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AOGV: Implementation of Hydraulics to Relieve Pressure Affected Intervention Tool

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Figure 1: Izomax Logo

Figure 2: UiS Logo

Abstract

In the following bachelor thesis options for simplification of the pressure affected intervention tool are examined. Considering industry standards and ease of use, a design of the Isolation Fasting Tool for Hydraulics is made to fit the 06” AOGV. For 3D design, technical drawings Autodesk Inventor is used, while simulations are done with Ansys Mechanical.

The thesis is separated in two branches Hydraulic and Mechanical Design. By utilizing the broad applications of hydraulics, the insertion prosses of the intervention tool is simplified. The prosses is automized and without the need for excessive manual labor.

An Isolating Cylinder Fasting Tool is designed to give two key qualities to the system. Firstly, it isolates the pressure inside the system to give the hydraulic cylinders air pressure at the head. Secondly, radial forces are eliminated to ensure that the cylinders work with linear forces.

By utilizing the hydraulic system to insert the pressure affected intervention tool the prosses needs less manual work, is more automized and reduces the risk for excessive force on the inside of the AOGV when the Isolation Spade is fully inserted.

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Nomenclature

AOGV:	Add On Gate Valve
CFT:	Cylinder Fasting Tool
HC:	Hydraulic Cylinder
TEL:	Top Envelope Lid
IT:	Isolating Tool

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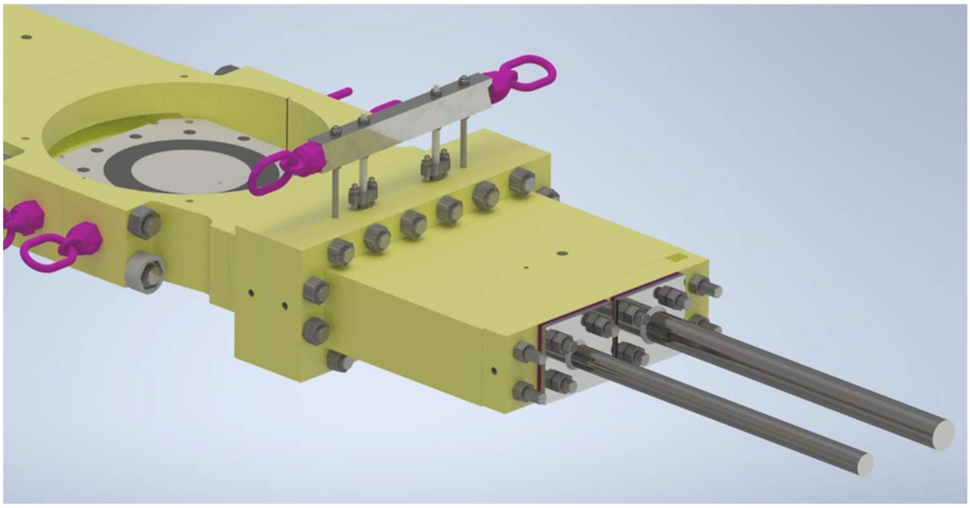


Figure 3: Assembly Render [1]

IK Group and Izomax

IK Group was founded in 1987. The company found its place in the energy industry in the early 2000s. In 2006 a management buyout shaped IK-group greatly. Since then, IK Group have undergrown massive growth. In 2011 IK-UK was developed for supplying the UK marked. Likewise, IK-Saudi was developed in 2013. In 2017 Sales offices were opened in Huston, Singapore, and Dubai. IK-Group is today an international company based in Stavanger, with employees in 8 countries. IK consists of engineers, mechanics, administration, and sales, totaling 215 employees.

IK-Group delivers solution in four business areas.

AOGV - Izomax	Pipeline services	Subsea	Topside Services

The AOGV department has grown massively and is from the 1-st of January it is a separate company owned by IK-Group. The company deliver tested and proven technology to a variety of costumers. Fluid cooperation between the engineers and mechanics is therefore a key business strategy. A general operation starts with documentation of the specific site. Then the engineering team find the ideal AOGV for the case. The next stage is mechanical testing with realistic parameters. After validation a team delivers the solution with the tested AOGV.

The AOGV concept was developed in 2016. Since than the technology has reached multiple milestones. In 2017 the first AOGV prototype was used. In 2018 the first high pressure AOGV was successfully executed (150 bar). As of 2019 the technology was patented. Despite of the success, the Izomax research and development team is striving for new solutions to improve the general design of the AOGV. Izomax is a company with a focus on development for the employees. The engineering culture facilitates effective innovation, where ease of use and cost are essential focus points.

Figure 4: IK Group/Izomax location

Motivation

The AOGV is an important step forward in flange tool technology. The patented tool is used for turning any flange pair into a valve. In practice it means that specific parts of a process plant are replaced/changed while the global plant is active. The economic and environmental benefits of this makes the AOGV a sought-after technology that contributes the carbon-neutral transition.

AOGV

AOGV- Add On Gate Valve technology install and uninstall an isolation spade on any active flange pair. This creates a zero-energy zone where maintenance, modification, and inspection work are carried out safely and efficiently, while production continues to run.

As described in Figure 5 which is a crop out of (G), the prosses of installing the AOGV starts by mounting flanges on either side of the flange pair. Av envelope structure that is pressure isolated is added to the system. The flanges are split, and pressure is uniformly spread in the AOGV gasket. An Isolation Spade is pushed by rods in location to seal the flange end. Finally, the AOGV is removed leaving a closed flange.

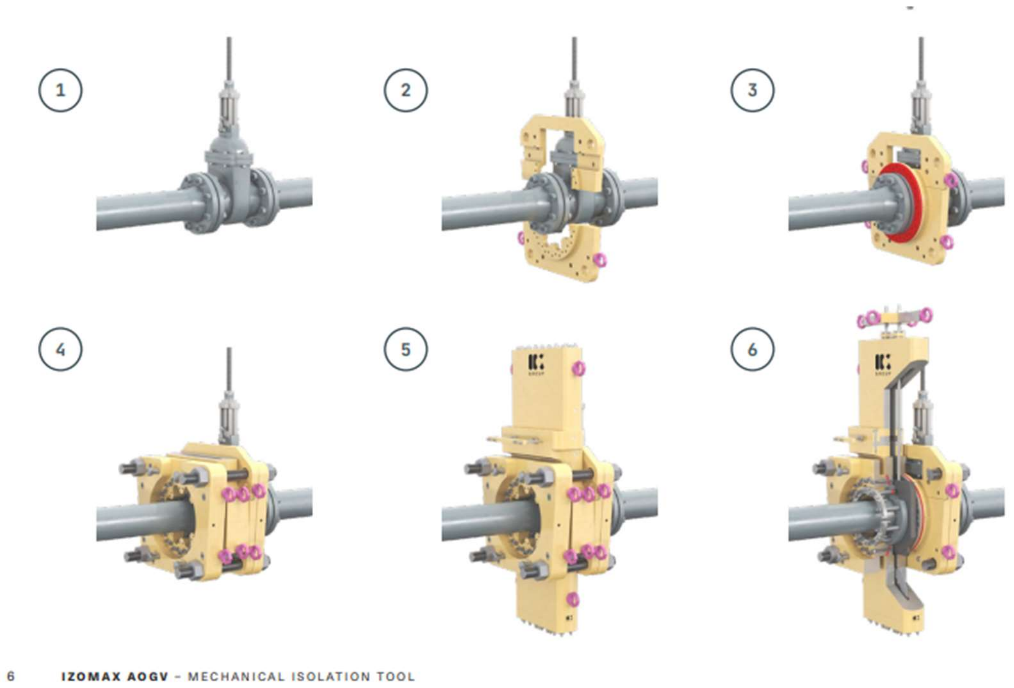


Figure 5: AOGV Description [2]

General Objective

Isolation Spade is pushed by rods inside a pressure chamber to fitting position. Forces on the rods necessary to overcome the internal pressures are in many cases larger than one can do by directly by hand. Methods for dealing with this include chains and trolleys, as seen in Figure 6.

In this thesis the general objective is to design a system using hydraulics that is implemented to the AOGV. A tool used to optimize the utilization of hydraulic cylinders, on the rods used for inserting the Isolation Spade are designed. In addition, a concept hydraulic system for AOGV use is discussed.

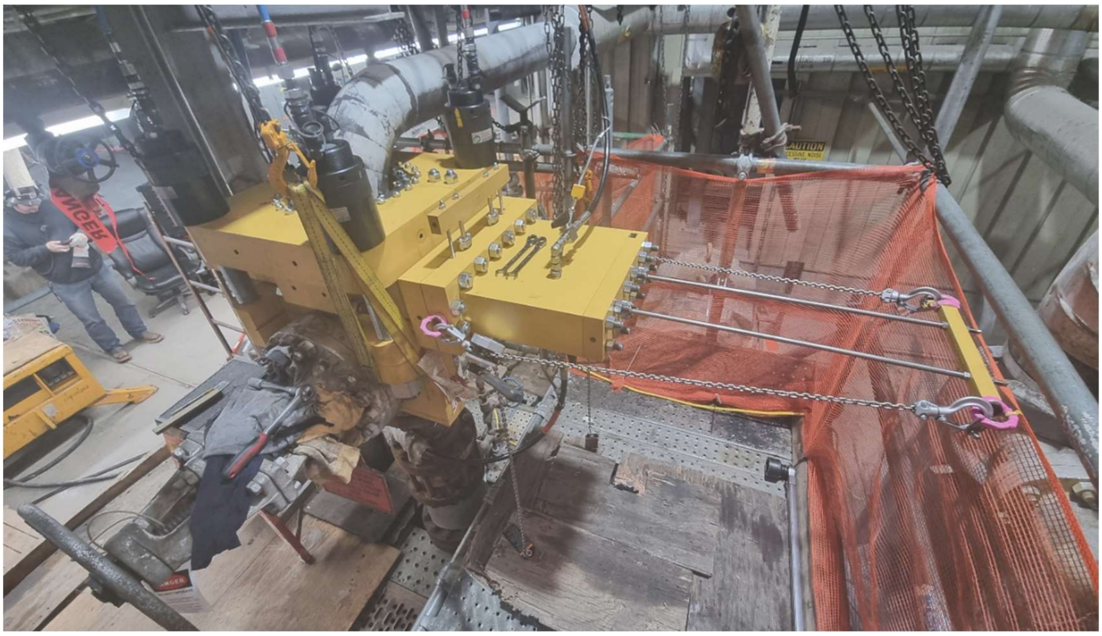


Figure 6: AOGV Under Operation [3]

1. Parameters

1.1.1 The 6” AOGV

Izomax deliver a AOGV lineup ranging from 1” to 36”. To simplify the thesis, the 6” AOGV is chosen as the design parameter model. An assembly drawing of the 6” model is in (H). Part number 1 (Envelope Assembly), and part number 5 (Spade Assembly) are directly relevant to the general objective of the thesis.

1.1.2 Envelope Assembly

In the assembly drawing for Envelope Assembly (I), part 1 is the TEL - Top Envelope Lid. Part 2 (Stuffing box) is fastened to the TEL and is in direct contact with the rods.

1.1.3 Top Envelope Lid

TEL drawing is shown in (J), and a render of the Stuffing box socket is in Figure 7. Steel type P355NL2 used for the TEL. Structural qualities of P355NL2 are described in the Table 1.

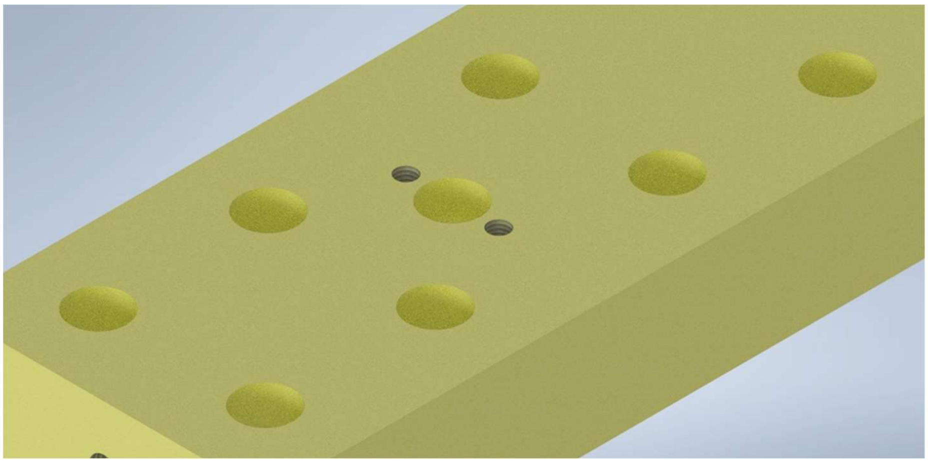


Figure 7: Render of Top Envelope Lid [4]

Table 1: Mechanical properties of P355NL2

Nominal Thickness (mm)	To 60	60 - 100	100 - 150
Rm – Tensile Strength (MPa)	490 - 630	470 - 610	460 - 600
Nominal Thickness (mm)	To 16	16 - 40	40 - 60
ReH – Minimum Yield Strength (MPa)	355	345	335

1.1.4 Stuffing Box

The Stuffing box is the part of the AOGV where the rods used to insert the isolation spade enter the system. The Stuffing Box assembly is shown in (K), and a cross section of the part is shown in Figure 8. They are connected to the AOGV by two bolts. The boxes have two critical functions AOGV. Firstly, they guide the rods on the isolation spade. The low contact friction IGUS contact baring backed by the metallic structure of the stuffing box aligns the rods at the entry point of the AOGV.

Secondly, it counteracts the internal pressure in the AOGV. NBR material has great isolating properties isolating for air, nitrogen and most relevant gasses that may be present in an application. The NBR seal is fitted on the stuffing box working on the rods. By utilizing bolts, the seal is pushed together between the bearing and the metal roof energizing the seal. The correct tightening level of the bolts are different between applications and is done by inspection. It is tightened to isolate the system, while giving the least amount of friction.

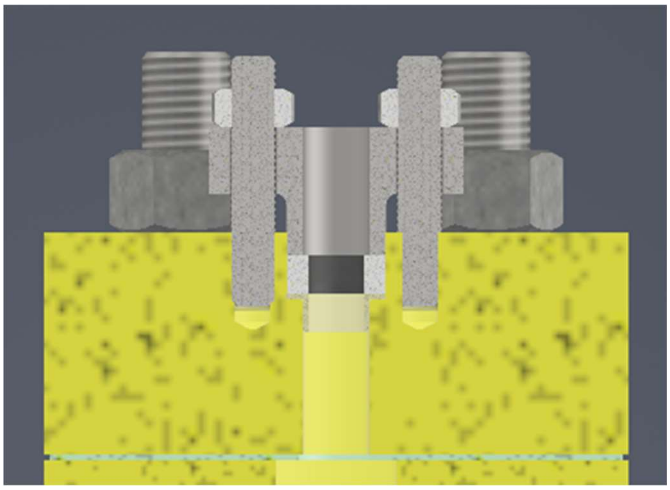


Figure 8: Stuffing Box when Mounted [5]

1.1.5 Spade Assembly

The spade isolation tool is shown in drawing (L). Part 1 is the spade. The spade is fitted between the flange pair under applications sealing the flange. When the spade is fitted between the flanges, bolts are threaded though the flanges and spade creating an isolating seal. The AOGV can then be removed.

The rods are seen as part 2 in the picture are fastened between the spade and the handle. The spade is moved on the inside of the AOGV by exerting force on the handle on the outside of the isolated system. The handles are fitted with rings (part 6) that are utilized for chains when trolleys are used for insertion.

1.1.6 AOGV Measurements

Figure 9 describes the length between the center of the flange and the center point of the Isolation Spade at initial position. Figure 10 shows the length between the center of the initial position of the Isolation Spade and the top surface of the TEL. Lastly, Figure 11 show the thickness of the TEL.

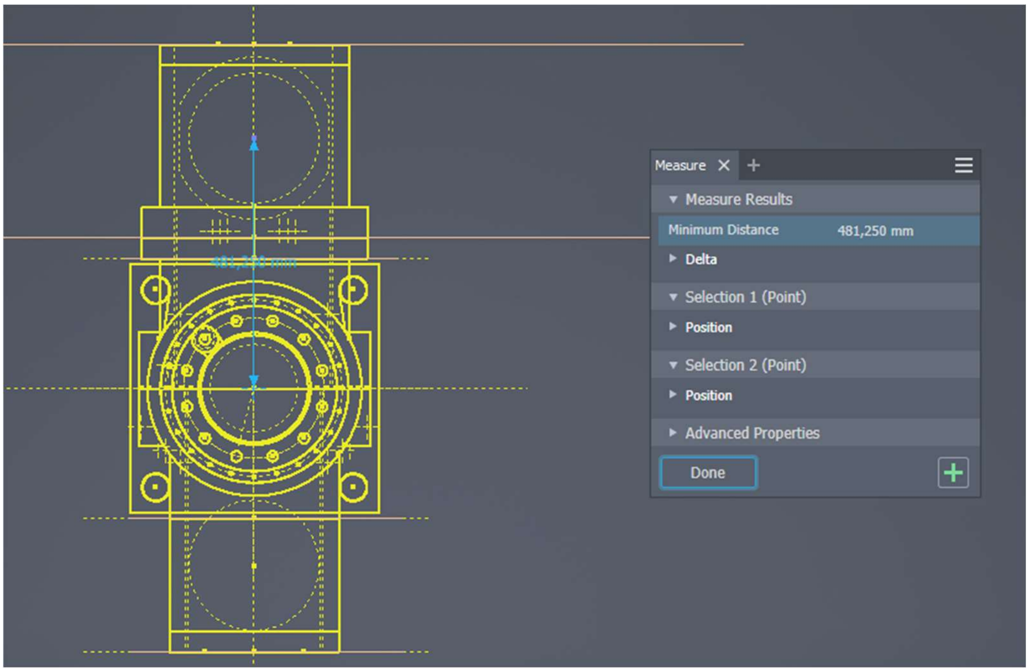


Figure 9: Measurement of Distance Between Initial and Final Center Points [6]

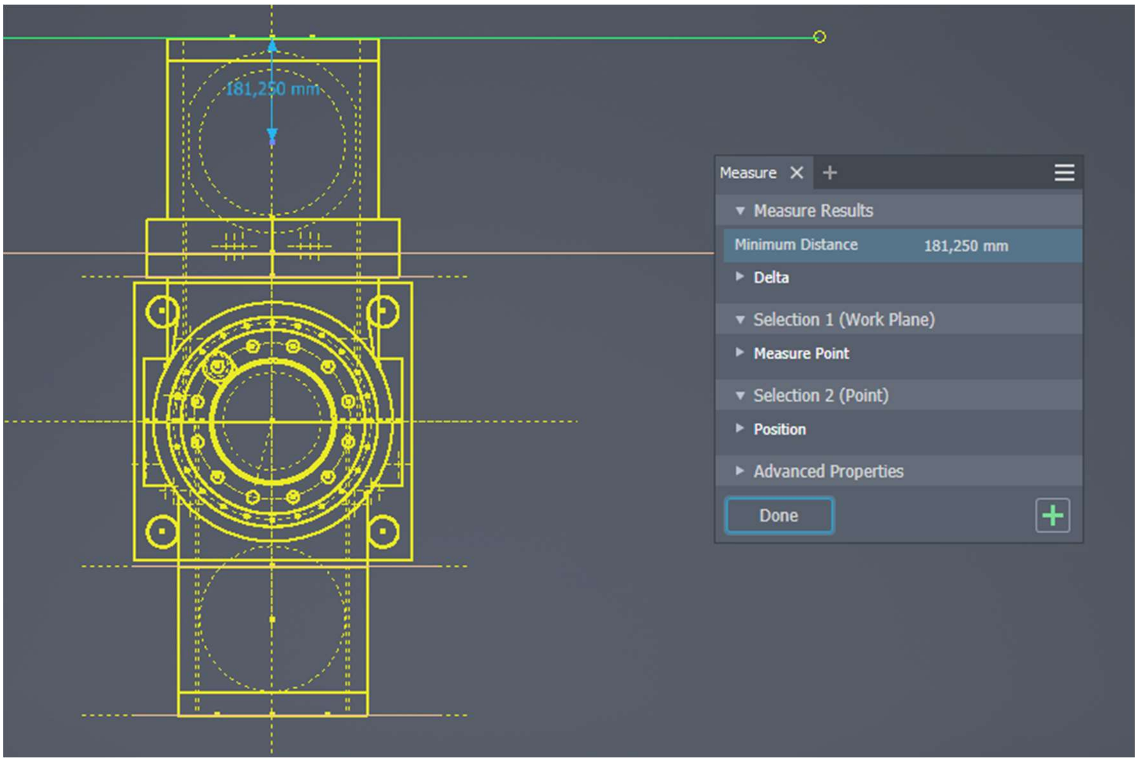


Figure 10: Measurement of Distance Between Top of TEL and Initial Center Point [7]

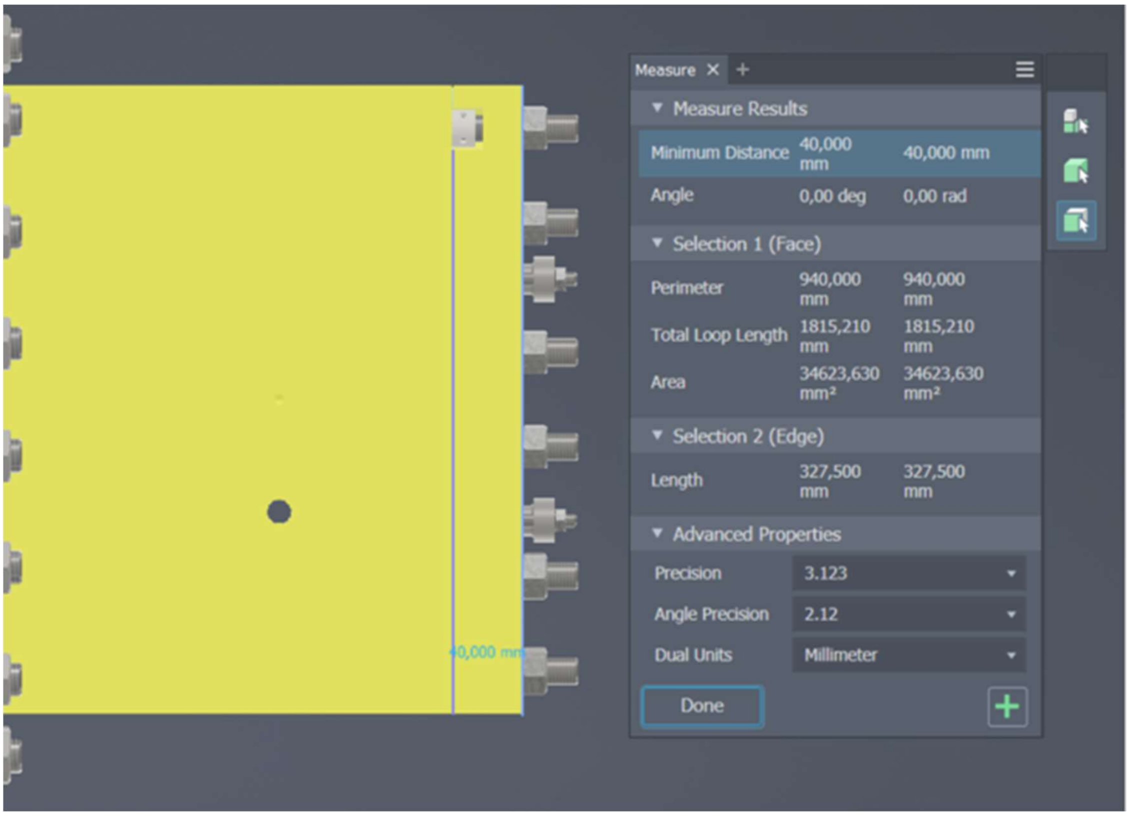


Figure 11: Measurement of Thickness of the TEL [8]

1.1.7 Rods

Rod diameters is 10 mm as shown in the drawing (M). Material properties of S165, which the rods are made of is shown in Table 2.

Table 2: Mechanical Properties of S165 Steel

Condition	**Rp0.2** – Yield Strength (MPa)	**Rm** – Tensile Strength (MPa)
Typical	720 - 850	950 - 1050

1.1.8 AOGV Pressure

The AOGV isolates a flange pair opening and maintains the internal pressure in the pipes. Applications have variating pressures depending on the application. The standard test pressure for the 06-inch AOGV is 5 MPa and the test pressure is 7.75

$$Design\ pressure\ (max\ allowable) = 5MPa$$
$$Test\ Pressure = 7.75\ MPa$$

1.1.9 Bolts and Pretension

The nut and bolt size used for fasting the TEL to the AOGV body is M16. Existing designs are used with a bolt pretension of 45 kN. Figure 12 is a crop-out of the AOGV assembly with bolts and nuts.

