigger on the inside than outside. For everything to fit together, the *second guide peg* of the Ball will enter inside the Stem. The *second guide peg* is the pins on both sides of the Ball, and they match the corresponding slots in the Sleeve and inside the Stem. The Stem is further explained in chapter *5.1.4 Stem and Interlock*.

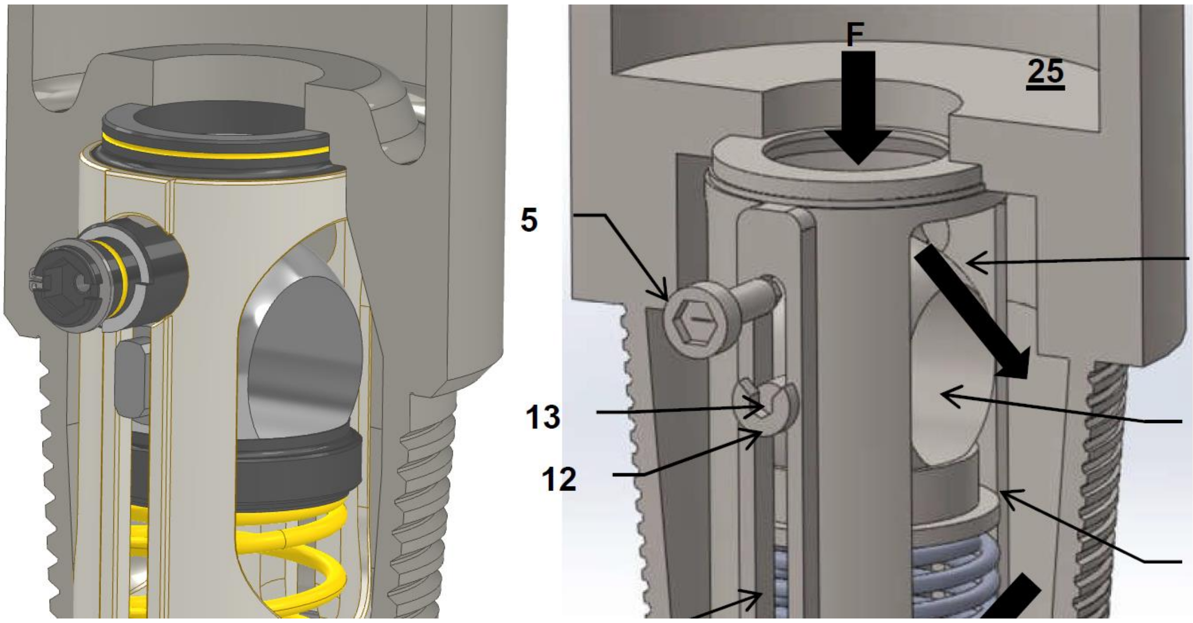



Figure 15: Comparison of guide system

For the valve to act as a check valve, the Ball must leave its Upper Seat during pump through operation. There is done assumptions and simplifications to calculate flow through a closed valve. The reason for calculating flow through the closed valve is that a rate of 800 litres per minutes is desirable as pump through capacity. It is therefore needed to check if that flow rate is possible to achieve with the design. Some of the parameters affecting the flow rate in addition to the dimensions are the mud specification, flow type (turbulent or laminar) and pressure. Kill mud is usually made on site and customized in each scenario it is used. Based on some inputs from the various range of density and viscosity of the kill mud character, it is used a density of 1800 kg/m$^3$ and a viscosity of 50 centipoise (0.05 Ns/m$^2$) in this calculation. The differential pressure needed to equalize the force from the Spring and start move the Ball is set to 3 bar. This means if there is 50 bar well pressure from below, it requires 53 bar pressure from above to equalize the force.

The calculation used to determine how far the Ball must leave the Upper Seat to sustain 800 litres per minute are only applicable for turbulent flows. It is therefore first calculated Reynolds number to check for turbulent flow as shown:

800 litres per minute is converted to SI unit m$^3$/s:

$$Q = 800\frac{\mathrm{l}}{\mathrm{min}} \cdot \frac{1\mathrm{m}^3}{1000\mathrm{l}} \cdot \frac{1\mathrm{min}}{60\mathrm{s}} = 0.0133\frac{\mathrm{m}^3}{\mathrm{s}} \tag{2}$$

The area of the ID inside the Body (marked A-9) is:

$$A = \frac{\pi \cdot d^2}{4} = \frac{\pi \cdot (61\mathrm{mm})^2}{4} = 2922.5\mathrm{mm}^2 = 0.002923\mathrm{m}^2 \tag{3}$$

Velocity in this area with a flow rate of 0.0133 m^3/s is:

$$V = \frac{Q}{A} = \frac{0.0133\frac{\mathrm{m}^3}{\mathrm{s}}}{0.002923\mathrm{m}^2} = 4.55\frac{\mathrm{m}}{\mathrm{s}} \tag{4}$$

Reynolds number is then calculated:

$$Re = \frac{\rho VL}{\mu} = \frac{1800\frac{\mathrm{kg}}{\mathrm{m}^3} \cdot 4.55\frac{\mathrm{m}}{\mathrm{s}} \cdot 0.061\mathrm{m}}{0.05\frac{\mathrm{Ns}}{\mathrm{m}^2}} = 9991.8 \tag{5}$$

The Reynold number is above 2300 which indicates turbulent flow in the area before the Ball if the flow is 800 litres per minute. It is then assumed turbulent flow between the Ball and Upper Seat because the area in that space is assumed to be smaller than the area of ID (A-9) in the Body.

Further is the formula *flow rate in a turbulent nozzle* used to find the smallest area needed to achieve 800 litres per minutes. The outflow coefficient "$\mu$" is set to 0.6 based on the geometry between the Ball and Upper Seat [7]. 3 bars are converted to 300000 Pa to fit the formula units.

The area is calculated as followed:

$$Q = \mu A \sqrt{\frac{2}{\rho}(p_1 - p_2)} \rightarrow A = \frac{Q}{\mu \cdot \sqrt{\frac{2 \cdot (\Delta p)}{\rho}}} = \frac{0.0133 \frac{m^3}{s}}{0.6 \cdot \sqrt{\frac{2 \cdot 3 \cdot 10^5 Pa}{1800 \frac{kg}{m^3}}}} = 0.001214 m^2 \tag{6}$$

The smallest area needed to sustain 800 litres per minute is equivalent to 1214 mm$^2$. To determine how far the Ball must leave the Upper Seat, Autodesk Inventor is used as calculation tool. The Ball is first moved slightly from the Upper Seat in Y-direction. This creates an opening area between the components where there is drawn a circular area shown blue in Figure 16. Then the Ball is moved furthermore in Y-direction until the area between the Ball and Seat reaches 1214 mm$^2$. At the point when 1214 mm$^2$ (1214.464 mm$^2$ in Figure 16) is achieved the Ball has moved approximately 10 mm from the Seat. This is a quite small distance which is beneficial in regards of the total weight. If the Ball would have needed a long moving distance to achieve sufficient flow area, this would have impacted the design of the lower part of the Body to be longer and thus heavier. Further is the assumed distance length used as one of the parameters calculating the Spring. It should be noted that the flow area is just a rough estimate and not used as an exact calculation.

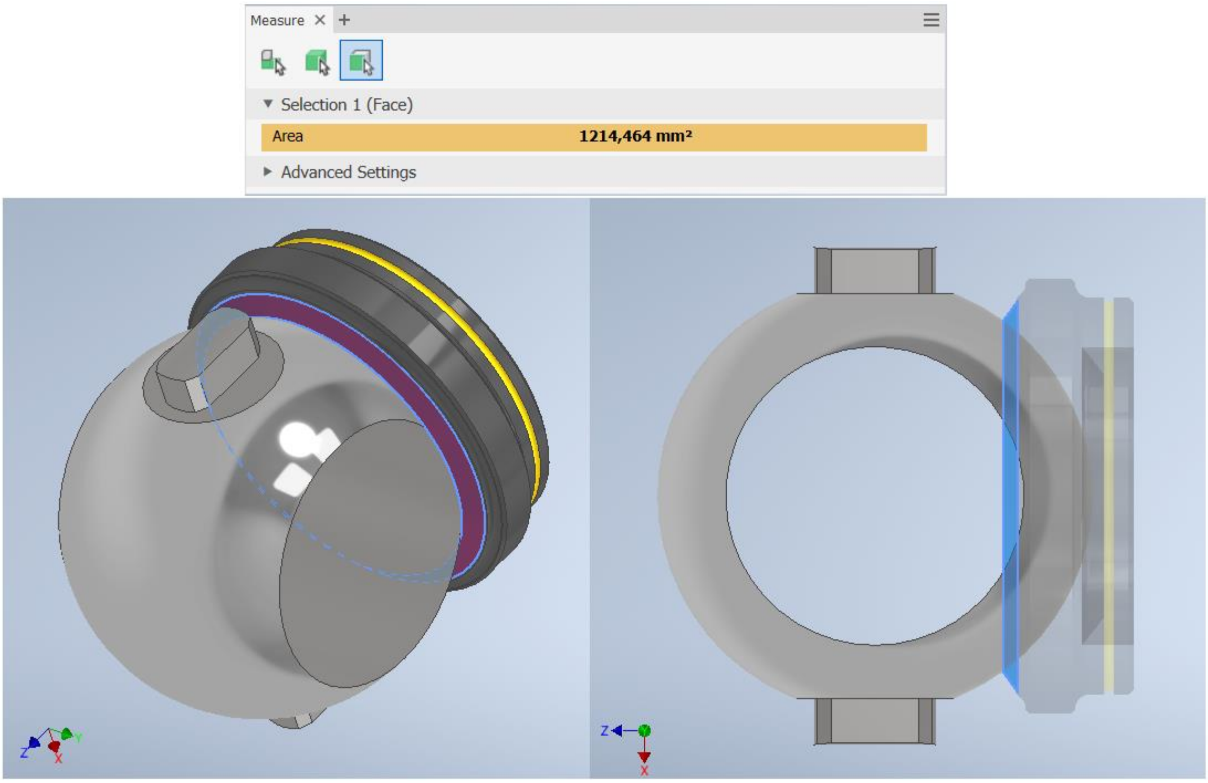



Figure 16: Flow area estimation

### 5.1.3 Spring

The Spring's function is to force the Ball towards its Upper Seat, while also being able to compress by applying pressure from above. The factors mainly considered are dimensions and force. The dimension is limited to the ID of the Sleeve and distance between the Lower Seat and the Retainer Cap seen in Figure 17. The force input is related to the pressure needed from above to start compressing the Spring.

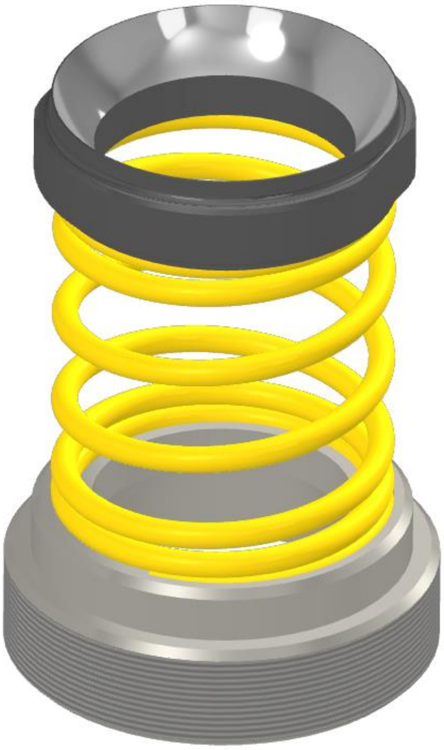


Figure 17: Spring interfaces

The pressure needed to start compressing the Spring is set to 3 bar differential pressure from above. This might be considered as a low pressure for its purpose, but it is set this way for two reasons. For an easier assembly phase, the compression force on the Spring cannot be too large. The pressure needed to compress the Spring during assembly, will be equal to the force needed for compression during downhole operations. Therefore, the force needed to compress the Spring cannot exceed the compression that makes it unfeasible to assemble. The second reason is related to the sealing arrangement between the Ball and Upper Seat which are intended to be metal-to-metal. The spring-force will constantly push the Ball towards the Upper Seat with the same force as the compression force. If the force is too large in combination with debris or other foreign objects on the seal surfaces, there are possibilities for damaging the sealing capabilities.

The force equal to 3 bars from above is calculated as followed:

$$\mathrm{F} = \mathrm{P} \cdot \mathrm{A} = 0{,}3\,\frac{\mathrm{N}}{\mathrm{mm}^2} \cdot 2923\mathrm{mm}^2 \approx 877\mathrm{N} \tag{7}$$

The Spring is designed with use of a *Compression Spring Component Generator* in Autodesk Inventor. This allows certain parameters to be dominating and the rest of the parameters will be calculated by the software. The dominating parameters can be seen in Table 1. The generator consist of two tabs, the *Design tab* (Figure 18) and the *Calculation tab* (Figure 19). Depending on the configuration some parameter might not be directly modified and will have a greyed-out background. In the first tab, parameters such as the Spring start and Spring stop specifications can be set. The generator allows for multiple "design types" which determines what parameters that will be dominant. This option can be found close to the top in the *Calculation tab*. The *Design type* closest to the dominating parameters, are the *F_8, D, Assembly dimensions*. However, the *Min. Load* is not possible to insert directly in this configuration. (Figure 19). To achieve a *Min. Load* of 877 N, the *Max. Load* is adjusted upwards until *Min. Load* reaches a desirable value.

| **Parameter** | **Value** | **Reason** |
|---|---|---|
| Min. Load Length | 87.8 mm | The length from Cap to Upper Seat while pump through closed. |
| Outside Diameter | 80.0 mm | Bigger than Cap ID and smaller than Sleeve ID |
| Min. Load | 877 N | Equal to 3 bar in this geometry. |
| Working Stroke | 10.0 mm | Based on calculations in 5.1.2 |

Table 1: Dominating Spring parameters

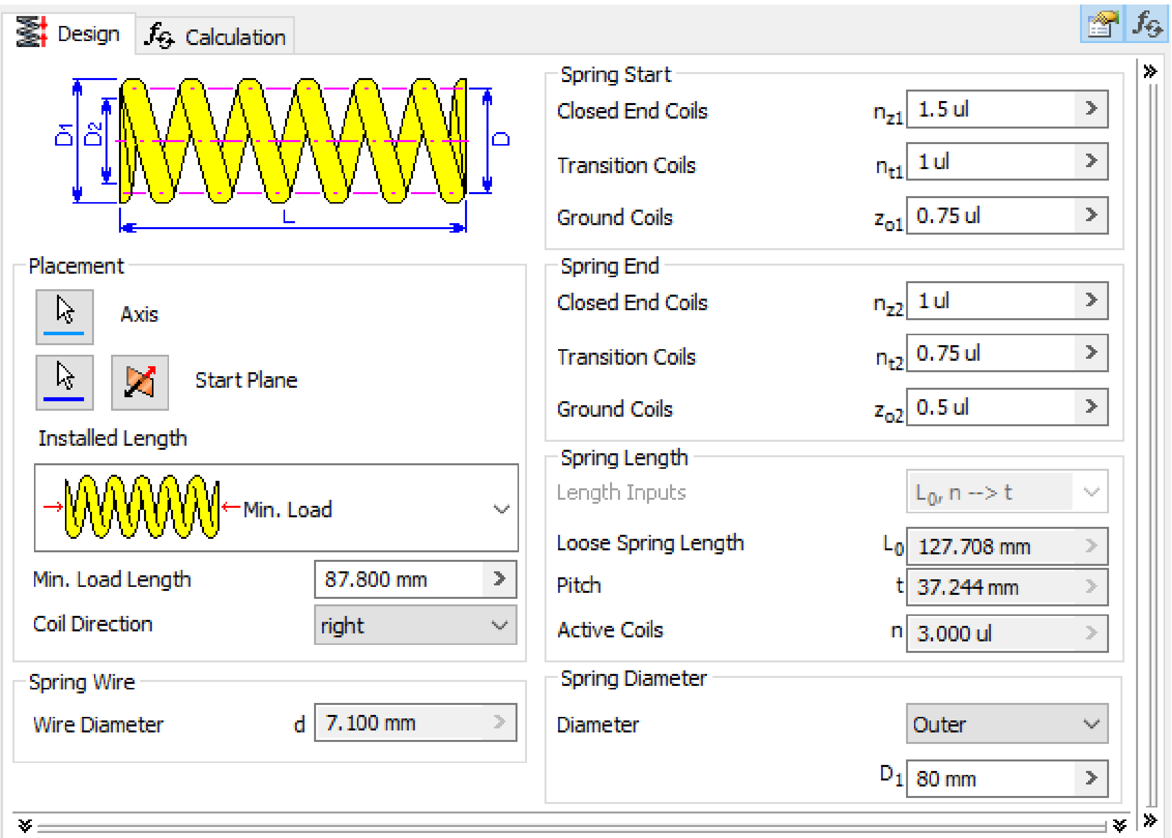



Figure 18: Compression Spring Component Generator; Design tab

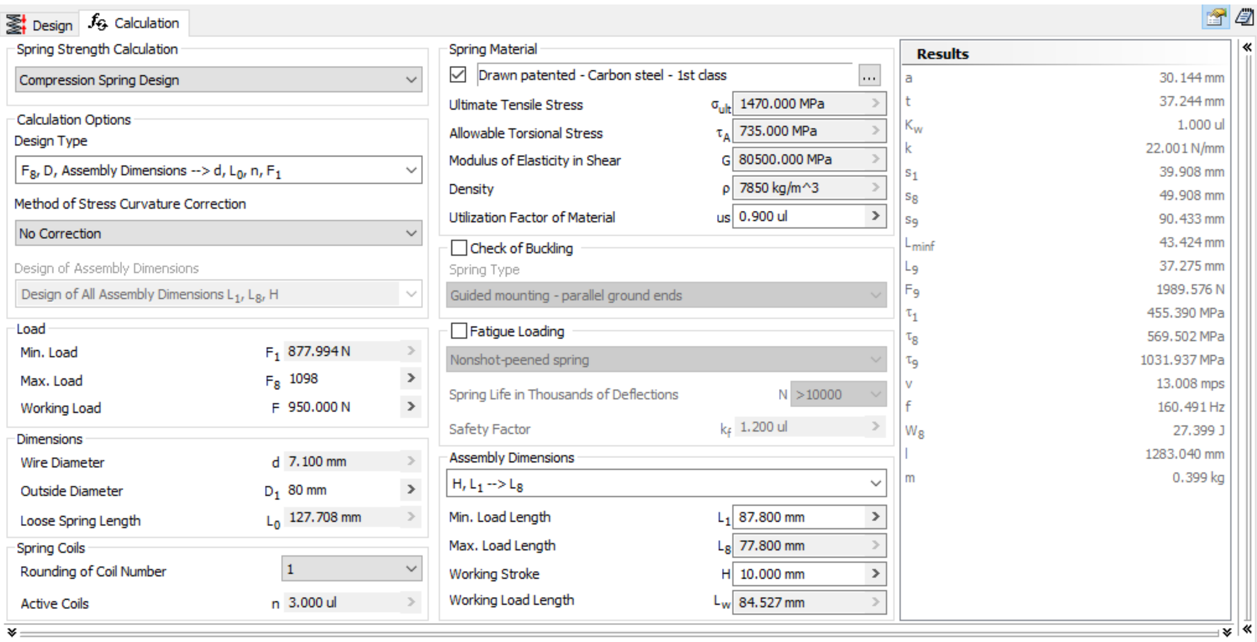



Figure 19: Compression Spring Component Generator; Calculation tab

The Spring generator calculates the Spring specifications from the input parameters. The full Spring specification sheet can be viewed in *Attachment 2*, the most relevant values will now be presented. The *Loose Spring Length* at 127.708 mm is the length of the Spring before installation into the SmartCock. The *Min. Load length* is already set to 87.8 mm, the length difference of 39.908 mm is the distance the Spring will have to be compressed during installation into the Body. Another important information is the *Wire Diameter* at 7.1 mm and *Active Coils* of three. This configuration is assumed to give sufficient flow through the Spring during pump through activity. The mass calculated by the generator do not consider that the ends are closed and grounded. This is a cut that makes the Spring fit the flat surfaces of the Lower Seat and Retainer Cap, seen in Figure 17 & Figure 20. The actual mass is calculated by its real volume in chapter *5.2 Material choice and assembly* .

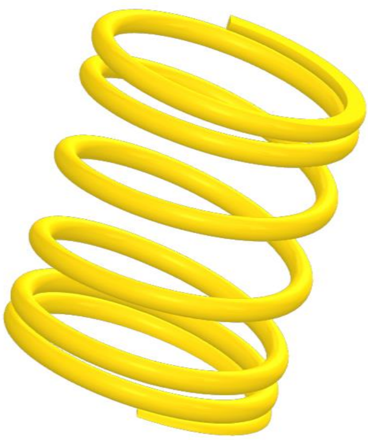


Figure 20: Spring

By adjusting the *Max. Load* to 1098N a *Min. Load* of 877N is obtained. The *Max. Load* is the force needed to achieve the 10mm distance between the Ball and Upper Seat during pump through. The calculation below shows the pressure equivalent to 1098 N:

$$P = \frac{F}{A} = \frac{1098\text{N}}{2923\text{mm}^2} \approx 0{,}376\frac{\text{N}}{\text{mm}^2} = 3.76\text{ bar} \qquad (8)$$

As a result of this calculation, the pressure range between closed and fully forced open valve is 3 to 3.76 bar.

### 5.1.4 Stem and Interlock

The Stem is used to operate the Ball Valve between open and closed position. The main factors considered in this design are functionality and forces. It should have a rotation limit, anti-rotation mechanism during downhole operation, and be adequately secured in the Body. Forces are considered since the Stem will act as a piston during pressure differentials between inside and outside of the Body. Dimensions are also considered since it is desirable with a sensible socket size to operate the Stem with. Weight factor is not considered in this part since it has a small influence on the total weight.

There are several features regarding the Stem, these can be seen in Figure 21. In the picture to the left can the operating socket for the hex key be seen, and in bottom of it there is a 6 mm threaded hole. The socket size is designed to be 5/8" and the threaded hole is used for fixing the Interlock, which is described later. The cut-out slots on top of the Stem are also for the Interlock. In the middle picture is the face and slot that interfaces with the Ball seen. In the picture to the right, two grooves are seen, the top one for a Circlip and the lower yellow one is for a soft seal.

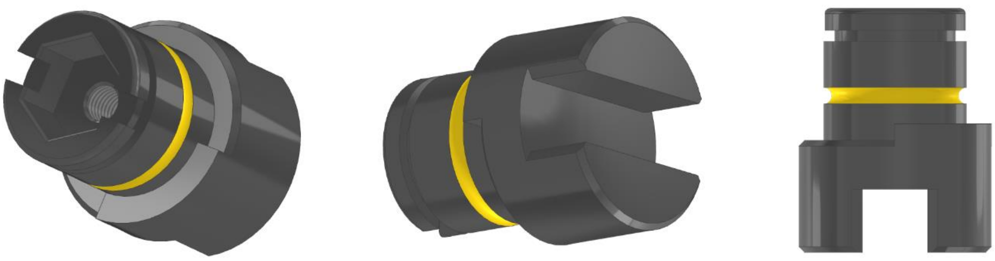


Figure 21: Stem

As mentioned in chapter *5.1.2 Ball Valve Assembly*, the Stem is designed to be assembled inside the Body and outwards, instead of opposite. Several alternatives were considered but to fulfil all functionalities this is the selected solution. When considering forces, the Stem will act as a piston, which means increased pressure leads to increased force trying to push the Stem out or in of the Body. SmartCock is intended to be rated as 10k psi equipment, this means FAT tests equal to 15k psi. The pressure tests from the outside of the Body is not a common test to carry out after FAT. Inside pressure test is usually carried out every other week offshore. That test is often 1.05 to 1.1 above working pressure. The reason to enlighten this is that the valve will most likely carry out many more inside pressure tests than outside. The Stem will either way have one side that is more resistant to deformation in this design, and it is decided to put the bigger contact surface on the inside to prevent it from bursting out of the Body. Firstly, it is a safety measure to prevent personal injuries during pressure tests. Secondly, it is believed to be easier to secure the Stem from the outside than inside with some sort of circlip system.

For safe and easy operation during use, it is desirable that the Stem is limited to only operate 90 degrees. This will give an effective operation between open and closed position in hectic situations. Different solutions have been considered to achieve this, some involving spring loaded systems and other fine mechanical techniques. Since SmartCock is intended to work in a though environment were functionality is critical, it is prioritized to keep functions simple and robust. The 90 degrees rotation solution is therefore chosen to be of a simple character. By lifting a quarter of the contact surface that faces the Body and cut half of the corresponding surface in the Body this will limit the rotation to only 90 degrees. This means that three quarters of the contact surface is active, and one quarter is always in “free air”. This will reduce the surface that absorb the force during pressures by a quarter. It is still estimated to be feasible in this design due to the generous contact surface originally.

Another functionality to suggest a solution for, is an anti-rotation mechanism of the Stem during downhole operation. It is decided to use an interlock function to prevent the Stem from rotating unintentionally. From an engineering point of view, an interlock is a function that interacts or engages with another system or part. This device, shown in Figure 22, is designed to secure the Stem when the Ball Valve is in a closed position. That is desirable if the SmartCock will leave the drill floor and be lowered into the well together with the drill string. If the pump through function is used the Ball Valve will leave its Upper Seat and the slot in the Stem. It is crucial that when the pump through is done the Ball returns to its Upper Seat. If the Stem has rotated out of position due to vibrations, then the pin/guide peg on the Ball might not enter the slot in the Stem. It will then not go to closed position and the valve will leak from below.

The Interlock is designed to fit inside the 5/8” operating socket on the Stem. The middle part will absorb rotational force and transmit it to the small pin that matches a slot on the outside of the Body, this restricts rotations of the Stem. The curvature of the top surface, shown in the right picture below, follows the curvature on the Body which give a smooth overall surface that reduces chances of snagging equipment during downhole operation. It will be held in place by a 6mm machined hex head bolt.

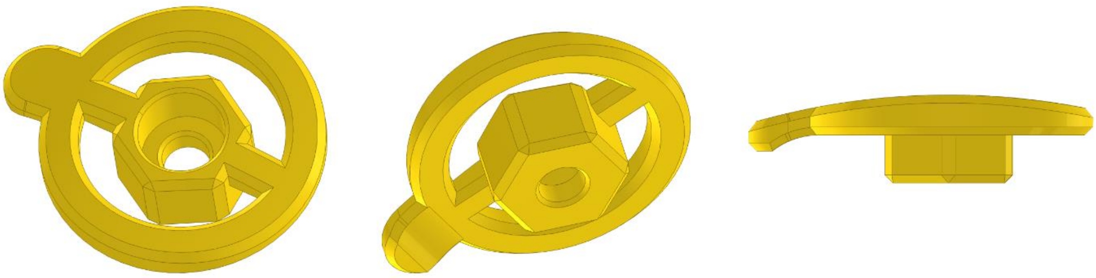


Figure 22: Interlock

Figure 23 shows a 6 mm hex head bolt and a Circlip. The 6 mm bolt used to secure the Interlock is suggested to be in a stainless-steel material. The bolt will not likely need customization for use in the SmartCock. The Circlip shown to the right is used to keep the Stem from translating into the Body. If there is differential pressure between inside and outside of the Body during downhole operation will the Stem act as a piston. If the pressure is greater on the outside, the Stem will be forced into the Body. The Circlip will prevent the Stem from move into the Body. The Circlip will need further research into what mechanical properties are needed. It is assumed that a Circlip or similar solution would need customization to fit its purpose.

Figure 23: 6 mm hex head bolt & Circlip

When the SmartCock is in standby on drill floor during drilling operations, the valve is initially in open position. The Stem and Circlip are always installed in the SmartCock while the Interlock and 6 mm bolt must be stored in a suitable place on the drill floor. In the Figure 24, the left picture shows the Stem in open position. The white marking on the Stem is intended to follow the direction of flow. After installation onto the drill string, the valve can be closed to the position shown in the middle picture. The white marking will then be in transversal direction of the flow direction. If the next operation is to go downhole then the Interlock will be installed as shown in the right picture.