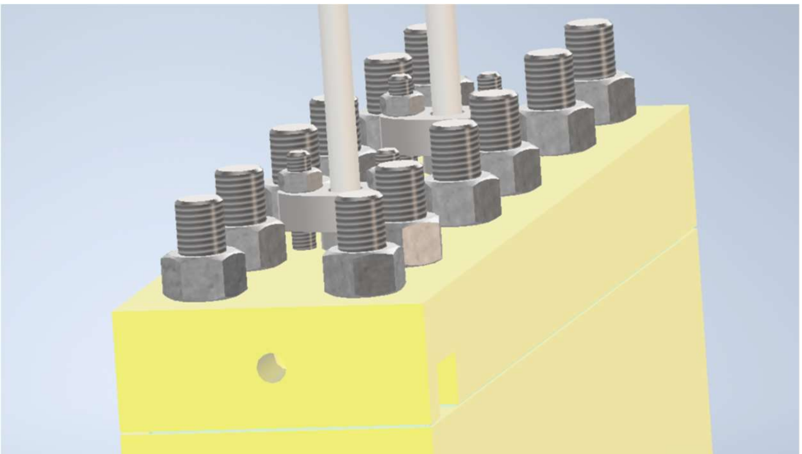


Figure 12: AOGV Assembly [9]

2. Theory

2.1 Mechanical Theory

2.1.1 Pressure Over Area

The pressure P over an area A equals the force F on a flat surface with uniform pressure *[10, s. 9]*.

$$F = P * A \qquad (1)$$

2.1.2 Moment

The moment M equals the force F times the distance d between force (for example centre of mass of a gravitational object) and the reference point *[11, s.112]*.

$$M = F * d \qquad (2)$$

2.1.3 Buckling

Slenderness of the rod γ, where L_k is the kink length and i is the radius of gyration. I is the moment of inertia, and A is cross-sectional area. *[12, s.33]*

$$\lambda = \frac{L_k}{i} \qquad (3)$$

were

$$i = \sqrt{\frac{I}{A}} \qquad (4)$$

The critical slenderness γ_1 *[13, s.34]* is derived by the E-modulus E *[14, s.6]* and Yield Strength R_e.

$$\gamma_1 = \sqrt{\frac{2\pi^2 E}{R_e}} \qquad (5)$$

were

$$E = 210\ 000 MPa$$

The kink length derived from the type of constraints and total length of rod is shown in Figure 13. *[15, s.33]*

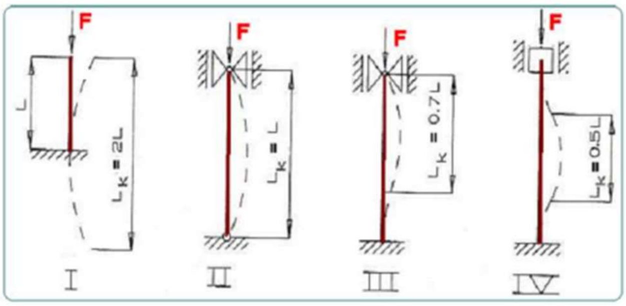


Figure 13: Types of Constrains

Figure 14 is a graphic visualisation of the implications of the deference between slenderness and critical slenderness. *[16, s.34]*

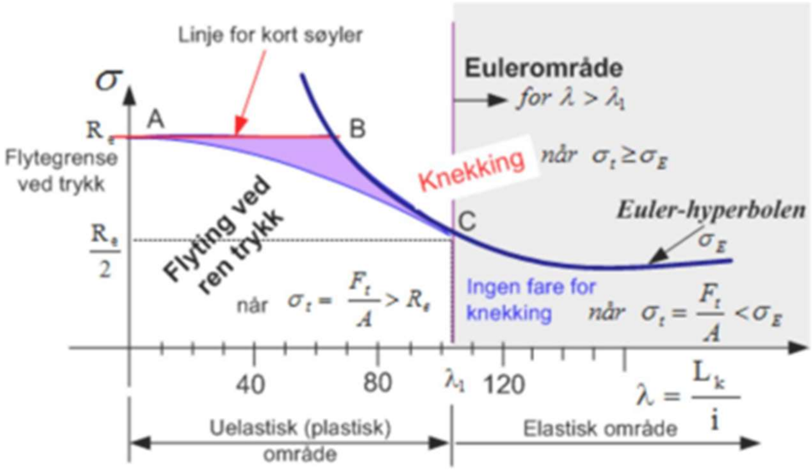


Figure 14: Slenderness and Critical Slenderness

Formela for no danger for buckling in the elastic area. *[17, s.34]*

$$\sigma_t = \frac{F_t}{A} < \sigma_e \qquad (6)$$

With the Euler stress

$$\sigma_e = \frac{\pi^2 E}{\gamma^2} \qquad (7)$$

2.1.4 Normal Stress from Moment

The normal stress by moment is found by formula 9. *[18, s.14, 15]*

$$\sigma_b = \frac{M}{w_b} \quad (8)$$

With the cross-sectional modulus

$$w_b = \frac{I}{c} \quad (9)$$

Were

$$c = Distance\ from\ nutrual\ axis\ to\ plane\ of\ element$$

$$I = \frac{\pi d^4}{64}\ , for\ circular\ cross\ section$$

2.1.5 Factor of Safety

The factor of safety against fracture under bending. [19, s.12]

$$n_f = 1.0 - 1.8$$

2.1.6 Friction Constants for Various Materials

Table 3: Frictional Constants for Various Materials [20, s. 116].

Material	Static Friction	Kinetic Friction
Steel on Steel	0.74	0.57
Copper on steel	0.53	0.36

2.1.7 Hole Clearance

Cut out of Descriptions of Preferred Fits Using the Basic Hole System is shown in Table 4. *[21, s.408]*

Table 4: Types of Hole Clearances

Type of Fit	Description	Symbol
Clearance	Loose running fit: wide commercial tolerances or allowances on external members.	H11/c11
	Free Running fit: Not or use where accuracy is essential, but good for large temperature variations, high running speeds, or heavy journal presses.	H9/d9
	Close Running fit: for running accurate machines and for accurate location at moderate speeds and journal presses.	H8/f7
	Slit fit: where parts are not intended to run freely but must move and turn freely and locate accurately.	H7/g6
	Location clearance fit: provides snug fit for location of stationary parts but can be freely assembled and disassembled.	H7/h6

Formula for Diameter of Hole and Shaft with Clarence fits c, d, f, g, h. *[22, s.409]*

$$D_{max} = D + \Delta D,\ \ D_{min} = D \qquad (10)$$

$$d_{max} = d + \ \delta_F,\ \ d_{min} = d + \ \delta_F - \Delta\text{d} \qquad (11)$$

- D = basic size of hole
- d = basic size of shaft
- δ_u = upper deviation
- δ_l = lower deviation
- δ_F = fundamental deviation
- ΔD = tolerance grade for hole
- Δd = tolerance grade for shaft

Table 5: International Tolerance Grades [23, s. 1038]

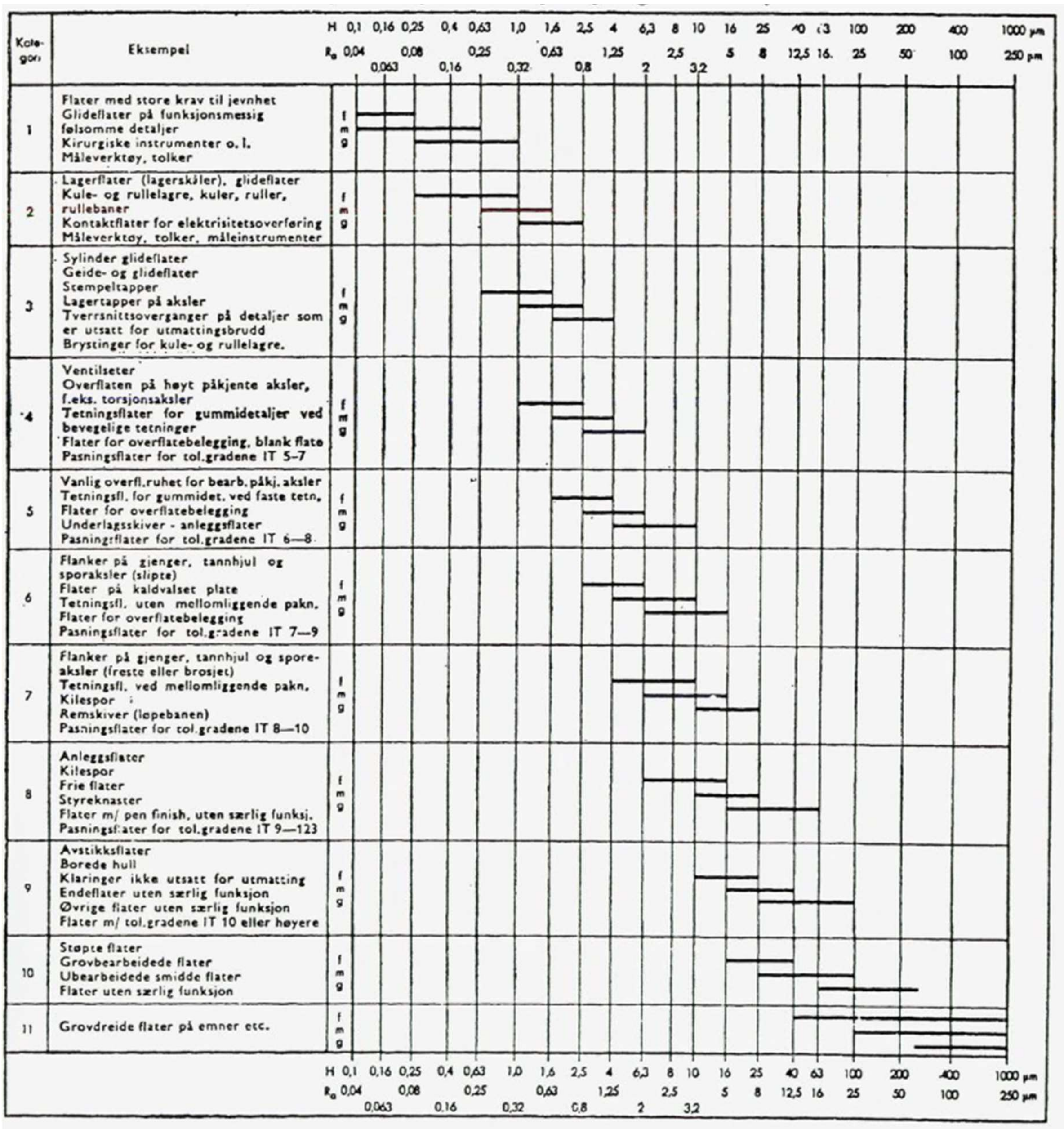
Basic Sizes	Tolerance Grades			
	IT6	IT7	IT8	IT11
10-18	0.011	0.018	0.027	0.110
18-30	0.013	0.021	0.033	0.130
30-50	0.016	0.025	0.039	0.160

Table 6: Fundamental Deviation of Shafts [24, s. 1039]

Basic Sizes	Upper-Deviation Letter			Lower-Deviation Letter
	c	g	h	k
10-14	-0.095	-0.006	0	+0.001
18-24	-0.110	-0.007	0	+0.002
24-30	-0.110	-0.009	0	+0.002
30-40	-0.120	-0.009	0	+0.002

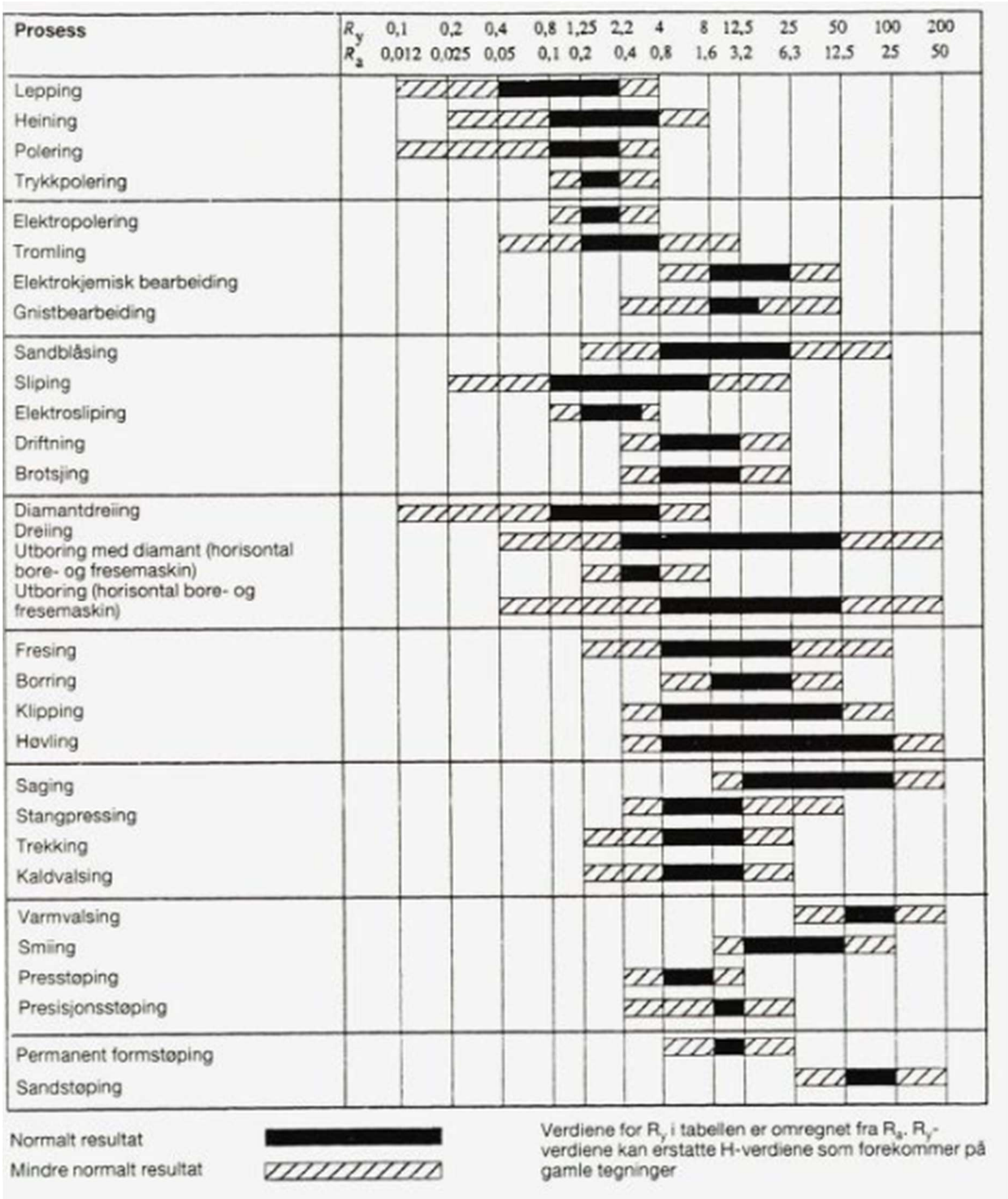
2.1.8 Surface Roughness

Table 7: Examples of suggested surface roughness for various constructions [25, s. 216]



Kategori	Eksempel	
1	Flater med store krav til jevnhet Glideflater på funksjonsmessig følsomme detaljer Kirurgiske instrumenter o. l. Måleverktøy, tolker	f m g
2	Lagerflater (lagerskåler), glideflater Kule- og rullelagre, kuler, ruller, rullebaner Kontaktflater for elektrisitetsoverføring Måleverktøy, tolker, måleinstrumenter	f m g
3	Sylinder glideflater Geide- og glideflater Stempeltapper Lagertapper på aksler Tverrsnittsoverganger på detaljer som er utsatt for utmattingsbrudd Brystinger for kule- og rullelagre.	f m g
4	Ventilseter Overflaten på høyt påkjente aksler, f.eks. torsjonsaksler Tetningsflater for gummidetaljer ved bevegelige tetninger Flater for overflatebelegging, blank flate Pasningsflater for tol.gradene IT 5–7	f m g
5	Vanlig overfl.ruhet for bearb.påkj.aksler Tetningsfl. for gummidet. ved faste tetn. Flater for overflatebelegging Underlagsskiver - anleggsflater Pasningsflater for tol.gradene IT 6—8	f m g
6	Flanker på gjenger, tannhjul og sporaksler (slipte) Flater på kaldvalset plate Tetningsfl. uten mellomliggende pakn. Flater for overflatebelegging Pasningsflater for tol.gradene IT 7—9	f m g
7	Flanker på gjenger, tannhjul og spore-aksler (freste eller brosjet) Tetningsfl. ved mellomliggende pakn. Kilespor Remskiver (løpebanen) Pasningsflater for tol.gradene IT 8—10	f m g
8	Anleggsflater Kilespor Frie flater Styreknaster Flater m/ pen finish, uten særlig funksj. Pasningsflater for tol.gradene IT 9—123	f m g
9	Avstikksflater Borede hull Klaringer ikke utsatt for utmatting Endeflater uten særlig funksjon Øvrige flater uten særlig funksjon Flater m/ tol.gradene IT 10 eller høyere	f m g
10	Støpte flater Grovbearbeidede flater Ubearbeidede smidde flater Flater uten særlig funksjon	f m g
11	Grovdreide flater på emner etc.	f m g

Table 8: Surface Roughness from various manufacturing method. [26, s. 217]



2.1.9 O-Ring

O-rings are a cheap and simple sealing solution that resist the pressures described in Figure 15.

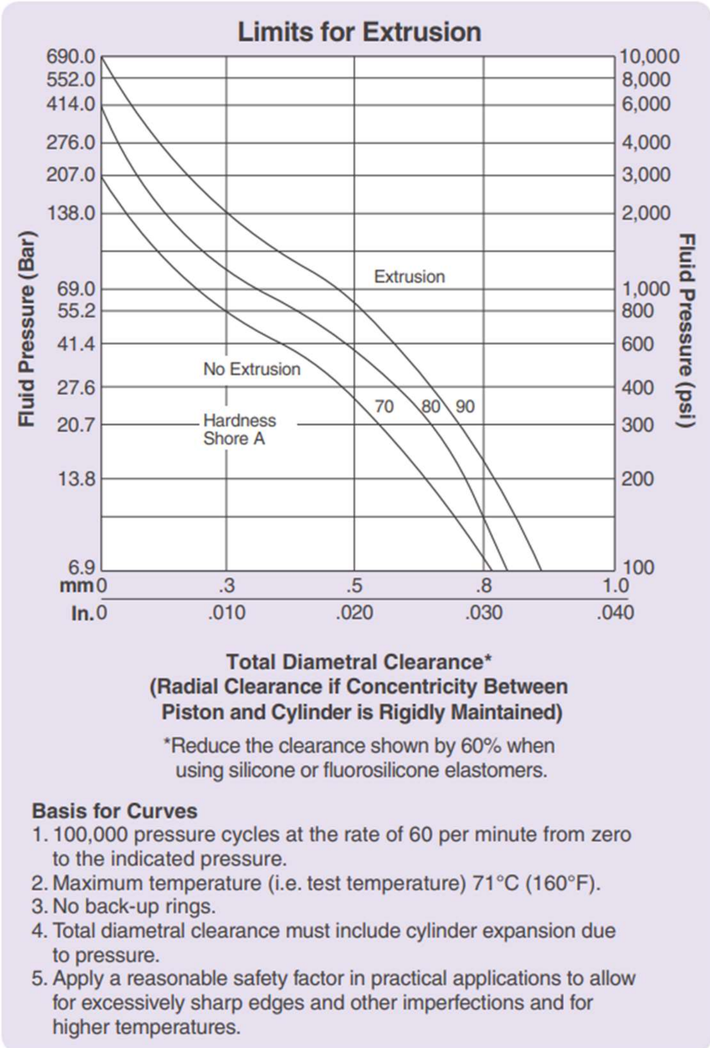


Figure 15: O-Ring Limits for Extrusion [27, 3-3]

2.1.10 The Trelleborg Quad Ring®

The Trelleborg Quad-Ring® (A) is a seal for dynamic high-pressure applications. It works well with rods with reciprocating motion, with a max speed of 0.5 m/s. The working pressure for reciprocating motion is 5MPa without a backup ring, and 30MPa with a backup ring. Two standard material types are presented in the Quad-Ring® catalog (B): NBR with a temperature range of -30°C to +100°C and FKM with a temperature range of -18°C to +200°C As seen in the Chemical Compatibility Guide both NBR and FKM materials work with nitrogen, as well as a multitude of other gasses.

Internal seals are mounted on the outer part of the cylinder and works on the rod. For an internal double backed Quad-Ring® on a 10 mm rod the groove dimensions are granted in Table 9.

Table 9: Quad ring Grove Dimensions

Dimensions	Grove	Grove Width	Radius
10.20x2.62	14.6	5.8	0.3

Figure 16: Trelleborg Quad Rings [28]

2.2 Hydraulic Theory

2.2.1 Characteristics of Hydraulics

The operational system is dependent on physical and tedious labour. By implementing hydraulics operations can be done more efficient. There are upsides and downsides to hydraulic systems. Downsides are the possibility of leakages and pressure loss, change of velocity because of temperature dependent viscosity of fluids, synchronisation of movements where marginal tolerances are accepTable. *[29, s 7 hydraulic]*

Positive attributes of mechanical systems are.

- Large forces form easily managed elements that need little maintenance.
- Regulations of speed are stepless. That gives the potential for fluid change in velocity.
- Regulations of power are stepless.
- High power from low area and mass.
- Automatic greasing.
- Safety valves reduces risk.

2.2.2 Problems of Radial and Bending Forces in a Hydraulic Cylinder

When used correctly hydraulic cylinders have a long lifetime. When used wrongly however the lifetime can be dramatically shortened. Side loading and rod bending are problems that can lead to lead to premature failure *[30].*

2.2.3 Flow Rate and Velocity

Flow rate Q and velocity V of a hydraulic cylinder, with cylinder area A *[31]*.

$$V = \frac{Q}{A*6} \qquad (12)$$

2.2.4 Hydraulic Flow Divider

The purpose of a flow divider is to separate the flow in a system. Generally, it is used for applications where actuators move in sync. There are two types of flow dividers. Gear flow dividers work by two gears releasing equal volumes of fluid. Being the more expensive option, the gear flow divider is the best performing option *[32]*.

The spool type work by pressure compensation. The spool on each flow side shifts to balance the pressure required to equalize the flow. Though lighter and cheaper than the gear flow divider, the spool type is more sensitive to pressure swings.

2.2.5 Directional Control Valves

The Directional Control Valve is used for controlling the flow in a hydraulic system. That way the direction of a cylinder can be controlled *[33, s 7 hydraulic].* Directional Control Valves are categorized with standard numeration. The first number in the combination gives the amount of exit points in the Directional Control Valve. In Figure 17 the amount of exit points is 4. Exit points 1 and 2 deliver flow in the opposite orientation of exit points 3 and 4 *[34]*.

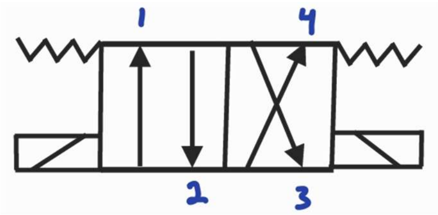
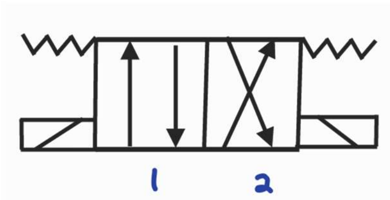


Figure 17: Direction Control Valve Schematic

The second number in the notation is the amount of flow paths provided in the valve. In Figure 18 which is a 4/2 Directional Control Valve there are two possible flow paths, giving the two options stated above. Figure 19, which is a schematic of a 4/3 valve also includes the neutral setting where no flow passes.



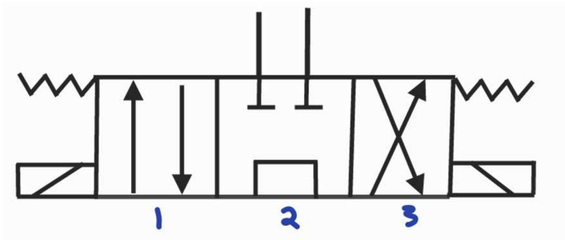


Figure 18: Constant Flow Schematic

Figure 19: Direction Control Valve with Neutral Position Schematic

Directional Control Valves can be pressure compensated. For non-compensated systems the flow from the valve fluctuates with varying pressure on the cylinder, which will have varying speed. This is because of variation in pressure at the exit of the DCV. Pressure compensated Directional control valves automatically regulates the flow for constant speed with a compensator spool, without altering the outlet pressure significantly *[35]*.

2.2.6 AdjusTable Pressure Relief Valve

The schematic of an adjusTable relief valve is shown in Figure 20 *[36, s 76]*. The squiggly line with an arrow describes a spring that is adjusted by compression. If the pressure in the dotted line overcomes the pressure limit of the spring the valve opens.