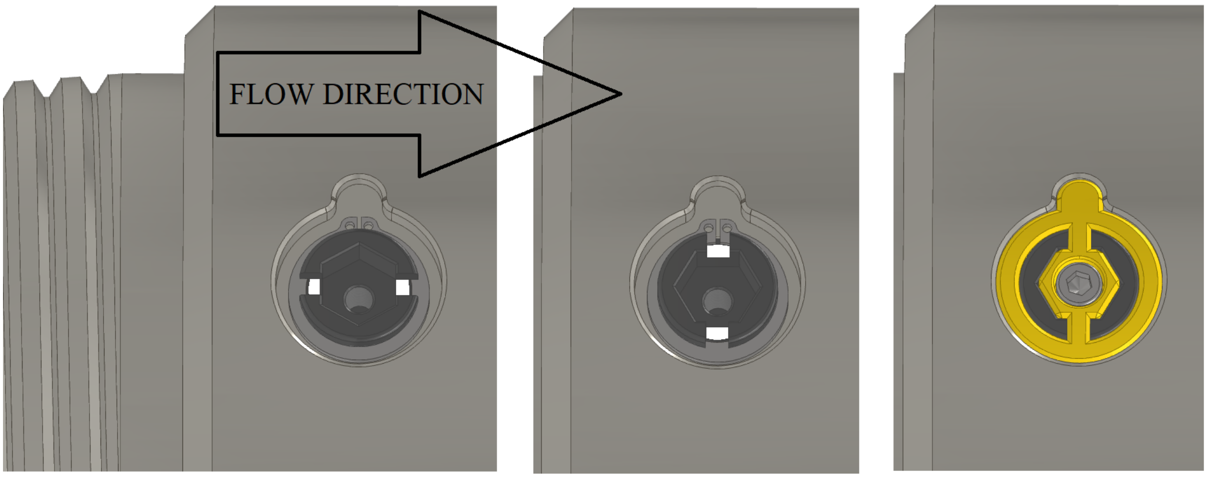



Figure 24: Marking of valve position

Figure 25 shows a half cut of the Body, and all the parts regarding the Stem functions are visible. The quarter lifted contact surface is seen middle to the left on the Stem. The Stem is secured from translate into the Body with the Circlip and the Interlock prevents it for rotation. The Stem is fitted to the Body through the Sleeve.

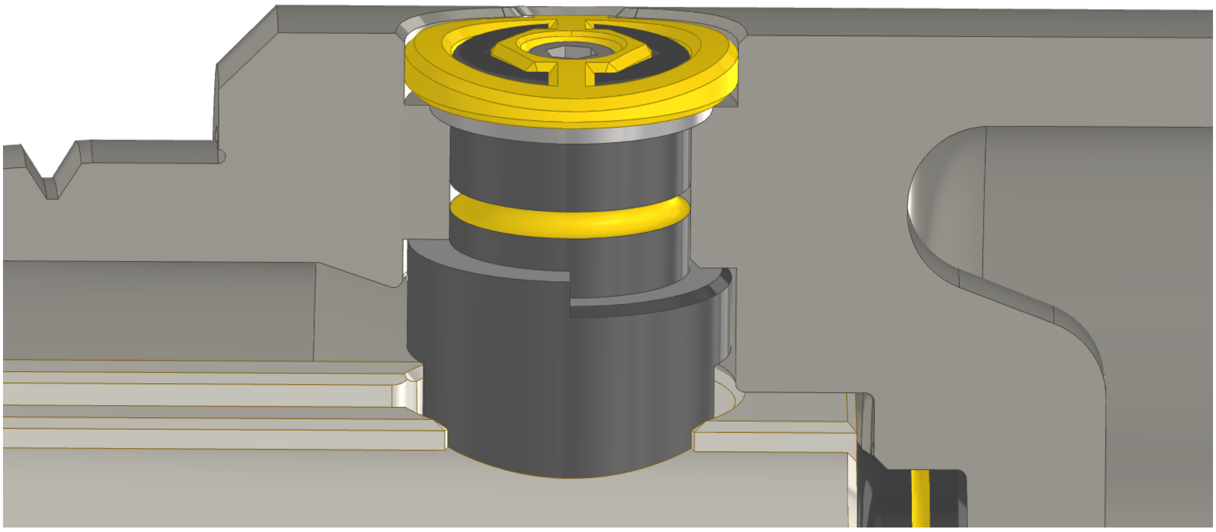


Figure 25: Full equipped, Stem components

### 5.1.5 Sleeve

The Sleeve's (referred to as valve cage in the patent description) main task is to house the Ball and Seats. The factors mostly considered in this design are related to dimensions and functions. The OD is limited by the opening in the Body and the ID is desirable to be as big as possible to achieve a large Ball Valve. The Sleeve will interact with many parts and are generally designed to serve their functions. To fulfil the patent description, it is important that the Sleeve only allows the Ball to rotate in upper position and not during pump through. The mud flow will also have to be led through the Sleeve. Some sort of anti-rotation support is needed keep it in place. It is assumed that the Sleeve will not absorb any great forces in this design, but it will transfer the make-up torque from the Retainer Cap to the Upper Seat. The weight factor is not considered since its functionalities requires a quite slim design initially. But when choosing material for this component it is optimal to choose a lightweight material as aluminum or titanium.

In Figure 26 below is the designed Sleeve shown. On the left picture is the hole for the Stem visible. The bottom is thickened to match the ID of the Body which will support the Sleeve together with a wall from the Retainer Cap. In the middle picture can the guiding slot for the Ball be seen. The guiding slot is open all the way from the bottom for the Ball to be installed into the Sleeve. Inside there is a flat surface where the guide slot is. That surface matches a flat face on the Lower Seat and Ball, to prevent rotation of them along longitudinal axis (normal to Ball operation axis). It should also be noted that the hole opening for the Stem is only designed on one side and the other side only matches the guide pin of the Ball to allow rotation in upper position. This means there is only one correct way to install the Sleeve into the Body. On the right picture is two big openings seen in front and back of the sleeve. That is to allow flow through the Sleeve during pump through. On top of the Sleeve on both sides there are a rectangular surface that is extruded and matches dedicated slots in the Body to prevent rotation of the Sleeve. No significant forces are estimated to occur in this direction, it is just to simply keep it in place. The Upper Seat is intended to be shrink-fitted to the top of the Sleeve. This is an interference-tolerance where one of the parts are heated or cooled before mounting. When the temperature equals, the parts will have a strong frictional connection.

Figure 26: Sleeve

Figure 27 shows the estimated flow path through the valve and Sleeve during pump through with closed valve. The Ball is not visible since it is hidden behind the Sleeve. After the mud has passed the Ball, it will escape the Sleeve, go between the Lower Seat and Body then enter back into the Sleeve through the Spring coils. The mud will leave SmartCock through the Retainer Cap and enter the drill string beneath.

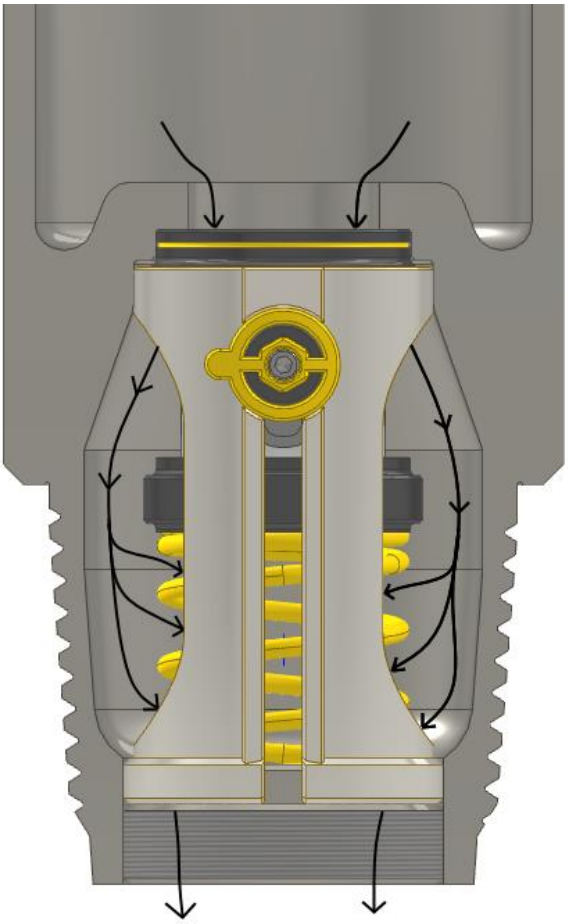


Figure 27: Flow pathway

During a technical appraisal meeting with MRC Global, an evaluation was done regarding whether the impact of fluids flowing through the Spring coils could lead to erosion issues on the Spring [14]. This is an issue that needs further investigation, but an assumption was made that the Spring should be able to withstand the erosion for the short period of time during the pump through sequence. The valve would have to go through a post operation maintenance or overhaul after use anyways. If the Spring collapses due to erosion, it could lead to functionality and integrity issues. If the Spring breaks it can be crushed to several pieces that can clog and disturb the circulation system. This could lead to serious integrity issues during a kick situation and is of course not desirable. With enough pressure from below, the Ball will still go back to sealing position without the Spring, but it will be very difficult to kill the well by use of the pump through function.

### 5.1.6 Retainer Cap

The Retainer Cap is designed to keep the internal parts in place and slightly compressed. The factors mainly considered are related to dimensions, functionality, and weight. The OD must correspond the ID of the Body. Some functions it must fulfil is to support the Sleeve and have a groove adapted for the Spring. Weight measures have been carried out due to excess goods possible to remove.

Figure 28 shows the designed Retainer Cap. On the picture to the left are two of center holes visible, they are interface spots for a spanner wrench to be used during installation to the Body. The Retainer Cap is designed to be threaded onto the Body. Thread specification are not carried out, but it is suggested that a fine thread application is beneficial. The Cap will absorb the force from the Spring and transfer it to the Body through the threads. On the right picture can the inner groove be seen, that is where the Spring will centralize and be supported. The outer edge above the threads, interfaces with the Sleeve. The elevated inner part creates a wall that support the Sleeve from collapsing inwards. Weight reduction is done by carving the front face seen on the left picture to a sensible balance between mass and function. For simplicity and galvanic corrosion reasons, it is recommended to have the Retainer Cap in the same material as the Body. This will also keep the weight low since the Body is in a lightweight material.

Figure 28: Retainer Cap

Figure 29 shows how the Retainer Cap interfaces with the different parts. In the left picture is the Cap installed to the Body. The middle picture shows how it will support the Spring. The right picture shows both Sleeve and the Spring supported by the Retainer Cap.

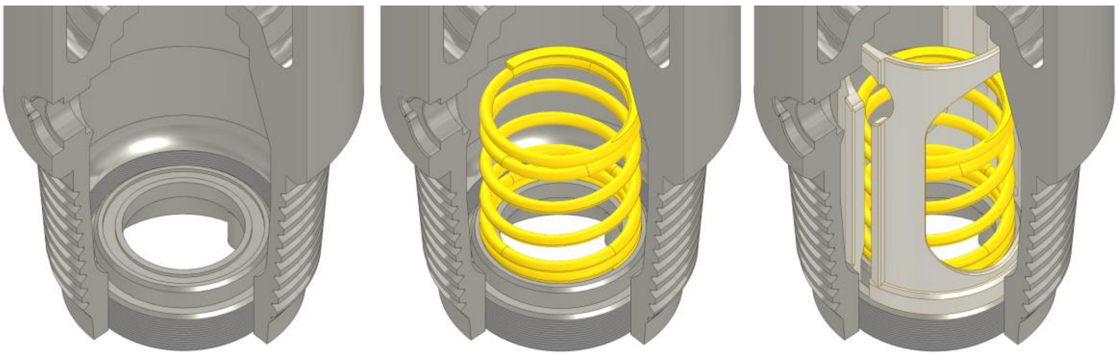


Figure 29: Retainer Cap interfaces

## 5.2 Material choice and assembly weight

After the prototype is created, continues the *Analysis and Optimization* phase in the Shigley's design process. One of the analysis carried out in this project is regarding the weight. The reason for this is that one of the main objectives in this project are to achieve a total weight below 25 kg. To accomplish that, materials for each component must be carefully chosen. The main factors considered are weight and force, which in this case is density and strength of the materials. When the project was handed over by Moonshine, titanium was suggested as main material for the assembly. This material is known for its relatively good strength-to-weight ratio. Like other materials, titanium also has many different alloys. For this project, a suitable titanium alloy must be chosen to fulfil SmartCock's purpose. Material choice of the Body is the most decisive factor regarding the total weight.

The first component to choose a material for is the Body. An analysis has been carried out regarding different titanium grades with use of a material datasheet provided by Moonshine, which can be viewed in *Attachment 4*. Three grades have been selected as potential options (listed in Table 2). There is benefits and drawbacks with all three of them. Grade 5 is the most widely used titanium, but the yield strength is considered to be too low in this case. Grade 21 has good mechanical properties, but it is described to be best suited for use above 300 ℃ and has slightly higher density compared to Grade 19. Grade 19 has good mechanical properties and has already been proven as a suitable alloy used in downhole production equipment. One of the reasons for this is the remarkable resistant to corrosion pitting in environments containing $H_2S$. Grade 19 is therefore the chosen material for the Body in this project.

| | **Titanium Grade 5** | | **Titanium Grade 19** | | **Titanium Grade 21** | |
|---|---|---|---|---|---|---|
| | Min value | Typical value | Min value | Typical value | Min value | Typical value |
| Yield Strength | 825 MPa | 910 MPa | 1105 MPa | 1150 MPa | 965 MPa | 1100 MPa |
| Ultimate Strength | 895 MPa | 1000 MPa | 1170 MPa | 1250 MPa | 1030 MPa | 1150 MPa |
| Hardness | - | 330-390 HV | - | 360-420 HV | - | 360-420 HV |
| Modulus of elasticity | - | 114 GPa | - | 102 GPa | - | 72-85 GPa |
| Charpy V-notch Impact | - | 20-27 J | - | 11-16 J | - | 103-110 J |
| Elongation in 50mm | 10 % | 18 % | 6 % | 9 % | 6 % | 10 % |
| Reduction in area | - | 20 % | - | 30 % | - | - |
| Density | - | 4.43 g/cm$^3$ | - | 4.82 g/cm$^3$ | - | 4.9 g/cm$^3$ |

Table 2: Material comparison

When comparing Titanium Grade 19 with requirements from NS-EN ISO 10424-1: *Rotary drill stem elements* (summarized in Table 3), there are two aspect that raise some uncertainty. The Charpy V-notch Impact value for the Grade 19 is lower than the minimum value that is stated in the standard. This needs further analysis due to different methods to measure the Charpy V-notch Impact between the material data sheet and the NS-EN ISO 10424-1. Elongation property in Grade 19 is also slightly below the specified value in the standard, but also here there are uncertainties in the different test methods. The hardness of 360-420 HV is estimated to be above the requirement of 285 HBW. The yield strength and tensile strength of Grade 19 is within the requirements.

| **ISO 10424-1: Table A9, OD range 177.8-279.4 mm** | |
|---|---|
| Yield Strength min. | 689 MPa |
| Tensile Strength min. | 931 MPa |
| Charpy V-notch Impact | Avg. 42 J, no single >32 J |
| elongation min. | 13 % |
| Brinell hardness min. | 285 HBW |

Table 3: ISO 10424-1 Material criteria, for OD 177.8 mm

The Ball Valve Assembly is chosen to be in AISI 316 LN. This alloy is often found in other ball valve assemblies used in the offshore sector. Another reason for the material choice is its high density, to give more leeway in material options in future designs. The Tungsten Carbide coating will typical have a metallic binder consisting of nickel, cobalt, chromium and molybdenum [13]. It is estimated that this coating will not have a considerable impact on the total weight.

The Sleeve is selected to be in a pure titanium alloy with low chemical composition of other elements. Grade 1 to 4 are typical pure titanium alloys with the same density of 4.51 g/cm$^3$. The mechanical properties generally raise with increased grade. Grade 3 is chosen to give a low weight and good mechanical properties, more detailed information about the alloy is found in *Attachment 4.*

The Retainer Cap is selected to be in Titanium Grade 19, the same material as the Body. This will be beneficial in regards of galvanic corrosion and thread properties. The Stem is also chosen to be in Grade 19 due to the forces that will be generated during pressures. Also, it is beneficial in regards of thermal expansion and galvanic corrosion.

The Spring material is chosen to be in a carbon steel alloy in regards of mechanical properties. It is possible to change the material into titanium on a later basis if needed but it is assumed to be more complex and expensive.

The Interlock is chosen to be in a stainless-steel material. It will most of the time be stored in a suitable place together with the 6 mm bolt, and when in use it should only prevent rotation due to vibrations. The 6 mm bolt and Circlip are selected in a steel material. It is not investigated further which steel material is most beneficial.

Table 4 shows a summary of all parts and their impact on the total weight. One can easily see that the Body have the largest impact with its 85.75% of the total weight. That is why it has been prioritized during weight reduction measures. With the selected materials in this design, the total weight ends up at 24.85 kg.

| Part | Weight | | Percent | Volume | | Material | Density | |
|---|---|---|---|---|---|---|---|---|
| Body | 21309 | g | 85.75 | 4420.97 | $cm^3$ | Titanium G19 | 4.82 | $g/cm^3$ |
| Upper Seat | 390 | g | 1.57 | 48.74 | $cm^3$ | AISI 316LN | 8.00 | $g/cm^3$ |
| Lower Seat | 393 | g | 1.58 | 49.14 | $cm^3$ | AISI 316LN | 8.00 | $g/cm^3$ |
| Ball | 1157 | g | 4.66 | 144.59 | $cm^3$ | AISI 316LN | 8.00 | $g/cm^3$ |
| Sleeve | 498 | g | 2.00 | 110.53 | $cm^3$ | Titanium G3 | 4.51 | $g/cm^3$ |
| Retainer Cap | 579 | g | 2.33 | 120.14 | $cm^3$ | Titanium G19 | 4.82 | $g/cm^3$ |
| Stem | 117 | g | 0.47 | 24.34 | $cm^3$ | Titanium G19 | 4.82 | $g/cm^3$ |
| Spring | 364 | g | 1.46 | 42.75 | $cm^3$ | Carbon Steel | 7.85 | $g/cm^3$ |
| Interlock | 31 | g | 0.12 | 3.85 | $cm^3$ | SS | 8.00 | $g/cm^3$ |
| 6 mm bolt | 7 | g | 0.03 | 0.89 | $cm^3$ | Steel | 7.85 | $g/cm^3$ |
| Circlip | 5 | g | 0.02 | 0.62 | $cm^3$ | Steel | 7.85 | $g/cm^3$ |
| Total | 24850 | g | 100.00 | | | | | |

Table 4: Total weight summary

Figure 30 shows how the SmartCock look like with this design. One quarter of the Body is cut in the picture for visualisation of the internal parts. The valve is designed in a way that makes it feasible to assemble. The components interact with each other to fulfil all functions, but engineering tolerances are not considered. That is not part of this project and will have to appear on a later basis.

Figure 30: SmartCock assembly

# 6. Simulation runs

Continuing the *analysing and optimization* phase in Shigley's design process, a new analysis related to forces and stress are carried out. In this analysis will Von Mises stress and displacement be the centre of attention. This chapter is divided into separate simulations to evaluate the digital prototype in different scenarios specified by the design criteria. In the simulation software *linear static* is used for the structural analysis. The simulation objectives in this project are related to the Body and Stem. Both components are chosen to be in Titanium Grade 19, which have a minimum yield strength of 1105 MPa and a modulus of elasticity at 102 GPa.

## 6.1 Tensile stress on Body

Tensile stress on the Body is carried out to check what pulling force the top drive can apply to the SmartCock before plastic deformation is reached. Usually, drill pipes and other related equipment has a specified tensile capacity. This is often directly related to the cross-section area of the pipe and the yield strength of the material as seen in formula *5* in chapter *3.2 Formulas*

Since the geometry of the Body is not a simple cylindrical tube, it might contain weak spots. It is therefore tested with the mathematically tensile capacity based on wall thickness to check for weak spots. The wall thickness of the Body is 10 mm and the tensile capacity is calculated as followed:

$$\mathrm{TC} = \frac{\pi \cdot (\mathrm{OD}^2 - \mathrm{ID}^2)}{4} \cdot \mathrm{YS} = \frac{\pi \cdot (177.8^2 - 157.8^2)\mathrm{mm}^2}{4} \cdot 1105\ \mathrm{MPa} \approx 5.83\ \mathrm{MN} \tag{9}$$

For this tensile stress test the force has been applied to the pin threads, while the box threads are constrained as shown in Figure 31 below.

**The following parameters are used for the 1st simulation:**

- The force is applied along the y-axis (marked green) with a magnitude of 5.83 MN.
- The mesh uses element size of 10 mm.
- The structural constrain (marked blue) is constraining translation.

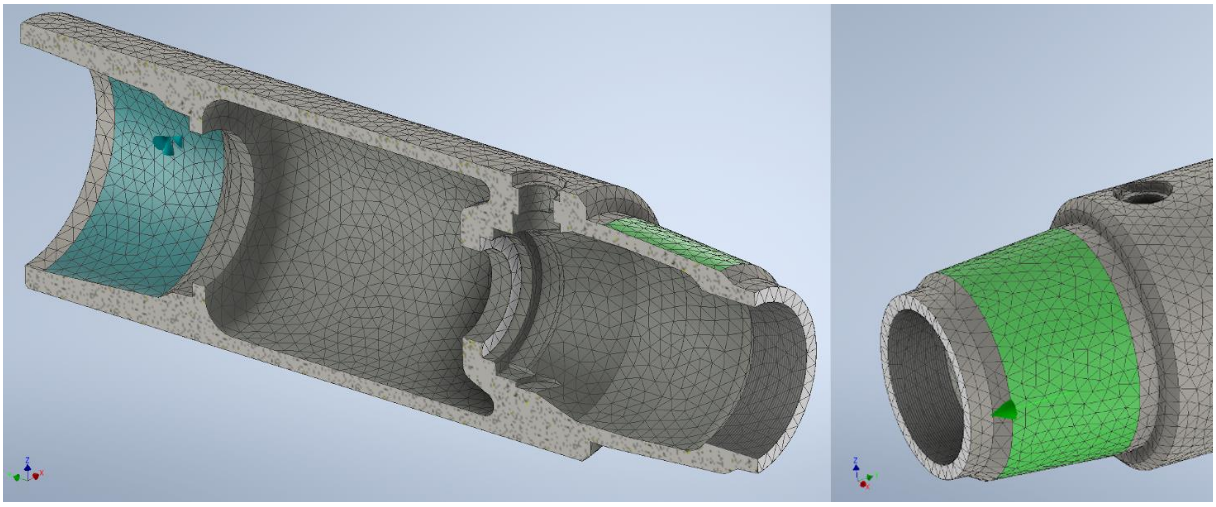


Figure 31: Tensile test; mesh, forces, and constraints

The simulation shows that the main part of the Body do in fact reach the yield strength of the material of 1105 MPa (Figure 32). However, the highest Von Mises stress during the test are found inside the Stem hole with a magnitude of 1878 MPa. This is above the material yield strength and will result in a permanent deformation, which is fatal to the Body's integrity. The thread areas have been neglected in this project, but the simulation indicates that the applied force might cause problems there as well.

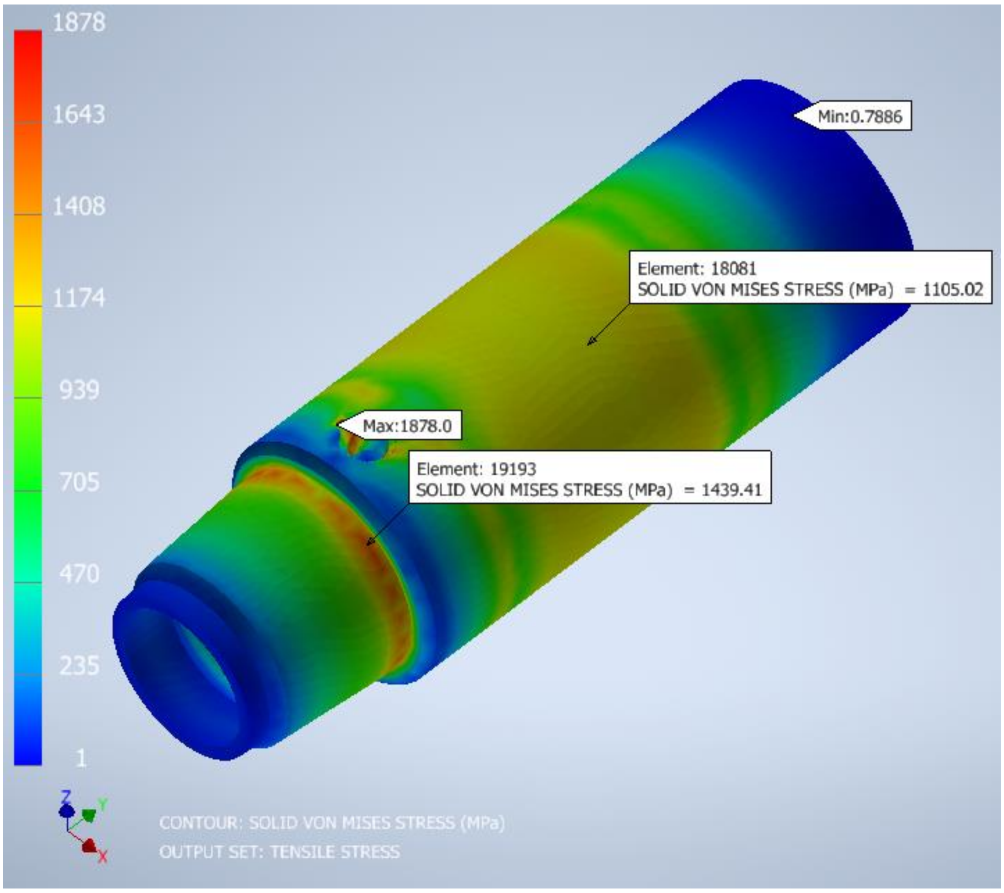



Figure 32: Tensile test (5.83 MN); Von Mises stress in MPa

The theoretical tensile capacity based on wall thickness is not achievable. The capacity will be decreased to a level that corresponds to another drill pipe system. National Oilwell Varco (NOV) delivers a broad range of drilling equipment, including titanium drill pipes. Based on information from their product brochure, a pulling force capacity of 3.3 MN is used on one of their 5 7/8" titanium drill pipe systems [15]. It is beneficial to find systems where the SmartCock is capable to withstand the systems specified pulling force, to assure that the SmartCock can be integrated with different systems.

**The following parameters are used for the 2nd simulation:**

- The force is applied along the y-axis with a magnitude of 3.3 MN.
- The mesh uses element size of 10 mm.
- The structural constrain is constraining translation.

Figure 33 shows that in the second test, maximum Von Mises stress is approximately 930 MPa and are located around the Stem hole similar to the previous test. The stress on the pipe body is 635 MPa. Reducing the force to 3.3 MN makes the results acceptable with the chosen material. The safety factor regarding tensile capacity is normally achieved by calculating allowed pulling force by use of a Drillers Data Handbook [16].

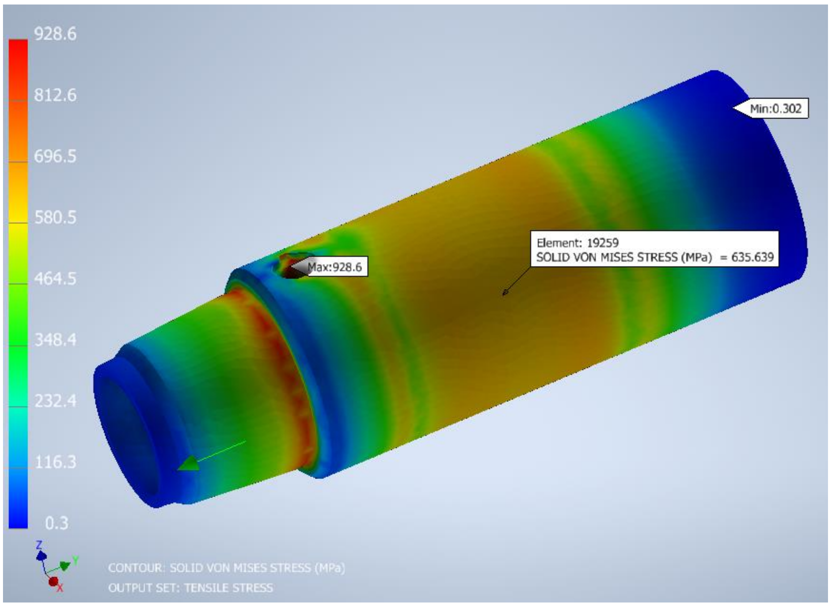



Figure 33: Tensile test (3.3 MN); Von Mises stress in MPa

The total deformation of the Body during 3.3 MN load is 2.2 mm (Figure 34, Left Side) and the Body's outer diameter is compressed by 0.5 mm (Figure 34, Right Side). These deformations are quite small and within acceptable displacement. In overall, the tensile capacity simulation is at an acceptable level.

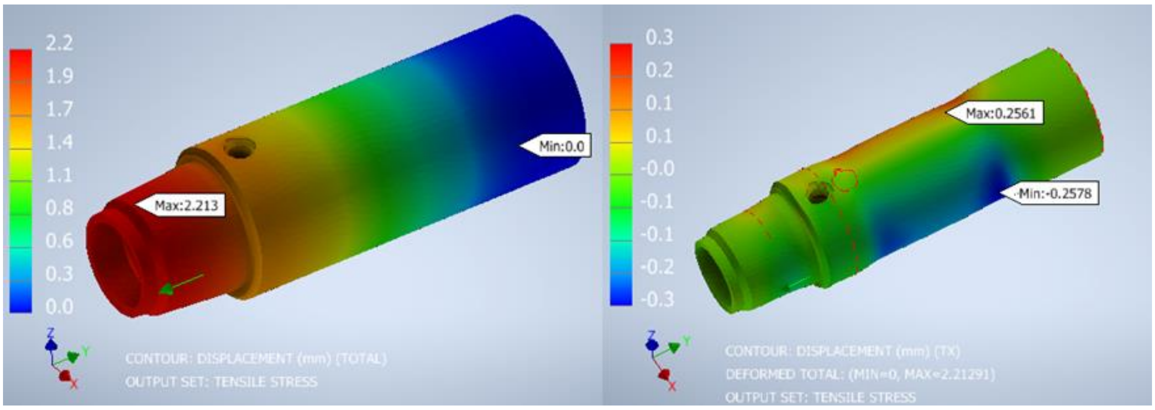



Figure 34: Tensile test; total displacement and displacement along x-axis in mm

## 6.2 Torsion on Body

The Body will generally experience torsional forces in two different scenarios. The first is from a roughneck during make-up torque onto a drill string. The second is from rotational forces that the top drive can apply to the drill string during operation. The make-up torque is generally lower than the maximum allowed torque that the top drive can apply to the drill string. It is therefore tested with the torsional forces applied by the top drive. The allowed torque from the top drive is normally determined by the weakest link/point in the system. To determine what torque the Body should be tested with, several different torsion capacities from similar sized drill pipes have been reviewed. A value of 160 kNm is chosen to be a sensible torque to test with, since it is the uttermost of what similar drill pipes handles.

The torsion test simulates the torque that the top drive would apply to the Body at max strength.

- The torque is applied at the top of the Body (marked green) around the y-axis, with a magnitude of 160 kNm
- The mesh uses element size of 10 mm.
- The structural constrain (marked blue) is constraining translation and rotation

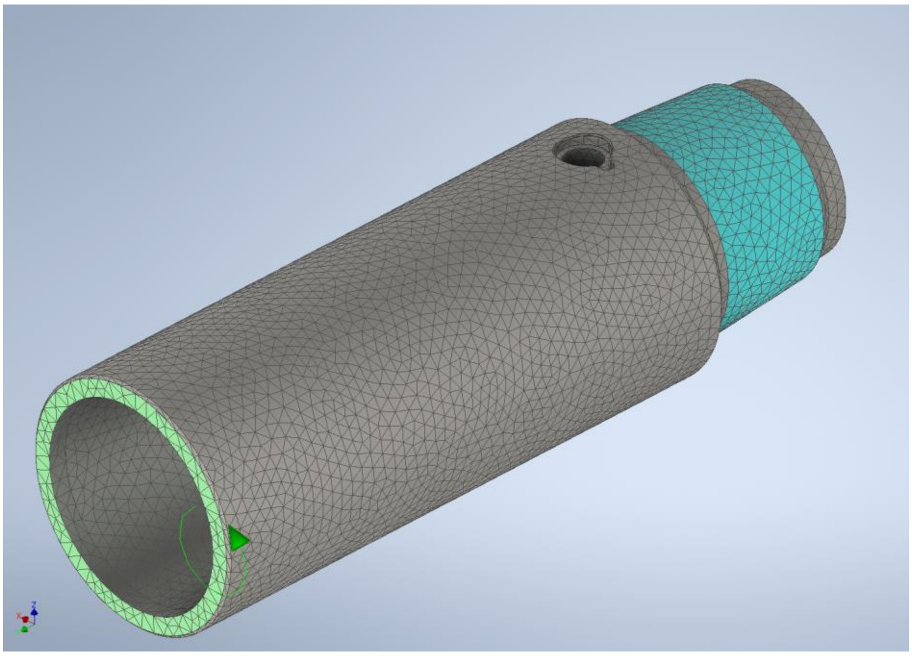


Figure 35 Torsion test; mesh, forces, and constraints

The maximum Von Mises stress from the torsion test is found in the pin thread area with a magnitude of 1370 MPa (Figure 36). This value is above minimum yield strength and would normally not been accepted. But in this design are the threads not applied yet, which makes it unfeasible to interpret the received value in this area. There are several variables to consider since the upper and lower make-up shoulder will absorb rotational force transmitted by the threads. Thread specifications and friction coefficient will be some of the factors that will impact the real value. This will need further investigation when the thread specifications are available. The value of 1370 MPa is therefore not considered to influence this test. The focus in this test is pinpointed towards the middle part of the Body to check if the wall thickness is thick enough and check the Stem hole for any weak spots due to torsional forces. The middle part of the Body shows a Von Mises stress of 650 MPa which is an acceptable value. The Stem hole has no weak spots that exceeds 650 MPa in this test. In general, the Body is well configured to cope with torsional forces, as this test shows. The safety factor regarding torsion capacity is covered in a Drilling Data Handbook.

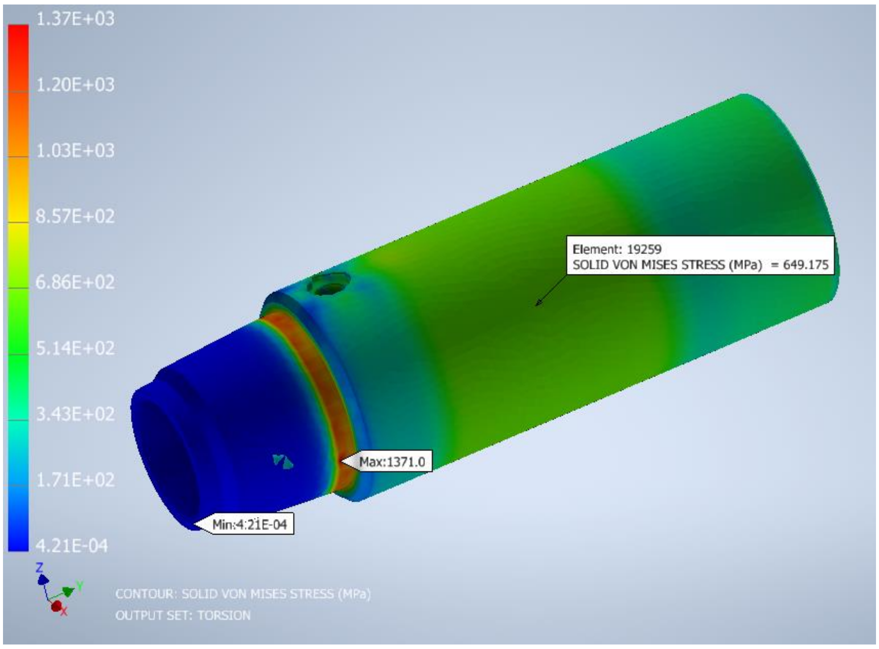



Figure 36: Torsion test; Von Mises stress in MPa

The total deformation (Figure 37) of 3.8 mm is at an acceptable level for this test. The displacement will increase with the Body length from where it is constrained to where the force is applied. It is not estimated that the deformation of 3.8 mm will cause issues regarding the integrity of the Body. In overall, the torsion simulation is at an acceptable level.

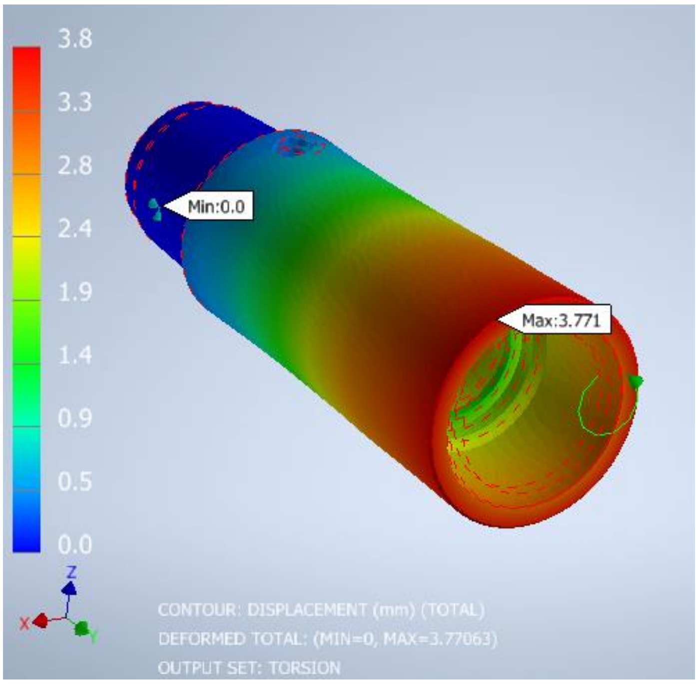



Figure 37: Torsion test; total displacement in mm

## 6.3 Clamp force on Body

The Body will have to withstand the clamping force caused by the roughneck during make-up and break-out of the SmartCock. In a real scenario, the roughneck will grip around the Body with a clamping force, then apply rotational force to either make-up or break-out the SmartCock from the drill string. In this test will only the clamp force be applied to check the Body for resistance to deformation. To carry out the test, it is designed some simplified clamps based on information provided by MHWirth. The designed clamps are imitations from a 2-grip clamp system as seen in Figure 38. The clamp design is not accurate to a particular specification, but it will give a fair approximation of how the clamp force will impact the Body.

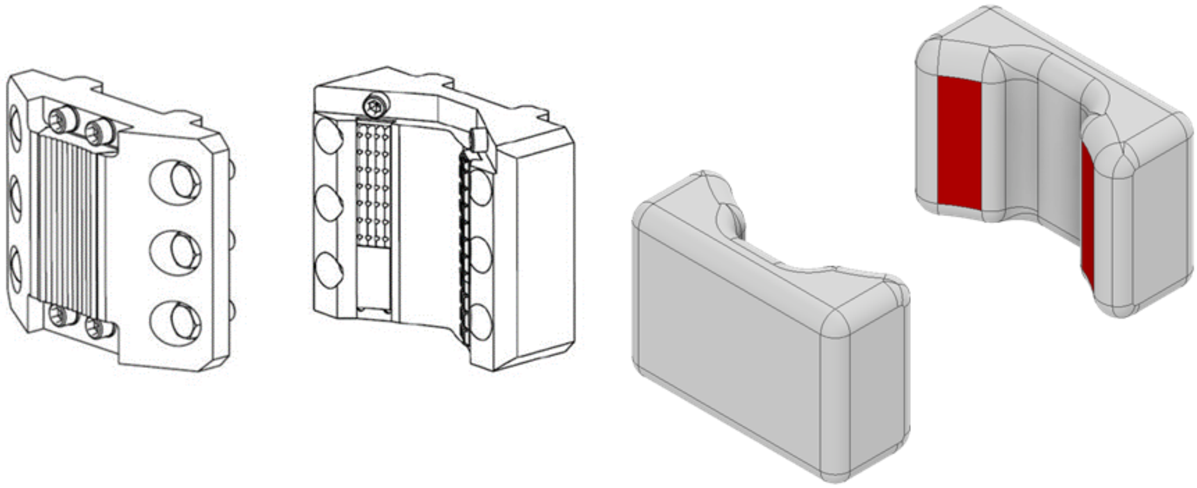


Figure 38: Clamp dies comparison (MHWirth to the left)

For the simulation of the clamp force, the clamp dies are attached to the Body as an assembly, directly at the weakest location on the Body (Figure 39). When performing an assembly simulation, the software requires to set surface contact parameters. The parameters that are set for this test is done with the focus on clamp force. The coefficient of friction needs further investigation for a real-life scenario, where the roughneck also would simultaneously apply the make-up/break-out torque in combination with the clamp force.

**The following parameters are used during simulation:**

- The surface contact between the clamps and the Body is set to “separation” with the following parameters:
    - Stiffness Factor: 1
    - Coefficient of Friction: 0
    - Penetration Surface Offset: 0 mm
- The mesh element size is set to 15 mm for both the clamps and the Body.
- A force is applied on both clamps (marked green) on the x-axis with a magnitude of 485 kN, giving a combined clamp force of 970 kN.
- The clamps are structurally constrained in the z- and y-axis allowing only movement along the x- axis
- The Body is structurally constrained on the lower threads (marked blue) preventing translation and rotation.

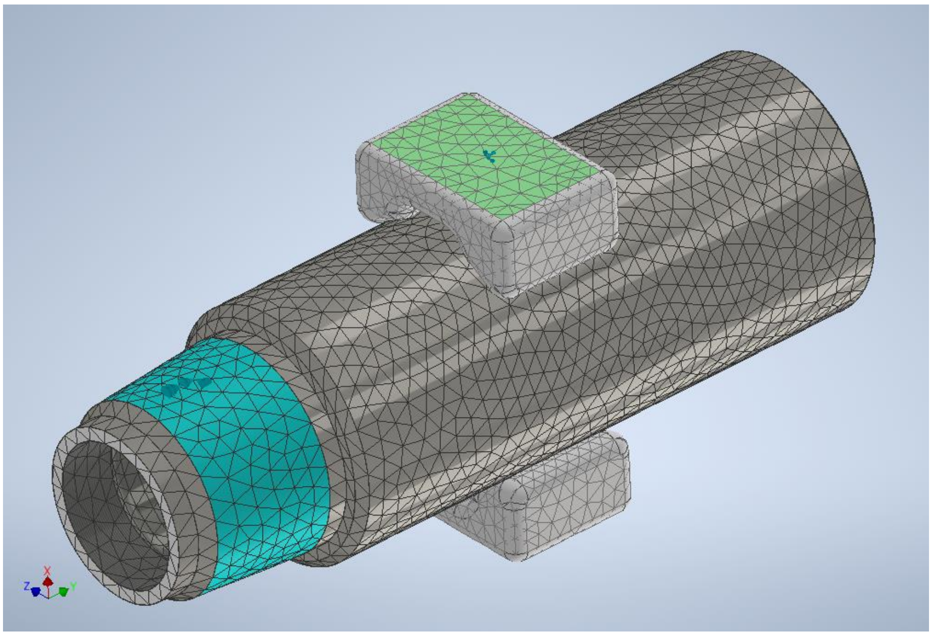


Figure 39: Clamp test; mesh, forces, and constraints

The maximum Von Mises stress on the Body is 722 MPa, (Figure 40) where the clamps are in contact with the Body. This will not cause any permanent deformation with the given material. If the clamp force is applied regularly it might cause material fatigue over time, but it is not assumed that the SmartCock will experience that many assemblies to a drill string. A stress of 722 MPa and a minimum yield strength of 1105 MPa gives a safety factor of 1.5.

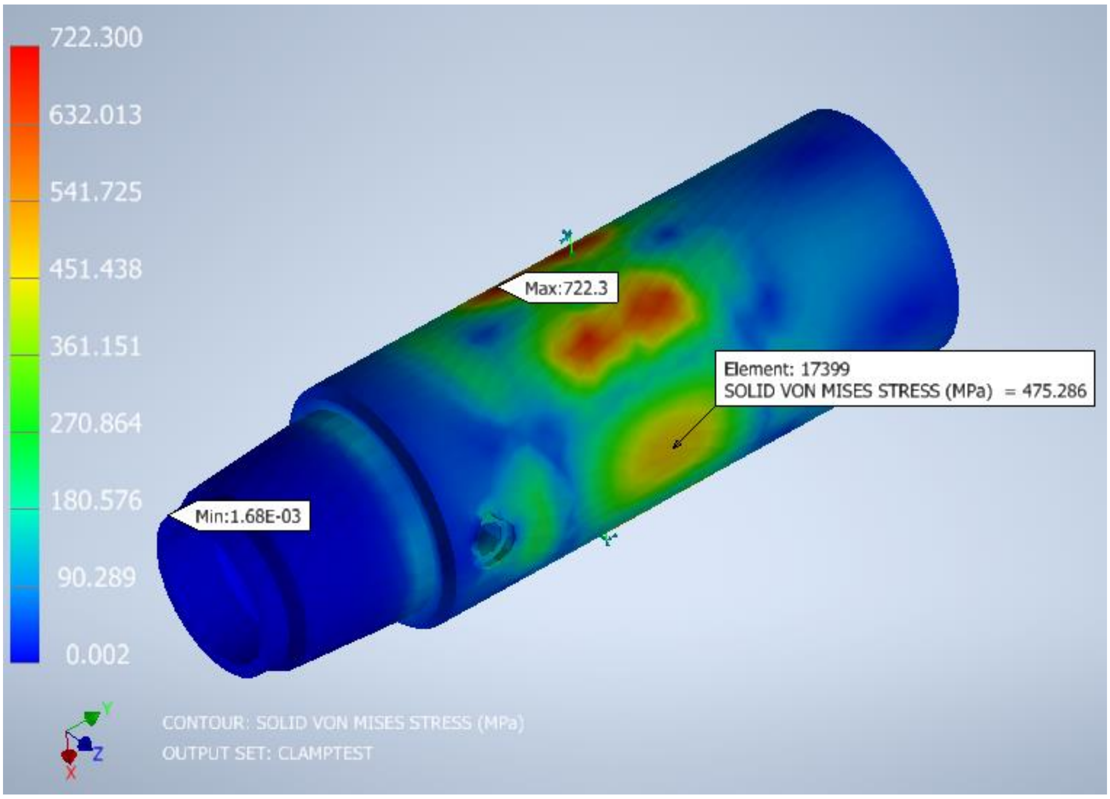



Figure 40: Clamp test; Von Mises stress in MPa

The maximum displacement is located between the dies, by collapsing the walls with 1.3 mm (Figure 41) in both directions. The figure below shows an exaggeration of how the deformation would look like. With use of 10 mm walls it is not considered to cause integrity issues with this deformation. In overall, the clamp force simulation is at an acceptable level.

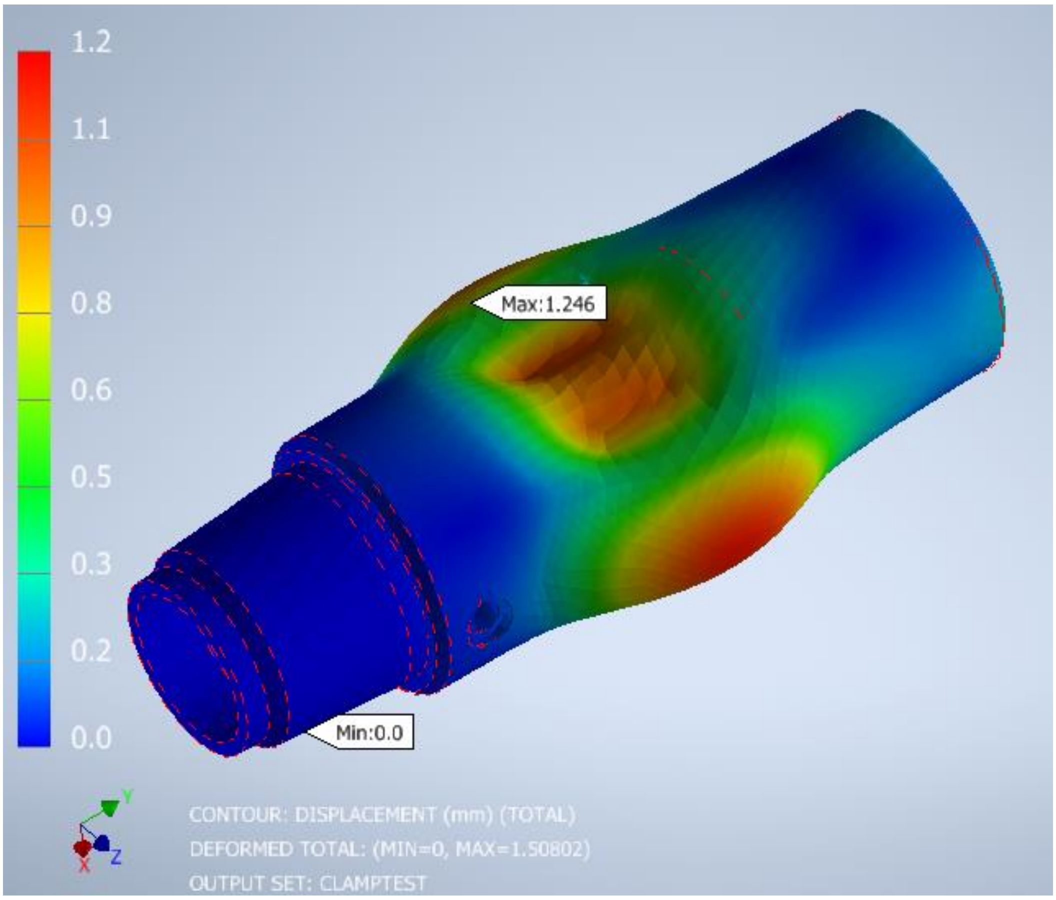



Figure 41: Clamp test; total displacement in mm (the displacement is NOT to scale)

## 6.4 Internal pressure on Body

The SmartCock is intended to be a pressure rated equipment designed for a 10k psi system. Since the SmartCock will have a check valve function when pressurised from above, it is only applicable to the test the valve from below. NS-EN ISO 10424-1 states that a 10k psi system valve, must perform a FAT with 15k psi to be approved. This project will not cover seals and test of sealing systems, but it will test the Body for structural integrity. This means that the force generated by the pressure will be applied to dedicated areas to check for stress. For the walls in the lower part of the Body is pressure directly added. For the Stem shoulder and Upper Seat shoulder are the force calculated based on the piston area that is created during pressures.

Force against Upper Seat shoulder:

$$p = \frac{F}{A} \rightarrow F = p \cdot \frac{\pi \cdot d^2}{4} = 103.42\ \text{MPa} \cdot \frac{\pi \cdot (80\text{mm})^2}{4} \approx 520\text{kN} \tag{1}$$

Force against Stem shoulder:

$$p = \frac{F}{A} \rightarrow F = p \cdot \frac{\pi \cdot d^2}{4} = 103.42\ \text{MPa} \cdot \frac{\pi \cdot (36\text{mm})^2}{4} \approx 105.3\text{kN} \tag{10}$$

**The following parameters are used for this simulation:**

- A pressure of 103 MPa (15k Psi) is applied inside the front section of the valve (marked green) which act normal to these faces.
- A force of 105.3 kN is applied to the Stem shoulder (marked yellow)
- A force of 520 kN is applied to the Upper Seat shoulder (marked red)
- The mesh element size is set to 8 mm.
- The Body is structurally constrained on the pin and box threads (marked blue) preventing translation and rotation.

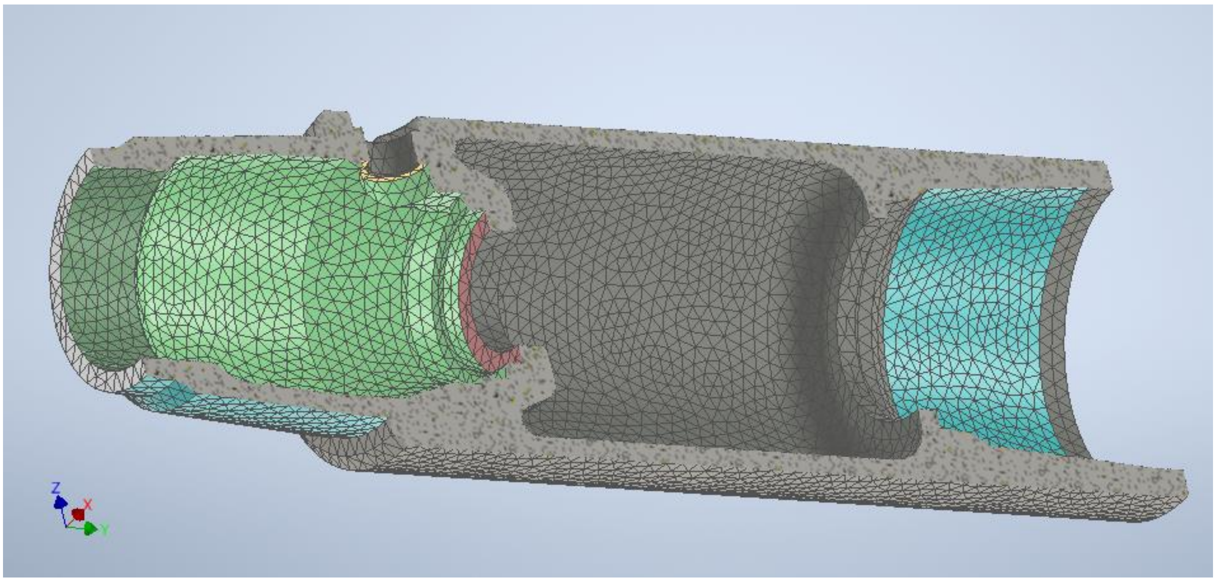


Figure 42: Internal pressure test; mesh, forces, and constraints

The maximum Von Mises stress of 673 MPa during this test is located on the Upper Seat shoulder as seen in Figure 43. According to NS-EN ISO 10424-1 is the minimum safety factor during FAT equal to 1.0. The FAT pressure is 1.5 times higher than the working pressure rating of 10k psi. This give a general safety factor of 1.5 during use within the working rate pressure. The stress of 673 MPa is therefore on an acceptable level with good margins. There are also forces generated by the Spring and the make-up torque of the Retainer Cap impacting the Upper Seat shoulder. These forces are of low influence compared to the force generated by the 15k psi and are neglected in this test.

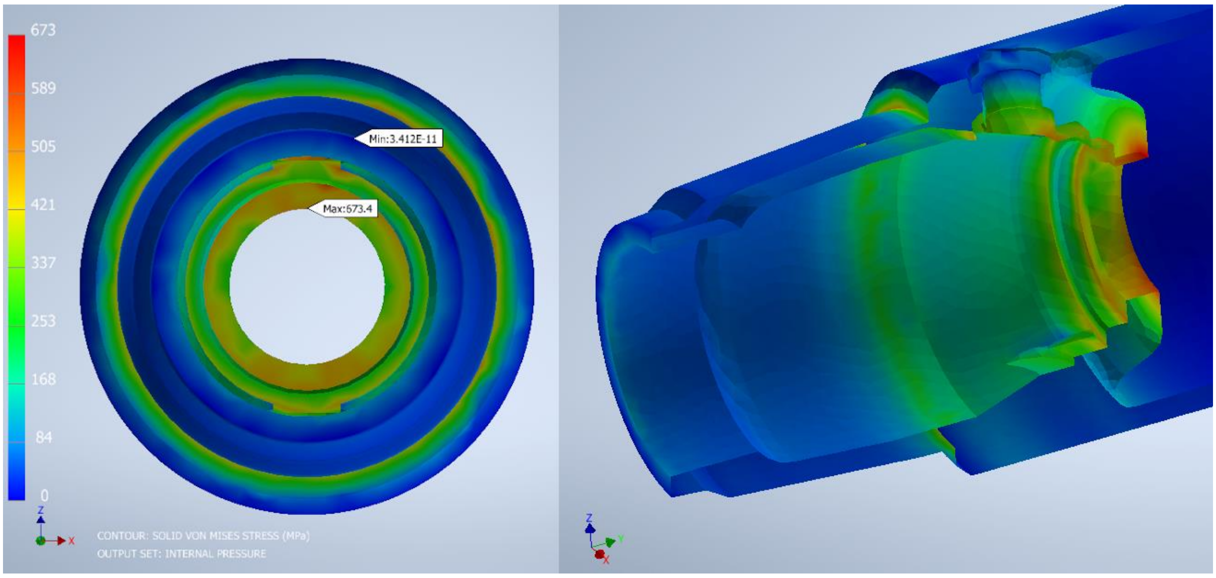



Figure 43: Internal pressure test; Von Mises stress in MPa

The maximum total displacement also occurs around the Upper Seat shoulder (Figure 44). A displacement of only 0.3 mm is acceptable. Low displacement is beneficial considering the soft seal arrangement that have to be fitted to this design on a later basis. In overall, the internal pressure simulation is at an acceptable level.

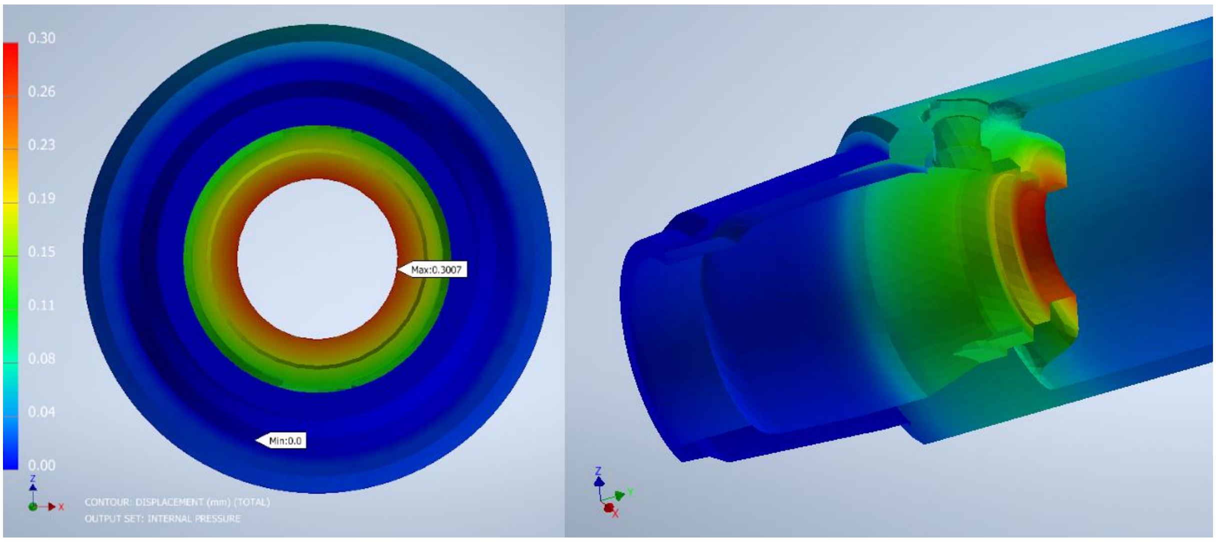



Figure 44: Internal pressure test; total displacement in mm

## 6.5 External pressure on Body

The SmartCock can also be exposed to external pressures during downhole operations. This will occur if the annulus pressure is greater than the pressure inside the drill string. It is desirable that all downhole equipment is capable of handling external pressures equal to the working pressure rate. In this simulation it will be applied pressure to the outer shell, with values similar to the FAT internal pressure test.