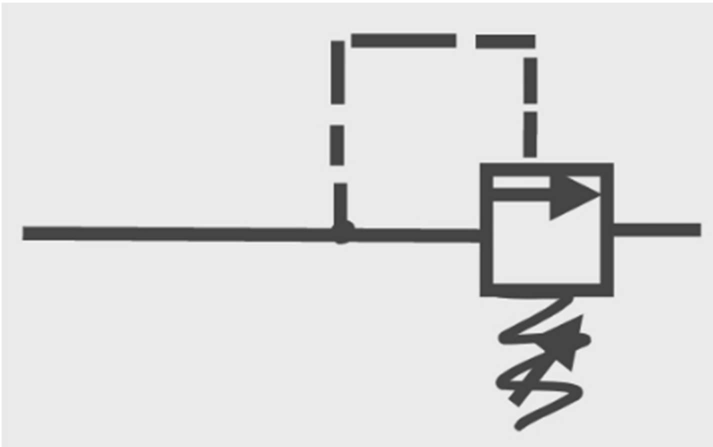


Figure 20: AdjusTable Pressure Relief Valve Schematic

2.2.7 Double Acting Hydraulic Cylinder

HC-s are shown as rectangles with a line through the centre describing the rod configuration *[37, s 64]*. Figure 21 shows a one-sided HC which means the rod exits the cylinder on one side. There are lines coming in from the sides on either side of the cylinder piston. This shows that the model is double acting, which means that the HC can be pushed by hydraulic fluid in either direction.

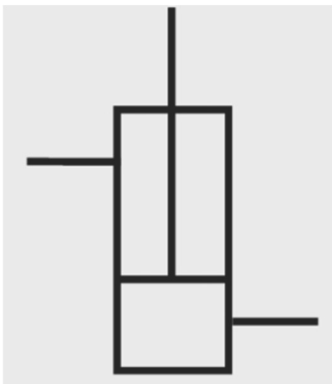


Figure 21: Hydraulic Cylinder Schematic

2.2.8 Motor and Pump

Hydraulic pumps convert mechanical energy to hydraulic energy. Figure 22 shows a schematic of a hydraulic pump system with three parts. Firstly, circle with a M is the motor that runs the pump. Secondly, the hydraulic pump is shown by the circle with the rectangle. Lastly, the open part on the bottom is the hydraulic oil source. *[38, s 63-64]*.

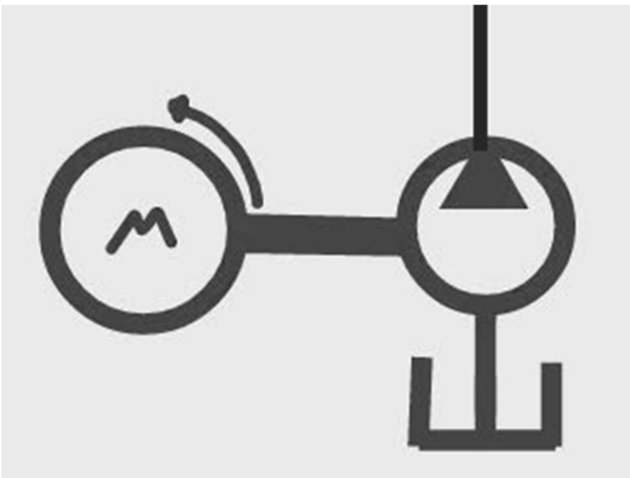


Figure 22: Motor and Pump Schematic

2.2.9 Gauge

Hydraulic gauges are used to measure and display the pressure of hydraulic fluid in a hydraulic system. Figure 23 is a schematic of a gauge is a circle with a crossing arrow *[39, s 65]*.

Figure 23: Gauge Schematic

3. Calculations and Design

3.1 Introduction Mechanical Tests

3.1.1 Mechanical Force-Pressure test

The goal of the mechanical test is to measure the forces necessary to insert an Isolation Spade in a 50 Bar pressurized AOGV. The test is done as an additional part of the regular AOGV testing conducted before an operation. The test is done on a 08" AOGV with rod diameter of 12 mm. The theoretical linear force is predicted at 1000N throughout the insertion prosses based on 3.2.14.

Before insertion the rods are marked with 50 mm intervals for which the spades are inserted (Figure 24). To measure the forces a weight is put in place on the chain between the top of the spade, and the middle of the AOGV (Figure 24). The pressure in the AOGV is at a constant 50 Bar. The internal volume in the system decreases as more of the rods enter the AOGV, so excess pressure is released continuously under testing.

Figure 24: Rod Markings [40]

Figure 25: Test Setup [41]

The chain comes in at the handle at an angle. The adjacent length is initially 975 mm, and the opposite length is 12.7 mm initially. The angle will then alter when the rods are inserted as the adjacent length changes. The angle of attack is negligible because the angles are minimal as shown in Figure 26.

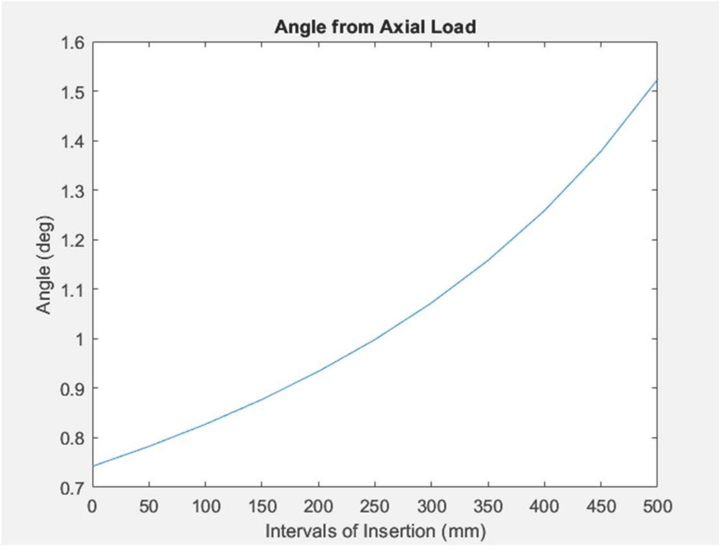
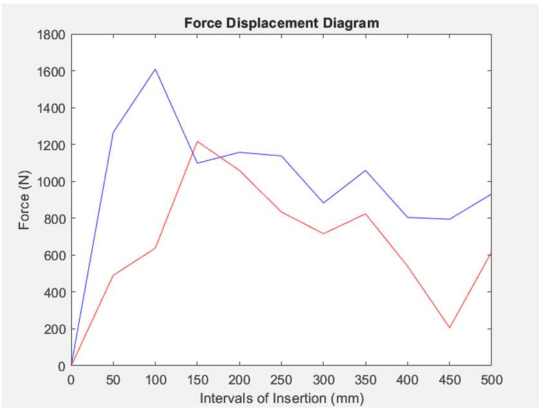


Figure 26: Angle of Attack

In Figure 27, the blue and red graphs the two first tests were done with a digital weight cell. The next two graphs from Figure 28 in with light blue and purple, are done with an analogue weight cell. Because of the geometry of the analogue weight cell the measurement of the fully extended rod was not measured. The initial force is not captured because there is no force applied to the chains initially.



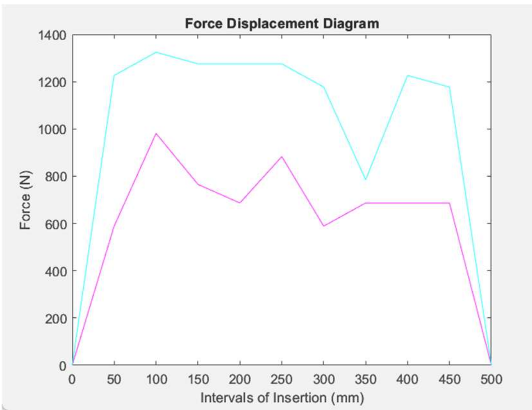
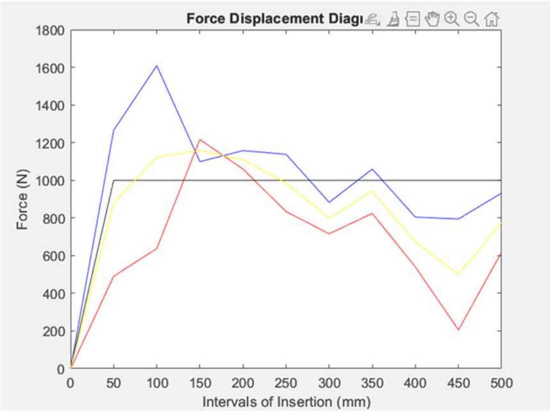


Figure 27: Force Displacement Diagram for Tests 1 and 2

Figure 28: Force Displacement Diagram for Test 3 and 4



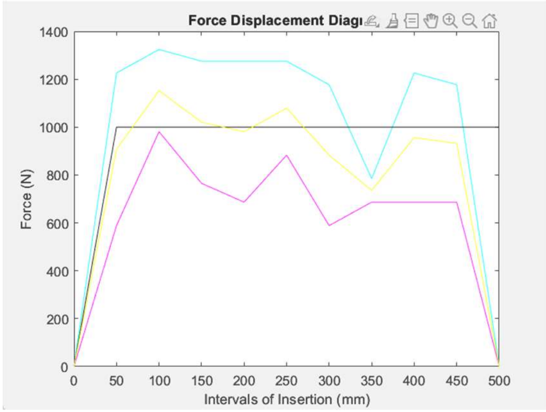


Figure 29: Test 1 and 2 with Theoretical and Average results

Figure 30: Test 3 and 4 with Theoretical and Average results

The black curves are the theoretical insertion force of 1000 N, and the yellow curves are the average of the tests (Figure 29 and 30). The general trajectory of the tests follows the back curve. This implies that the theoretical force of 1000N is applicable. However, the test results differ greatly between the tests and at the different points of insertion. The hypothesis for this is that the force difference between the chains is large. Tests one and two were inserted by a different person then tests three and four. There are clear correlations between the pairs implying that the technique of the manual work on the pulleys results in the offset from the theoretical force.

Another takeaway is that the rods are not inserted perfectly parallel. This is a point of improvement where a hydraulic system could deliver a consistent flow that wound make the insertion parallel.

3.1.2 Mechanical friction test

The friction of the system was tested by inserting the AOGV without the pressure. Only the chain on one side was used to push the spade into the AOGV. The force ranged between 130 N and 177 N. The maximum friction per rod that needs to be overcome is, therefore.

$$\text{Friction per rod } = \frac{117}{2} = 88.5\ N$$

3.2 Mechanical Calculations and Design

3.2.1 Case for a New Stuffing Box

The stuffing box works well with the existing system where there are no parts, other than the stuffing box that are fastened to the top of the AOGV. In (K) the NBR 85 seal is shown. In Figure 31 the NBR seal is activated by the two bolts forcing the stuffing box downwards, activating the seal. When implementing hydraulic cylinders to insert the rods they need to be fastened to the top of the AOGV. This will be a problem because the assembly would be a two-step prosses where the stuffing box is fastened first, with the cylinders fastened after. A solution to this problem is by utilizing seals activated in grooves.

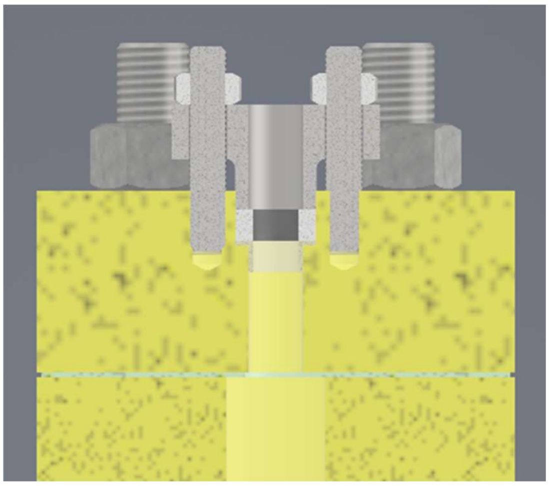


Figure 31: 6" AOGV Stuffing Box [42]

AOGV operations are often utilized in tight environments. A key priority is therefore to minimize the size of the AOGV. Between the top of the TEL and the top of the bolts there are about 30 mm. This distance is utilized differently in the new design, where there is no longer a need for seal activating bolts. A new fastening method for the hydraulic cylinder is at a minimum distance to the top of the TEL to keep the overall size of the AOGV to a minimum.

3.2.2 Concept for the Isolating Fastening Tool for Hydraulics

By opting for a groove activating sealing solution, the only bolts needed are the ones already used for fastening the TEL – Top envelope Lid to the body of the AOGV. There are three seals in the design where two work on the rods, and one between the IT – Isolation Tool and the TEL which seals any pressure leakage from the bearing opening. The housing of the seals is called the IT, and it is fitted in the TEL and fixed by the ITL - Isolation Tool Lid.

A bespoke HC – Hydraulic Cylinder is welded to the CFP - Cylinder Fasting Plate and fastened with the bolts form the existing system. These are the only bolts needed for the system which is less than the existing design. There are more seals in the Isolating Fasting Tool for Hydraulics design. However, the seals can be pre-assembled on the IT before an operation, reducing the operational assembly time.

Another key change in the new concept is the addition of a second bearing. This will reduce the radial forces for the cylinder to work on two ways. Firstly, the bearing themselves will counteract the moment and radial forces. Secondly, the separation of the parts along the rod will improve the working forces for the HC. Figure 32 is a schematic of the Isolation Fasting tool for hydraulics concept.

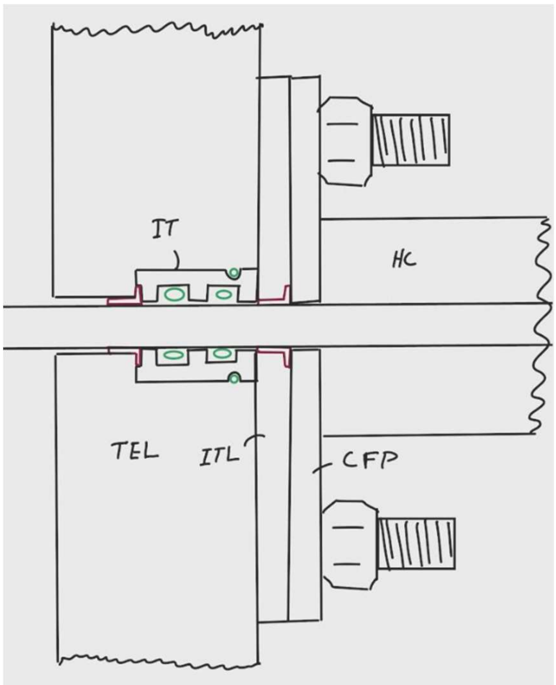


Figure 32: Isolating Fasting Tool for Hydraulics Concept Schematic

3.2.3 Main Pressure Isolating Seal

Standard cylinders are design to work with air as the external medium. By utilizing seals, the internal pressure of the AOGV is neutralized. The maximum test pressure of the 6” AOGV is at 7.7 MPa, which is the minimum sealing capability needed. AOGV applications are used for multiple types of mediums including nitrogen. The seals therefore have a broad chemical capability.

The Quad-Ring as presented in Trelleborg Quad-Ring® Catalogue (A) is a great seal for dynamic rod applications and high pressure. Quad rings (Figure 33) can be used with or without backup rings. For reciprocating motion, the maximum pressure without a backup ring 5 MPa. That is below the maximum testing pressure. Using two backup rings, the maximum pressure is 30 MPa. For the main seal the Quad-Ring with two backup rings is therefore chosen.

In the Trelleborg Quad-Ring® Catalogue there are two standard options of for materials to choose from: NBR and FKM. As seen in the Trelleborg Chemical Compatibility Guide (B) both NBR and FKM are compatible with a multitude of materials including nitrogen. One key difference is the temperature capabilities of the material. There are applications in sub-zero environments which gives the NBR material the upper edge with the capability of use in thirty below zero the existing Stuffing Box design the material of the seal is NBR. This has been tested thoroughly by Izomax, with good results. NBR is therefore the chosen material for the Quad Ring.

Figure 33: Trelleborg Quad Rings [43]

Dimensioning of the seals are dependent on whether the application is male or female. The seal works on the rod, so a female grove design is correct. Grove designs for double backed quad rings are granted in catalogue (A) and stated in the Table below. Quad ring type QRAR4112 is chosen with two BP2300100 backup rings mounted on either side of the quad ring. Figure 34 is a half section view of the Quad ring configuration. The BP2300100 are compatible based on the support suggestion in catalogue (A). The CAD file used is accrued from Trelleborg Sealing Solution CAD service.

Table 10: Grove Dimensions for Quad Ring with Double Support

Dimensions	Grove Diameter	Grove Width Diameter	Radius
10.20x2.62	14.6 mm	5.8 mm	0.3mm

Figure 35 shows a sketch of the groove design. The revolute function in Autodesk Inventor is used on the profile, with the construction line as the centreline of the function. Take the distance between the centreline and groove hight line at 7.03 mm. Doubling this shows that the diameter is 14.6 mm.

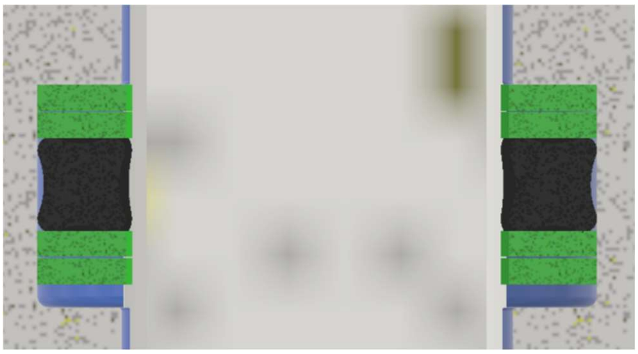


Figure 34: Quad rings with Support Rings [44]

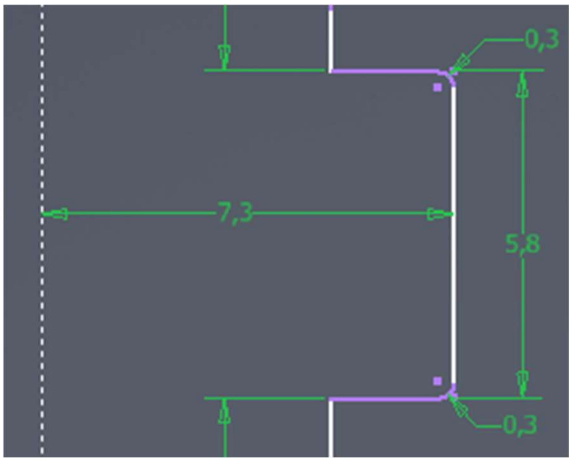


Figure 35: Quad Ring Grove Dimension

3.2.4 Rod Seal

Quad seals with backup rings can maintain the full pressure. Another one directional seal will however reduce leaking risk. The typical choice for rod seal is a spring-loaded type. In this instance the rod size is a limiting factor in rod seal selection. By using a special ordered O-Ring loaded rod seal the solution works well for 10 mm rods.

From advisement form the sealing company Otto Olsen, pressures over 7.7 MPa is managed with a multitude of gasses with the NBR70 material s3 seal. A limitation of the s3 seal is the temperature limitations. Woking between 4-30 degrees Celsius. For applications in artic and tropical environments other solutions should be considered.

Key attributes that make the s3 seal an efficient support, is the one directional properties of the seal. That means that if pressures are trapped between the seals at disassembly the s3 will easily let the pressures out, which a quad ring would not.

A section view of the seal is shown in Figure 36, and the groove of the s3 is described in the Figure 38. Otto Olsen delivered the dimensioning of the seal and is shown at their approval. Figure 37 shows the sketch for the groove, and the second picture is a description of the s3 rod seal granted by Otto Olsen for this solution.

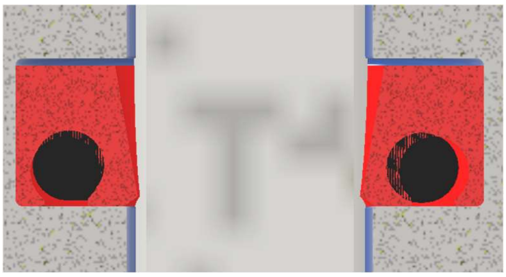


Figure 36: S3 Rod Seal [45]

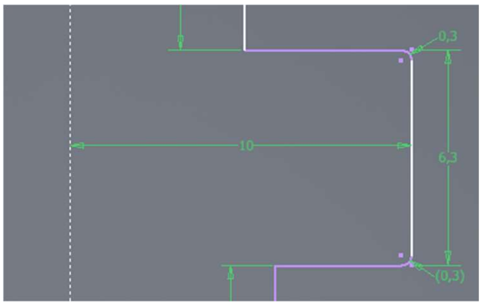
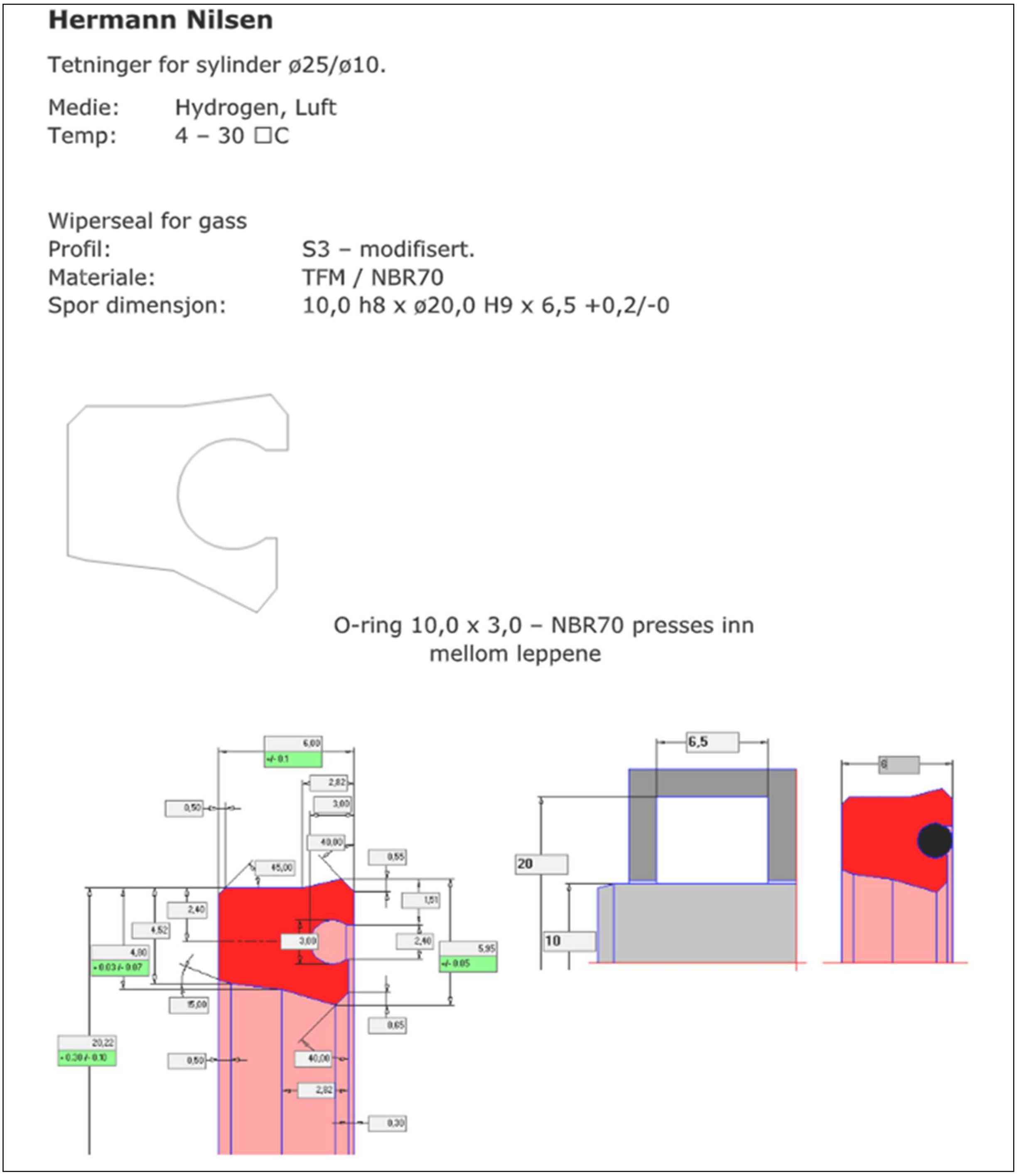


Figure 37: S3 Grove Dimensions



Hermann Nilsen

Tetninger for sylinder ø25/ø10.