**The following parameters are used during simulation:**

- A pressure of 103 MPa (15k Psi) is applied on the outside wall (marked green)
- The mesh element size is set to 10 mm.
- The Body is structurally constrained on pin and box thread area (marked blue) preventing translation and rotation.

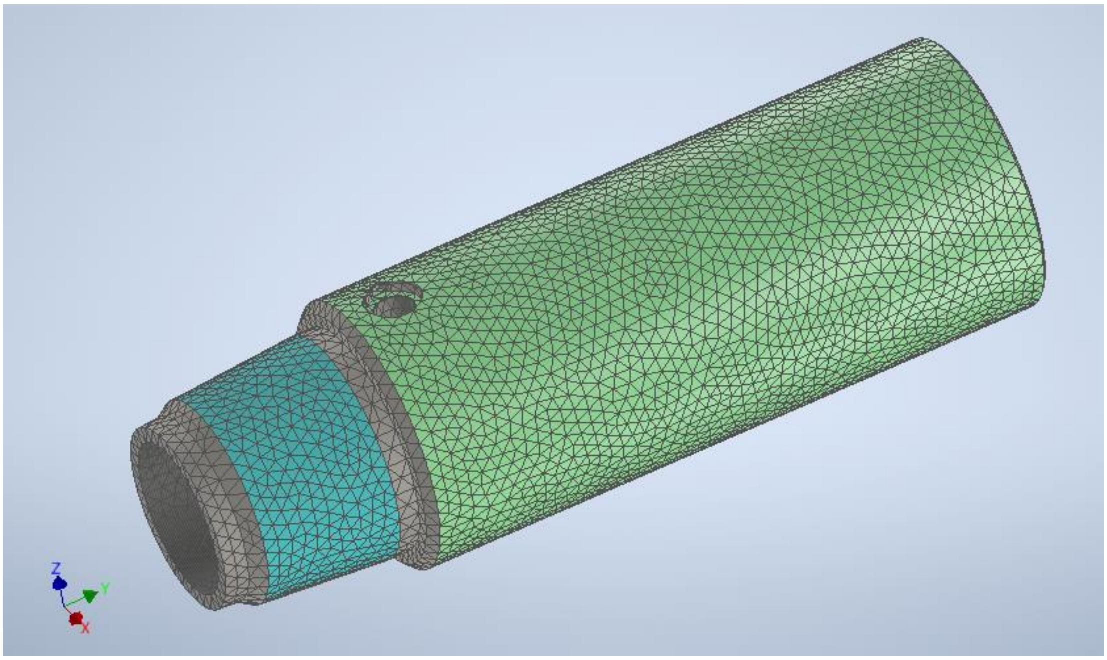



Figure 45: External pressure test; mesh, forces, and constraints

The external pressure simulation shows a maximum Von Misses stress of 975 MPa located inside the Body seen in Figure 46.This point is close to the arc of the weight reduction cut out (A-14 in the Body). The stress is within the yield strength of the material and therefore acceptable. This gives a safety factor above 1.5 due to the FAT pressure used for the simulation. The outside wall has a general stress of 720 MPa.

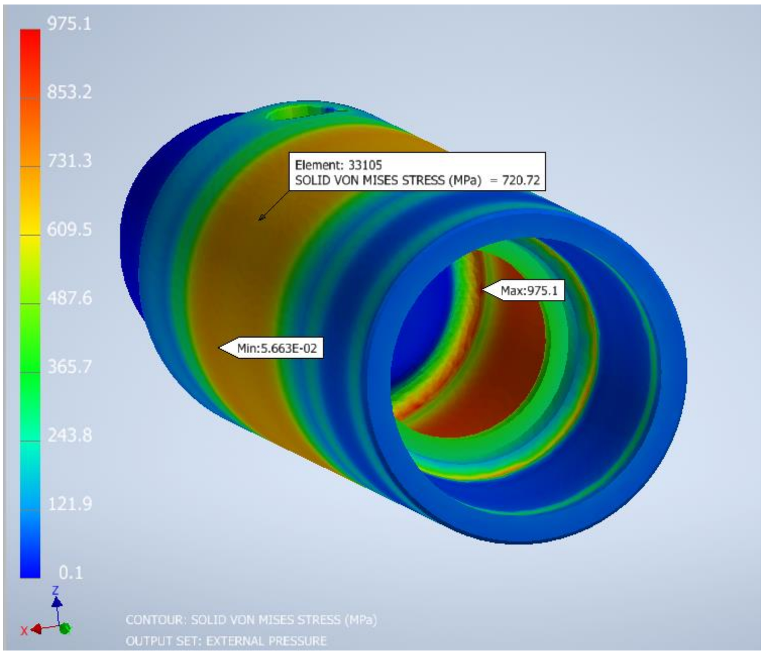



Figure 46: External pressure test; Von Mises stress in MPa

The total displacement of 0.7 mm is considered to be at an acceptable level. The displacement is generally found in the middle part of the Body as seen in Figure 47. This is sensible since the slightest internal support is situated in the middle of the Body. In overall, the external pressure simulation is at an acceptable level.

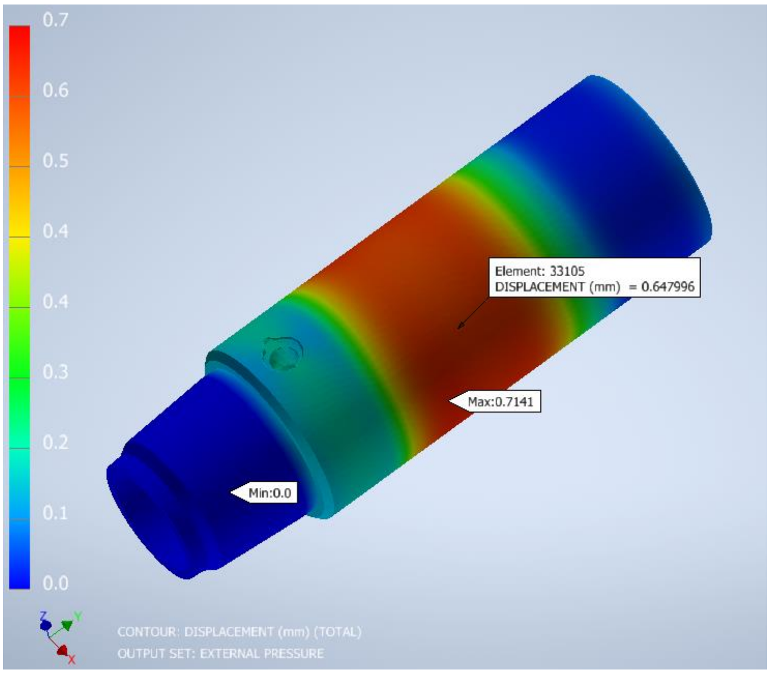



Figure 47: External pressure test; total displacement in mm

## 6.6 Forces on Stem due to internal pressure

When the SmartCock is pressurised from the inside, the Stem will act as a piston. This will generate a force that the contact surfaces must withstand. If the Stems structural integrity fails during an internal pressure test it could be launched out of the Body and cause severe injury or damage. The Stem will be simulated with forces equal to FAT pressure of 15k psi. The OD of sealing area on the Stem is 26 mm. This is used to calculate the piston area that will try to push the Stem out of the Body.

The force generated by pressure is calculated as followed:

$$\mathrm{p} = \frac{\mathrm{F}}{\mathrm{A}} \rightarrow \mathrm{F} = \mathrm{p} \cdot \mathrm{A} = 103.42 \frac{\mathrm{N}}{\mathrm{mm}^2} \cdot \frac{\pi \cdot (26\mathrm{mm})^2}{4} \approx 54909\ \mathrm{N} \approx 54.9\ \mathrm{kN} \tag{11}$$

**The following parameters are used during simulation:**

- A Force of 54.9 kN is applied on the bottom of the Stem
- The mesh element size is set to 5 mm
- The Stem is structurally constrained on the shoulder face, preventing translation along the y-axis and rotation around the x- and y-axis (marked blue)
- The Stem is structurally constrained on the Body, preventing translation along the x- and y-axis as well as preventing rotation around these axes.

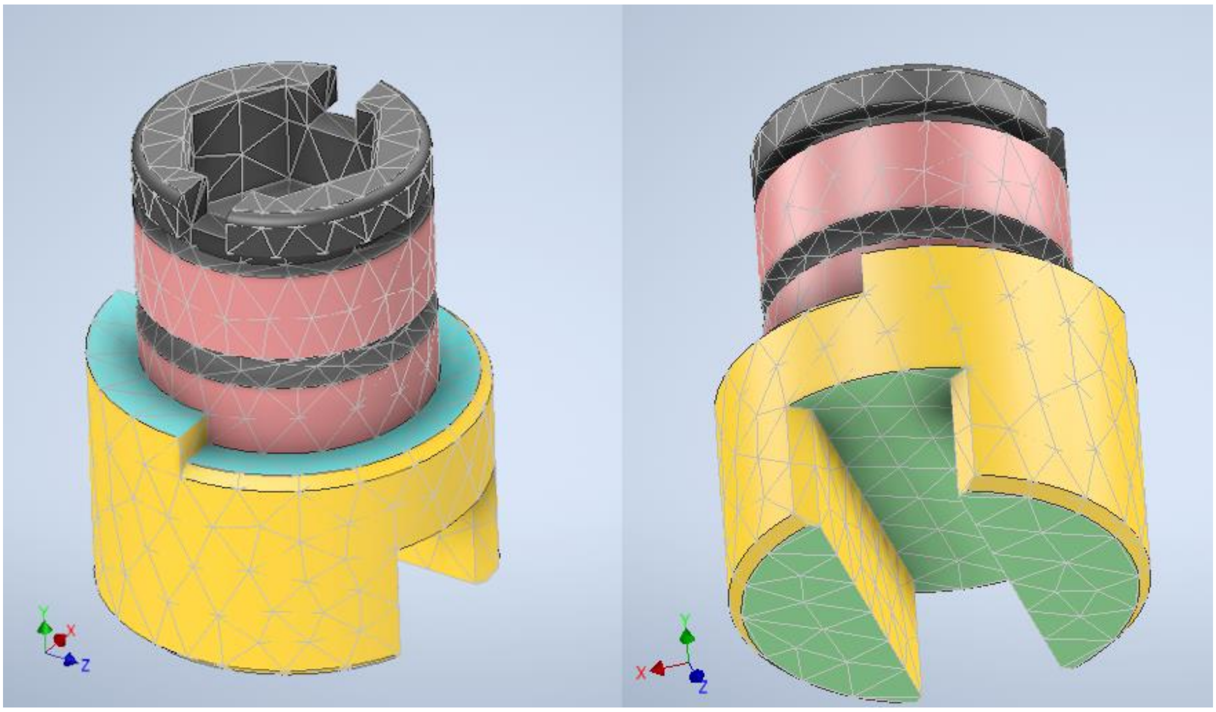


Figure 48: Internal pressure test on Stem; mesh, forces, and constraints

The maximum Von Mises stress on the Stem during the pressure test is 359 MPa (Figure 49). The largest stress is generally located in the angle between the contact surface and the middle part of the Stem. With the chosen material this will give a safety factor of 3.0 during the FAT test. It is required a safety factor of 1.0 during FAT which makes this test acceptable with good margins. Due to the 90-degree rotation limit design, a quarter of the surface will not be in contact with the Body and therefore not absorb any forces. This means that the remaining three-quarter surface will take up the force. With the generous safety factor of 3.0 it is concluded that the one quarter missing will not cause an integrity problem for the Stem.

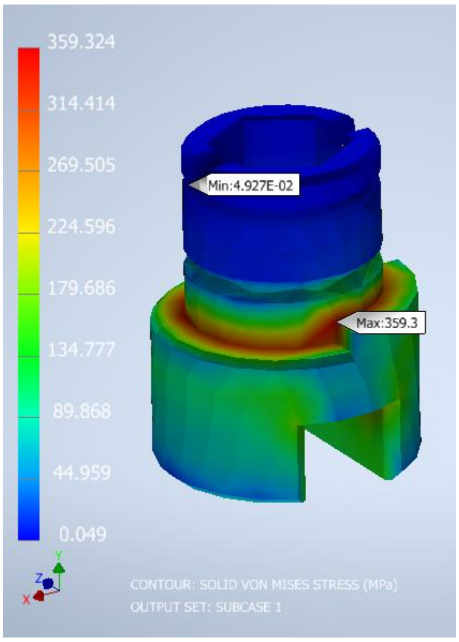



Figure 49: Internal pressure test on Stem; Von Mises stress in MPa

The maximum total displacement is only 0.026 mm (Figure 50, Left) and a displacement of 0.010 mm is shown along the x-axis at the bottom (Figure 50, Right). The picture to the right shows an exaggeration of the actual deformation. This are acceptable levels that will not cause issues regarding function or integrity. In overall, the internal pressure simulation on the Stem is at an acceptable level.

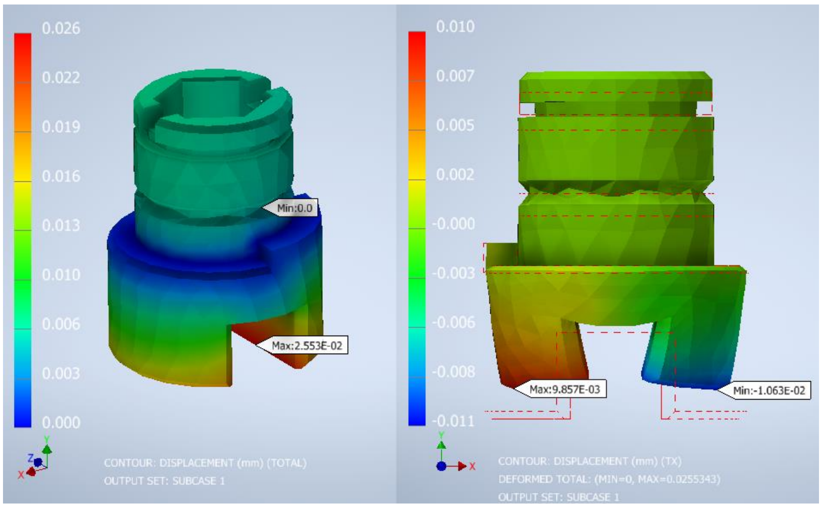



Figure 50: Internal pressure test on Stem; displacement in mm

## 6.7 Forces on Stem due to external pressure

The Stem will act as a piston during external pressures too. Instead of being forced outwards, external pressures will try to push the Stem inwards into the Body. According to NS-EN ISO 10424-1, the external stem-seal should be tested during the design verification process.

> *"The low-pressure test shall be at 1,7 MPa (250 psi) and the highest-pressure test shall be a minimum of 13,8 MPa (2 000 psi) but may be higher, up to the rated working pressure, at the manufacture's discretion."* [11]

The test described above is generally related to check the Stem-seal for leakage. Although the seal test will not be performed here, a pressure test is still conducted to make sure that the Stems structural integrity still remains as desired during the test. The Stem will in this test only be constrained from the Circlip and the pressure will be applied at all faces that are exposed to the external pressure. The test pressure is set to working pressure which is 10k psi.

**The following parameters are used during simulation:**

- A pressure of 69 MPa (10 000 psi) is applied on the faces marked yellow.
- The mesh element size is set to 3 mm
- The Stem is structurally constrained on the Circlip face, preventing translation along the y-axis (marked blue).
- The Stem is structurally constrained on the Body, preventing translation along the x- and y-axis as well as preventing rotation around these axis (marked red).

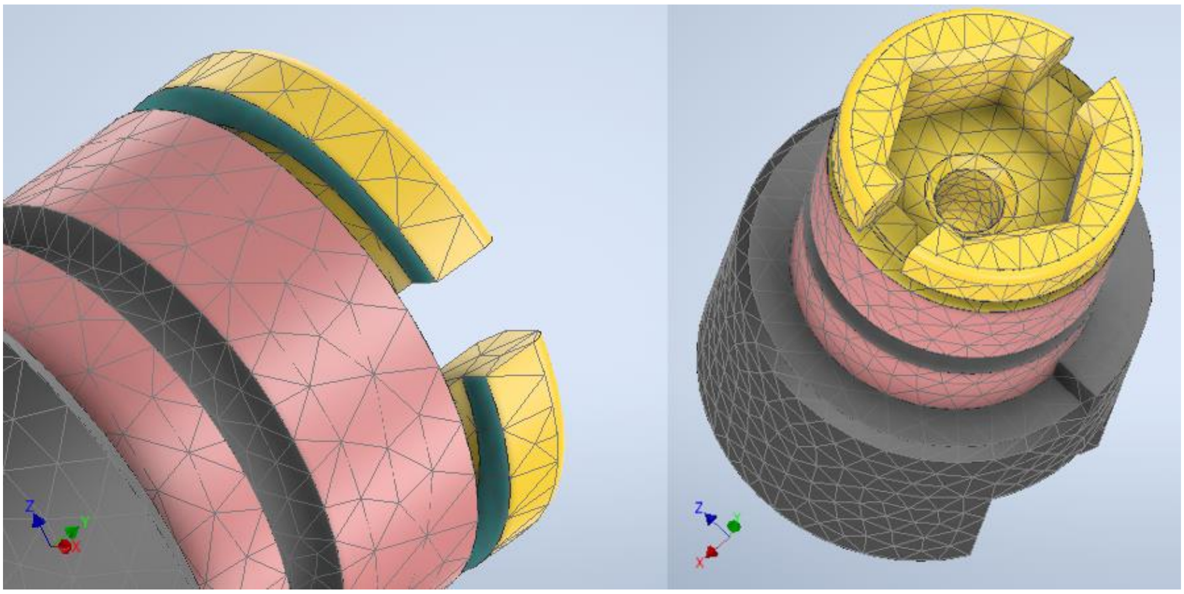


Figure 51: External pressure test on Stem; mesh, forces, and constraints

The maximum Von Mises stress is 793 MPa and is found at the corners where the cuts for the Interlock interface seen in Figure 52. These corners will be the weakest points in regards of deformation during external pressure. The pressure used in this test is equal to the working pressure which means there is no safety factor considered. The value of 793 MPa is within the minimum yield strength and will give a safety factor of 1.4 with the given material. The surface that interfaces with the Circlip will generally be exposed for a stress of 350 MPa, which are significantly lower than the edges.

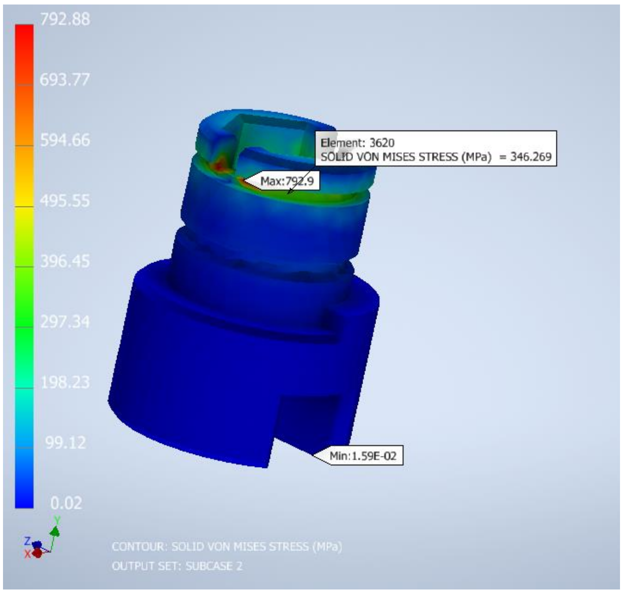



Figure 52: External pressure test on Stem; Von Mises stress in MPa

The maximum total displacement of only 0.016 mm prevents the Stem from translating into the Body (Figure 53). This will help keep the Stem in its desired position and will not interfere with the seal area towards the Body. In overall, the external seal test is at an acceptable level.

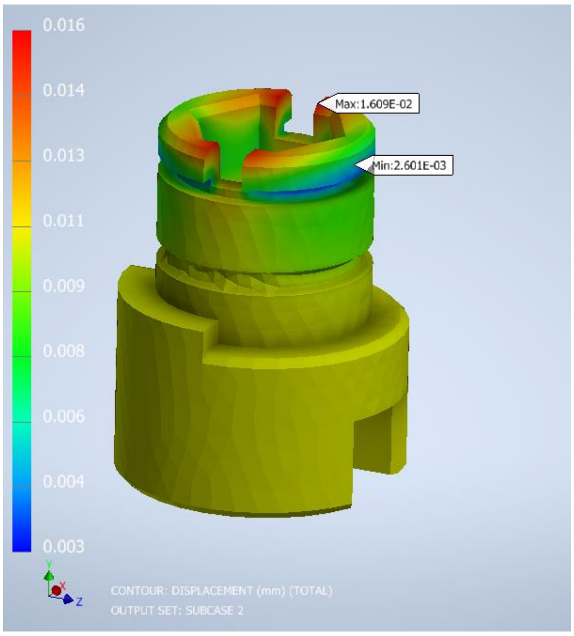



Figure 53: External pressure test on Stem; total displacement in mm

# 7. Conclusion

The aim of this project has been to further develop the design of SmartCock regarding the technology that is patented and achieve a weight below 25 kg. As a part of this development it is commenced a design process to clarify if the SmartCock technology is suitable to use as a drill string safety valve, while also being under 25 kg. To check for its suitability the following four factors have been considered; force, dimension, weight and function.

The SmartCock is designed by use of a CAD software and in accordance with criteria from Moonshine and standards. The patent description and criteria from Moonshine is used as framework during modelling to ensure that the prototype is within the described technology. API Standard 53 *Well Control Equipment Systems for Drilling Wells* and NS-EN ISO 10424-1 *Rotary Drill Stem Elements* are interpreted and implemented into the design process for the SmartCock to easier comply with regulations regarding barrier control in the oil & gas industry. As a result of the design process has several improvements been implemented to the SmartCock. One of them is the decision to install the Stem from inside which makes the solution more suitable to withstand internal pressure.

When choosing material for the Body the strength-to-weight ratio has been a key driver to consider. Titanium Grade 19 fulfil most criteria except elongation and material notch toughness described in NS-EN ISO 10424-1. It is still uncertain whether those criteria are relevant for the intended use of SmartCock and will need further investigation. All components have been given a dedicated material and the total weight of SmartCock is 24.85 kg in this design.

Simulation analysis show that the Body and Stem are capable to withstand the forces from the pressure tests stated in NS-EN ISO 10424-1. The simulations show good performance regarding tensile and torsion capacity which is beneficial in terms of integrating SmartCock to different drill string systems. The results regarding clamp force from a roughneck is positive, which indicates that the clamp force will not cause plastic deformation of the Body with the chosen material.

A technical appraisal meeting with MRC Global was held (digitally) to get a professional assessment on the Ball Valve Assembly. It was concluded that the seat surfaces that are in contact with the Ball and the Ball itself should be coated with a tungsten carbide coating. This coating is often used for valves that experience high amount of forces during operation to avoid scratches that can interfere with the sealing capability.

The plan was to create a 3D-printed version of the design to test the function regarding the Ball Valve Assembly and for Moonshine to use as an advertising product. Unfortunately, due to COVID-19 pandemic this objective had to be neglected. As a substitute for the physical 3D-printed version it has been made an animation video of the design. The animation covers all aspects of SmartCock from assembly to function and surrounding environment. Moonshine can use this animation as a part of their advertising strategy.

In conclusion to whether the SmartCock is suitable to use as a drill string safety valve, there is yet to find a drawback that would put an end to the continuation of the development. Further it is concluded that a weight below 25 kg is achievable by focusing on mass reduction measures and choosing Titanium Grade 19 for some of the main components.

**Further work**

This technology is without a doubt an interesting intruder that can challenge today's solutions in the drill string safety valve market. This project has developed a design basis for the SmartCock technology that easily can be customized and adapted to fit future purposes. There are many steps left in the design process to be complete. A natural next step is to further develop the Ball Valve Assembly to ensure full integrity of valve function and sealing. Even though API Standard 53 and NS-EN ISO 10424-1 are used as guidance in this project, it is recommended that a thorough research is carried out to determine all relevant standards that the SmartCock must fulfil to function as a drill string safety valve.

# 8. References

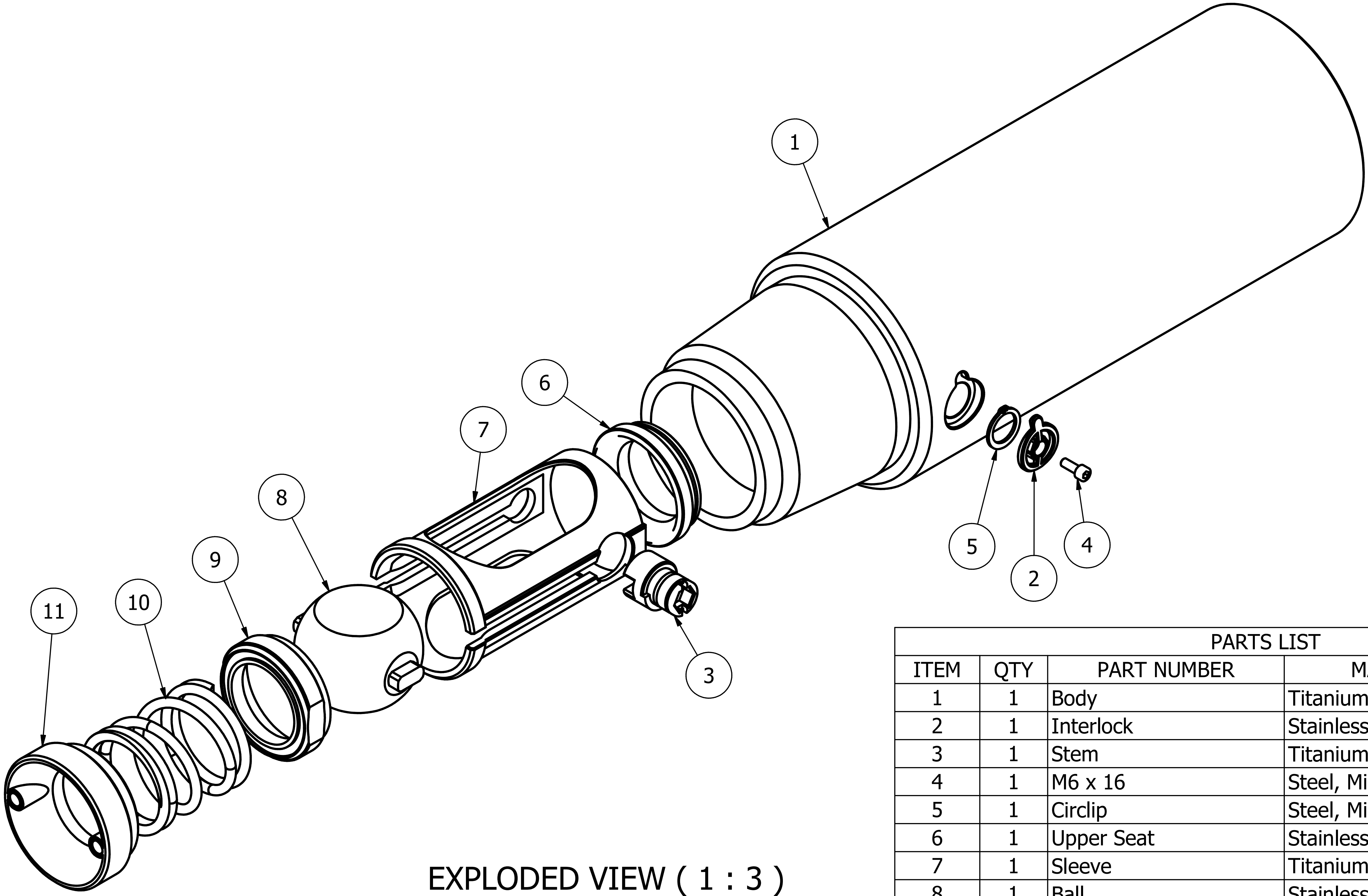

[1] Moonshine Solutions AS, *Smartalizer*, 2016. http://smartalizer.no/ (accessed Apr. 12, 2020).
[2] WIPO, *What is a patent?*, 2020. https://www.wipo.int/patents/en/ (accessed Apr. 12, 2020).
[3] *Well integrity in drilling and well operations*, NORSOK Standard D-010, Juni 2013. Rev 04.
[4] *Well control equipment systems for drilling wells*, API standard 53, Dec 2018. 5th edition.
[5] K. Ludvigsen, *Barrier innvendig i borestrengen*, 2017. https://ndla.no/nb/subjects/subject:6/topic:1:182849/topic:1:163658/resource:1:162436 (accessed Apr. 15, 2020).
[6] K. Lalit Narayan, K. Mallikarjuna Rao, and M. M. M. Sarcar, *Computer Aided Design and Manufacturing*, 1st ed. Prentice-Hall of India Private Limited, 2008.
[7] K. Brautaset, *Innføring i oljehydraulikk*, 1st ed. Oslo, Norway: Gyldendal, 2015.
[8] Drilling Formulas, *Drillingformulas*, 2014. http://www.drillingformulas.com/basic-of-drillpipe-tensile-capacity-and-its-calculation/ (accessed Mar. 05, 2020).
[9] Arbeidstilsynet, *Tungt arbeid*, 2020. https://www.arbeidstilsynet.no/tema/ergonomi/manuelt-arbeid/tungt-arbeid/ (accessed Apr. 13, 2020).
[10] MHWirth AS, *Personal communication*, Jan. 27, 2020.
[11] *Rotary drilling equipment Part 1: Rotary drill stem elements*, NS-EN ISO 10424-1:2004. .
[12] DP-Master Manufacturing, *DP-Master Manufacturing Company Products & Services*, 2013. http://www.dpmaster.com.sg/DP%20Master%20w%20Product%20Intro%20Presentation%2028AUG13.pdf (accessed Apr. 03, 2020).
[13] Equinor Europe, *TR2000-Piping and Valve Specification System*, 2010. https://tr2000.statoil.com/TR2000/integration/get_datasheet.jsp?DS=MDS&ID=ZH201&companyID=&areaID=&opn=&issue= (accessed May 09, 2020).
[14] MRC Global Norway AS, *Personal communication*, May 08, 2020.
[15] National Oilwell Varco, *High strength, High toughness drill stem products*, 2012. https://wbtapps.nov.com/gp/catalog/catalog.cshtml?puid=GaWFzBZ$3oL35C (accessed Apr. 05, 2020).
[16] L. V. Lidal, *Bruk av Drilling Data Handbook*, 2018. https://ndla.no/nb/subjects/subject:6/topic:1:182849/topic:1:147875/resource:1:145894 (accessed May 16, 2020).