Medie: Hydrogen, Luft
Temp: 4 – 30 □C

Wiperseal for gass
Profil: S3 – modifisert.
Materiale: TFM / NBR70
Spor dimensjon: 10,0 h8 x ø20,0 H9 x 6,5 +0,2/-0

O-ring 10,0 x 3,0 – NBR70 presses inn mellom leppene

Figure 38: S3 Technical Sheet by Otto Olsen [46]

3.2.5 O-rings

Pressure along the rod is cancelled by the s3 and quad ring. There is however a leakage point out of the connection at the IGUS bearing. The solution to this is a O-Ring. It is located near the top for easy assembly, see Figure 39. To keep the TEL as simple as possible the O-Ring is mounted in a groove on the outside of the IT, making it a male configuration. As seen in the limits of extrusion graph (2.1.9) the O-Ring works at pressures over 138 bar when the diametrical clearance is as low as 0.013mm wish is the case in this circumstance (see 3.2.12).

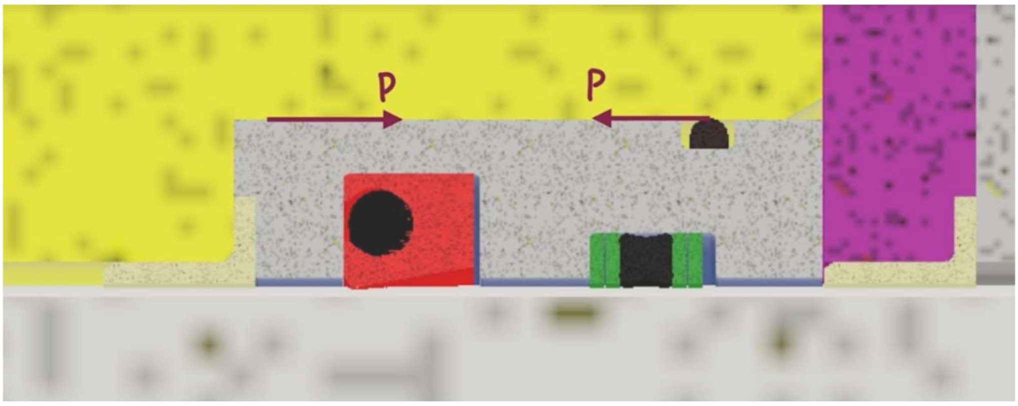


Figure 39: Pressure Illustration

In Parker O-Ring Handbook (F) there it a Table describing the dimensions of the grove. The (d3, d9) = 25 mm configuration is the correct option because the diameter size of the IT is approximately 25 mm. The O-Ring for this application is then part number 2-020. 2-020 has a thickness of 1.78 mm. In the groove design for this configuration the groove diameter is 22.4 mm (Figure 40). The length of the groove along the length of the IT is 2.4 mm. The sketch of the groove is seen in the picture below. As seen in in the Table "Design Dimensions for O-Ring" in (F) the radius of 0.2 mm is selected based on the seal thickness of 1.72 mm and the length of groove.

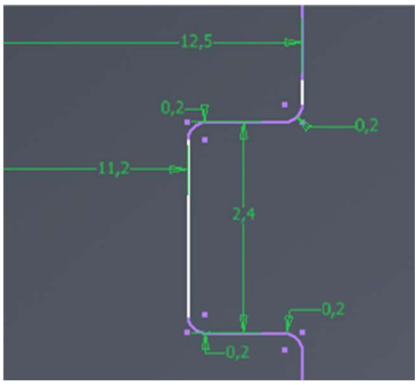


Figure 40: O-Ring groove dimensions

3.2.6 Low friction Bearings

Existing AOGV-s are designed with a single IGUS Iglide® J350. J350 bearings are easily replaceable making it a cheaper offer then metal bearing with higher production cost. As stated in IGUS Iglide® J350 Catalogue (C) the bearing also works well for counteracting moments in the rods. Coefficient of friction on the IGUS are also great being between 0.10 and 0.20. In compression Copper on Steel has a coefficient of friction of 0.36 (2.1.6).

Hydraulic cylinders optimally work with linear forces only (2.2.2). Figure 41 is the existing design, and Figure 42 is the new design. A key improvement in the bracket design is the addition of a second journal bearing. This way moments are counteracted by the separated radial forces as well as the internal moments counteracted internally in the individual bearings.

Ease of assembly is a priority with the bearings. They are simply putt loosely in place and fixed by the surrounding parts. The innermost bearing being fixed between the TED and the IT, while the other is fixed between the ITL and the CFP.

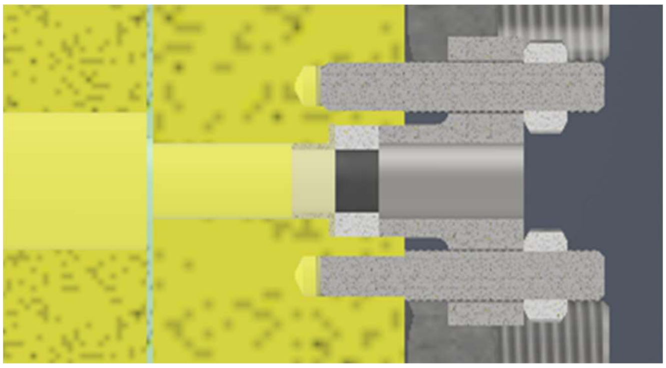


Figure 41: Stuffing Box for non-Hydraulic Applications [48]

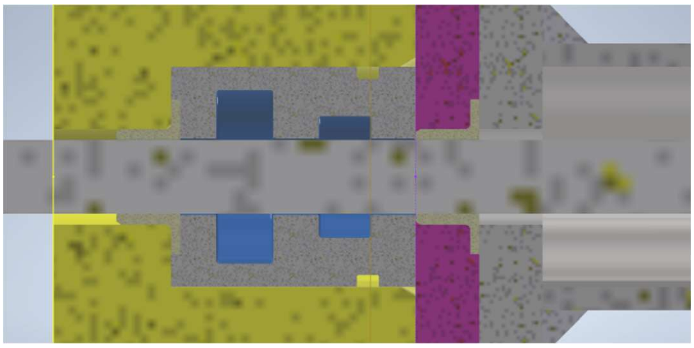


Figure 42: Double Bearing Locations

3.2.7 Isolation Tool Design

The IT design seen in Figure 43 is designed with two key attributes. Firstly, the part isolates the internal pressures of the AOGV. A Quad Ring (3.2.3) and a S3 rod seal (3.2.4) is mounted in groves located inside the IT structure. To stop the pressure leaking through the bearing and out on the outside of the IT an O-Ring (3.2.5) is used. The O-Ring groove is located on the outer surface if the Isolation Tool. All the groove seals are integrated in the IT design. This simplifies the assembly process because the Quad Ring, S3 Rod seal, and O-Ring all are mounted on a single part before assembling the Isolating Fasting Tool for Hydraulics.

Secondly, the IT is designed for the rods to slide through the structure. By utilizing IGUS bearings on either side of the IT the rod run linearly through the inner hole. IGUS Iglide® J350 bearings are designed with a flat surface on one side to hold the bearing in place. The innermost end if the Isolation Toll is designed with a to fit the top face IGUS (Figure 41). The hole is dimensioned (3.2.11) for the rod to run freely without contact, while limiting the hole size for a satisfactory seal performance.

The material selection of the Isolation Tool is the same as for the Top Envelope Lid because it simplifies the overall production prosses. P355NL2 (Table 1, 1.1.3) is a mechanically strong steel with mechanical properties fitting the task of housing high pressure seals. The outermost surface has a tight clearance with the Top Envelope Lid hole (3.2.12). Because of this a surface quality limit on $R_a = 1.6$ is set (3.2.13).

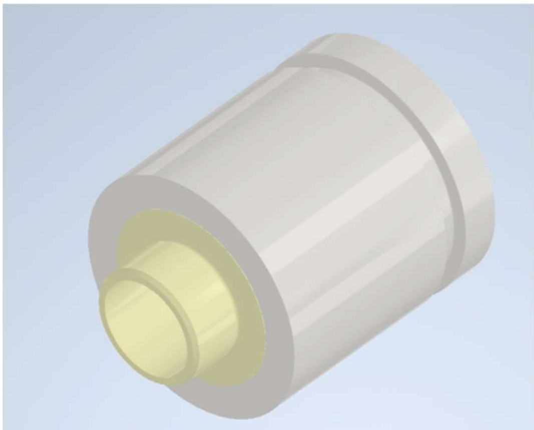


Figure 43: Isolating Tool with IGUS Bearing

3.2.8 Top Envelope Lid Design

Several design changes are made to accommodate the IT design. Firstly, the hole dimensions are changed to fit the IT. The IT has a larger outside diameter and is longer in length. This small reduction in TEL volume is negligible in terms of structural properties because the volume change is minimal. Correspondingly with the outer surface if the IT, the hole surface of the TEL has a surface roughness of $R_e = 1.6$.

As you can see in Figure 44. On the top edges of the Top Envelope Hole a 60-degree chamfer is added for smooth assembly of the IT. The Stuffing Box bolt holes from the existing design are removed as they are no longer needed.

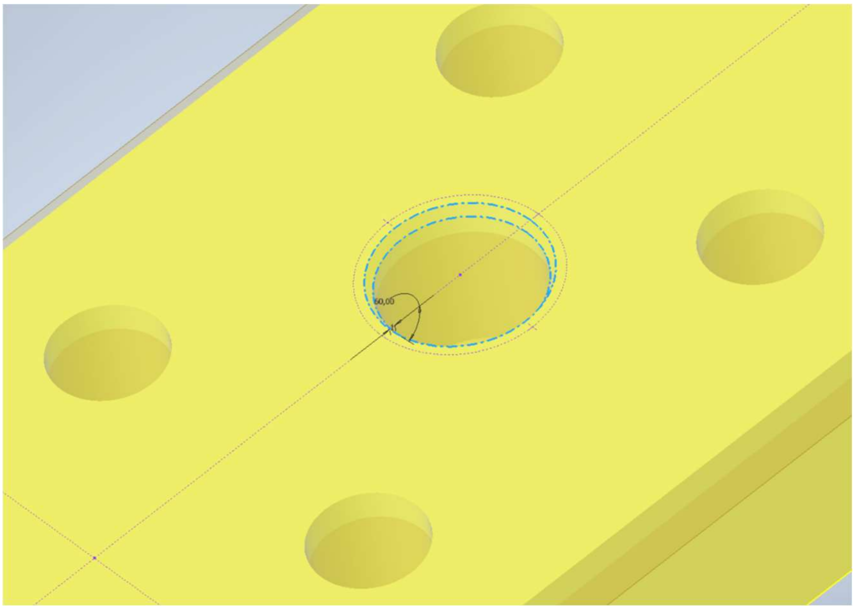


Figure 44: Top Envelope Lid Design

3.2.9 Isolating Tool Lid Design

The ITL (Figure 45) has two main functions. Firstly, it hods the IT in place by surface contact. From simulations of forces from the IT on the ITL (3.2.20), the thickness of the plate is 7 mm. The outer diameters of the lid are similar in width as the TEL. The length of the ITL is such that the distance between the bolt holes and the edge is 22.5 mm. This is sufficient for M16 bolts by simulation. Secondly, the upper part of the hole is designed to fit the IGUS Bering (Figure 46).

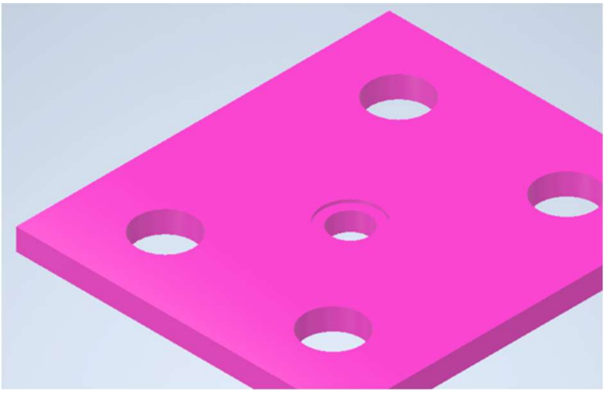


Figure 45: Isolation Tool Lid Design

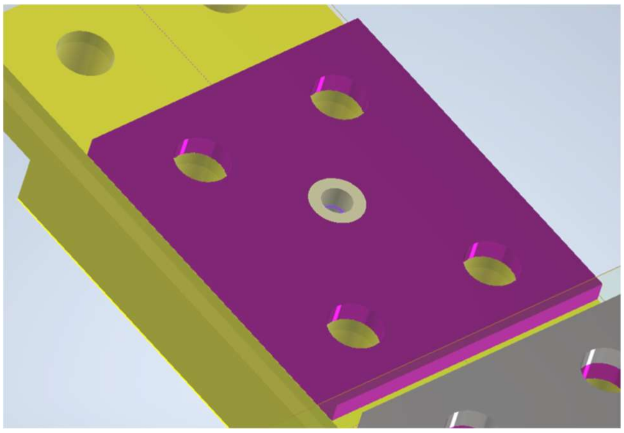


Figure 46: ITL with IGUS Bearing

3.2.10 Cylinder Fastening Plate

The CFP is a flange that is welded to the hydraulic cylinder. The CFP is mounted on top of the ITL (Figure 47). Together they carry the load from the bolts (3.2.20). The Cylinder Fastening Pate fixes the IGUS bearing in the hole on the ITL.

The thickness and outer dimensions are the same on the CFP as the ITL. This simplifies production as both parts are cut from the same 7 mm plate of P355NL2 (1.1.3).

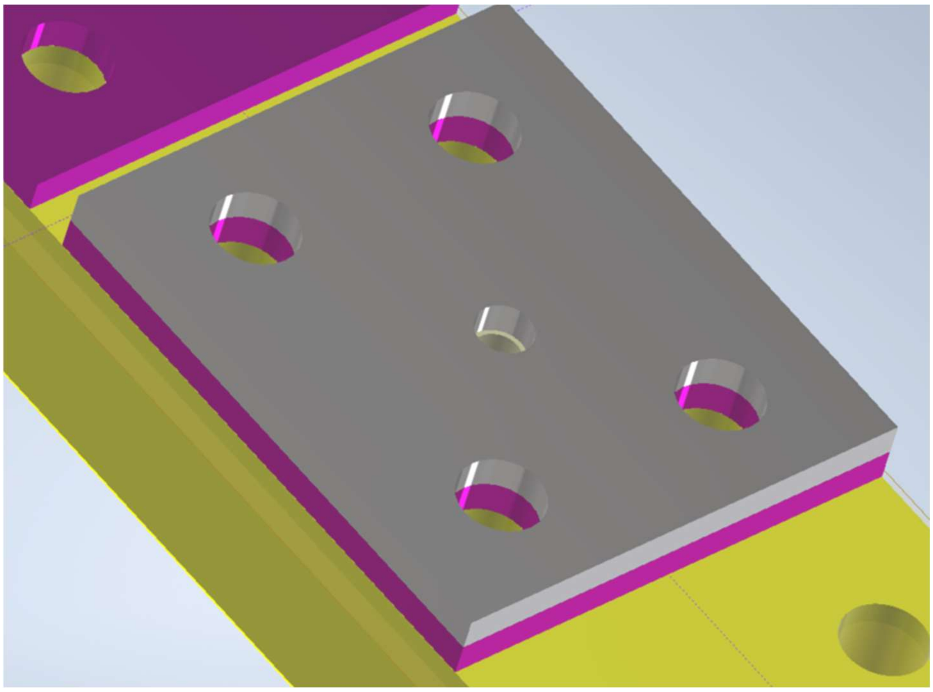


Figure 47: Cylinder Fasting Tool Design

3.2.11 Fitting Isolation Tool and Rod

For ideal seal function the fitting between the rods are at a minimum. However, the parts cannot be in contact. The inner diameter of the IT is dimensioned thereafter. From (2.1.7) a loose running fit is chosen (H11/c11). Using the Tables and formulas in (2.1.7) for a 10 mm hole the following numbers are found

$$\Delta D = 0.110 \quad \delta_F = -0.095 \quad \Delta d = 0.113$$

Which gives

$$D_{max} = D + \Delta D = 10 + 0.110 = 10.110\, mm$$
$$D_{min} = D = 10\, mm$$

$$d_{min} = d + \delta_F - \Delta d = 25 - 0.095 - 0.113 = 9.792 mm$$
$$d_{max} = d + \delta_F = 25 - 0.095 = 9.905\, mm$$

The diametrical difference between D_{max} and d_{max} is 0.205 mm, while the difference between D_{min} and d_{min} is 0.208 mm. The IT carry the load of the rod, so the largest difference is chosen. The Rod will in some operations be subjected to a larger temperature from inside the AOGV and might slightly expand. An additional 0.3 mm is therefore added to the IT hole.

$$Diameter\ IT = 10.5\, mm$$
$$Diameter\ Rod = 10\, mm$$

3.2.12 Fitting between bracket and Top Envelope Lid

There are high pressures working in the inner part of the bracket. For that reason, the bracket needs a tight fit for seals to work efficiently. It is however a clearance fit because the seals work on the pressure and assemblies are simple. From (2.1.7) the clearance fit is optimal because the bracket is not moving when assembled. The fitting type us therefore H7/h6. The basic size is of 25 mm.

ΔD is 0.021 δ_F = 0 Δd= 0.013

Which gives,

$$D_{max} = D + \Delta D = 25 + 0.021 = 25.021\, mm$$
$$D_{min} = D = 25\, mm$$

$$d_{min} = d + \delta_F - \Delta d = 25 + 0 - 0.013 = 24.987\, mm$$
$$d_{max} = d + \delta_F = 25 + 0 = 25\, mm$$

Top Envelope Lid hole diameter is the defining parameter for clearance selection because it is more standardized than the bracket. Therefore, the chosen diameters for the shaft and hole are

$$Diameter\ Hole\ in\ Lid = 25\, mm$$
$$Diameter\ Isolsting\ Bracket = 24.987\, mm$$

The IT is assembled and disassembled with no pressure or temperature form the AOGV because gasses are only released when the full assembly is complete. Therefore, the fine clearance of 0.013 mm is used.

3.2.13 Surface Roughness

In the Fitting between bracket and Top Envelope Lid (3.2.12), the clearance is at a minimum. For this reason, a Surface Roughness limit is set on both surfaces. Because the tolerance grade is set to H7/h6, the tolerance grade is IT6. Categories 4 and 5 in Table 7. (2.1.8) are for IT6 fitting applications. To limit the costs in production the largest acceptable category is chosen. Category 5 is the largest acceptable surface roughness category. A R_a surface roughness of 1.6 is chosen which is in the middle of category 5 spectrum.

3.2.14 Force from pressure

To find the working pressure under the installation of the isolation plate the formula *(1)* is used. From (1.1.8) the test pressure is 7.75 MPa.

Because the internal forces on the Isolation Spade inside the AOGV body cancels out, the force-pressure-area is derived from the Stuffing Box (1.1.4) The diameter of isolating surface is 10 mm. This gives the area 78.54 mm^2 from geometric calculations. This gives the theoretical force on each rod of:

$$F_{pressure} = 678.4\ N$$

3.2.15 Rod Calculations on Isolating Fasting Tool for Hydraulics

The case for rod calculations is when the AOGV is horizontal because that maximizes the bending moment in the rods. The bearings are simplified as perfect journal bearings in the rod calculations (Figure 49). Figure 48 shows the rods with Isolation Spade in horizontal position.

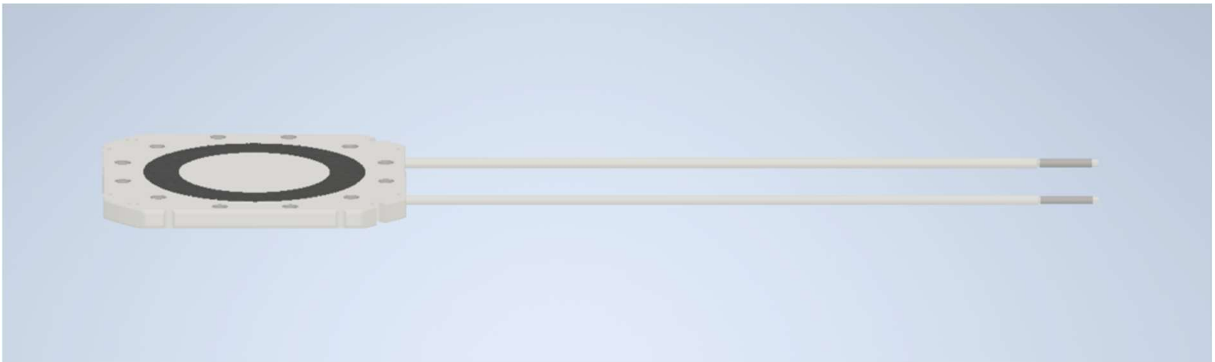


Figure 48: Spade and Rods in Horizontal Position [48]

To organize effective rod calculations, the rod is separated in parts. Part number 1 is the longest at 497.6mm when fully extended (Table 11). Because force is applied from the cylinders and the This part will be evaluated for both bending and buckling.

Table 11: Length of Rod in Part 1.

Reference	From	To	Length (mm)
Figure 9 (1.1.6)	Centre Isolation Spade at max insertion	Centre Isolation Spade at min insertion	481.3
Figure 10 (1.1.6)	Centre Isolation Spade at min insertion	Top of TEL	181.3
Figure 11 (1.1.6)	Top of TEL	Bottom of TEL	- 40
4.12 (1.1.5)	Half the length of Isolating Spade		- 290/2
SUM	Bottom end of rod at max insertion	Bottom of TEL	477.6

Part number 2 is where the force applied by the pressure pushes the rod by the seals. However, no calculations are made on the rod in area 2, because the distance between the bearings are small.

In area 3 there are only forces along the rod. Therefore, a buckling calculation is done. The length of the rod on the cylinder side is unknown because it will vary between cylinder types. The IT where there are two seals has a length of 34 mm (Figure 50). A distance between the lid and maximum extended cylinder is therefore set at 34 mm to simulate the head seals of the hydraulic cylinder. The most relevant calculation in area 3 is when the cylinder is minimally extended. The length of the rod in area 3 is therefore 477.6 + 34 = 511.6 mm.

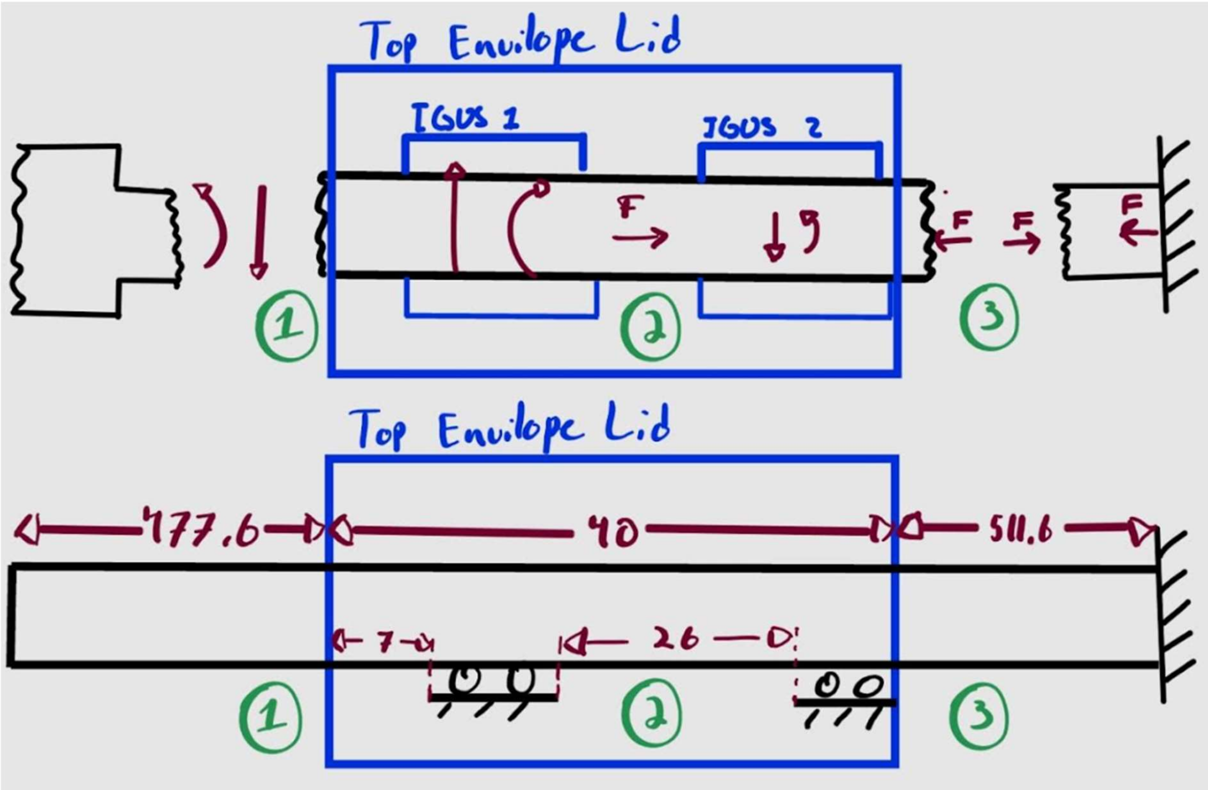


Figure 49: Rod Calculation Schematic

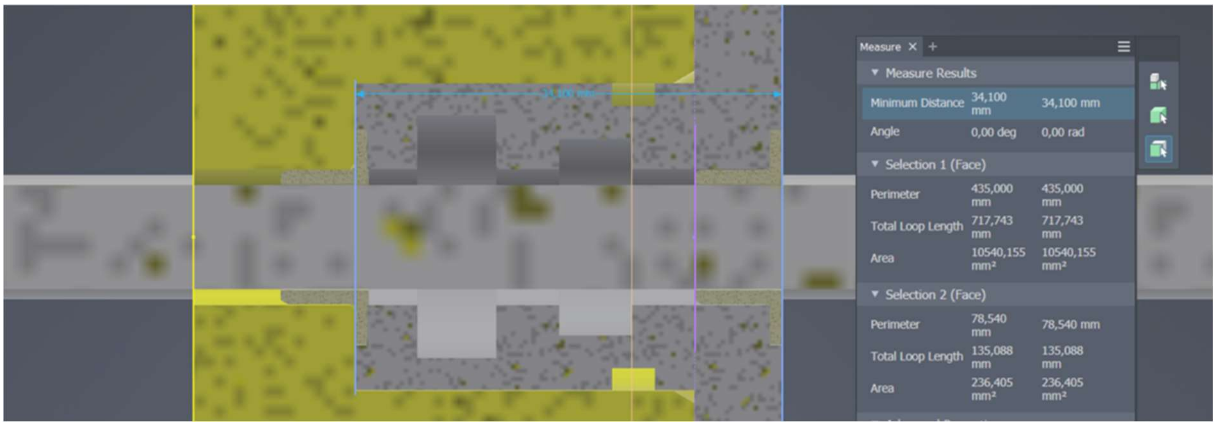


Figure 50: Cylinder Length Prediction Visualization

3.2.16 Bending of Part 1

The bending of the rod is an important case because the Isolating Spade has a weight of 12.2 kg working at the end of a fully extended rod. Using the formula for moment *(2)* we find a moment diagram for the rod. The extended side is loaded with radial force and moment, and the other is fixed by the bearing. The moment from the spade on the extended end is found by the weight multiplied with half the length of the isolation spade. Moment from the own weight is found by the weight corresponding to the length divided by 2.

$$9.81 * 12.2 * \left(\frac{290}{2} + x\right) = M_{spade}$$

$$9.81 * 0.4 * \frac{x}{2} = M_{rod}$$

$$M_{total} = \frac{M_{spade}}{2} + M_{rod} = 61.803(x + 140.397)$$

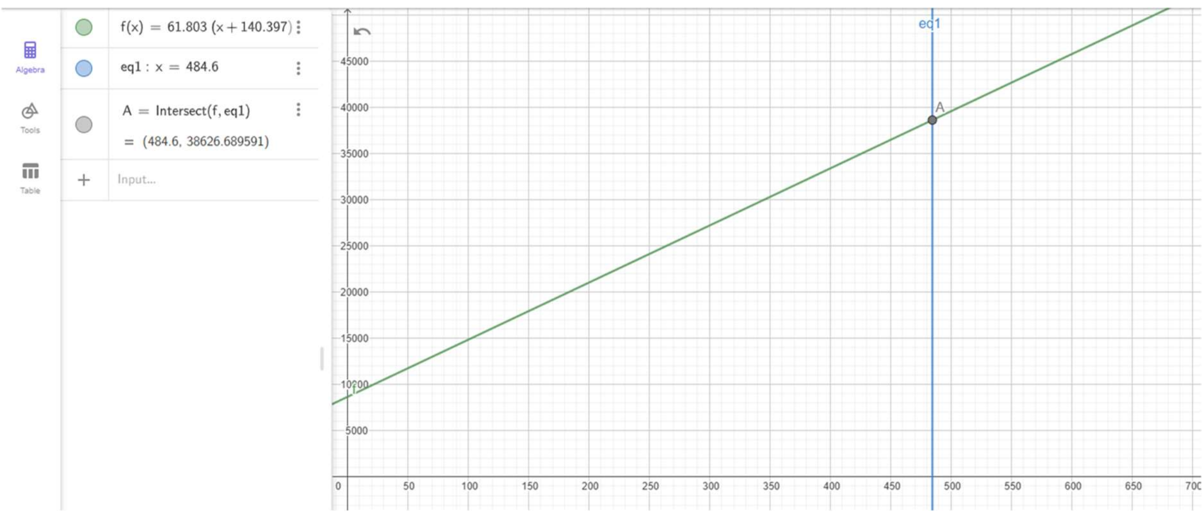


Figure 51: Bending moment diagram in GeoGebra

The maximum moment will be at the maximum inserted rod length form the innermost bearing x = 477.6+7 = 484.6 The maximum moment as seen in the GeoGebra cut out (Figure 51) is 38627 N/mm. Using formulas *(8)* and *(9)* the maximum bending stress is found.

$$\sigma_b = \frac{M}{w_b} = \frac{38627}{(\frac{\pi * 10^4}{64 * 5})} = 393\ MPa$$

The yield of the S165M material is 720 MPa. The result of 393 MPa gives safety number of n = 1.83. In 2.1.5 it is stated that the standard factor of safety against fracture under bending is n_f = 1.0 - 1.8. Because thar may occur some variability in forces a safety factor of 1.83 is reasonable.

3.2.17 Buckling in Area 1

The Isolation Spade is fastened in a socket at full extension. This would function as a pinned location for the buckling calculations. By comparison to (3.2.18) there is a much lower load, and the risk for buckling would be negligible.

If the spade fails to enter the socket properly a case of buckling with force on a free end might happen. This case of buckling has a lower max force then a pinned connection because of the kink length being double the rod length (Figure 13). Buckling calculations are therefore done using formulas (4), (3), (7), (6).

$$\mathrm{i} = \sqrt{\frac{\mathrm{I}}{\mathrm{A}}} = 2.5$$

$$\gamma = \frac{L_k}{i} = \frac{970}{2.5} = 388$$

$$F_{Max} < \frac{\pi^2 E}{\gamma^2} * \pi r^2 = 1081\ N$$

As the max force only will account for a small contact force of 50 N 3.3.9. Therefore, there is no risk for buckling in area 1.

3.2.18 Buckling in Area 3

Buckling will most likely happen inside the hydraulic cylinder because the applied force is much larger than the forces on the rods upon impact inside the AOGV. Firstly, the kink length is derived. from the total length of the rod inside the cylinder which is 511.6 mm from (3.2.15). Inside the cylinder the end point of the rod is rotation fixed and translation fixed. Therefore, Euler-type 3 is used.

$$L_k = 0.7 * 511.6 \approx 358\ mm$$

Then, formula (4) and (30) is used to find the slenderness

$$i = \sqrt{\frac{\left(\frac{\pi r^4}{4}\right)}{\pi r^2}} = 2.5$$

$$\gamma = \frac{L_k}{i} = \frac{372}{2.5} = 140.8$$

Secondly, the critical slenderness is derived by $E = 1.2 * 10^5$ (2.1.3) and R_e = 720 MPa (1.1.7) which is the minimal yield noted in the Table for the properties of S165M.

$$\gamma_1 = \pi \sqrt{\frac{2E}{R_e}} = \pi \sqrt{\frac{2 * 2.1 * 10^5}{720}} = 75$$

The slenderness is larger than the critical slenderness. Therefore, the rods are in the Euler area. To find the maximum force that can be applied before buckling, formulas *(7)* and *(6)* for buckling in the elastic area are used.

$$\frac{F_t}{\pi r^2} < \frac{\pi^2 E}{\gamma^2} = 55.25\ MPa$$

$$F_t < 4340\ N$$

In this concrete example the cylinder could apply a force on the rods much greater than the necessary force for insertion with is about 700 N (3.2.14). We conclude that the risk for buckling of the rods is small enough that the rods used in the system are 10 mm.

3.2.19 Simulations of IGUS Bearings

To ensure structural integrity from contact forces on the of the bearings and rod the system is simulated using the finite element tool Ansys Mechanical. Irrelevant parts for the simulation case like the full AOGV model, is not included to reduce likelihood of complications and simulation times. The rods with the isolation spade, bearings, isolation tools and the isolation tool fasting plate are relevant parts because they effect the internal forces of the bearings by exerting the forces on the IGUS-es or by carrying the loads of the bearing.

The forces applied to the system are the gravitational forces of all the parts. For this the "Standard Earth Gravity" force tool in Ansys Mechanical is used (Figure 52). The direction of the gravitational force is in the positive x direction wish translates to a horizontal AOGV assembly which is the case where the maximum bending forces occur. The system is fixed in the internal area of the bolt holes because the TEL is fastened accordingly (Figure 53).

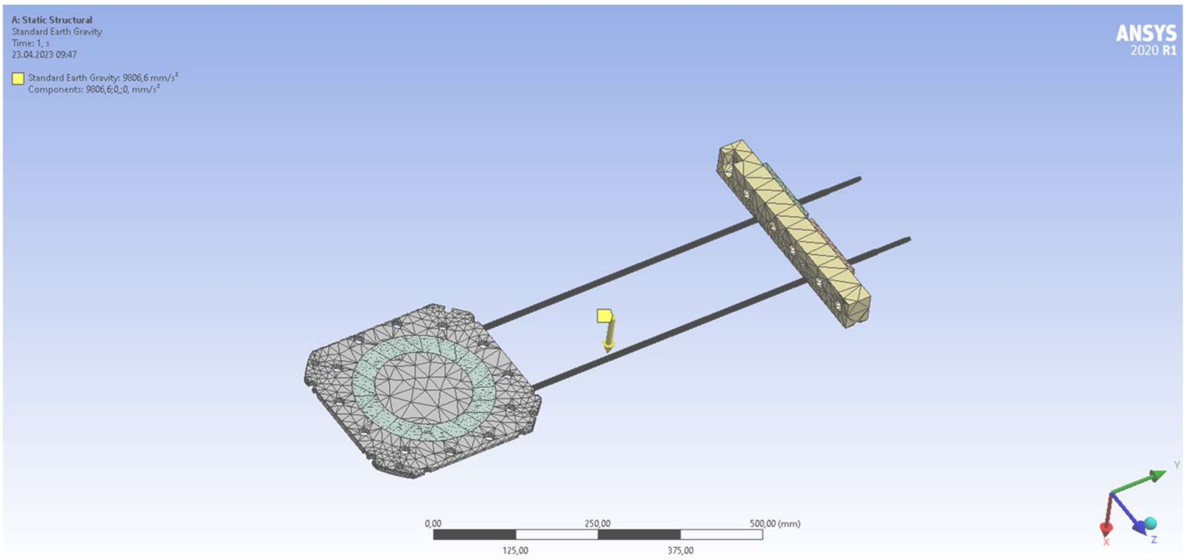


Figure 52: Gravitational Force Setup

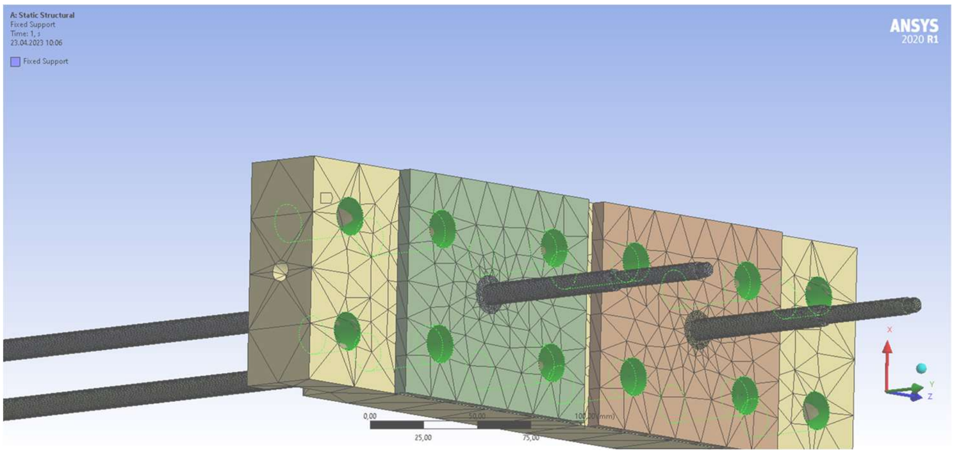


Figure 53: Fixed Support Setup

Mesh configuration will have varying effect on the simulation results depending on how important the local internal force on the part is. The two parts with the highest mech density are the rod and the bearings which have a sizing of 1 mm (Figure 52). The data form local parts on the other IT body, IT lid and TEL is of less relevance, so the mesh is larger. These parts alto have huge volumes which results in large simulation times.

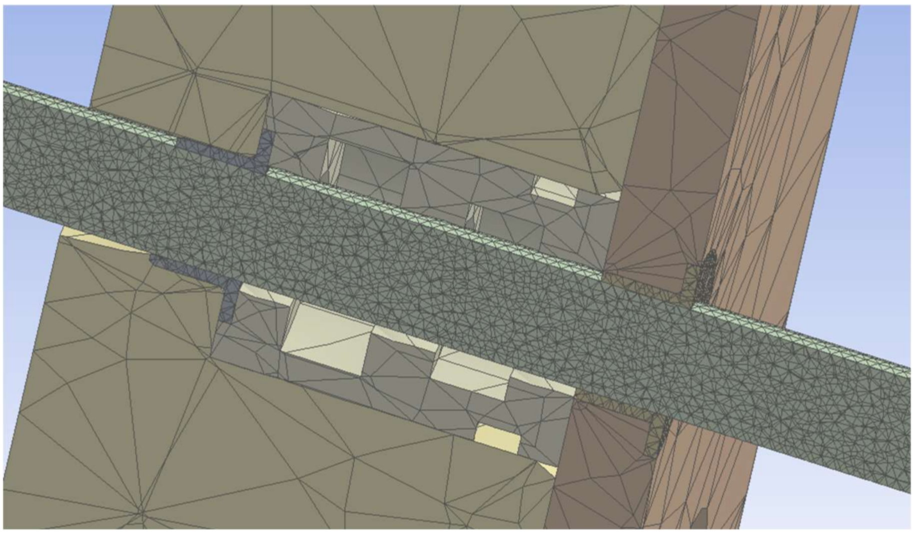


Figure 54: Mesh Setup

To achieve a realistic journal bearing loads the rods and bearings are jointed with the "Joint" function in Ansys Mechanical. The inner surface of the bearing is chosen as the reference surface, and the outer surface of the rod is chosen as the mobile part. Connection type setting is set to cylindrical which applies a moment and radial forces between the parts. Both parts are set to the "deformable" setting to ensure that the bending forces are realistically represented (Figure 55).

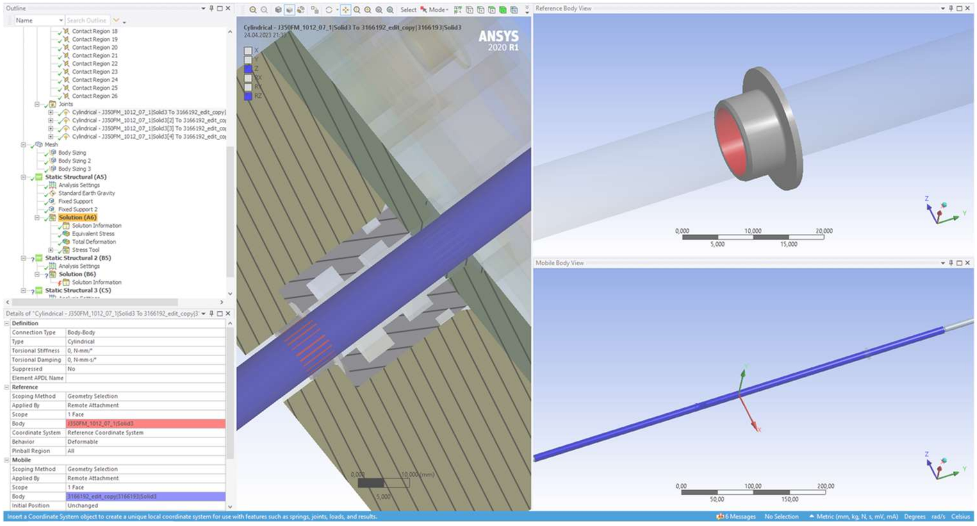


Figure 55: Journal Bearing Setup

From the IGUS Iglide® J350 Catalogue (C) the compressive strength of the Bearing material is 8.702 Psi which translates to 60 MPa. By setting the data output to Von Mises Equivalent Stress (MPa). The internal forces in the bearing are shown. The elements are set to be colored red at stresses above 16 MPa. As the results show there are no red areas on the IGUS Bearing (Figure 56 and 57). Maximum stress in the bearing appears to be 10 MPa. That gives a factor of safety of $n_b = 6$, which is an acceptable result. There is no risk of contact damage on the rod because the rod material is much stronger. To comment on other parts of the simulation, the red areas in the beam are below 400 MPa because the scale is heavily shifted. The maximum stresses occur on the fasting point between the Rod and Insertion Spade (Figure 58), which is not a part of the thesis.

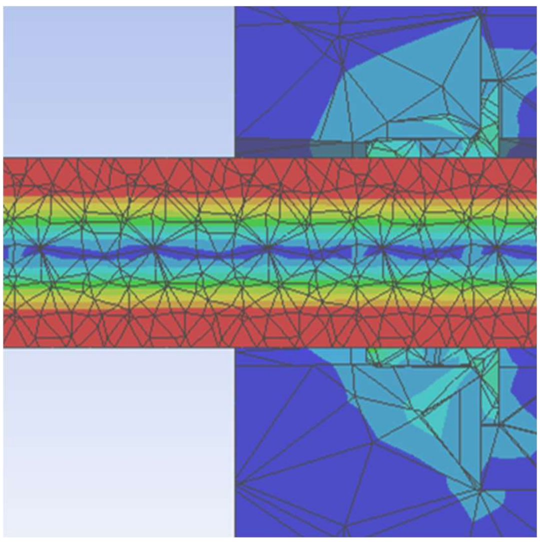


Figure 56: Result of Simulation in a Vertical Cros-Section Along the Rod

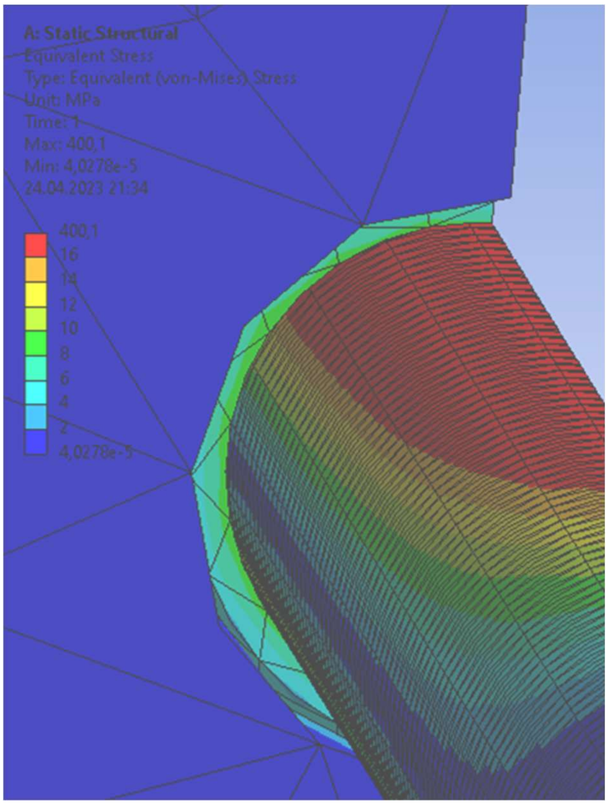


Figure 57: Result of Simulation on the Outermost Surface

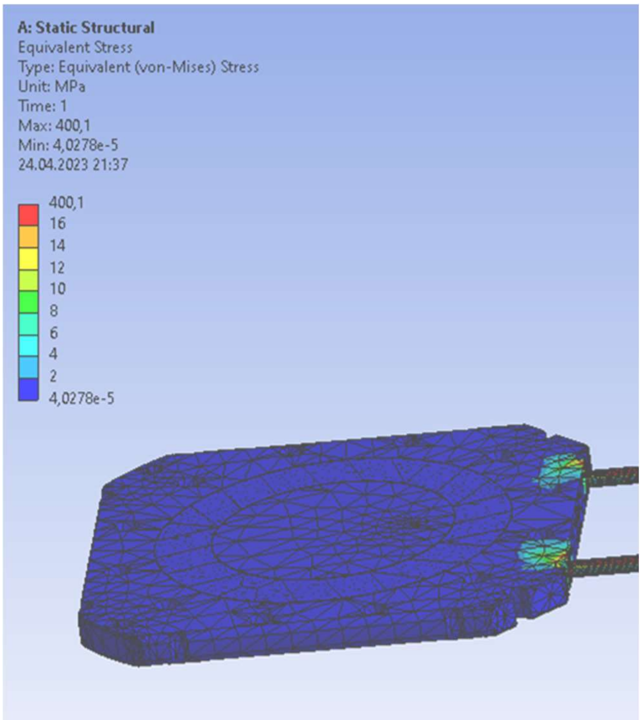


Figure 58: Visualization on the maximum Von Misses Stresses

3.2.20 Insertion Tool Lid and Hydraulic Cylinder Fasting Plate Simulation

Simulations are used for the ITL and the CFP because of the three-dimensional nature of the case. A force of 678.4 N (3.2.14) works on the location of the rod seal (Figure 59). Both parts are connected to the TEL using the existing nut and bolt configuration which has a pretension at 45kN (1.1.9). The force is applied from the bottom surface on the nuts on the CFP top surface (Figure 60). The TEL is the fixed component, while having no connection with the isolation tool to ensure that the pressure forces are applied to the ITL.

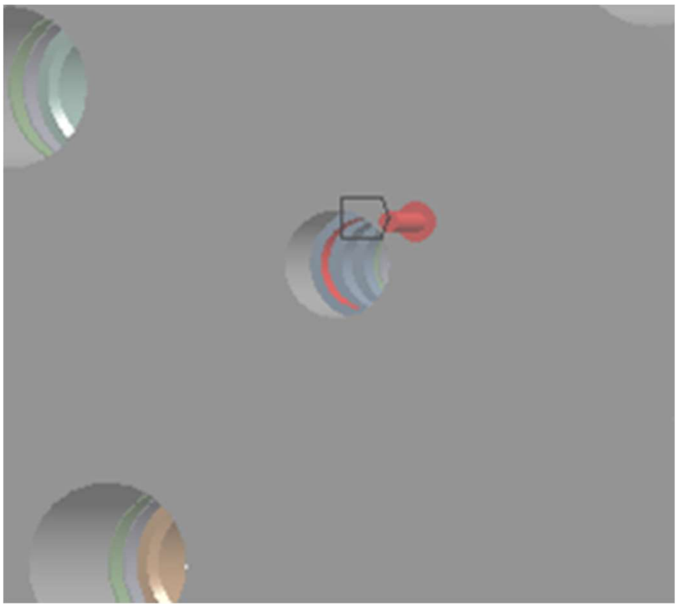


Figure 59: Location of Force form Seals

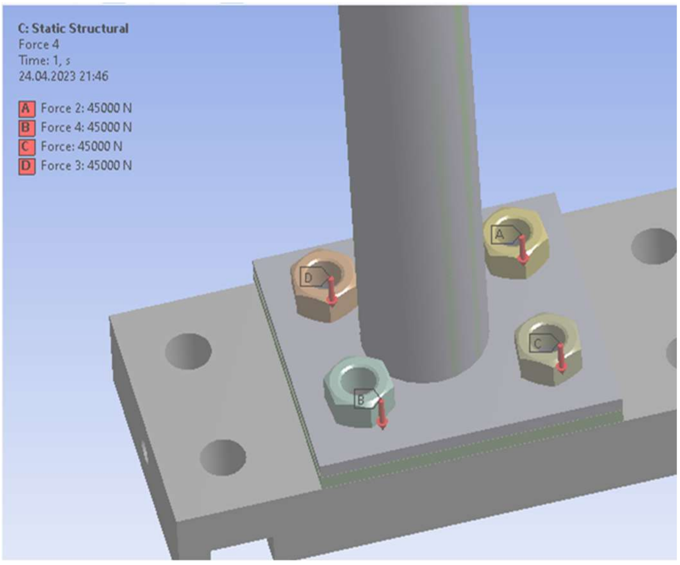


Figure 60: Pretension Forces

From visual inspection the limiting design factor is the pretension of the bolts. This is logical as the stresses from the IT are minimal in comparison to the pretension. The stresses around the bottom of the bolts are 264 MPa at a maximum (Figure 61). The yield strength of P355NL2 steel is 345 MPa for part with 16-40 mm thickness. The factor of safety is therefore $n_{lid} = 1.30$ which is sufficient. There is also negligible deformation in the system which supports the design of the ITL and the CFP (Figure 62).