# List of figures

# List of tables

# Attachment 1 - Technical drawings

| **Draw Nr.** | **Name** |
|---|---|
| 00 | SmartCock Assembly |
| 01 | Body |
| 02 | Interlock |
| 03 | Stem |
| 06 | Upper Seat |
| 07 | Sleeve |
| 08 | Ball |
| 09 | Lower Seat |
| 11 | Retainer Cap |

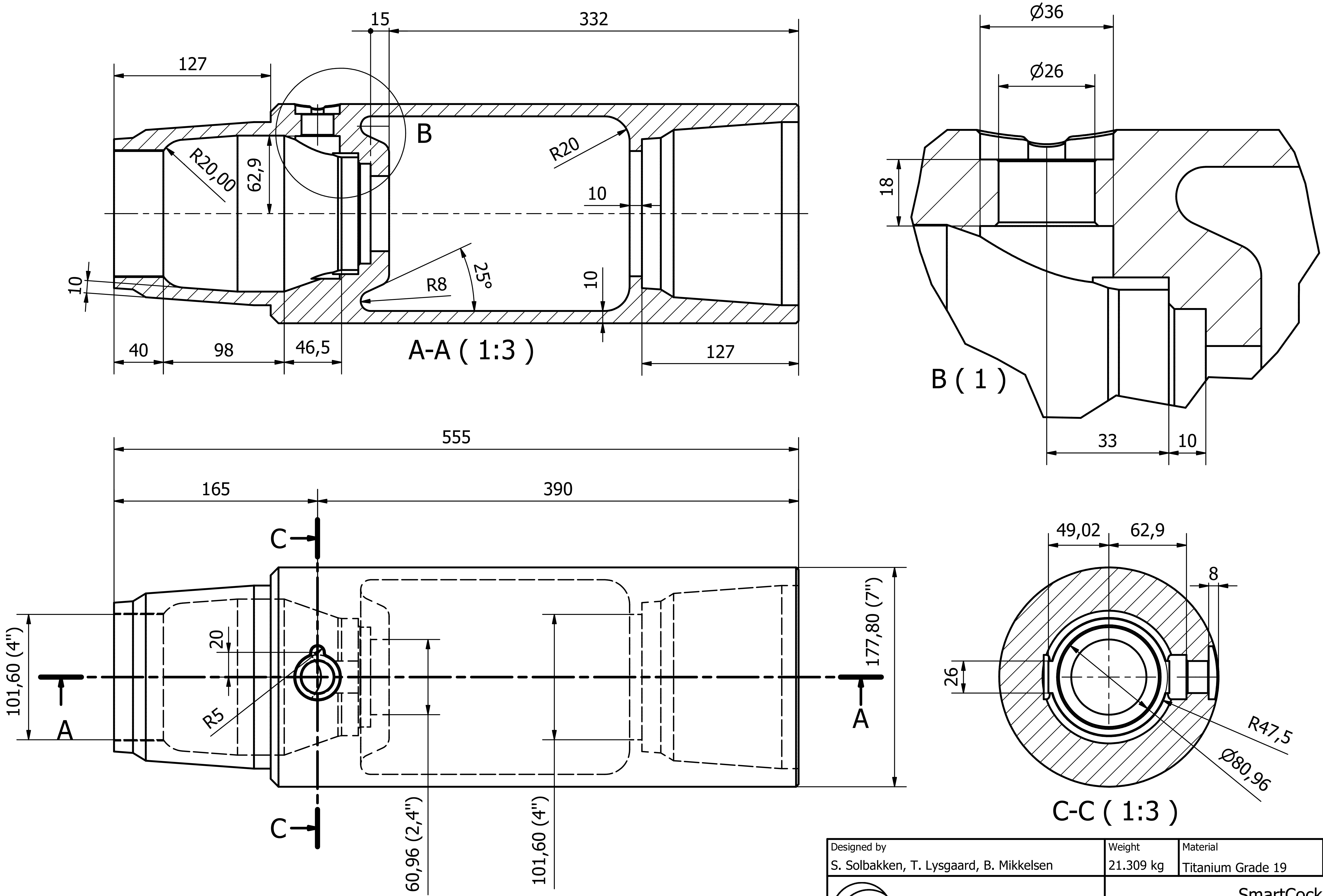



EXPLODED VIEW ( 1 : 3 )

| PARTS LIST | | | | |
|---|---|---|---|---|
| ITEM | QTY | PART NUMBER | MATERIAL | MASS |
| 1 | 1 | Body | Titanium Grade 19 | 21,309 kg |
| 2 | 1 | Interlock | Stainless Steel | 0,031 kg |
| 3 | 1 | Stem | Titanium Grade 19 | 0,117 kg |
| 4 | 1 | M6 x 16 | Steel, Mild | 0,007 kg |
| 5 | 1 | Circlip | Steel, Mild | 0,005 kg |
| 6 | 1 | Upper Seat | Stainless Steel | 0,390 kg |
| 7 | 1 | Sleeve | Titanium | 0,499 kg |
| 8 | 1 | Ball | Stainless Steel | 1,157 kg |
| 9 | 1 | Lower Seat | Stainless Steel | 0,404 kg |
| 10 | 1 | Compress Spring | Steel | 0,364 kg |
| 11 | 1 | Retainer Cap | Titanium Grade 19 | 0,581 kg |

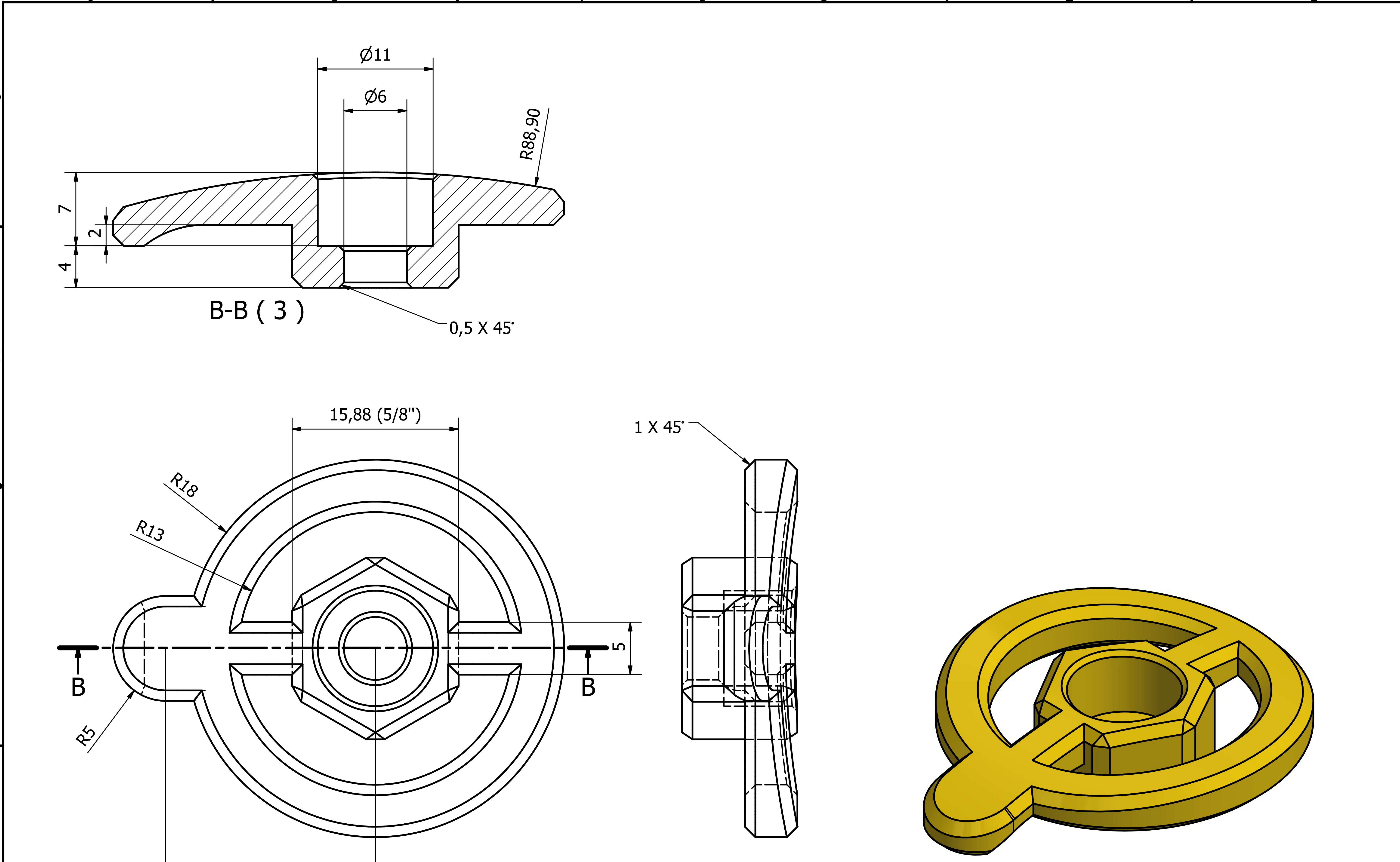
| Designed by | Total Weight | Scale | Date |
|---|---|---|---|
| S. Solbakken, T. Lysgaard, B. Mikkelsen | 24,851 kg | 1:3 | 15/05/2020 |

| | | | |
|---|---|---|---|
| Moonshine Solutions AS | SmartCock Assembly | | |
| | SmartCock | Draw Nr. 00 | Sheet 1 / 1 |

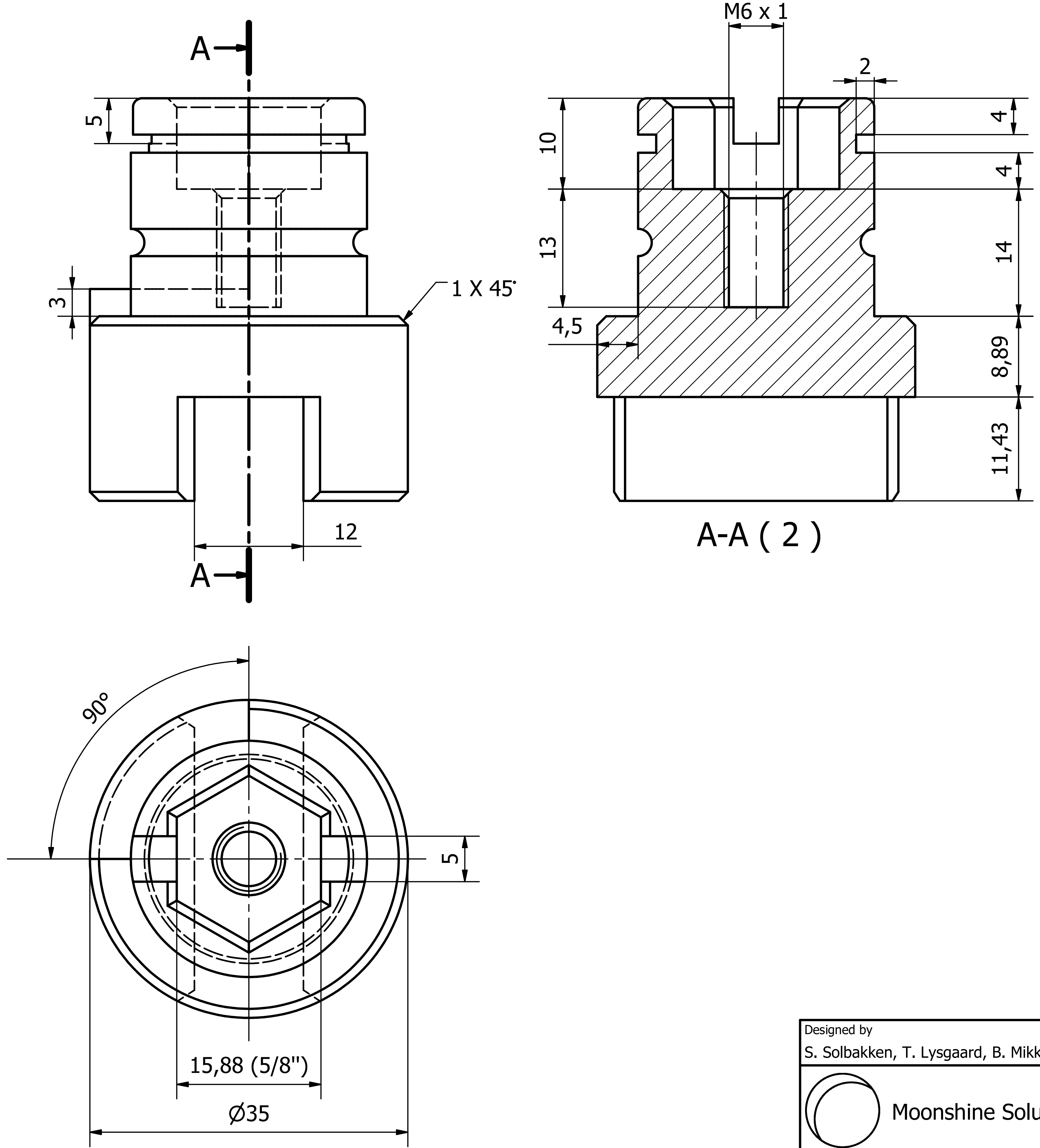


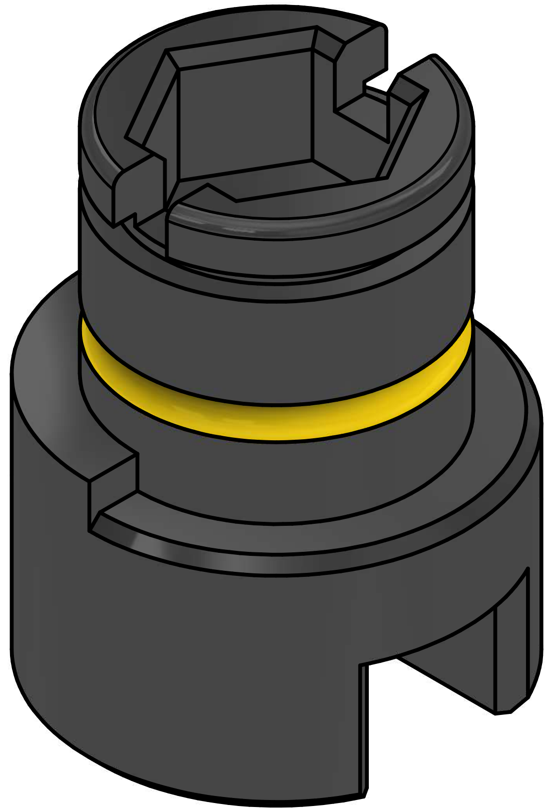

| Designed by | Weight | Material | Scale | Date |
|---|---|---|---|---|
| S. Solbakken, T. Lysgaard, B. Mikkelsen | 21.309 kg | Titanium Grade 19 | 1:3 | 17/05/2020 |

| | | | |
|---|---|---|---|
| Moonshine Solutions AS | SmartCock | | |
| | Body | Draw Nr. 01 | Sheet 1 / 1 |

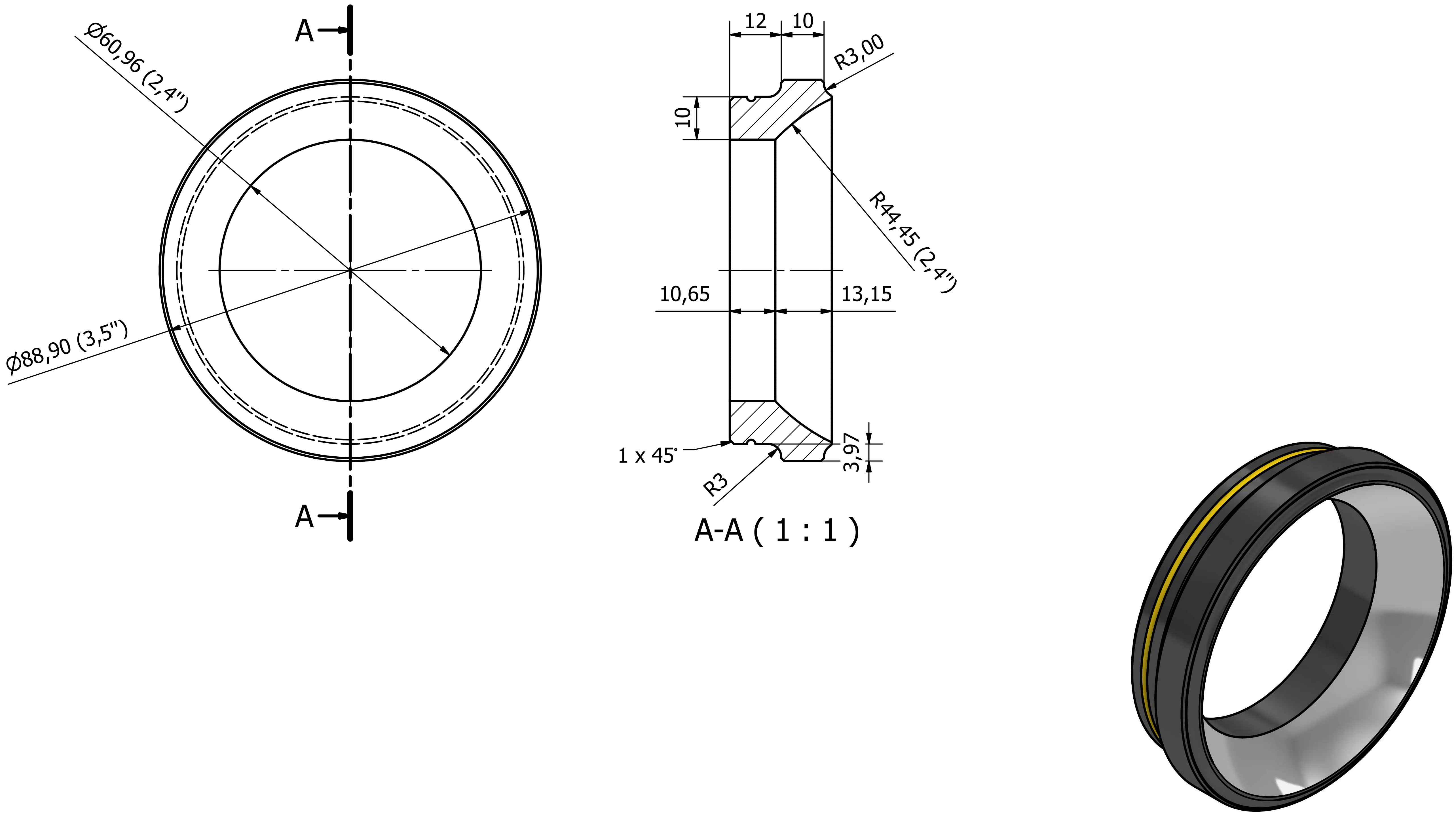



| Designed by | Weight | Material | Scale | Date |
|---|---|---|---|---|
| S. Solbakken, T. Lysgaard, B. Mikkelsen | 0.031 kg | Stainless Steel | 3:1 | 18/05/2020 |
| Moonshine Solutions AS | SmartCock | | | |
| | Interlock | | Draw Nr. 02 | Sheet 1 / 1 |

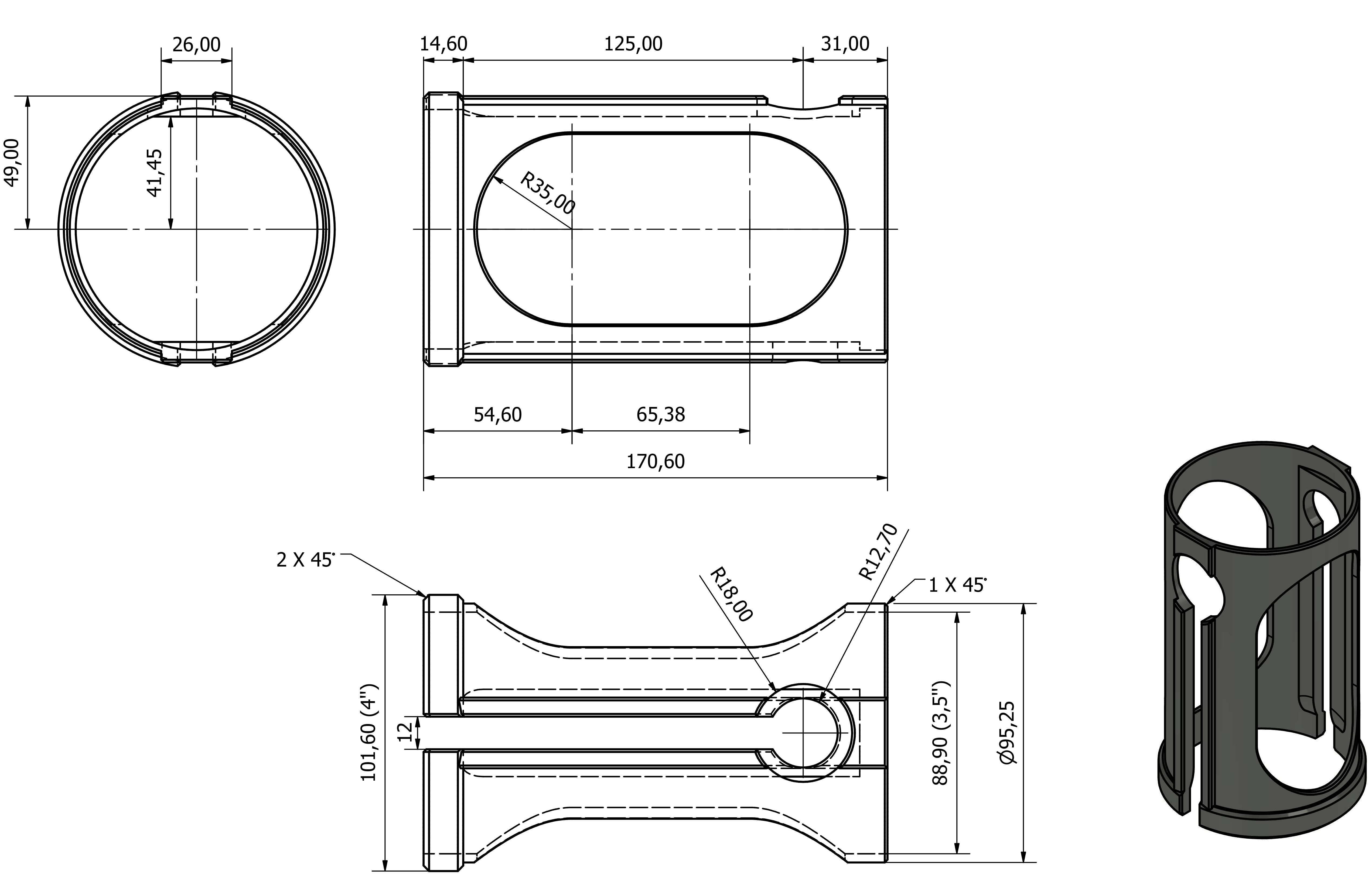


| Designed by | Weight | Material | Scale | Date |
|---|---|---|---|---|
| S. Solbakken, T. Lysgaard, B. Mikkelsen | 0.117 kg | Titanium Grade 19 | 2:1 | 18/05/2020 |
| Moonshine Solutions AS | SmartCock | | | |
| | Stem | | Draw Nr. 03 | Sheet 1 / 1 |

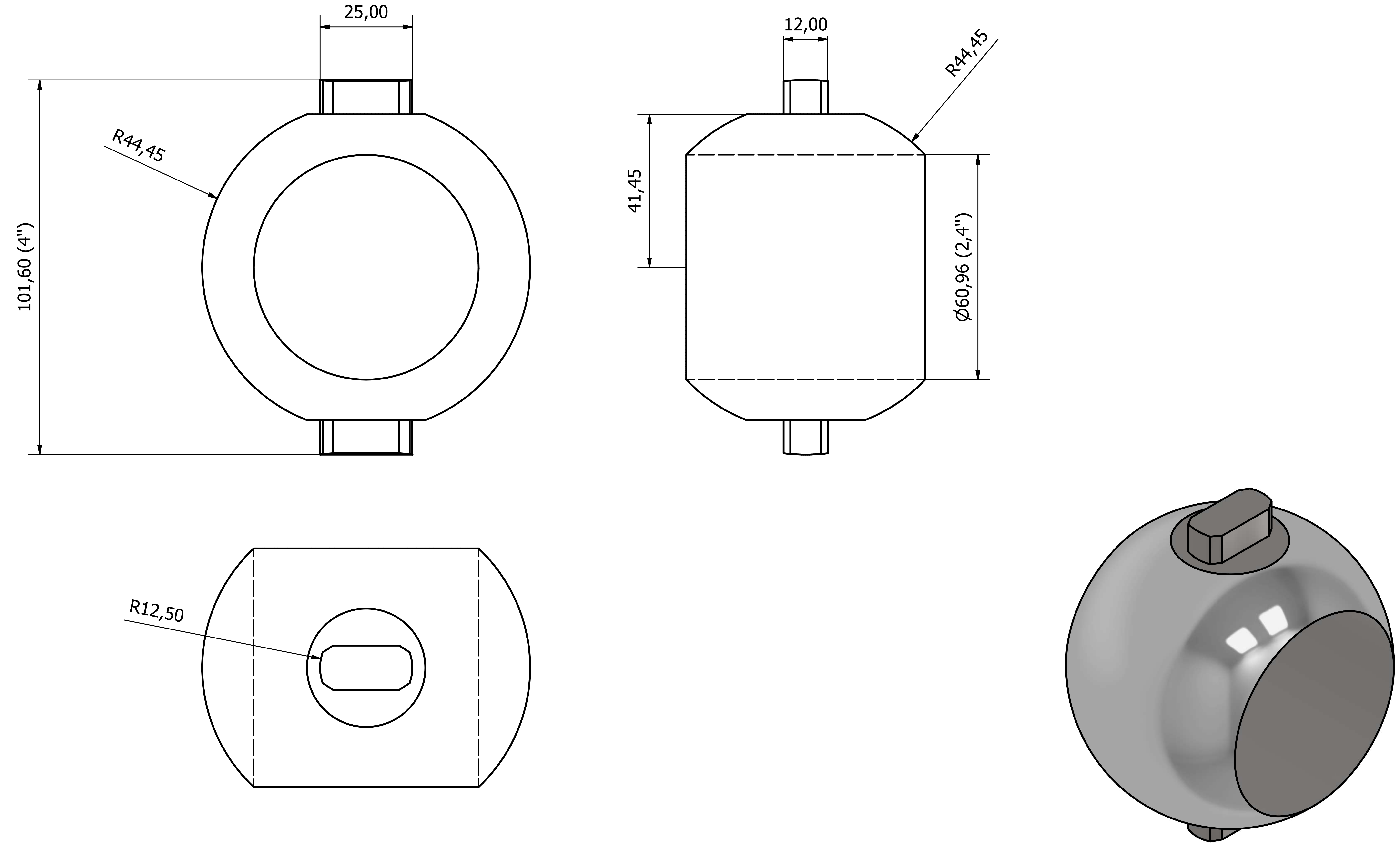



| Designed by | Weight | Material | Scale | Date |
|---|---|---|---|---|
| S. Solbakken, T. Lysgaard, B. Mikkelsen | 0.390 kg | Stainless Steel | 1:1 | 18/05/2020 |
| Moonshine Solutions AS | SmartCock | | | |
| | Upper Seat | | Draw Nr. 06 | Sheet 1 / 1 |

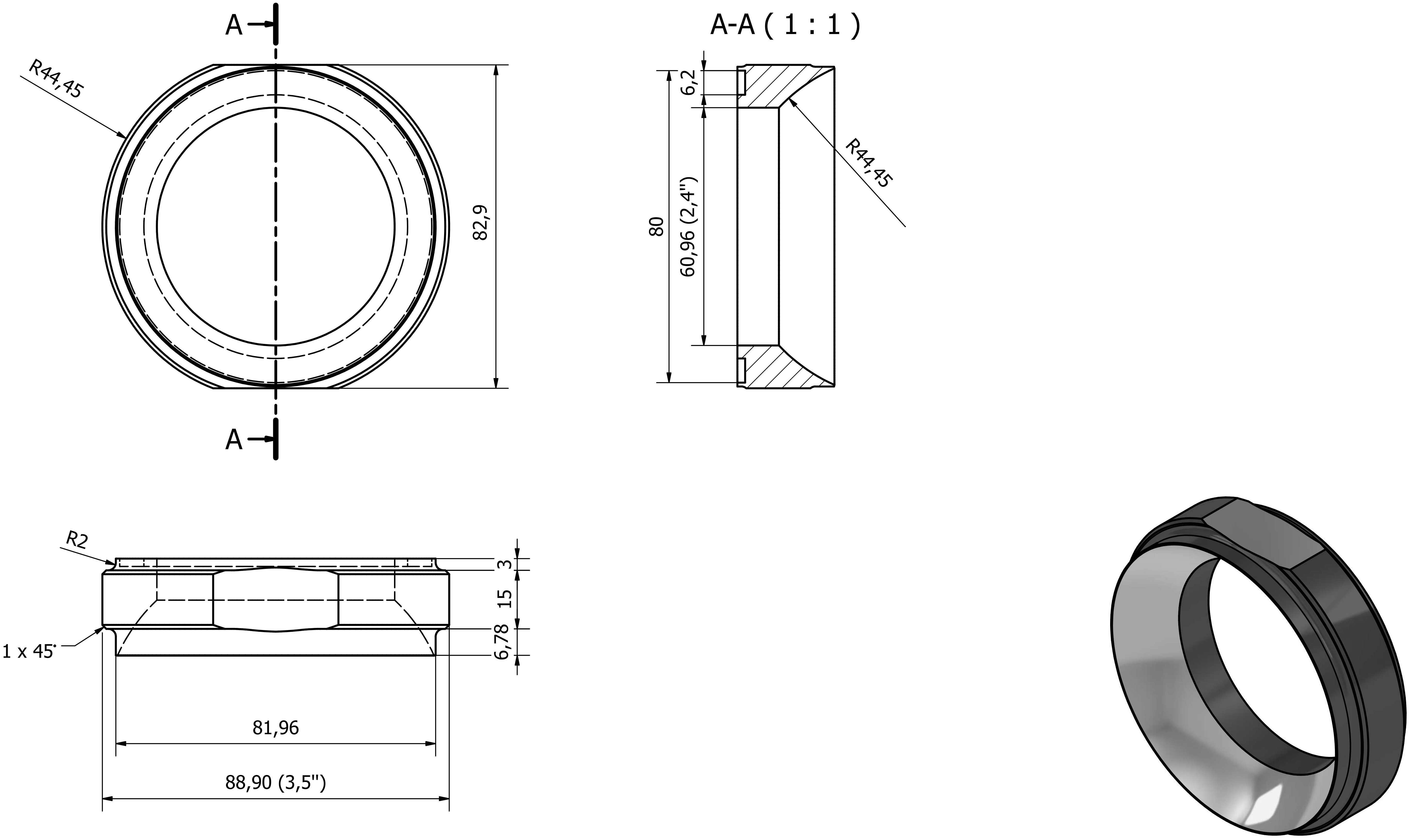



| Designed by | Weight | Material | Scale | Date |
|---|---|---|---|---|
| S. Solbakken, T. Lysgaard, B. Mikkelsen | 0.499 kg | Titanium | 2:3 | 18/05/2020 |

| | | | |
|---|---|---|---|
| Moonshine Solutions AS | SmartCock | | |
| | Sleeve | Draw Nr. 07 | Sheet 1 / 1 |

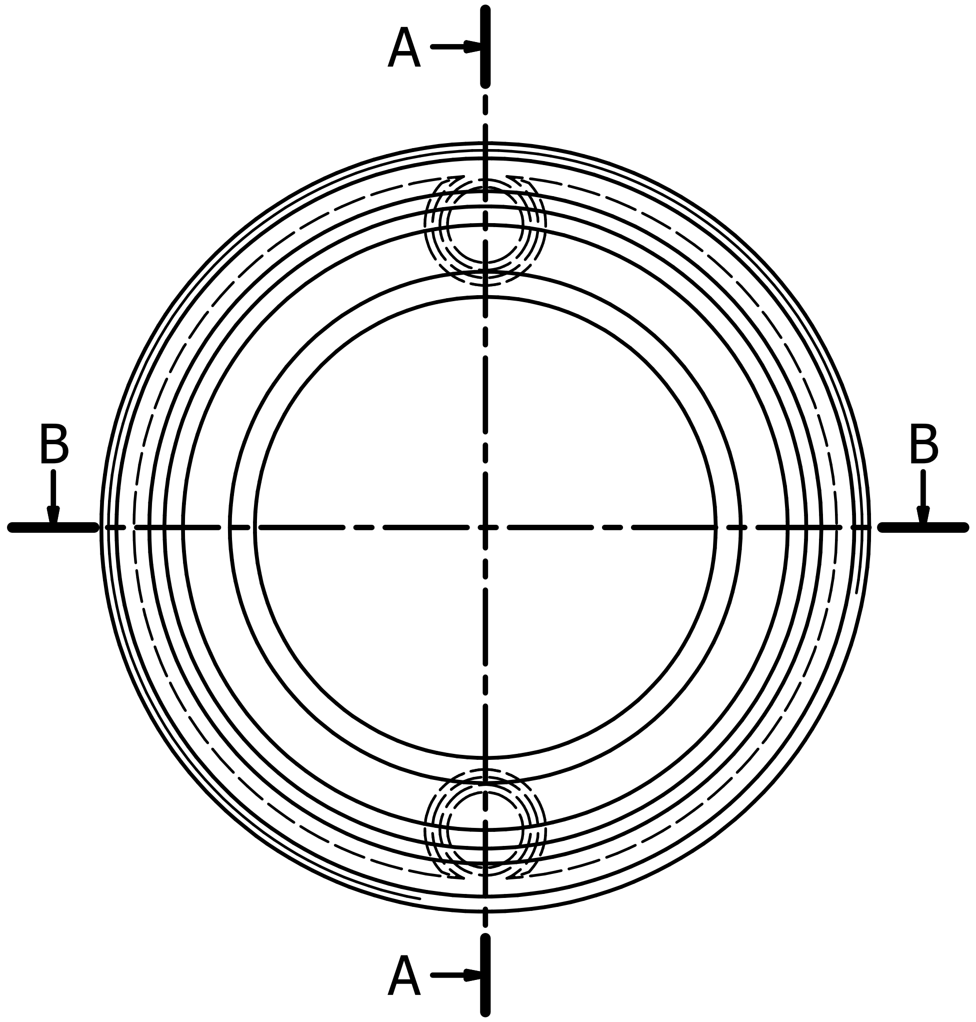


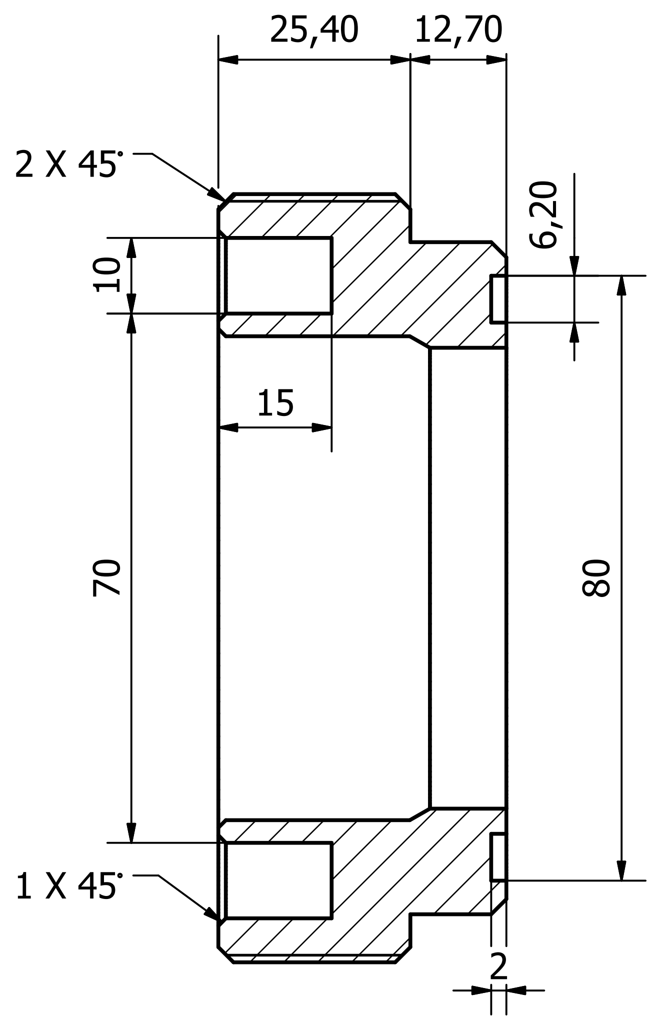

| Designed by | Weight | Material | Scale | Date |
|---|---|---|---|---|
| S. Solbakken, T. Lysgaard, B. Mikkelsen | 1.157 kg | Stainless Steel | 1 | 18/05/2020 |
| Moonshine Solutions AS | SmartCock | | | |
| | Ball | | Draw Nr. 08 | Sheet 1 / 1 |




| Designed by | Weight | Material | Scale | Date |
|---|---|---|---|---|
| S. Solbakken, T. Lysgaard, B. Mikkelsen | 0.404 kg | Stainless Steel | 1 : 1 | 18/05/2020 |

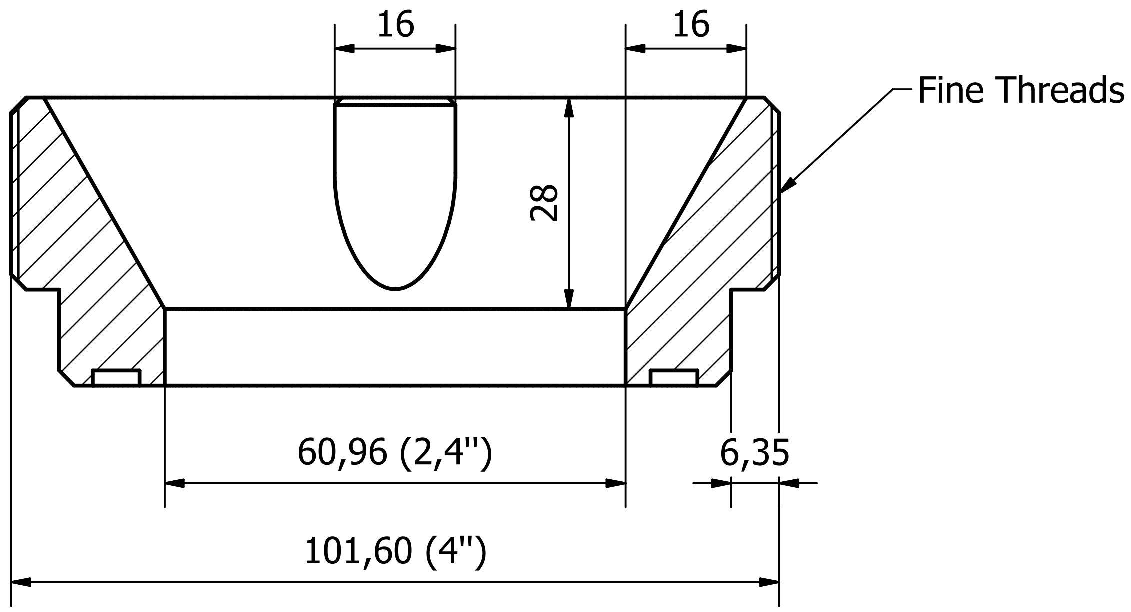
| | | | |
|---|---|---|---|
| Moonshine Solutions AS | SmartCock | | |
| | Lower Seat | Draw Nr. 09 | Sheet 1 / 1 |

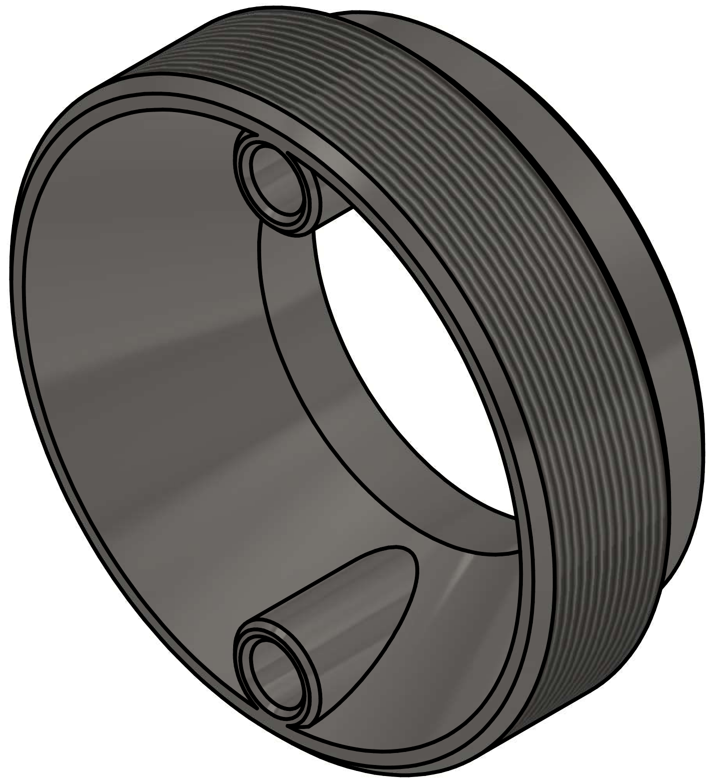



A-A ( 1 : 1 )

B-B ( 1 : 1 )



| Designed by | Weight | Material | Scale | Date |
|---|---|---|---|---|
| S. Solbakken, T. Lysgaard, B. Mikkelsen | 0.581 kg | Titanium Grade 19 | 1 : 1 | 18/05/2020 |
| Moonshine Solutions AS | SmartCock | | | |
| | Retainer Cap | | Draw Nr. 11 | Sheet 1 / 1 |

# Attachment 2 - Spring specification

## Guide

| | |
|---|---|
| Spring Strength Calculation | Compression Spring Design |
| Design Type | $F_8$, D, Assembly Dimensions --> d, $L_0$, n, $F_1$ |
| Method of Stress Curvature Correction | No Correction |

## Spring Load

| | | |
|---|---|---|
| Min. Load | $F_1$ | 877.994 N |
| Max. Load | $F_8$ | 1098.000 N |
| Working Load | F | 950.000 N |

## Spring Dimensions

| | | |
|---|---|---|
| Loose Spring Length | $L_0$ | 127.708 mm |
| Wire Diameter | d | 7.100 mm |
| Pitch of Free Spring | t | 37.244 mm |
| Outside Spring Diameter | $D_1$ | 80.000 mm |
| Mean Spring Diameter | D | 72.900 mm |
| Inside Spring Diameter | $D_2$ | 65.800 mm |
| Spring Index | c | 10.268 ul |

## Spring Coils

| | | | | |
|---|---|---|---|---|
| Active Coils | n | 3.000 ul | | |
| Rounding of Coils Number | 1 | | | |
| Coil Direction | right | | | |
| Spring Ends | | | | |
| Params | Start | | End | |
| Closed End Coils | $n_{z1}$ | 1.500 ul | $n_{z2}$ | 1.000 ul |
| Transition Coils | $n_{t1}$ | 1.000 ul | $n_{t2}$ | 0.750 ul |
| Ground Coils | $z_{o1}$ | 0.750 ul | $z_{o2}$ | 0.500 ul |

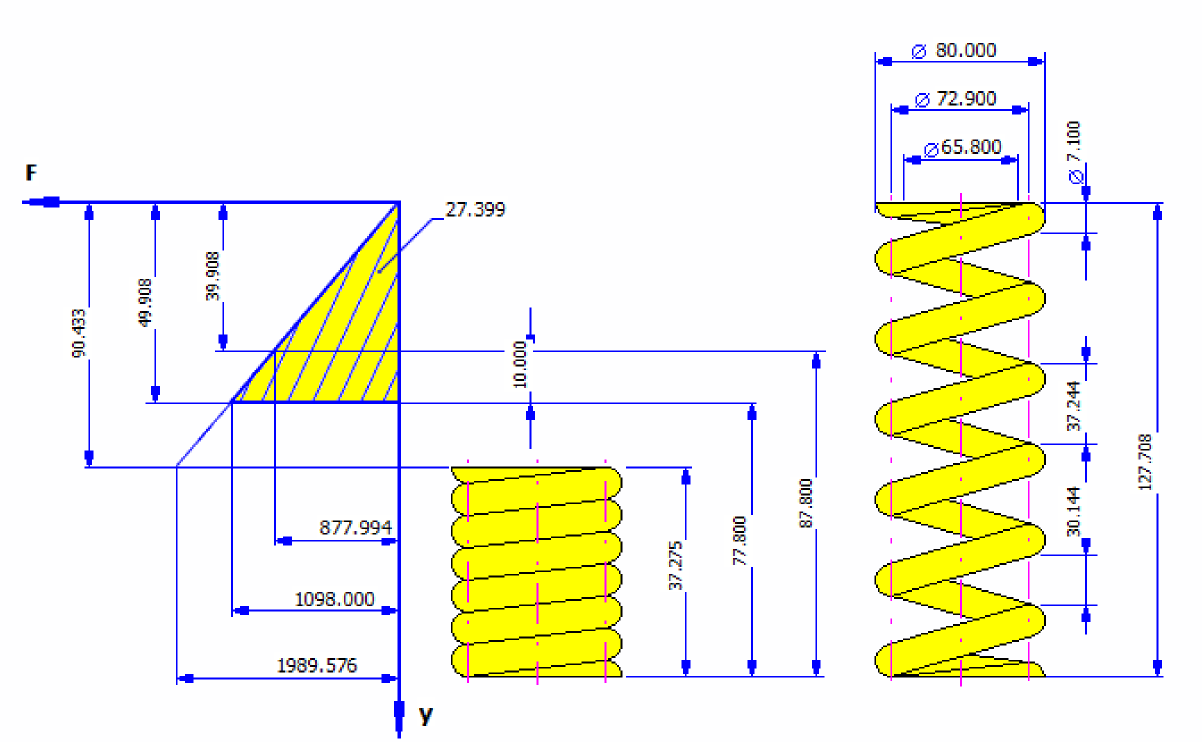
## Assembly Dimensions

| | | |
|---|---|---|
| Min. Load Length | $L_1$ | 87.800 mm |
| Max. Load Length | $L_8$ | 77.800 mm |
| Working Stroke | H | 10.000 mm |
| Working Load Length | $L_w$ | 84.527 mm |
| Installed Length | L | 87.800 mm |

## Spring Material

| Drawn patented - Carbon steel - 1st class | | |
|---|---|---|
| Ultimate Tensile Stress | $\sigma_{ult}$ | 1470.000 MPa |
| Allowable Torsional Stress | $\tau_A$ | 735.000 MPa |
| Modulus of Elasticity in Shear | G | 80500.000 MPa |
| Density | ρ | 7850 kg/m^3 |
| Utilization Factor of Spring Material | us | 0.900 ul |

## Working Diagram



## Results

| | | |
|---|---|---|
| Space between Coils of Free Spring | a | 30.144 mm |
| Pitch of Free Spring | t | 37.244 mm |
| Stress Concentration Factor | $K_w$ | 1.000 ul |
| Spring Constant | k | 22.001 N/mm |
| Min. Load Spring Deflection | $s_1$ | 39.908 mm |
| Total Spring Deflection | $s_8$ | 49.908 mm |
| Limit Spring Deflection | $s_9$ | 90.433 mm |
| Limit Test Length of Spring | $L_{minf}$ | 43.424 mm |
| Theoretic Limit Length of Spring | $L_9$ | 37.275 mm |
| Spring Limit Force | $F_9$ | 1989.576 N |
| Min. Load Stress | $\tau_1$ | 455.390 MPa |
| Max. Load Stress | $\tau_8$ | 569.502 MPa |
| Solid Length Stress | $\tau_9$ | 1031.937 MPa |
| Critical Speed of Spring | v | 13.008 mps |
| Natural Frequency of Spring Surge | f | 160.491 Hz |
| Deformation Energy | $W_8$ | 27.399 J |
| Wire Length | l | 1283.040 mm |
| Spring Mass | m | 0.399 kg |
| Spring Check Result | | **Positive** |

## Summary of Messages

16:25:33 : Calculation indicates design compliance!

# Attachment 3 - Patent description

# Certificate of registration of patent

## Patent number: 343784

Proprietor: MOONSHINE SOLUTIONS AS

The patent was granted in Norway in: 2019.06.03

Inventor: Helge Hope, Gimlebakken 27, 5052 BERGEN, Norge
Alf Breivik, Aplabrotet 6, 5730 ULVIK, Norge
Jan Georg Tveiterås, Gyldenprisveien 20, 5056 BERGEN, Norge

This is to certify that the Norwegian Industrial Property Office, in accordance with the Norwegian Patents Act of 15 December 1967, has granted a patent for the enclosed invention.

REGISTRERT RETTIGHET · I NORGE ·

Per A. Foss

Per A. Foss
Director

# An improved drill string safety valve device

## Field of the invention

The invention concerns the field of drilling into fluid reservoirs in earth formations. More specifically, the invention concerns an improved valve for controlling pressures inside a drill string when drilling into a fluid reservoir in an earth formation. The invention is particularly useful for controlling pressure variations inside a drill string when drilling into subterranean reservoirs.

## Background of the invention

Drilling wells into subterranean reservoirs containing hydrocarbon fluids and/or water, be it on an onshore location or a subsea seabed, requires a continuous monitoring and control of the fluid pressure inside the drill string. Hydrocarbon wells are subjected to rapid pressure differentials that may cause great damage and be catastrophic for the drilling rig and rig personnel if uncontrolled. Safety valves that are connected to, and form a part of, the drill string are essential components for ensuring the safety of drilling operations. Such valves may be used as safety valves on the rig floor as well as down-hole, to manage safe operations by controlling kicks and preventing back-flow of the drilling mud inside the drill string during drilling operations.

Various types of drilling safety valves exist. One example is the so-called "Kelly Valve", which is a manually operated ball valve used to close the bore of the drill string and stop backflow. The Kelly Valve is designed for high-pressure conditions and can hold pressure from both directions, and is normally screwed into the top of a drill string, below the top drive, where it may be operated by a drill floor worker. The Kelly Valve is also often referred to as a "Full Opening Safety Valve" (FOSV), because when the ball valve is in the open position; the flow path through the valve has a smooth inside diameter. During drilling operations, Kelly Valves matching the applicable sizes of drill pipe, drill collar, tubing, etc. need to be available on drill floor, ready to be stabbed in. When abnormal situations occur and a predefined pressure limits is exceeded, the drill floor worker can close the Kelly Valve to stabilize the pressure inside the drill string and avert a potential kick. One example of a Kelly Valve is disclosed by US 3 086 746.

Another type of drilling safety valve for installation onto the drill string, is the so-called "Inside Blow-out Preventer Valve" (IBOP Valve), also commonly referred to as a "Gray Valve". The IBOP Valve is a check valve, which allows pumping through the valve and into the drillstring, but prevents upward flow.

One problem associated with the known drilling safety valves is their size and weight. Typically, each valve may weigh approximately 100 kg, requiring at least two drill floor workers to be located on the drill floor in order to install the valve into the drill string. The installation is time consuming, which is undesirable in an emergency situation. It is therefore an object of the invention to provide a smaller, and thus lighter, drilling safety valve than those of the prior art.

**Summary of the invention**

The invention is set forth and characterized in the main claim, while the dependent claims describe other characteristics of the invention.

It is thus provided a drill string safety valve device, comprising a body with a through-going flow bore and connectors at respective ends of the flow bore for connection to tubulars, characterized by a valve member movably arranged in said flow bore and configured for being set in one of two states, wherein a first state of the valve member allows fluid to flow in both directions though the flow bore, and wherein a second state of the valve member allows fluid flow through the flow bore in only one direction.

In one embodiment, the valve member comprises guide-and-support means whereby the valve member may be moved between the two states. The valve member may be movably arranged on a first support member and the valve member is configured to bear against a first valve seat. In one embodiment, the first valve seat is arranged on the first support member. The drill string safety valve device comprises a resilient element, such as a spring, configured to exert a force on the valve member.

In one embodiment, the valve member comprises a rotatable ball valve having a through-going flow bore.

In another embodiment, the valve member comprises a valve head movably connected to a second support member, and the second support member is movably connected to the first support member.

The drill string safety valve device further comprises operating means configured for operating the valve member between said two states.

The drill string safety valve device body is in one embodiment a tubular body having an outer diameter corresponding to the outer diameter of the tubulars to which it may be connected.

A common principle of both of the embodiments of the invented safety valve device is its dual functional capability: as check valve and as a full opening safety valve. The change between these two states (or configurations) is effectuated by moving the valve member (e.g. ball valve or the valve head) inside the valve body. Preferably, the valve member is moved along the body longitudinal axis. The valve member is preferably moved with respect to the valve cage.

The invented safety valve device is therefore in effect a combination of the above mentioned Kelly Valve and the Grey Valve, and as such provides greater flexibility and improved logistics, compared to the safety valves of the prior art. The invented safety valve device may be made compact, and weights on the order of 25 kg have been envisaged. However, the invention shall not be limited to weights or dimensions.

## Brief description of the drawings

These and other characteristics of the invention will become clear from the following description of an embodiment, given as a non-restrictive example, with reference to the attached schematic drawings, wherein:

Figure 1 is perspective view of an embodiment of the invented safety valve device;

Figure 2 is an exploded perspective view of the valve device illustrated in figure 1;

Figure 3 is an enlargement of the encircled area "A" in figure 2;

Figure 4 is a sectional drawing of an embodiment of the invented valve device, in a forced-closed state;

Figure 5 is a perspective, and partly cutaway, view of an embodiment of the invented valve device, in a forced-closed state, and thus corresponding to figure 4;

Figure 6 is a sectional drawing of the invented valve device, in a forced-open state;

Figure 7a is a perspective, and partly cutaway, view of the invented valve device, in a forced-open state, and thus corresponding to figure 6;

Figure 7b is a perspective, and partly cutaway, view of the valve device and state as illustrated in figure 7a, but seen from a view diametrically opposite to that of figure 7a;

Figure 8 is a sectional drawing of the invented valve device in a static and locked-open state;

Figure 9a is a perspective, and partly cutaway, view of the invented valve device in a static and locked-open state, and thus corresponding to figure 8;

Figure 9b is a perspective, and partly cutaway, view of the valve device and state as illustrated in figure 9a, but seen from a view diametrically opposite to that of figure 9a;

Figure 10a is a sectional drawing of an alternative embodiment of the invented valve device, in a static and locked-open state;

Figure 10b is a sectional drawing of the valve cage, inner valve cage and valve member of the embodiment of the invented valve device illustrated in figure 10a (i.e. in a static and locked-open state), and the where the view is rotated 90° around the body longitudinal axis y, compared to the view in figure 10; and

Figure 10c corresponds to figure 10b, but shows the valve member in a forced-closed state.

**Detailed description of a preferential embodiment**

The following description will use terms such as “horizontal”, “vertical”, “lateral”, “back and forth”, “up and down”, ”upper”, “lower”, “inner”, “outer”, “forward”, “rear”, etc. These terms generally refer to the views and orientations as shown in the drawings and that are associated with a normal use of the invention. The terms are used for the reader’s convenience only and shall not be limiting.

Referring to figure 1, the invented safety valve device 1 comprises in the illustrated embodiment a tubular, elongated, body 2, with an axial, through-going, flow bore 25 and oppositely arranged openings 18, 19. A pin 4 and a box 3 are arranged at opposite axial ends of the body. In a practical application, the body 2 outer diameter OD will correspond to the outer diameter of the drill string joints (not shown) to which the safety valve will be connected. In use, the pin 4 is connected to a drill string joint (not shown) below the valve body 2, and the box 3 is connected to a drill string joint (not shown) above the valve body 2. Such connection means and methods are well known to the skilled person, and need therefore not be described in more detail here. The tubular, elongated, body 2 with an outer diameter corresponding to the drill string outer diameter, enables running the valve device downhole as a part of the drill string.

Also seen in figure 1 is the head of a valve actuator bolt 5. The actuator bolt 5 extends through an access bore 20 (see figure 2) in the body 2. In the illustrated embodiment, the head has a shape which is compatible with an Allen wrench whereby the bolt may be operated by e.g. a drill floor worker, but the skilled person will understand that the bolt head may have other shapes. Reference numbers 6a and 6d point to indicator markings on the valve body 2, where 6a indicates a locked-open valve (allowing backflow) and 6b indicates a check valve configuration (discussed below).