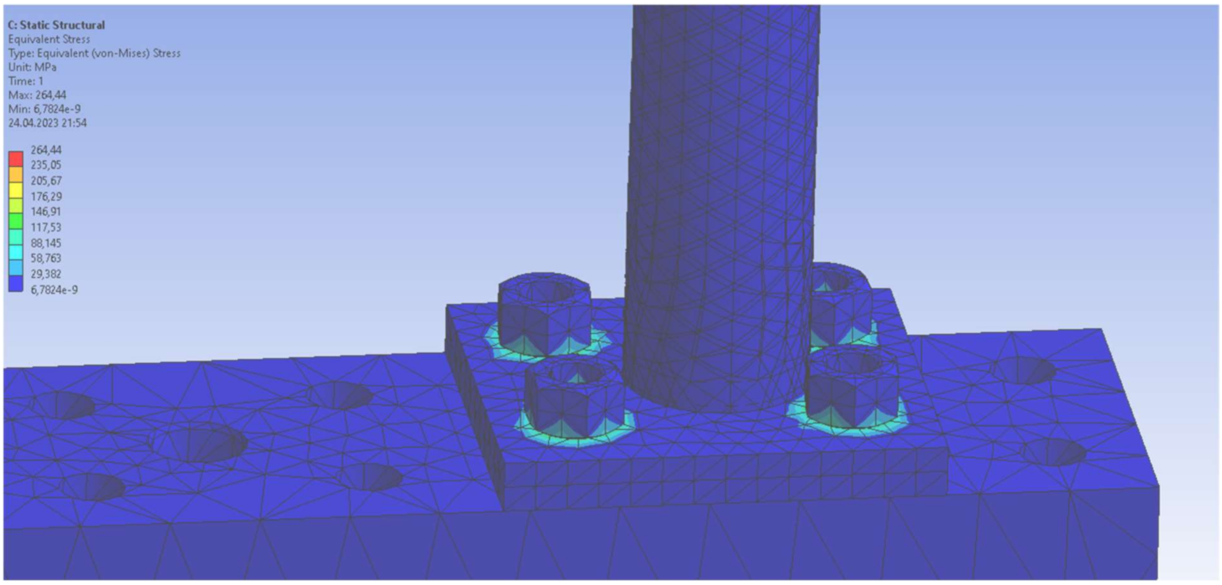


Figure 61: Von Mises Stresses on CFP and ITL

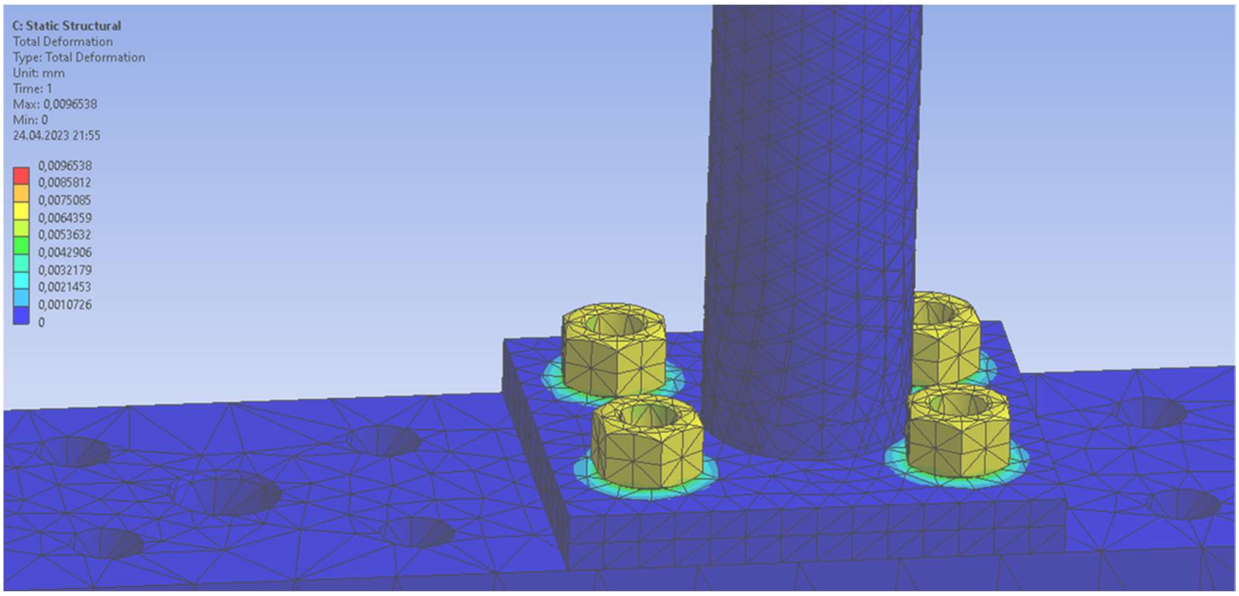


Figure 62: Deformation of CFP and ITL

3.3 Hydraulics

3.3.1 The Potential of Hydraulics

The main objective of the thesis is to simplify the insertion of the spade in the AOGV. From the points stated in (2.2.1) the potential utility for a hydraulic system makes it the ideal option to research. There are three key advantages. Firstly, a hydraulic system utilizes large amounts of force. This removes the necessity for heavy manual labour when the AOGV Is used. This has implications of ease of use and safety.

Secondly, the possibility for a responsive and dynamic system. When using the pulley system, the insertion happened in steps. The point of intersection with the spade may happen on a pull risking hitting the insides of the AOGV with excessive speed. Using hydraulics, the speed is low and fluid, and the process will stop immediately upon impact because a valve stops the additional force. The last reason is that the incompressible fluids hold the pressure constant, eradicating the need for chains to fix the system statically. Figure 63 is a render visualizing a potential hydraulic configuration.

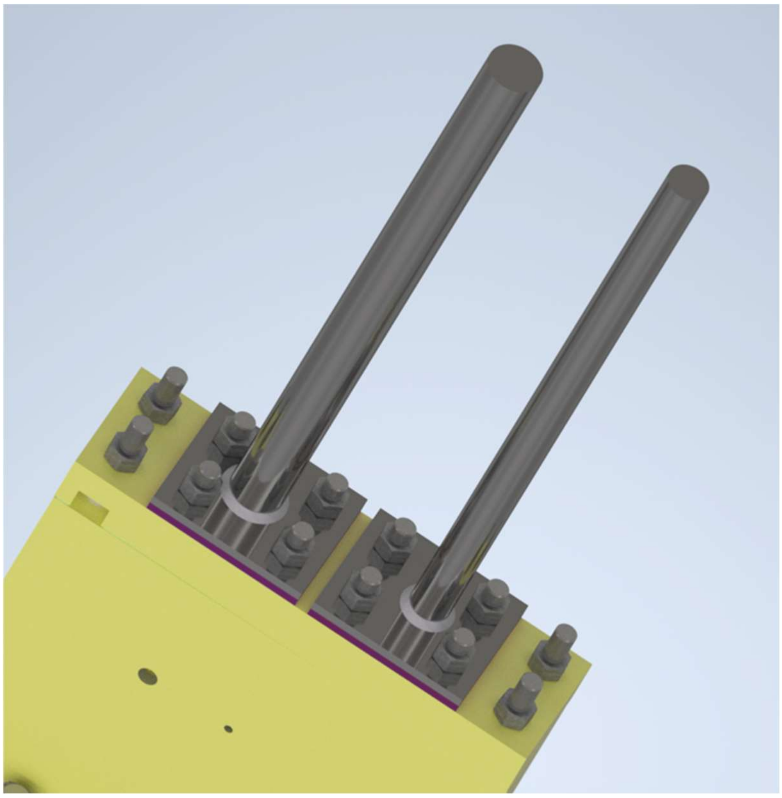


Figure 63: Render of AOGV with Hydraulic Cylinders [49]

3.3.2 Choice of hydraulic configuration

There are two systems disused in the thesis. Both have some key characteristics. Firstly, double acting cylinders (2.2.7) are utilized. Slow and steady control in both directions is needed for a precise application under varied conditions, and with double acting cylinders steady the system is controlled for insertion and extraction with controlled flow parameters.

Another key attribute of the system is the pressure relief valves (2.2.6). The Isolation Spade is in the correct place when it touches the bottom wall. When it does, relief valves levitate the pressure. This way no excessive force is applied by the cylinder when the Isolation Spade is in place.

Lastly, the parallel insertion of the rods is automized by delivering equal flow to both cylinders. A general principle in hydraulics is that hydraulic oil always takes the path of least resistance. Simply feeding both cylinders without any pressure adjusting measure will make the system greatly exposed for small variable differences in friction. The two concepts differ in how this challenge is solved.

3.3.3 Hydraulic System Concept with a Flow Divider

One way to insert two cylinders simultaneously is to use hydraulic flow dividers (2.2.4). The flow divider separates the flow of hydraulic oil from the motor. With equal flow entering both cylinders the rods move simultaneously. Figure 64 is a schematic of a hydraulic system utilising a spool type flow divider.

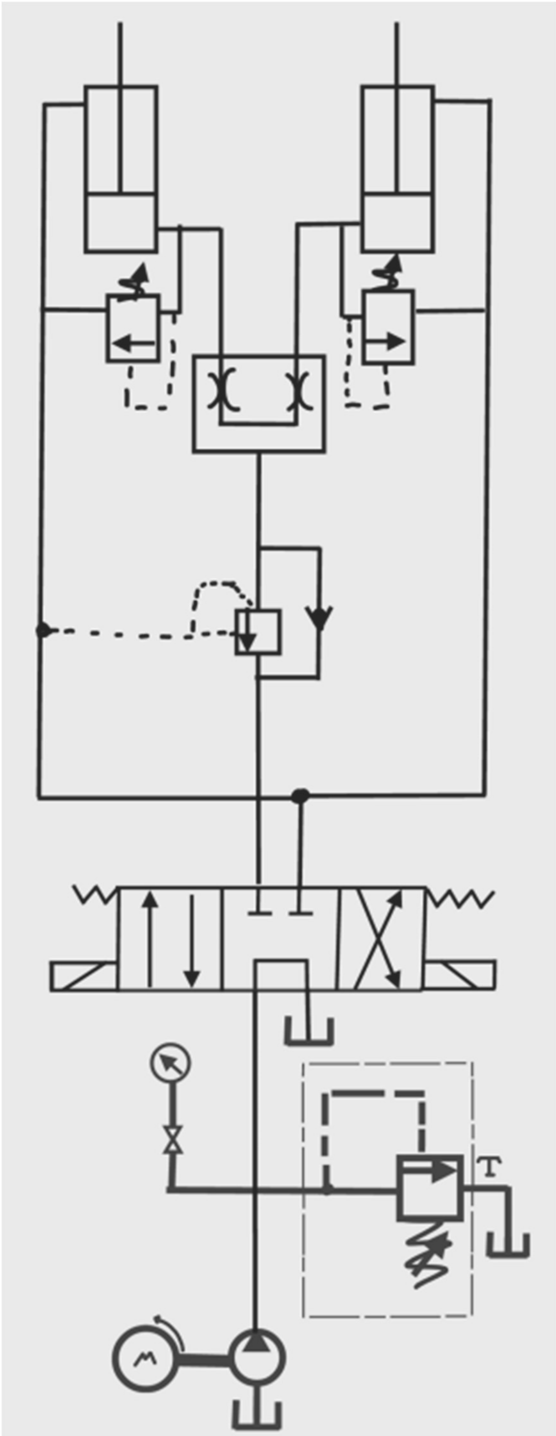


Figure 64: Hydraulic Concept with Flow Divider Schematic

There are many flow dividers on the market, but one limiting factor is the minimum flow rate. In (D) the lowest standard flow rate is 2 l/min. The rods in the AOGV needs to be inserted slowly to recuse the risk of failure. Cylinders with small volumes therefore has a flow rate too small for standard flow dividers. In the Table under different flow rates are described based on bore diameter and insert time, using formula *(12)*. Because the flow is separated to two cylinders, the input flow of the flow divider is double the cylinder flow. From Figure 65 and 66 it is shown that the minimum bore diameters, which is 35 mm for a 30 second insertion is larger than applicable for a 10 mm rod.

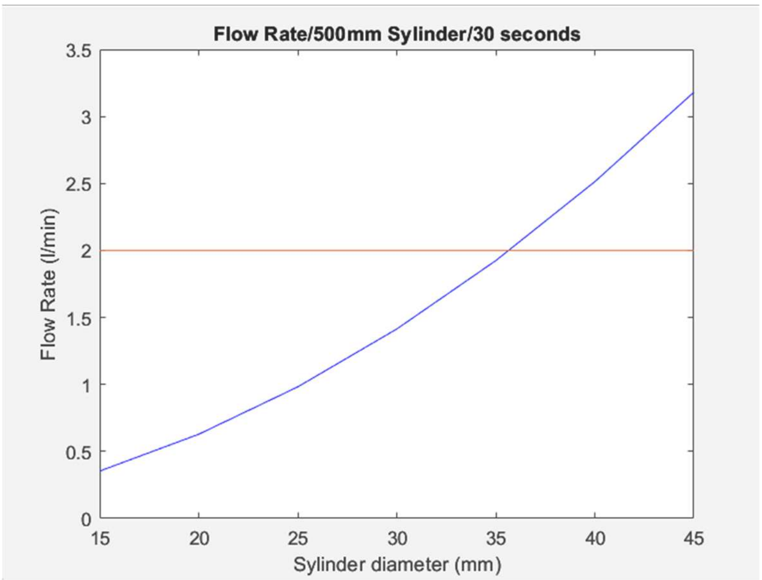


Figure 65: Flow rate of HC with Variating Bore Diameters.30 Seconds

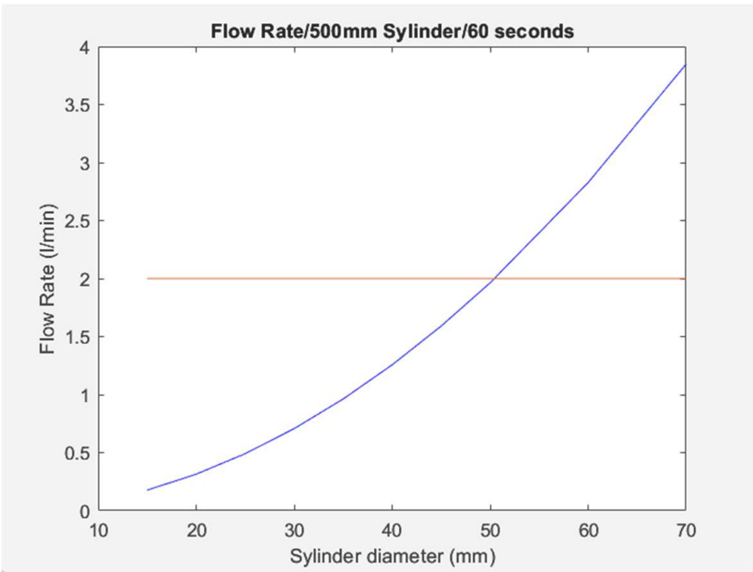


Figure 66: Flow rate of HC with Variating Bore Diameters.60 Seconds

3.3.4 Hydraulic Concept Regulated by Pressure Area

Hydraulic fluids are incompressible. This means that in theory movement in the first cylinder will instantly move the second cylinder. Because the area in the cylinder on the head side of the piston is smaller than the back side the out-pressure is larger. For the cylinders to be synchronised the bore size of the cylinders must be different. Figure 67 is a schematic of a hydraulic system concept utilising area differences. Table 12 describes the bore sizes for the first and the second cylinder. The bore size on the second cylinder is derived from the area of the exiting part of the first hydraulic cylinder. Figure 68 is a render of a potential HC assembly.

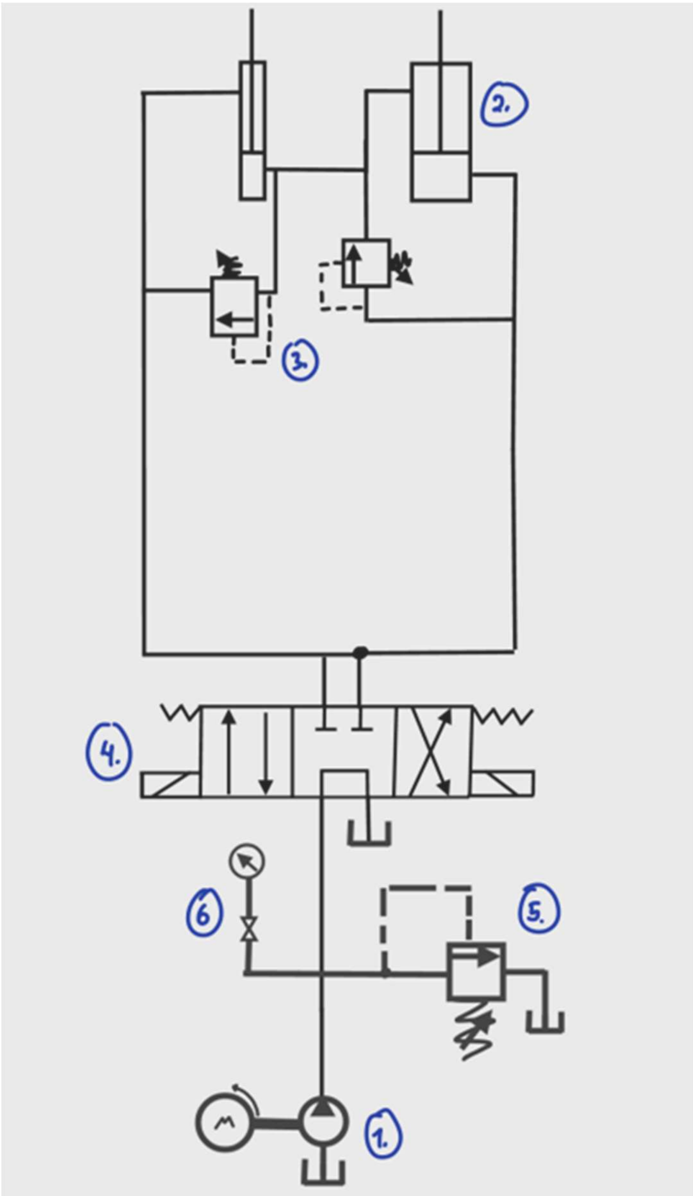


Figure 67: Area Controlled Hydraulic System Schematic

Table 12: Rod Areas with Corresponding Insertion Area

Rod Ø(mm)	Bore 1 (mm)	Area inn 1 (mm^2)	Area out 1 (mm^2)	Bore 2 (mm)	Area inn 2 (mm^2)	Area out 2 (mm^2)
10	25	491	177	15.0	177	19.63
10	32	804	380	22.0	380	113
12	32	804	314	20.0	314	50.3
12	40	1256	616	28.0	615	201

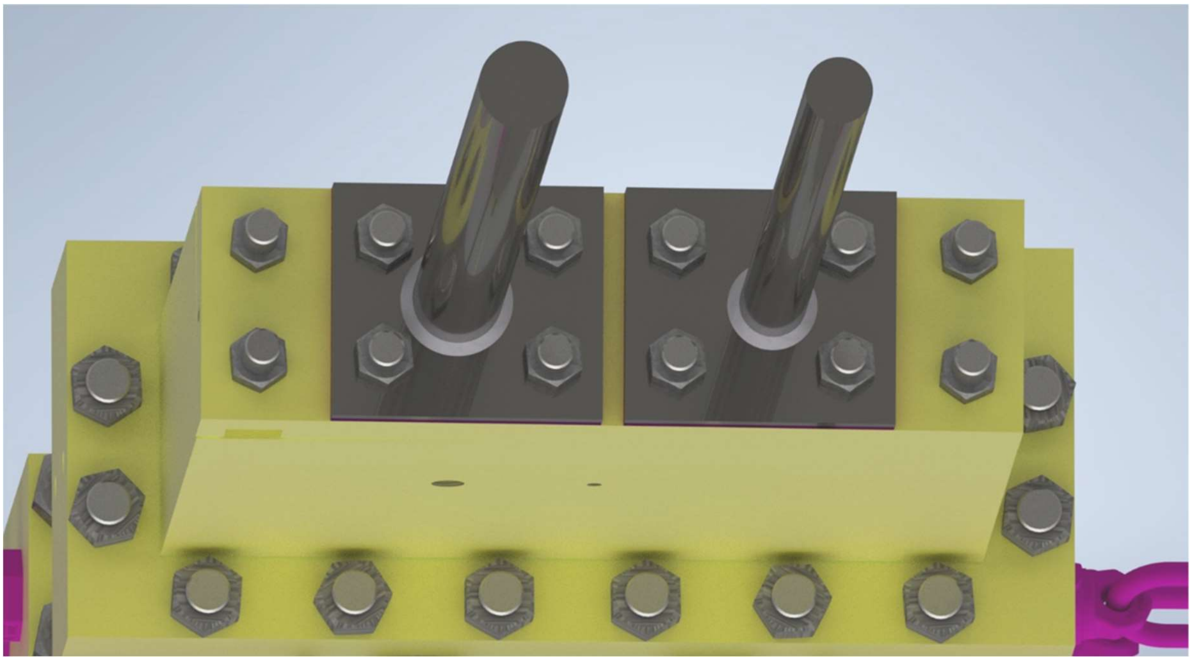


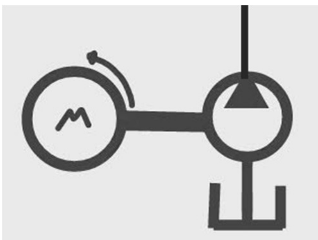
Figure 68: Hydraulic Cylinder Render [50]

3.3.5 Pump and Bore size 1

To reduce entry cost the Resato P80-260 7 Bar pump is chosen, as it is already in Isomax's inventory (Figure 69). As seen in the Resato P80 Datasheet (E), the maximum pressure is 1820 bar. That is enough force to push a 25 mm diameter cylinder with a force of just under 9000 kN. Strictly the pump is unnecessarily powerful. However, there are multiple of it in inventory and the pressure is regulated by the internal motor. From (3.3.12) we know that the maximum system pressure is 2 MPa so the pump is set at this pressure. In Table 13 the hydraulic pump specifications are granted.

Figure 69: Resato Pump in Izomax Workshop [51]

Table 13: Hydraulic Pump Configuration

Category	Selection	Characteristics
Max Pressure	1820 bar	More than enough
Set Pressure	2 MPa	System pressure
Flow Rate	0.29 l/min	Liters of hydraulic fluid the pump delivers during one minute
Schematic	 *Figure 70: Schematic Pump*	

The flow rate from the pump is constant. Cylinder diameter is then be determined by the inserting time of the intervention tool. Resto P80-260 7 Bar has a flow rate of 0.29 l/min, as seen in the Resato P80 Datasheet (E). The operation is most successful if the isolation plate is inserted first try. Therefore, the velocity of the insertion tool is slow. The ideal timeframe of insertion of spade to fasting point is 1 minute. A as shown in Table 11 the stroke length is 477.6 mm. This gives the average velocity of 0.00802 m/s.
Using the formula *(12)* we find the ideal cylinder diameter.

$$A = \frac{0.29}{0.00796\ \frac{m}{s} * 6} = 6.072\ mm^2$$

$$r = \sqrt{\frac{A}{\pi}} = 1.385\ cm = 13.85\ mm$$

$$d = r * 2 = 13.85 * 2 = 27.7\ mm$$

For simplification in production, and initially choosing the smaller cylinder the chosen bore size is 25 mm. This results in a cylinder velocity of 0.00986 m/s, which translates to an installation time of 48.8 seconds.

$$s = \frac{0.00986 m/s}{0.481 m} = 48.8\ seconds$$

3.3.6 Second Bore Size

The smaller bore size is determined to be 25 mm. The other cylinder, which is first in the hydraulic system is found by area calculations and shown in Table 14.

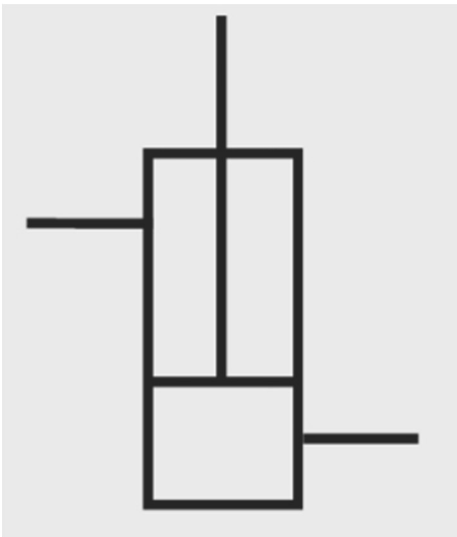
Table 14: Second Bore Size

Rod Ø(mm)	Bore 1 (mm)	Area inn 1 (mm^2)	Area out 1 (mm^2)	Bore 2 (mm)	Area inn 2 (mm^2)	Area out 2 (mm^2)
10	35.0	962.1	491	25	491	177

3.3.7 Hydraulic Cylinders 2

In the system there are two cylinders with different bore areas that work parallelly. The cylinders are welded to Cylinder Fasting Plates and are structurally self-carrying. See Figure 72 for potential cylinder design The rods are inside the cylinder AOGV, and therefore there is no way to track the location visually. Another addition to the system is a position sensor in the HC. In Table 15 below the HC properties are granted.

Table 15: Hydraulic Cylinder Configuration

Category	Selection	Characteristics
Location	Fastened to CFP	The HC is to be fastened on the CFP.
Fastening method	Weld	HC is welded on the CFP.
Static structural	Self-carrying	HC carries self-weight.
Seals	Standard	Because there is air pressure at HC head standard seals for hydraulic oil is used.
Stroke length	500 mm	Covers the necessary 477.6 mm with 22.4 mm extra for adjustments.
Bore Sizes	35 mm	
	25 mm	
Position sensor	Necessary for rod positional awareness.	
Schematic	 *Figure 71: Hydraulic Cylinder Schematic*	

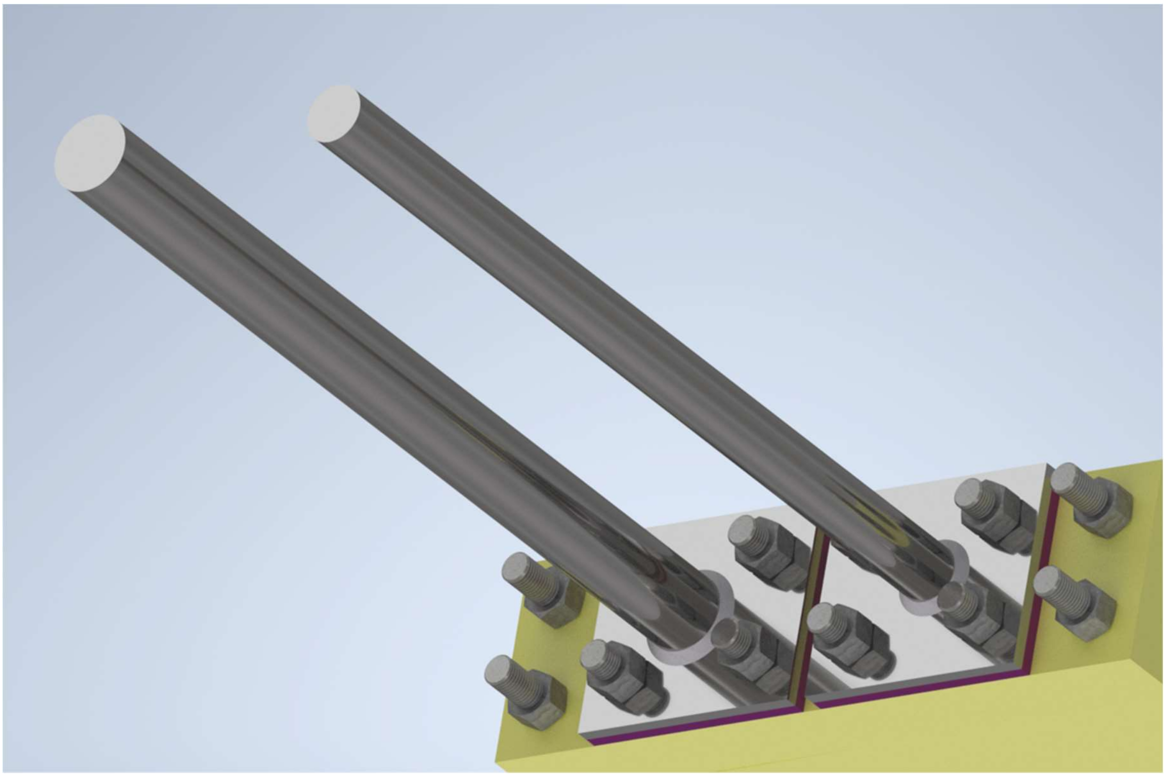


Figure 72: Hydraulic Cylinder Configuration

3.3.8 Running Pressures of the Cylinders

The pressure necessary to insert the cylinders is based on the force to counteract the pressure and friction force. Using the linear force $F_{Pressure}$ from (3.2.14), the frictional force from 3.1.2 and formula *(12)*, the necessary pressure in the 25 mm cylinder is.

$$P_{25_{insert_{min}}} = \frac{F_{pressure}}{A_{25_{bore}}} + \frac{F_{friction}}{A_{25_{bore}}} = 1.38\ MPa + 0.18\ MPa = 1.56\ MPa.$$

$$P_{35_{insert_{min}}} = \frac{F_{pressure}}{A_{35_{bore}}} + \frac{F_{friction}}{A_{35_{bore}}} = 0.705\ MPa + 0.0919\ MPa = 0.797\ MPa.$$

3.3.9 Impact Pressures of the Cylinders

The release pressure valves that open when the isolation plate touches the inside of the AOGV. This are fined tuned valves that open when a pressure of excess pf the working pressure is recognized. A 50 N force on the bottom of the AOGV translates to the following pressures for the 35 mm and 25 mm cylinder sizes using formula *(12)*.

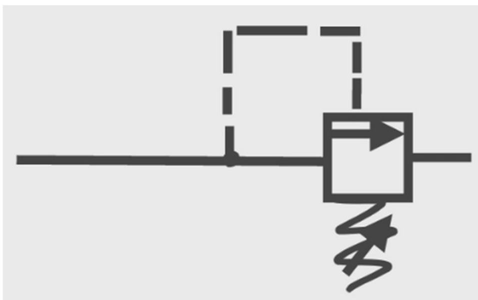
$$P_{25_{reliefValve}} = 0.102\ MPa$$
$$P_{35_{reliefValve}} = 0.0520\ MPa$$

3.3.10 Pressure Relief Valves 3

From (2.2.6) we know that the adjustable pressure relief vale opens at a selected pressure. The opening pressure of the pressure relief valve equals the running pressures (3.4.8) and the maximum contact pressures (3.4.9). In table 16 key release valve specifications are given.

$$P_{25_{reliefValve}} = 1.56\ MPa + 0.102\ MPa = \ 1.662\ MPa$$
$$P_{35_{reliefValve}} = 0.797\ MPa + 0.0520\ MPa = \ 0.849\ MPa$$

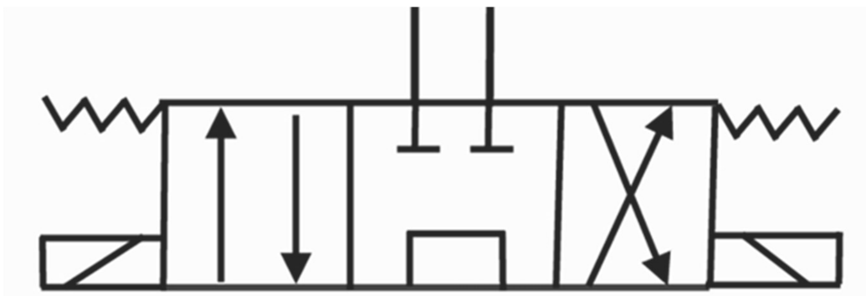
Table 16: Relief Valve Configuration

Category	**Selection**	**Characteristics**
Relief pressure for 25 mm bore HC	1.662 MPa	Valves are set to relief the pressure when the spade is in contact with the internal surface.
Relief pressure for 35 mm bore HC	0.849 MPa	
Flow rate compatibility	0.29 l/min	Flow delivered from the pump
Schematic	 *Figure 73: Adjustable Pressure Relief Valve for Limiting HC Pressure*	

3.3.11 Directional Control Valve 4

As stated in (2.2.5) the Direction Control Valve is needed to deliver flow from the pump to the HCs. There are numerus options on the market and the type selected is dependent on the nature of the movement of the cylinder. One key attribute of the hydraulic system is that the operator has manual control. Manual controlled directional control valves with springs is chosen. The cylinders that are used are double acting, so the valves must have the ability to deliver flow to both ends of the cylinder. Table 17 shows the essential specifications of the systems directional control valve.

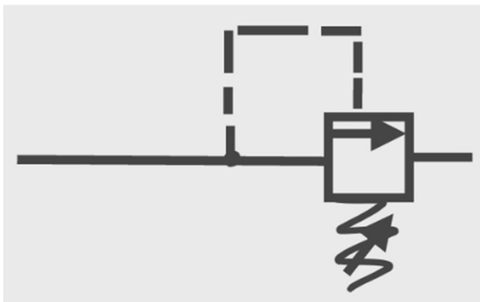
Table 17: Control Valve Configuration

Category	Selection	Characteristics
Flow rate compatibility	0.29 l/min	Flow delivered from the pump
Type	“4/3” Double acting	Deliver flows in both directions, with three settings.
Manual control	Spring activated lever	No flow when static, and flow regulation.
Pressure Regulation	Pressure compensated	From (2.2.5) a pressure compensated Valve ensures steady follow in the system.
Schematic	 *Figure 74: Directional Control Valve Schematic*	

3.3.12 System Pressure Valve 5

The system pressure valve limits the system pressure to limit the risk of explosions in the system. The pressure limit in the cylinder for inserting the rods is 1.56 MPa. Another valve will limit the pressure on each cylinder and will be closer to the pressure limit. The system pressure valve therefore will open at a pressure of 2.0 MPa. This is a pressure-limit well below max pressures of most standard hydraulic components. Table 18 includes crucial system pressure valve selections.

Table 18: System Pressure Valve Configuration

Category	Selection	Characteristics
Flow rate compatibility	0.29 l/min	Flow delivered from the pump
Release pressure	2.0 MPa	Mechanism to limit risk of an explosion
Schematic	 *Figure 75: Adjustable Relief Valve for System Pressure*	

3.3.13 Gauges and Other Sensory Equipment 6

For any hydraulic systems sensory control is necessary to be informed on key statistics. Pressure gauges capture the internal pressure in the system and are a key part in situational understanding. A schematic of a gauge is shown in Figure 23.

3.4 Mechanical drawings

3.4.1 Isolating Tool

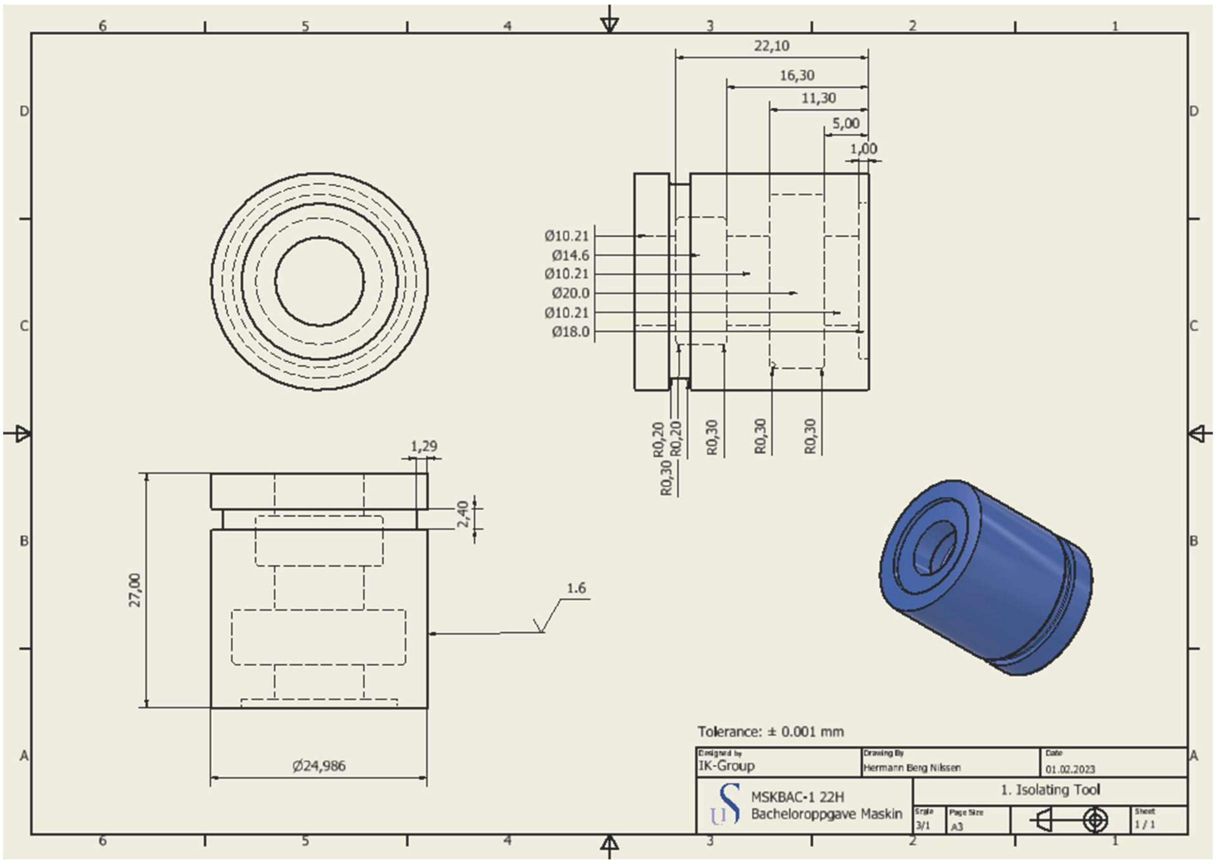


Figure 76: Isolating Tool Drawing

Table 19: Isolating Tool Specifications

Category	Selection	Characteristics
Material	Steel type: P355NL2	High strength material used for the existing TEL.
Method of Production	CNC Machining Lathe	A high level of precision for the internal parts of the IT.
	Filing	Reducing hard edges
	Buffing	Smooth edges and surface finish.
Mass	70 grams	Found from CAD
Surface Roughness	1.6	In Table 8 (2.1.8) the surface roughness from various production methods. The surface form the lathe might be sufficient but filing and buffing is also done to ensure sufficient surface roughness.

3.4.2 Isolating Tool Lid

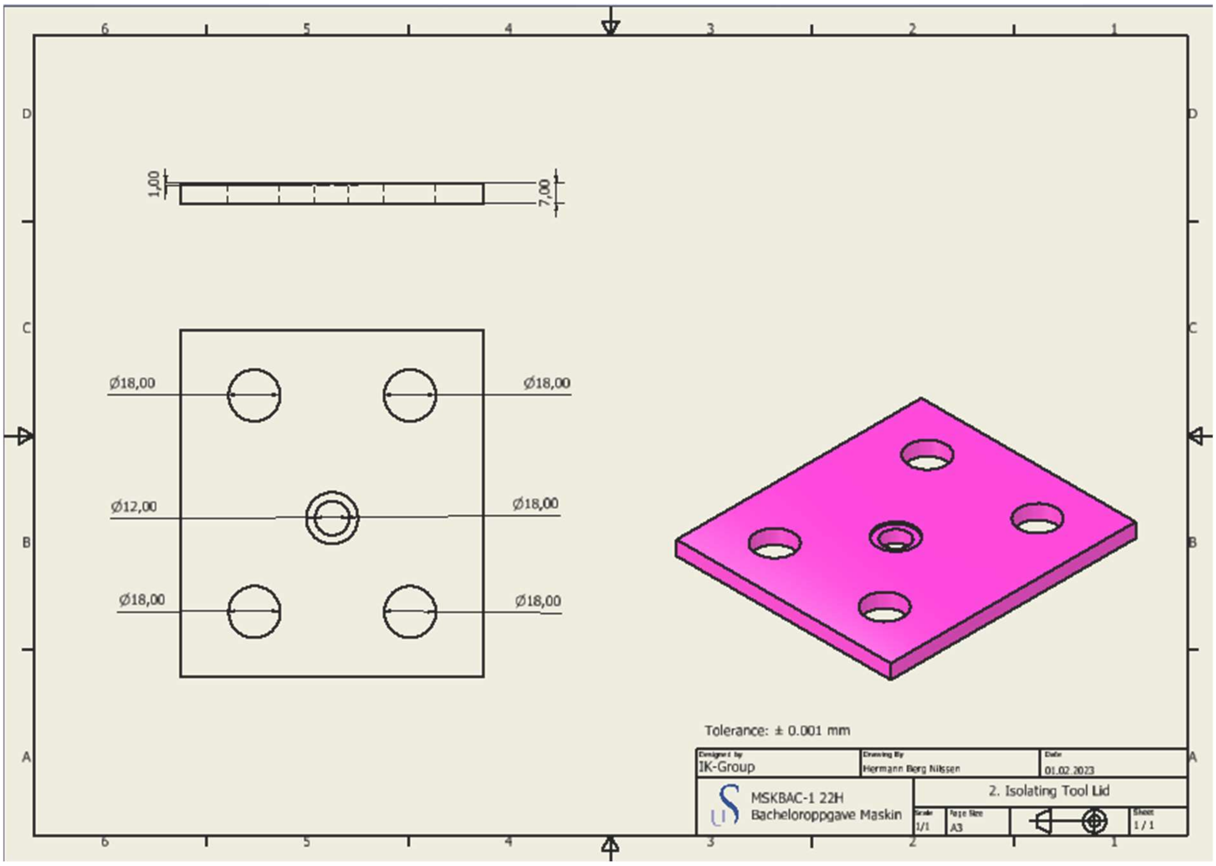


Figure 77: Isolating Tool Lid Drawing

Table 20: Isolation Tool Lid Specifications

Category	Selection	Characteristics
Material	Steel type: P355NL2	High strength material used for the existing TEL.
Method of Production	Flame cutting	Cutting the outer dimensions of a 7 mm thick plate.
	Milling	For holes at the centre of the part.
	Filing	Reducing hard edges
Mass	629 grams	Found from CAD

3.4.3 Cylinder Fastening Plate

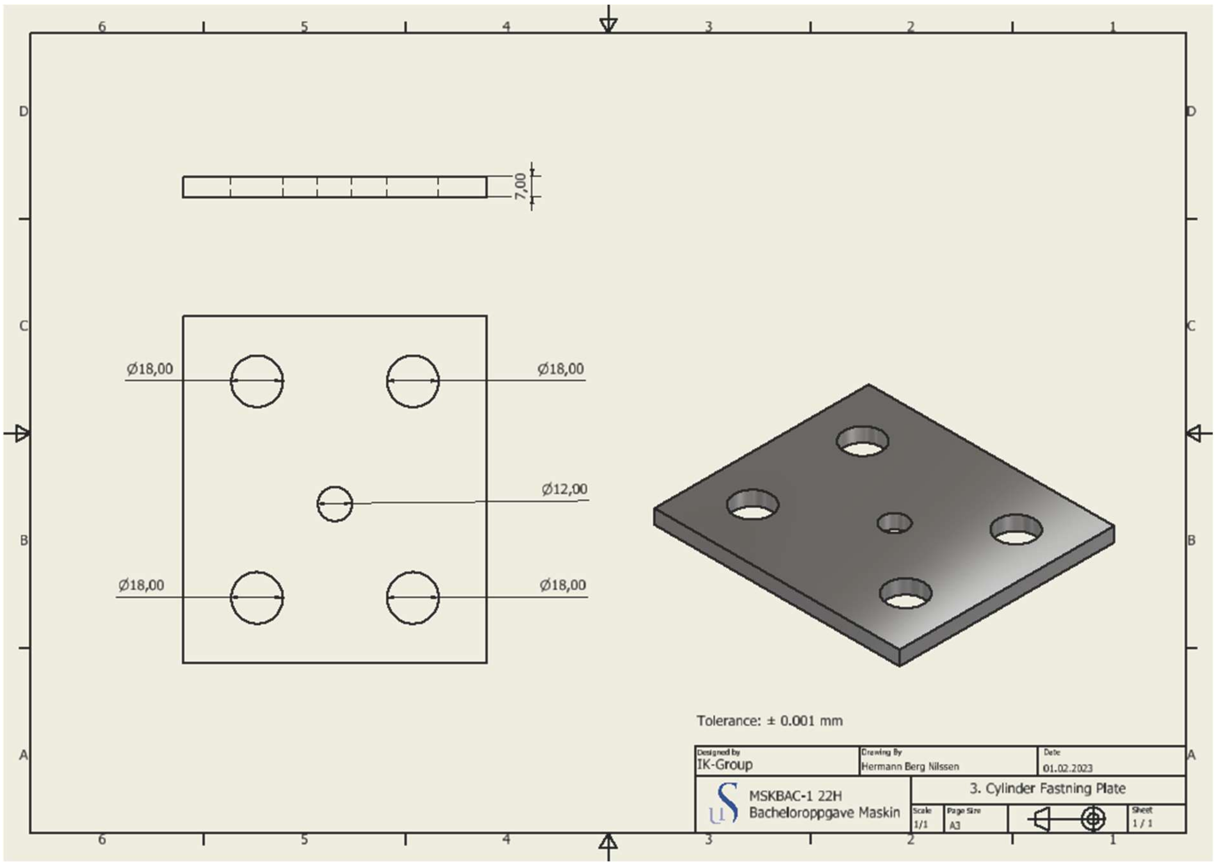


Figure 78: Cylinder Fasting Plate Drawing

Table 21: Cylinder Fasting Plate Specifications

Category	Selection	Characteristics
Material	Steel type: P355NL2	High strength material used for the existing TEL.
Method of Production	Flame cutting	Cutting the outer dimensions of a 7 mm thick plate.
	Milling	For holes at the centre of the part.
	Filing	Reducing hard edges
Mass	630 grams	Found from CAD

3.4.4 Top Envelope Lid Modifications

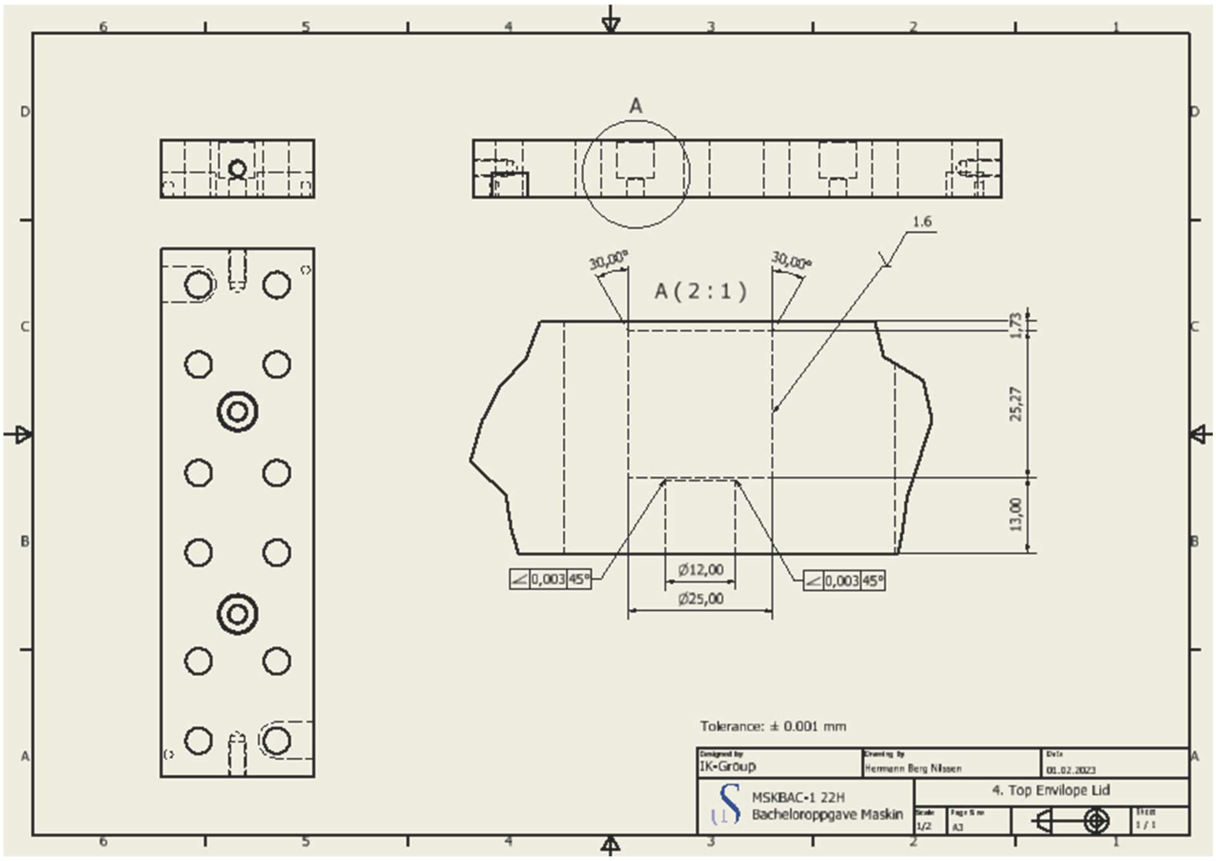


Figure 79: Top Envelope Lid Modifications drawing

Table 22: Top Envelope Lid Modification Specifications

Category	Selection	Characteristics
Material	Steel type: P355NL2	High strength material used for the existing TEL.
Method for Production of TEL modification	Milling	For holes at the centre of the part
	Filing	Reducing hard edges
Mass	10.6 kg	Found from CAD
Surface Roughness	1.6	In Table 8 (2.1.8) the surface roughness from various production methods. The surface form the lathe might be sufficient but filing and buffing is also done to ensure sufficient surface roughness.