Figures 2 and 3, are exploded views of parts inside the valve body. It should be noted that seals, threads and other fasteners for assembly are not shown in the figures, as these features are well known for the skilled person and need therefore not be illustrated or described for the purpose of elucidating the invention. Figures 2 and 3 show a valve cage 9, having an elongated, tubular body, and configured for assembly inside the valve body 2. The valve cage 9 comprises flow openings 26a,b at opposite axial ends of the cage, and lateral flow openings 22. The valve cage 9 also comprises a pair of elongated, guide slots 23, 24 arranged at diametrically opposite sides of the cage and extending along the cage 9 (and valve body 2) longitudinal axis.

A ball valve 7 comprises a pair of guide pegs 12, 14 arranged at diametrically opposite sides of the ball valve. The first guide peg 12 is configured to be slidably arranged in the first guide slot 23, and the second guide peg 14 is configured to be slidably arranged in the second guide slot 24, as is explained in more detail below. The first guide peg 12 comprises at its free end a receptacle 13 having a geometry which is compatible with the above mentioned actuator bolt 5. The ball valve 7 may thus be operated (rotated) by inserting the actuator bolt in the receptacle 13 and then rotating the bolt.

A first (upper) valve seat 10 is arranged at one end of the cage 9 and configured for sealing engagement with the ball valve 7. A second (lower) valve seat 11 is arranged at the opposite end of the cage 9 and configured for sealing engagement with the ball valve 7. Integral with the second valve seat 11 is a first (upper) spring abutment ring 15. (It should be understood, however, that the second valve seat 11 and first spring abutment ring 15 may be separate parts.) Arranged between the first spring abutment ring 15 and a second (lower) spring abutment ring 16 is a coil spring 8 A retainer ring 17 serves to secure the above mentioned parts inside the valve body 2 flow bore 25 when assembled. Required threads and/or locking member to secure the retainer ring 17 to the valve body 2 are not illustrated, as such devices are well known in the art.

Figures 4 and 5 illustrate the safety valve device 1 in an assembled state. The retainer ring 17 is fixed to the valve body 2 and holds the valve parts in place inside the flow bore 25. The upper end of the valve cage 9 and the first (upper) valve seat 10 bear against a flow bore neck 27. The spring 8 forces the ball valve 7 along the body longitudinal axis y, against the first (upper) valve seat 10, and the ball valve thus prevents back-flow (i.e. upwards in figure 4). The ball valve 7 comprises a through-going flow bore 21, but in the ball valve position shown in figures 4 and 5, the flow bore 21 is not aligned with the valve device flow bore 25, whereby flow through the ball valve flow bore 21 is prevented.

In this "forced-closed" configuration, wherein the ball valve is forced against the upper valve seat 10 by the spring force, the safety valve device 1 has the characteristics of a Grey Valve (i.e. a check valve configuration), preventing back flow through the valve body 2. It should be understood that the spring 8 will have to be designed and dimensioned (including the required resilience and stiffness) according to the intended use of the valve device.

Figures 6, 7a and 7b show a valve configuration similar to that of figures 4 and 5, but here a fluid F is applied from above, with pressure sufficient to overcome opposing force in the spring 8. The ball valve 7 is thus pressed downwards by the fluid F, whereby the fluid may flow around the ball valve 7 and out through the cage openings 22 into the flow bore outside the cage, and back in through the cage openings 22 and out of the valve body through the opening 18. The arrows indicate the flow path. Thus, these figures may be illustrative of a situation in which a drill fluid is pumped through the valve device and into the drill string below the valve device. Thus, in this "forced-open" configuration, the ball valve is forced away from the upper valve seat 10 by the fluid force.

As discussed above, and as readily apparent from figures 7a and 7b, the first guide peg 12 is configured to be slidably arranged in the first guide slot 23 (see figure 7a), and the second guide peg 14 is configured to be slidably arranged in the second guide slot 24 (see figure 7b). As shown in figure 7b, the free end 14’ of the second guide peg 14 is shaped to allow movement in the second guide 24 slot only when the second guide peg is in certain orientation. In the illustrated embodiment, the peg free end 14’ has an oblong cross-section with two opposite flat surfaces and is dimensioned such that the peg may travel in the second guide slot 24 only when flat surfaces are aligned with the guide slot, as shown in figure 7b. Referring to figure 7a, it will be understood that the ball valve 7 may be rotated by rotating the first guide peg 12, in the illustrated embodiment by rotating the actuator bolt 5 when the bolt is inserted in the receptacle 13. It will also be understood that such ball valve rotation is not possible when the second guide peg 14 free end 14’ is in the second guide slot (as shown in figures 7a and 7b). The ball valve 7 may only be rotated when the guide peg free end 14’ is in the enlarged portion 24’ (see figure 7b) at the upper end of the second guide slot 24.

This configuration is illustrated in figure 9b, where the ball valve 7 has been rotated a quarter-turn, such that the guide peg free end 14’ is lodged in the enlarged portion 24’ and prevented from moving in the second guide slot 24. Figures 8, 9a and 9b thus illustrate the invented valve device in a static and "locked-open" state, in which the ball valve 7 is prevented from moving inside the valve cage. However, the ball valve 7 through-going flow bore 21 is aligned with the valve device flow bore 25 when the ball valve is in the position shown in figures 8, 9a and 9b. Therefore, in this configuration, the valve device has an open, through-going bore (21 and 25) and fluid F (e.g. drill fluids) may flow upwards (backflow) through the valve device as indicated by the arrows. When the ball valve is in this position, the valve device may be installed onto the drill string, in case of upward flow, and subsequently closed to stop the upward flow.

In use, the invented valve device 1 may be placed in the top of a drill string, below the top drive, and set in a state as shown in figures 8, 9a and 9b, i.e. with the ball valve flow bore 21 aligned with the valve device flow bore 25 and thus allowing uninhibited flow through the valve device. This state would be similar to the operation of a conventional Kelly Valve. When abnormal situations occur and a predefined pressure limits is exceeded, the drill floor worker may engage the actuator bolt 5 and rotate the ball valve to the state illustrated in figures 6, 7a and 7b. In this state, the ball valve flow bore 21 does not permit flow through the ball valve, but the ball valve is movable in the guide slots 23, 24. In this state, the invented valve device is similar to an IBOP Valve (or Gray Valve), i.e. a check valve which allows pumping through the valve into the drillstring but prevents upward flow.

An alternative embodiment of the invented valve device is schematically illustrated in figures 10a, 10b and 10c. In this embodiment, the ball valve (7, discussed above) has been replaced by a dome-shaped valve head 29 which is configured to seal against the upper valve seat 10. The valve head 29 is supported, via a stem 33, by an inner valve cage 28 which is slidably arranged inside the valve cage 9. Although not illustrated, it should be understood that the inner valve 28 comprises lateral flow openings similar to the lateral flow openings 22 in the valve cage 9, as described above. Figures 10a and 10b show the valve device in a static and locked-open state; i.e. the valve head 29 is retracted (lowered) from the upper valve seat 10 in order to allow flow through the valve device. Functionally, therefore, this configuration corresponds to the configuration illustrated in figures 8, 9a and 9b and discussed above.

The inner valve cage 28 may be moved (along the body longitudinal axis y) inside the valve cage 9. As an example, this movement may be accomplished by a rack-and-pinion mechanism 30, 31, wherein the pinion 31 is operated (turned) by an actuator (not shown) extended through the access bore 20. Turning the pinion 31 will move the inner valve cage 28 by interaction with the rack 30.

In figure 10c, the pinion 31 has been turned so as to move the inner valve cage 28 upwards until the valve head 29 upper dome-shaped portion bears against the upper valve seat 10. A spring 32 is arranged between the valve head lower side and the inner valve cage and thus serves to force the valve head toward the valve set 10. Functionally, the configuration illustrated in figure 10c corresponds to the configuration illustrated in figures 4 and 5 and discussed above (i.e. a check valve configuration).

When a pressure of a magnitude sufficient to overcome the force in the spring 23, the valve head 29 may be moved a distance *d*, away from the upper valve seat and towards the inner valve cage. This pressure may for example be in the form of a fluid, indicated by the arrow F in figure 10c. Functionally, this configuration (which is not illustrated per se) corresponds to the configuration illustrated in figures 6, 7a and 7b and discussed above (i.e. a check valve configuration).

A common principle of both of the embodiments of the invented valve device is the dual capability of functioning both as check valve and as a full opening safety valve, and wherein the change between these two configurations may be effectuated by moving a valve member (e.g. the ball valve 7 or the valve head 29) inside the valve body 2. Preferably, the valve member is moved along the body longitudinal axis y. The valve member is preferably moved with respect to the valve cage.

It should be understood that the invented safety valve device may be made of any material suitable for the intended use, for example stainless steel.

Although the invention has been described with reference to a coil spring 8, it should be understood that other resilient members may be equally applicable.

**Claims**

1. A drill string safety valve device (1), comprising a body (2) with a through-going flow bore (25) and connectors (3, 4) at respective ends (19, 18) of the flow bore for connection to tubulars, **characterized by** a valve member (7; 29) movably arranged in said flow bore (25) and configured for being set in one of two states, wherein a first state of the valve member allows fluid to flow in both directions though the flow bore, and wherein a second state of the valve member allows fluid flow through the flow bore in only one direction.

2. The drill string safety valve device of claim 1, wherein the valve member (7; 29) comprises guide-and-support means (9, 12, 14, 23, 24) whereby the valve member may be moved between the two states.

3. The drill string safety valve device of any one of claims 1-2, wherein the valve member (7; 29) is movably arranged on a first support member (9).

4. The drill string safety valve device of any one of claims 1-3, wherein the valve member (7; 29) is configured to bear against a first valve seat (10).

5. The drill string safety valve device of any one of claims 3-4, wherein the first valve seat (10) is arranged on the first support member (9).

6. The drill string safety valve device of any one of claims 1-5, further comprising a resilient element (8; 32) configured to exert a force on the valve member (7; 29).

7. The drill string safety valve device of any one of claims 1-6, wherein the valve member comprises a rotatable ball valve (7) having a through-going flow bore (21).

8. The drill string safety valve device of any one of claims 1-6, wherein the valve member comprises a valve head (29) movably connected to a second support member (28), and the second support member (28) movably connected to the first support member (9).

9 The drill string safety valve device of any one of claims 1-8, further comprising operating means (5, 13; 31, 32) configured for operating the valve member (7; 29) between said two states.

10. The drill string safety valve device of any one of claims 1-9, wherein the body (2) is a tubular body having an outer diameter (OD) corresponding to the outer diameter of the tubulars to which it may be connected.

**Abstract**

A drill string safety valve device (1) has a body (2) with a through-going flow bore (25) and connectors (3, 4) at respective ends (19, 18) for connection to tubulars, such as pipe joints. A valve member (7; 29) is movably arranged in said flow bore (25) and configured for being set in one of two states: a first state of the valve member allows fluid to flow in both directions though the flow bore, and a second state of the valve member allows fluid flow through the flow bore in only one direction. The valve member may be a rotatable ball valve (7) with a through-going flow bore (21).

(Figure 4 to be published with the abstract)

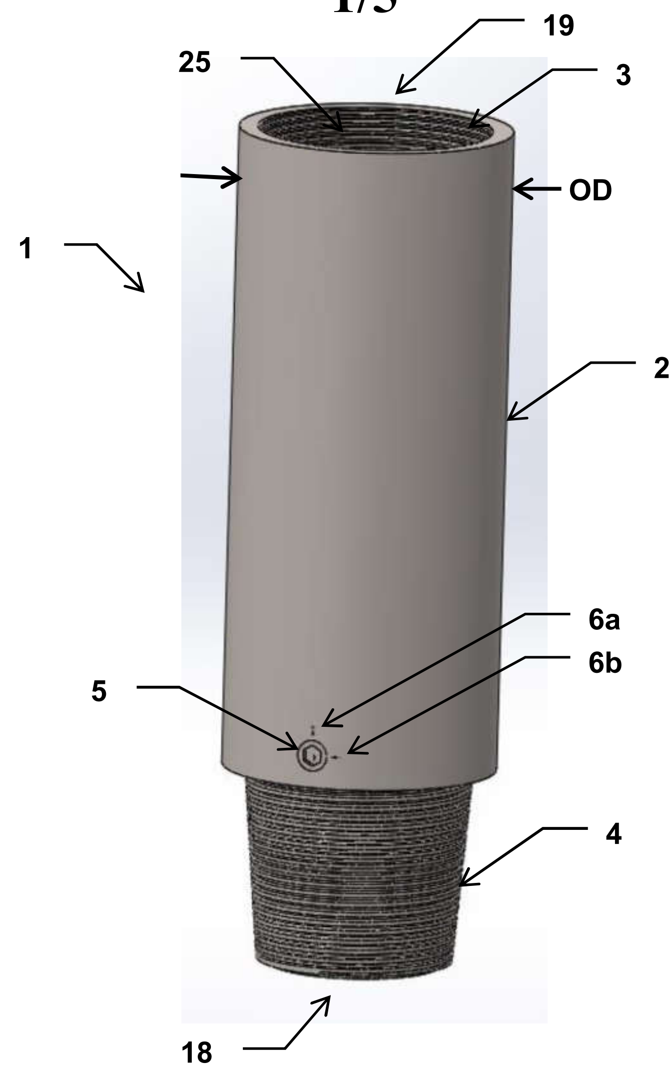



Fig. 1

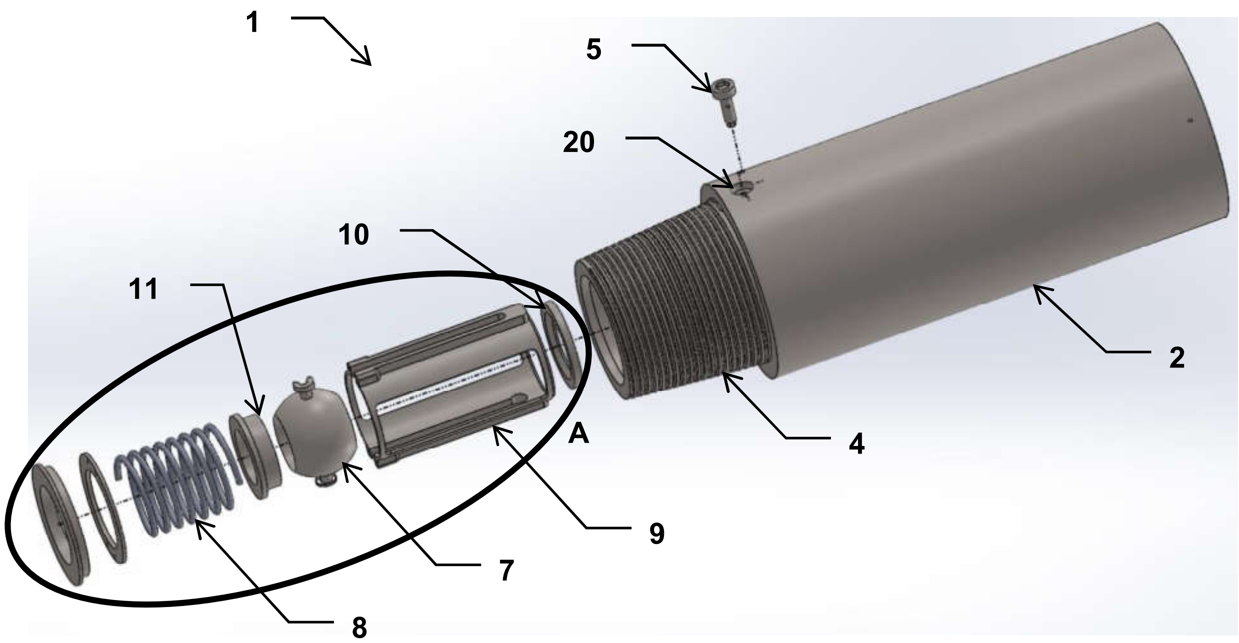



Fig. 2

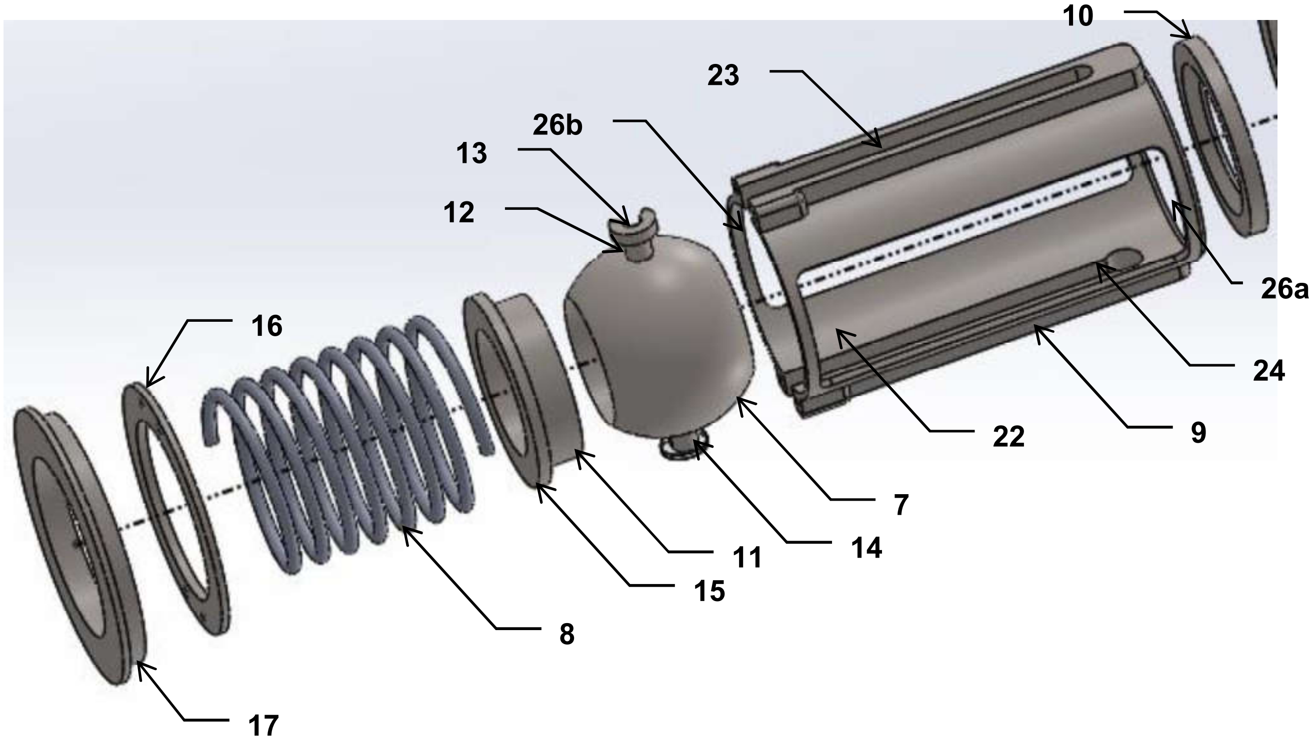



Fig. 3

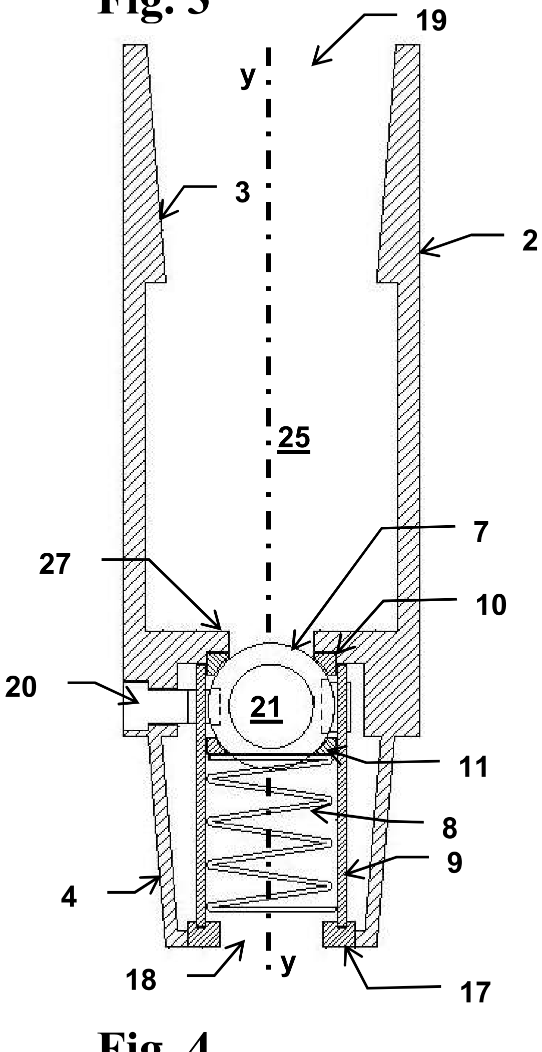



Fig. 4

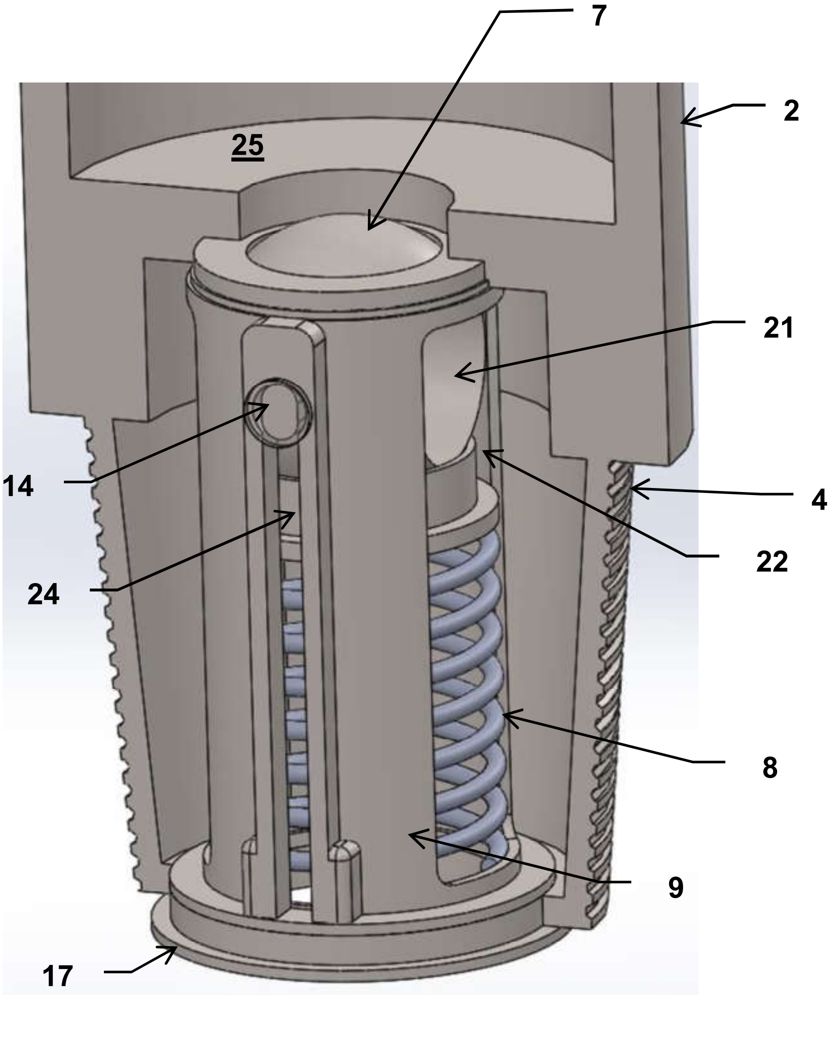



Fig. 5

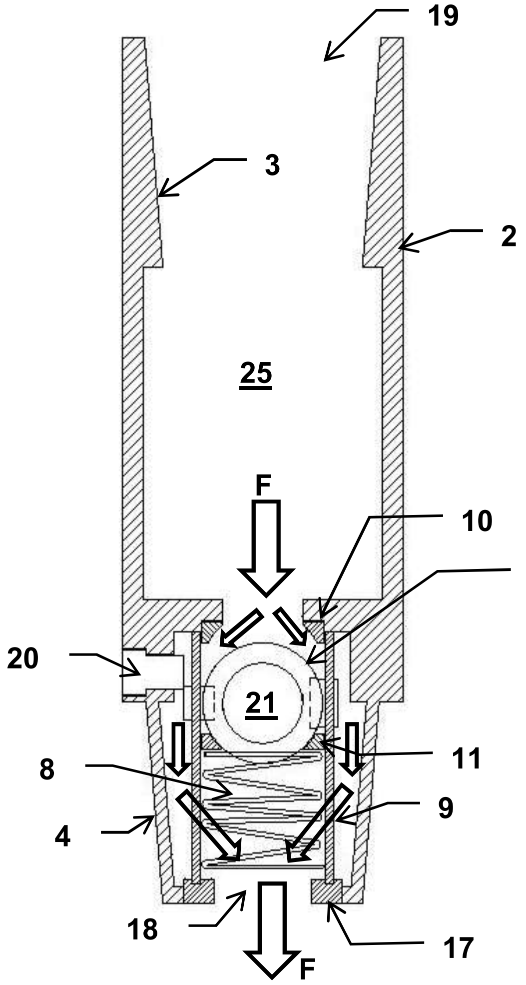



Fig. 6

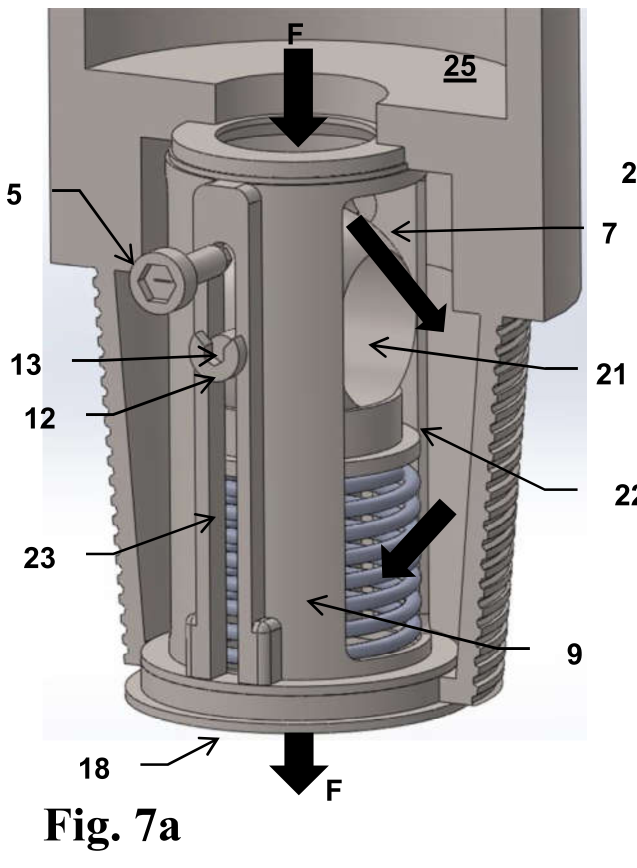



Fig. 7a

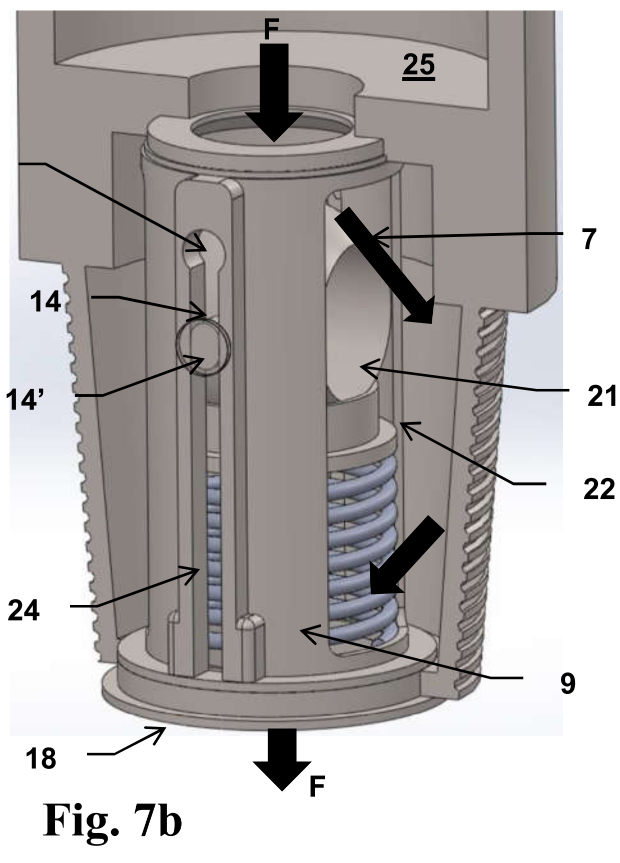



Fig. 7b

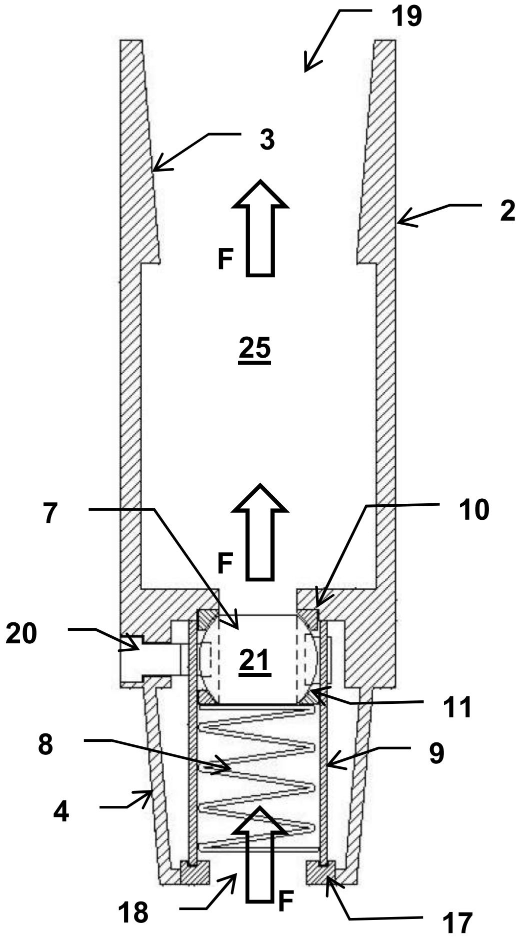



Fig. 8

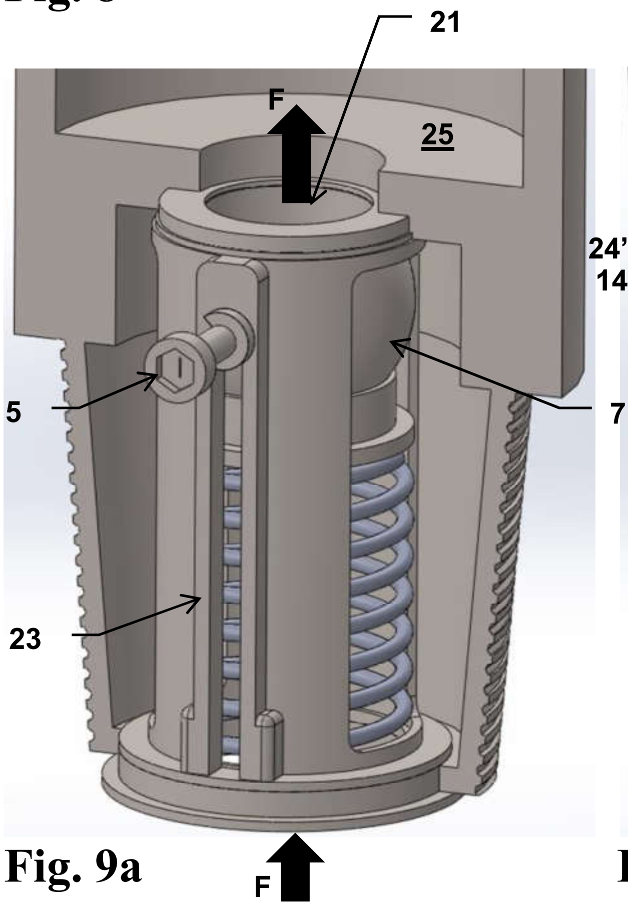



Fig. 9a

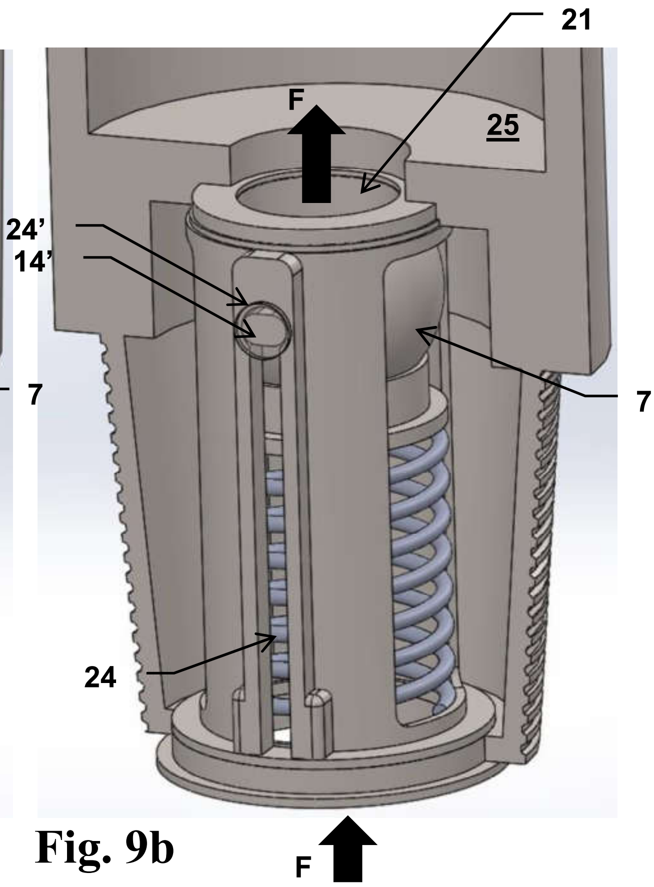



Fig. 9b

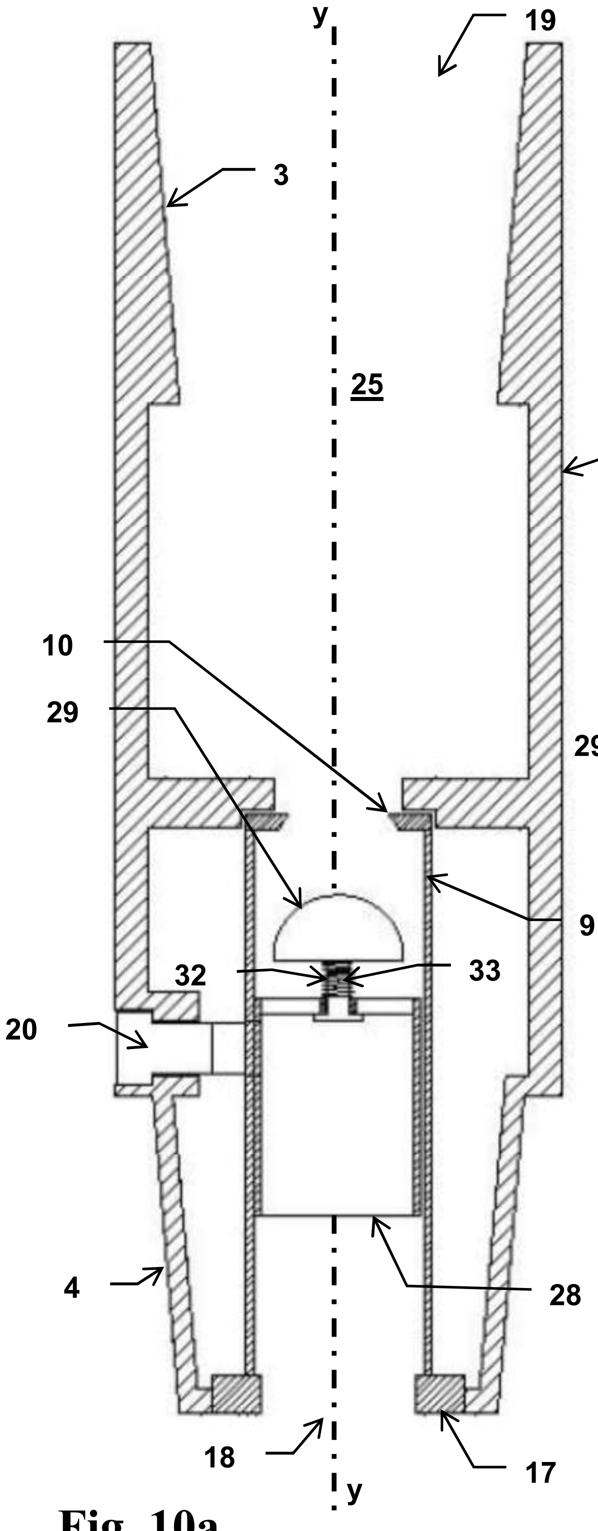



Fig. 10a

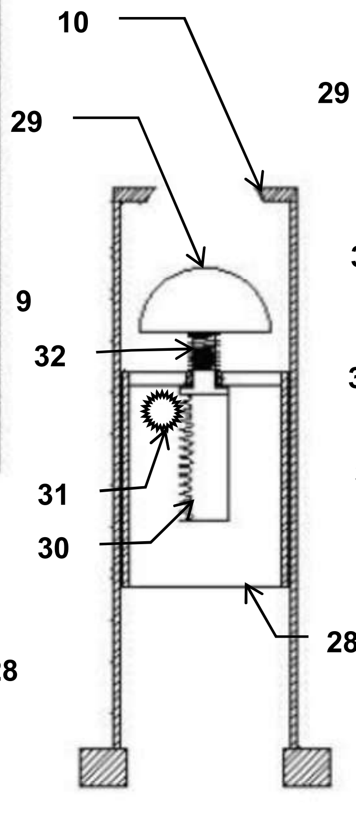



Fig. 10b

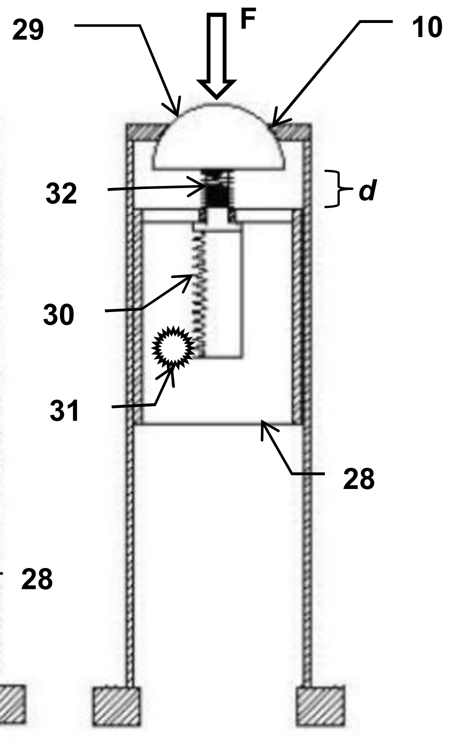



Fig. 10c

# Attachment 4 - Material datasheet

## TITANIUM Grade 3

Commercial pure titanium

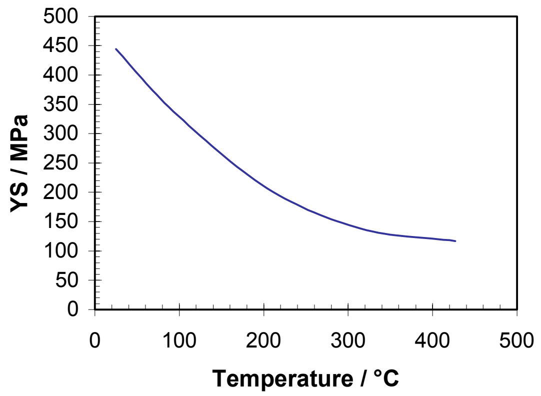
The data given is for information not for design

| **Chemical composition (weight %)** (Maximum values unless range is shown) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| O | N | C | H | Fe | Al | V | Ni | Mo | Others | Residuals |
| 0.35 | 0.05 | 0.08 | 0.015 | 0.30 | | | | | | 0.4 |

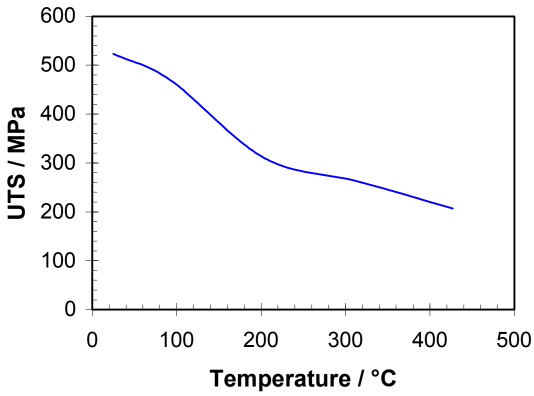
Unalloyed titanium offering optimum ductility and cold formability with useful strength, high-impact toughness, and excellent weldability. Highly corrosion resistant in oxidizing and mildly reducing environments, including chlorides.

| **Mechanical properties at room temperature** | **Minimum values** | **Typical values** |
|---|---|---|
| Yield Strength | 380 MPa | 460 MPa |
| Ultimate Strength | 450 MPa | 595 MPa |
| Elongation in 50 mm, A5 | 18 % | 25 % |
| Reduction in Area | 30 % | % |
| Hardness | | 180-220 HV |
| Modulus of elasticity | | 103 GPa |
| Charpy V-Notch Impact | | 24-48 J |

**Yield strength vs. temperature**




**Tensile strength vs. temperature**




| **Fatigue properties at room temperature** (Stress to cause failure in $10^7$ Cycles) Rotating bend | | | Direct stress limit | | |
|---|---|---|---|---|---|
| Smooth | $K_t$=1 | 380 MPa | Smooth | $K_t$=1 | 280 MPa |
| Notch | $K_t$=3 | 165 MPa | Notch | $K_t$=3 | 123 MPa |

| Physical properties | | |
|---|---|---|
| Melting point, ± 15 °C | 1680 | °C |
| Density | 4.51 | g/cm3 |
| Beta transus, ± 15 °C | 920 | °C |
| Thermal expansion, 20 - 100 °C | 8.6 | $*10^{-6}$ $K^{-1}$ |
| Thermal expansion, 0 - 300 °C | 9.2 | $*10^{-6}$ $K^{-1}$ |
| Thermal conductivity, room temperature | 17 | W/mK |
| Thermal conductivity, 400 °C | 16 | W/mK |
| Specific heat, room temperature | 0.54 | J/gK |
| Specific heat, 400 °C | 0.60 | J/gK |
| Electrical resistivity, room temperature | 56 | µW*cm |
| Poisson's ratio | 0.34-0.40 | |

| Heat treating | | Temperature | Time |
|---|---|---|---|
| Annealing | air-cooled | 650-760 °C | 6 min - 2 hours |
| Stress relieving | air-cooled | 480-595 °C | 15 min - 4 hours |

**Weldability – excellent**
Grade 3 has very good weldability. Being substantially single phase material, the microstructure of the alpha phase is not affected greatly by thermal treatments or welding temperatures. Therefore, the mechanical properties of a correctly welded joint are equal to, or exceed those of the parent metal and show good ductility

**Available mill products**
Bar, billet, ingot, extrusions, plate, sheet, strip, tubing, wire, pipe, forging, casting

**Typical Applications**
Equivalent to Grade 1 and 2, and eminently suitable where high strength is needed

| **Industry specifications** | **ASTM Grade 3, AMS 4900, JIS Grade 3, TIC, RMI 55, ST-70** |
|---|---|
| Sheet and plate | ASTM B265 Gr3, MIL-T-9046 CP-2, AMS 4900 |
| Bars and billets | ASTM B348 Gr3 |
| Tube | ASTM B337 Gr3, ASTM B338 Gr2 |
| Forging | ASTM B381 Gr3 |
| Casting | ASTM B367 Gr3, ASTM F467 Gr3 |
| | ASTM F468 Gr3, DIN 3.7055 |

## TITANIUM Grade 5

Ti - 6Al - 4V Alpha-beta alloy

The data given is for information not for design

**Chemical composition (weight %)** (Maximum values unless range is shown)

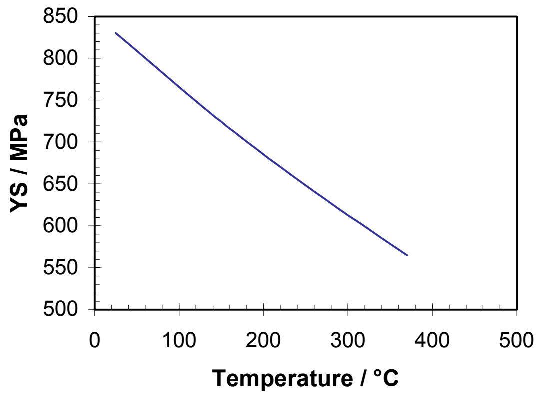
| O | N | C | H | Fe | Al | V | Ni | Mo | Others | Residuals |
|---|---|---|---|---|---|---|---|---|---|---|
| 0.20 | 0.05 | 0.08 | 0.015 | 0.40 | 5.5-6.75 | 3.5-4.5 | | | | 0.4 |

This is the most widely used titanium alloy. It has very high strength but relatively low ductility. The main application of this alloy is in aircraft and spacecraft. Offshore use is growing. The alloy is weldable and can be precipitation hardened.

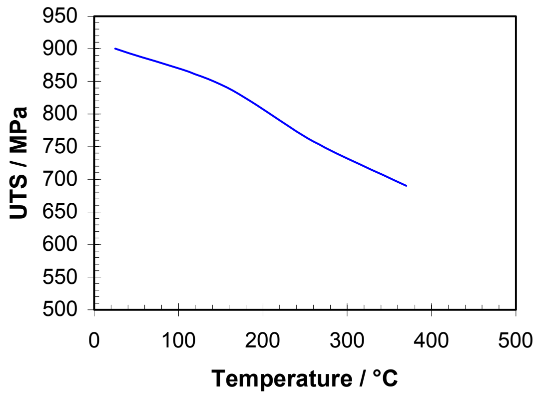
**Mechanical properties at room temperature**

| | Minimum values | Typical values |
|---|---|---|
| Yield Strength | 825 MPa | 910 MPa |
| Ultimate Strength | 895 MPa | 1000 MPa |
| Elongation in 50 mm, A5 | 10 % | 18 % |
| Reduction in Area | 20 % | % |
| Hardness | | 330-390 HV |
| Modulus of elasticity | | 114 GPa |
| Charpy V-Notch Impact | | 20-27 J |

**Yield strength vs. temperature**




**Tensile strength vs. temperature**




**Fatigue properties at room temperature** (Stress to cause failure in $10^7$Cycles)

| Rotating bend | | | Direct stress limit | | |
|---|---|---|---|---|---|
| Smooth | $K_t$=1 | 430-520 MPa | Smooth | $K_t$=1 | 376 MPa |
| Notch | $K_t$=3 | MPa | Notch | $K_t$=3 | 270 MPa |

| Physical properties | | |
|---|---|---|
| Melting point, ± 15 °C | 1650 | °C |
| Density | 4.43 | g/cm3 |
| Beta transus, ± 15 °C | 995 | °C |
| Thermal expansion, 20 - 100 °C | 9.0 | $*10^{-6}$ $K^{-1}$ |
| Thermal expansion, 0 - 300 °C | 9.5 | $*10^{-6}$ $K^{-1}$ |
| Thermal conductivity, room temperature | 6.6 | W/mK |
| Thermal conductivity, 400 °C | 13 | W/mK |
| Specific heat, room temperature | 0.57 | J/gK |
| Specific heat, 400 °C | 0.65 | J/gK |
| Electrical resistivity, room temperature | 171 | µW*cm |
| Poisson's ratio | 0.30-0.33 | |

| Heat treating | Temperature | Time |
|---|---|---|
| Solution treating temperature | 950-970°C | 1hour |
| Ageing temperature | 480-595°C | 4-8hours |
| Annealing | 710-790°C | 1-4hours |
| Stress relieving | 480-650°C | 1-4hours |

**Weldability – good**
Since the two-phase microstructure of alpha-beta titanium alloys responds to thermal treatment, the temperatures encountered during the welding cycle can affect the material being welded.

**Available mill products**
Bar, billet, extrusions, plate, sheet, strip, wire

**Typical Applications**
Compressor blades, discs and rings for jet engineers, aircraft components, pressure vessels, rocket engine cases, offshore pressure vessels.

| **Industry specifications** | **ASTM Grade5, ST-Al40, AMS4911D, MIL-T-9047G** |
|---|---|
| Sheet and plate | ASTM B265 Gr5, AMS 4911 |
| Bars and billets | ASTM B348 Gr5 |
| Bars, billets and forging (+circular forging) | AMS 4928, AMS 4965, AMS 4967 |
| Extruded products | AMS 4935 |
| Castings | ASTM B367 Gr5 |

# TITANIUM Grade 19

Beta-C

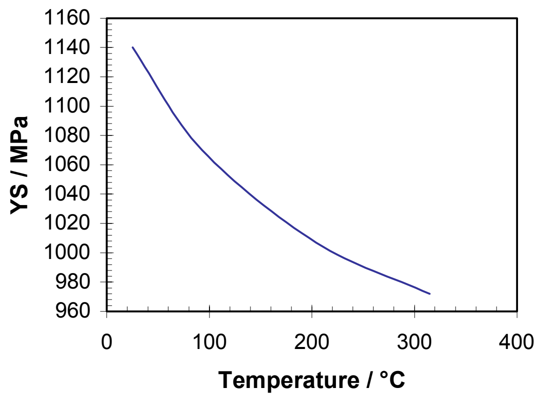
The data given is for information not for design

| Chemical composition (weight %) (Maximum values unless range is shown) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| O | N | C | H | Fe | Al | V | Cr | Mo | Zr | Residuals |
| 0.12 | 0.03 | 0.05 | 0.015 | 0.3 | 3.0-4.0 | 7.5-8.5 | 5.5-6.5 | 3.5-4.5 | 3.5-4.5 | 0.4 |

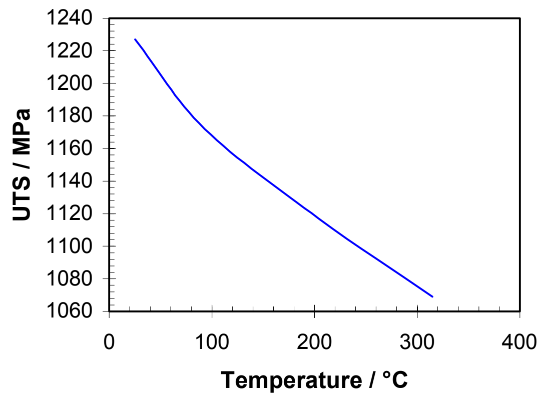
This metastable-beta alloy is strip-producible and cold-formable. The alloy is age-hardenable to a wide range of strengths. Beta-C is also being evaluated as extruded tubular for deep sour well applications. The alloy is superior resistant to general Corrosion pitting-, crevice- and stress Corrosion cracking in high temperature environments containing $FeCl_3$, NaCl, $CO_2$ and $H_2S$.

| Mechanical properties at room temperature | Minimum values | Typical values |
|---|---|---|
| Yield Strength | 1105 MPa | 1150 MPa |
| Ultimate Strength | 1170 MPa | 1250 MPa |
| Elongation in 50 mm, A5 | 6 % | 9 % |
| Reduction in Area | % | 30 % |
| Hardness | | 360-420 HV |
| Modulus of elasticity | | 102 GPa |
| Charpy V-Notch Impact | | 11-16 J |

**Yield strength vs. temperature**




**Tensile strength vs. temperature**




| **Fatigue properties at room temperature** (Stress to cause failure in $10^7$ Cycles) | Direct stress fatigue limit | | | |
|---|---|---|---|---|
| | Smooth | $K_t$=1 | 600 | MPa |
| | Notch | $K_t$=3 | 275 | MPa |

| Physical properties | | |
|---|---|---|
| Melting point, ± 15 °C | 1650 | °C |
| Density | 4.82 | g/cm3 |
| Beta transus, ± 15 °C | 730 | °C |
| Thermal expansion, 20 - 100 °C | 8.3 | $*10^{-6}$ $K^{-1}$ |
| Thermal expansion, 0 - 300 °C | 9.5 | $*10^{-6}$ $K^{-1}$ |
| Thermal conductivity, room temperature | 6.2 | W/mK |
| Thermal conductivity, 400 °C | | W/mK |
| Specific heat, room temperature | 0.52 | J/gK |
| Specific heat, 400 °C | | J/gK |
| Electrical resistivity, room temperature | 160 | µW*cm |
| Poisson's ratio | 0.34 | |

| Heat treating | | Temperature | Time | |
|---|---|---|---|---|
| Annealing | air-cooled/water | 705-760°C | 6min - | 30min |
| Stress relieving | air-cooled | 705-760°C | 6min - | 30min |
| Solution treating | water quench | 815-925°C | 1hour | |
| Aging | air-cooled | 455-540°C | 8hours - | 24hours |

**Weldability – fair**
Beta - C is weldable in annealed or solution treated Condition. The weld has often low strength and good ductility. To gain fully strength in metastable alloys it is important to weld the alloy in annealed Condition. The welds have to be cold worked (sandblasting, hammering) after welding. At the end the welds goes through solution treatment and aging. This treatment usually gives the weld a satisfactory ductility.

**Available mill products**
Billet, billet, plate, sheet, ingot, wire, pipe, forging

**Typical Applications**
Heavy sections where deep hardening and high strength with good fracture toughness are required. The alloy has been used for springs, fasteners, torsion bars, sheet, rivets and foil applications. Tubing and casing down hole production equipment.

**Industry specifications**

| | |
|---|---|
| Sheet and plate | MIL-T-909046 B-3 |
| Bar, billett and forging blank | MIL-T-9047 |
| | ASM 4957 |
| | ASM 4958 |

# TITANIUM Grade 21

Beta - 21S

The data given is for information not for design

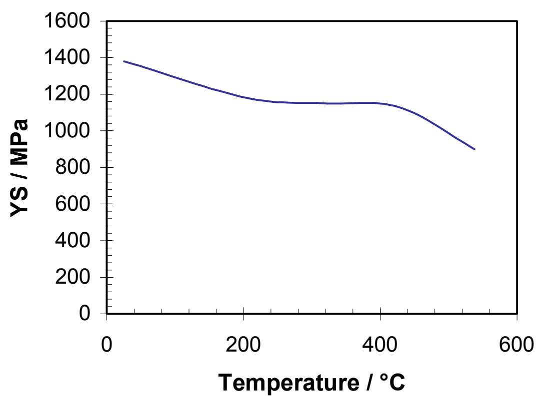
| **Chemical composition (weight %)** (Maximum values unless range is shown) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| O | N | C | H | Fe | Mo | Nb | Al | Si | Others | Residuals |
| 0.15 | 0.05 | 0.05 | 0.015 | 0.4 | 14.0-16.0 | 2.2-3.2 | 2.3 | 3.5 | | 0.4 |

Beta - 21S is a metastable beta alloy that offers high specific strength and good formability, and has been designed for improved oxidation resistance, elevated temperature strength, creep resistance and thermal stability. It is most useful for application above 300 ºC. Because it can be economically rolled to foil and is compatible with most fibres, it is also well suited for metal matrix composites.

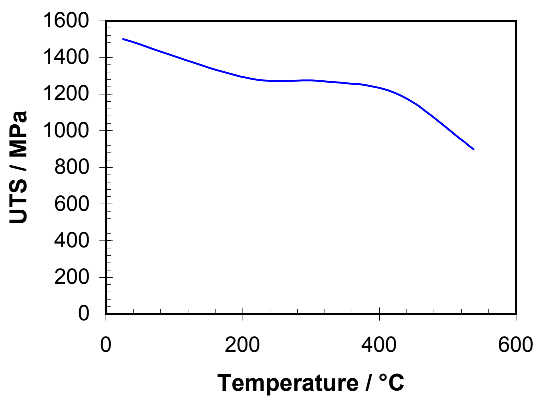
| **Mechanical properties at room temperature*** | **Minimum values** | **Typical values** |
|---|---|---|
| Yield Strength | 965 MPa | 1100 MPa |
| Ultimate Strength | 1030 MPa | 1150 MPa |
| Elongation in 50 mm, A5 | 6 % | 10 % |
| Reduction in Area | % | % |
| Hardness | | 360-420 HV |
| Modulus of elasticity | | 72-85 GPa |
| Charpy V-Notch Impact | | 103-110 J |

* Aged at 600 °C

**Yield strength vs. temperature**




**Tensile strength vs. temperature**



| Physical properties | | |
|---|---|---|
| Melting point, ± 15 °C | | °C |
| Density | 4.9 | g/cm3 |
| Beta transus, ± 15 °C | 815 | °C |
| Thermal expansion, 30 °C | 7.1 | $*10^{-6}$ $K^{-1}$ |
| Thermal expansion, 200 °C | 7.9 | $*10^{-6}$ $K^{-1}$ |
| Thermal conductivity, room temperature | 7.6 | W/mK |
| Thermal conductivity, 400 °C | 16.9 | W/mK |
| Specific heat, room temperature | 0.5 | J/gK |
| Specific heat, 400 °C | 0.6 | J/gK |
| Electrical resistivity, room temperature | 135 | µW*cm |
| Poisson's ratio | 0.34 | |

| Heat treating | | Temperature | Time |
|---|---|---|---|
| Annealing | air-cooled | 815-845 °C | 3-30 min |
| Stress relieving | air-cooled | 510-650 °C | 8-16 hours |

**Weldability – fair**

**Available mill products**
Forging, tube

**Typical Applications**
Warm airframes of engine structures, honeycomb, fasteners, metal matrix composites. TIMETAL 21S is useful for applications from 230°C to 600°C. The alloy is resistant to aircraft hydraulic fluids. It is well suited for metal matrix composites because it can be economically rolled to foil, is compatible with most fibers, and is sufficiently stable up to 820°C.

**Industry specifications** TIMETAL 21S