3.4.5 Drawing of Isolating Fastening Tool for Hydraulics

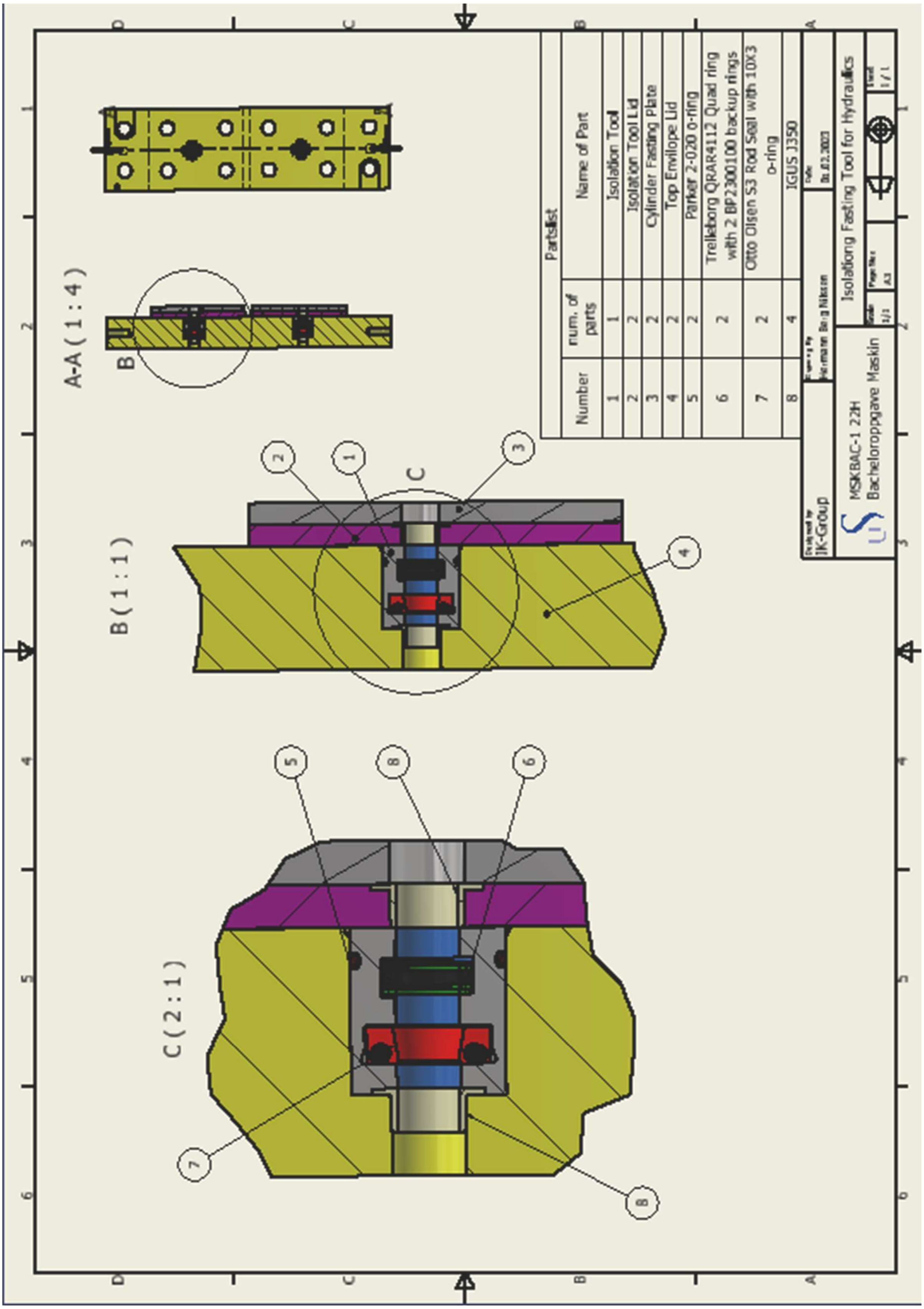


Figure 80: Isolating Fasting Tool for Hydraulics Assembly Drawing

Conclusion and Discussion

The general objective of the thesis is to implement a hydraulic system to insert the Isolation Spade in the 06” AOGV. An Isolation Fasting Tool for Hydraulics is designed to isolate the AOGV and facilitate for easy HC use. Quad Ring seals, S3 rod seals and O-rings are implemented in the fastening tool design, resulting in an atmospheric pressure environment on the HC opening.

The new seal design also overcomes the need for seal activation by bolts. This reduces the space necessity on the upper surface of the Top Envelope Lid. This makes the design a space efficient tool for hydraulic cylinder fasting. Calculations on rods and bearings has been conducted, with the results clearly indicating sufficient structural integrity of the system.

Hydraulic systems with a flow divider and area flow correlation have been discussed. The latter system is broken down in parameterized parts. The systems are designed for a simple insertion that stops automatically when inserted by a pressure sensitive system.

Further work before the systems is integrated in the AOGV lineup include mechanical testing of the Isolation Fasting Tool for Hydraulics. This could be done without a hydraulic cylinder initially. Flow and pressures rates of the system are lower than the standard of the industry. Specially ordered hydraulic components that comply to the granted specification would have to be ordered for a full-scale test.

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5. Appendix

A Trelleborg Quad Ring® Catalog

[52, s.1]

Applications

FIELDS OF APPLICATION

Quad-Ring® can be used for a wide range of different applications. It is used predominantly for dynamic sealing functions. Its use is always limited by the pressure to be sealed and the velocity.

Dynamic applications:

- For sealing of reciprocating pistons, rods, plungers, etc.
- For sealing oscillating, rotating or spiral movements on shafts, spindles, rotary transmission leadthroughs, etc.

Static applications:

- As a radial-static seal, e.g. for bushings, covers, pipes, etc.
- As an axial-static seal, e.g. for flanges, plates, caps, etc.
- As an energizer element for elastomer energized hydraulic seals where there is a risk of the O-Ring twisting.

QUAD-RING® FOR ROTARY APPLICATIONS

In applications with small cyclic periods of activity, Quad-Ring® can also be used for sealing rotating shafts. The following points according to the rotary seal principle should be observed:

The rotary seal principle is based on the fact that an elongated elastomer ring contracts when heated (Joule effect). With the normal design criteria, the seal ring inside diameter d_1 will be slightly smaller than the shaft diameter, and the heat generated by friction would cause the ring to contract even more. This results in a higher pressure on the rotating shaft so that a lubricating film is prevented from forming under the seal and even higher friction occurs. The result would be increased wear and a premature failure of the seal.

Using the rotary seal principle, this is prevented by the seal ring being selected so that its inside diameter is approximately 2 to 5% larger than the shaft diameter to be sealed. The installation in the groove means that the seal ring is compressed radially and is pressed against the shaft by the groove diameter. The seal ring is thus slightly corrugated in the groove, a fact which helps to improve the lubrication.

The rotary seal principle can be neglected at peripheral speeds of less than 0.5 m/s.

When using the Quad-Ring® as a rotary seal, the use of a suitable surface coating is recommended. Please note the information given in our Seal-Glide® brochure or contact your local Customer Solutions Center for further details.

TECHNICAL DATA

Quad-Ring® can be used for a wide range of applications. The choice of a suitable material is determined by the temperature, pressure and media. In order to assess the suitability of Quad-Ring® as a sealing element for a given application, the interaction of all the operating parameters have to be taken into consideration.

Working pressure, dynamic application:

Reciprocating
up to 5 MPa (50 bar) without Back-up Ring
up to 30 MPa (300 bar) with Back-up Ring

Rotating
up to 1 MPa (10 bar) without Back-up Ring
up to 3 MPa (30 bar) with Back-up Ring

Working pressure, static application:
up to 5 MPa (50 bar) without Back-up Ring
up to 40 MPa (400 bar) with Back-up Ring

Please note the permissible extrusion gaps, see Table 4.

Speed:

Reciprocating:		up to 0.5 m/s
Rotating:	briefly	up to 2.0 m/s

Operating temperature range:
depending on material and media resistance, for:

General applications, NBR:	-30 °C to + 100 °C
General applications, FKM:	-18 °C to + 200 °C

When assessing the application criteria, the transient peak and continuous operating temperature and the cyclic duration factor must be taken into consideration. For rotating applications, the increases in temperature due to frictional heat must also be taken into account.

Media:
Trelleborg Sealing Solutions offers a range of materials to seal against practically all liquids, gases and chemicals. Please note when selecting the material for your application, refer to our material selection tools such as our Chemical Compatibility Guide.

[52, s.5]

Materials

The available standard elastomer materials are shown in Table 1.

If no particular specifications are given for the material, NBR (Nitrile Butadiene Rubber) in 70 Shore A will be supplied.

Table 1: Standard materials for Quad-Ring®

Material-Type	NBR Nitrile Butadiene Rubber	FKM Fluorocarbon Rubber
Material code	N7004	V7002
Hardness Shore A (±5)	70	70
Color	Black	Black
Operating temperature range (°C)	-30 °C to +100 °C	-18 °C to +200 °C
Description	Standard material for hydraulics and pneumatics. Mineral oil-based hydraulic fluids, animal and vegetable oils and fats, aliphatic hydrocarbons, silicone oils and greases, water up to +80 °C	Mineral oils and greases, flame retardant liquids, aliphatic, aromatic and chlorinated hydrocarbons, petrol, 99 octane petrol, diesel fuels, silicone oils and greases

Other materials and specialized compounds are available on request.

When used in a real-world setting, conditions and media may vary, affecting material properties or operating temperature ranges. In application testing should be carried out to verify performance.

Characteristics and Inspection of Elastomers

HARDNESS

One of the most frequently named properties regarding polymer materials is hardness. Even so, the values can be quite misleading.

Hardness is the resistance of a body against penetration of an even harder body of a standard shape at a defined pressure.

There are two procedures for hardness tests regarding test samples and finished parts made out of elastomer materials:

1. Shore A / D in accordance with ISO 868 / ISO 48-4 ASTM D 2240 - Measurement for test samples
2. Durometer IRHD (International Rubber Hardness Degree) in accordance with ISO 48 / ASTM 1414 and 1415 - Measurement of test samples and finished parts

The hardness scale has a range of 0 (softest) to 100 (hardest). The measured values depend on the elastic qualities of the elastomers, especially on the tensile strength.

The test should be carried out at temperatures of 23 ±2 °C (73.4 ±2 °F) - no earlier than 16 hours after the last vulcanization process. If other temperatures are being used, this should be mentioned in the test report.

Tests should only be carried out with samples that have not been previously stressed mechanically.

HARDNESS TESTS IN ACCORDANCE WITH SHORE A/D

The hardness test device for Shore A (indentor with pyramid base) is a sensible choice for hardness range 10 to 90. Samples with a larger hardness should be tested with the Shore D device (indentor with spike).

Test specimen:
Diameter min. 30 mm (1.181 inch)
Thickness min. 6 mm (0.240 inch)
Upper and lower sides smooth and flat

When thin material is being tested, it can be layered to ensure a minimum sample thickness is achieved, up to a maximum of 3 layers. All layers must be at minimum 2 mm (0.080 inch) thick.

The measurement is done at five different places at a defined distance and time.

[52, s.6]

Installation Recommendation / Quad-Ring® with Back-up Ring for Reciprocating Applications - Internal

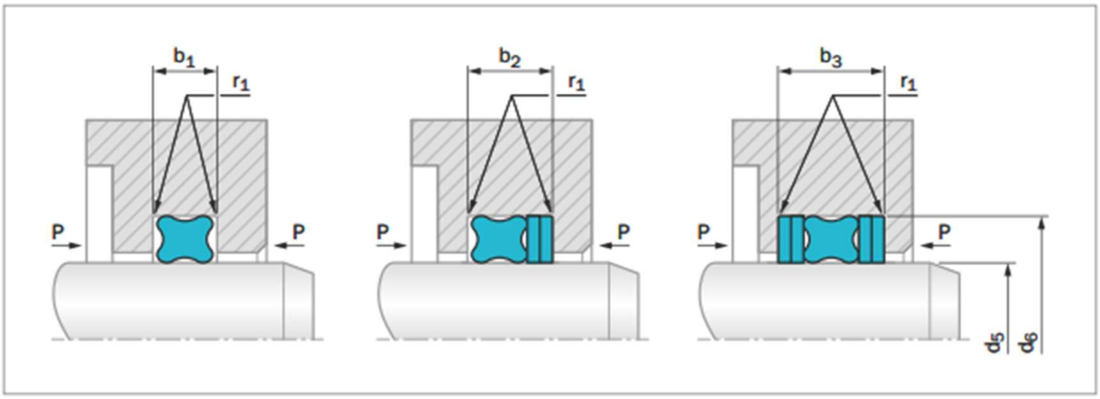


Figure 17: Installation Drawing

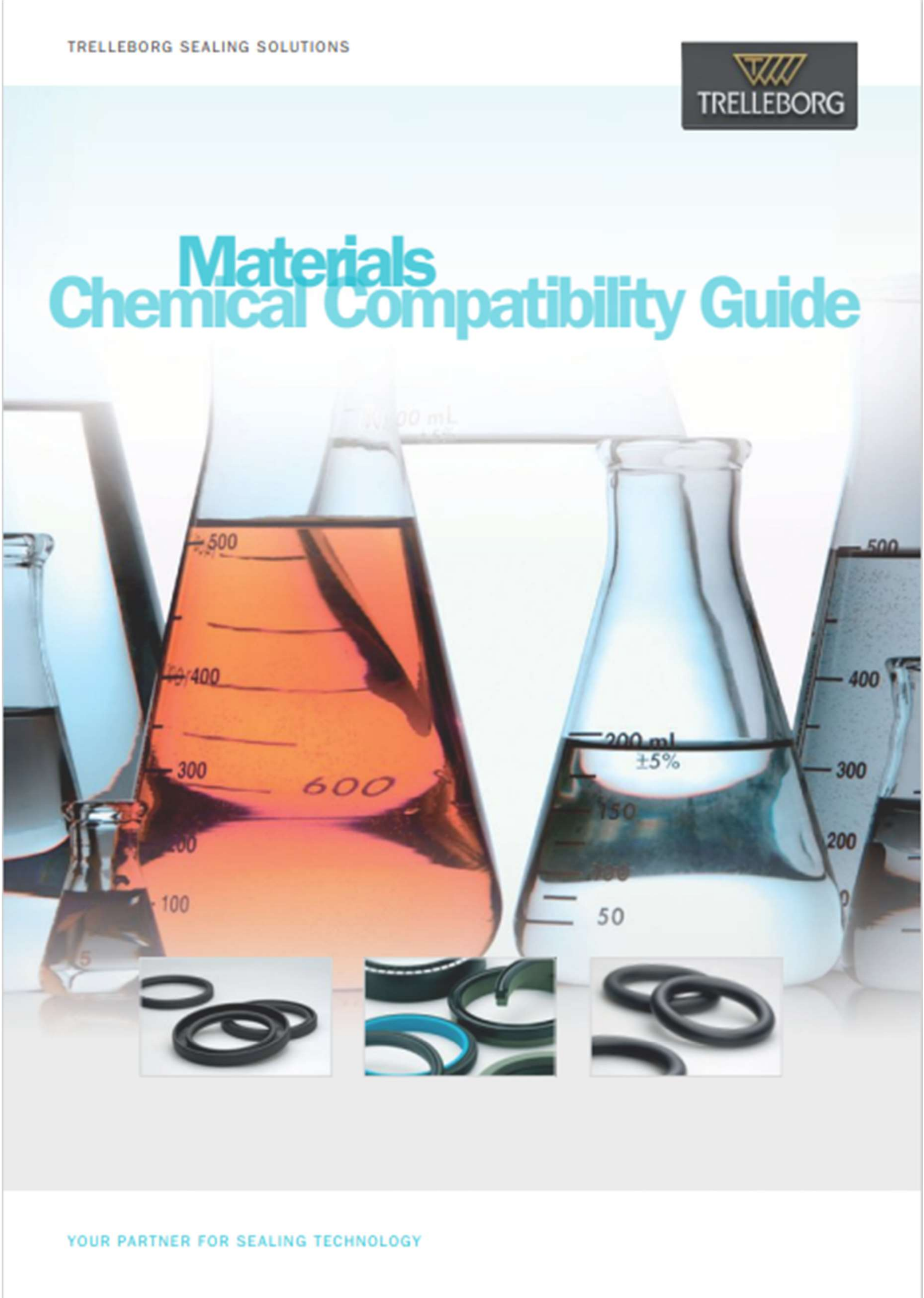
The following data regarding Back-up Rings and groove widths b_2 and b_3 are examples only. The use and the suitability of a Back-up Ring type, as well as the design of the appropriate groove widths b_2 and b_3, should be verified and adapted based on the application. For further information, please refer to the O-Ring and Back-up Rings Catalog.

Table 7: TSS Part No. / Installation Dimensions

Rod	Quad-Ring® Part No.	Dimensions	Back-up Ring, Spiral Part No.	Groove-Ø	Groove Width			Radius1)
d_5 f7				d_6 h9	b_1 +0.2	b_2 +0.2	b_3 +0.2	r_1
4.0	QRAR04008	4.47x1.78	BP1500040	7.0	2.0	3.4	4.8	0.2
5.0	QRAR04009	5.28x1.78	BP1500050	8.0	2.0	3.4	4.8	0.2
6.0	QRAR04010	6.07x1.78	BP1500060	9.0	2.0	3.4	4.8	0.2
8.0	QRAR4012A	8.20x1.78	BP1500080	11.0	2.0	3.4	4.8	0.2
10.0	QRAR4111A	10.20x2.62	BP2300100	14.6	3.0	4.4	5.8	0.3
12.0	QRAR04112	12.37x2.62	BP2300120	16.6	3.0	4.4	5.8	0.3
14.0	QRAR04113	13.94x2.62	BP2300140	18.6	3.0	4.4	5.8	0.3
15.0	QRAR4114A	14.70x2.62	BP2300150	19.6	3.0	4.4	5.8	0.3
16.0	QRAR4115A	16.20x2.62	BP2300160	20.6	3.0	4.4	5.8	0.3
18.0	QRAR4210A	18.20x3.53	BP32D0180	24.4	4.0	5.4	6.8	0.4
20.0	QRAR04211	20.22x3.53	BP32D0200	26.4	4.0	5.4	6.8	0.4
22.0	QRAR04212	21.83x3.53	BP32D0220	28.4	4.0	5.4	6.8	0.4
25.0	QRAR04214	24.99x3.53	BP32D0250	31.4	4.0	5.4	6.8	0.4
28.0	QRAR04216	28.17x3.53	BP32D0280	34.4	4.0	5.4	6.8	0.4
30.0	QRAR04217	29.74x3.53	BP32D0300	36.4	4.0	5.4	6.8	0.4
32.0	QRAR04218	31.34x3.53	BP32D0320	38.4	4.0	5.4	6.8	0.4
35.0	QRAR04220	34.52x3.53	BP32D0350	41.4	4.0	5.4	6.8	0.4
36.0	QRAR04221	36.09x3.53	BP32D0360	42.4	4.0	5.4	6.8	0.4
40.0	QRAR04326	40.64x5.33	BP4900400	49.8	6.0	7.7	9.4	0.4
42.0	QRAR04326	40.64x5.33	BP4900420	51.8	6.0	7.7	9.4	0.4

[52, s.21]

B Trelleborg Material Compatibility Guide



[53, s.1]

Chemical Compatibility Guide

N

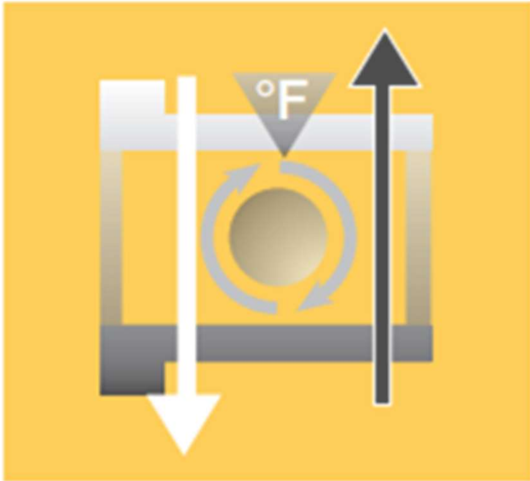
Chemical	ACM	AU	CR	EPDM	FFKM (Isolast®)	FKM	FKM Resifluor™ 500	FVMQ	HNBR	NBR	VMQ
Naphtha	B	B	U	U	A	A	A	B	U	U	U
Naphthalene	U	U	U	U	A	A	A	B	U	U	U
Naphthenic Acid	-	-	U	U	A	A	A	A	B	B	-
Naphtolen ZD	U	-	U	U	A	A	A	-	B	B	U
Natural Gas	A	B	B	U	A	A	A	A	A	A	A
Neats Foot Oil	A	A	U	B	A	A	A	A	A	A	B
Neon Gas	A	A	A	A	A	A	A	A	A	A	A
Nickel Acetate	U	U	B	A	A	U	A	U	B	B	U
Nickel Chloride	C	C	B	A	A	A	A	A	A	A	A
Nickel Nitrate	-	-	A	A	A	A	A	-	A	A	A
Nickel Sulfate	U	C	A	A	A	A	A	A	A	A	A
Nitrating Acids	U	U	U	A	A	U	A	U	U	U	U
Nitric Acid, concentrated	U	U	U	U	A	B	A	U	U	U	U
Nitric Acid, fuming	U	U	U	U	A	B	A	U	U	U	U
Nitro Benzene	U	U	U	U	A	U	A	U	U	U	U
Nitro Glycerin	U	U	C	A	A	A	A	U	U	U	U
Nitro Glycol	U	U	B	A	A	A	A	U	U	U	U
Nitro Methane	U	U	U	B	A	U	A	U	U	U	U
Nitro Propane	U	U	U	B	A	U	A	U	U	U	U
Nitro Toluene	U	U	U	U	A	U	A	U	U	U	U
Nitrogen Gas	A	A	A	A	A	A	A	A	A	A	A
Nitrogen Tetroxide	U	U	U	U	-	U	A	U	U	U	U
Nonanol	-	U	-	A	A	A	A	-	U	U	B
Nut Oil	A	B	B	U	A	A	A	A	A	A	B

O

Chemical	ACM	AU	CR	EPDM	FFKM (Isolast®)	FKM	FKM Resifluor™ 500	FVMQ	HNBR	NBR	VMQ
Octadecane	B	B	B	U	A	A	A	A	A	A	U
Octal	U	B	U	B	A	B	A	C	U	U	C
Octane	U	U	U	U	A	A	A	B	B	B	U
Octanol (Octylalcohol)	U	U	B	A	A	A	A	B	B	B	B

[53, s.27]

C IGUS Iglide® J350 Catalogue



Endurance runner: high dimensional stability at high temperatures

Can be used with various shafts and loads

iglide® J350

When to use it?

- When a wear-resistant bearing for rotational movement at medium and high loads is required
- When a cost-effective plain bearing for high temperatures is required
- When press-fit up to +302°F is necessary
- When the bearing is exposed to shock loading

When not to use?

- When continuous operating temperatures are higher than +356°F
  iglide® X
- When the lowest friction is required
  iglide® J
- When a cost-effective plain bearing with low friction is required
  iglide® D, iglide® R
- For high rotational speeds
  iglide® J

[54, s.1]

iglide® J350

Bearing technology | Plain bearing | iglide® J350

Ø
4 – 60mm
1/8 - 2 in.

Material available as:

Bar stock, round bar
Page 761

Bar stock, plate
Page 783

tribo-tape liner
Page 791

Piston rings
Page 685

Two hole flange bearings
Page 709

Molded special parts
Page 721

igubal® spherical balls
Page 965

Endurance runner: high dimensional stability at high temperatures

Can be used with various shafts and loads

An outstanding plain bearing for rotating applications – and for a wide range of different shaft materials: with iglide® J350 plain bearings, the service life can often be increased for applications between 2 and 7,252psi. In addition, the high temperature resistance makes it a very versatile material.

- Recommended for steel shafts
- Continuous operating temperatures up to +356°F
- Suitable for medium and high loads
- Suitable for rotating applications
- Self-lubricating
- Maintenance-free

Typical application areas

- Automation
- Mechanical engineering
- Automotive
- Glass industry

Available from stock
Detailed information about delivery time online.

Online ordering
Including delivery times, prices, online tools

Descriptive technical specifications			
Wear resistance at +73°F	–	■■■■□	+
Wear resistance at +194°F	–	■■■■■	+
Wear resistance at +302°F	–	■■■■□	+
Low coefficient of friction	–	■■■■□	+
Low moisture absorption	–	■■■■■	+
Wear resistance under water	–	■■■■□	+
High media resistance	–	■■■■□	+
Resistant to edge pressures	–	■■■□□	+
Suitable for shock and impact loads	–	■■■□□	+
Resistant to dirt	–	■■■□□	+

Online product finder
www.igus.com/iglidefinder

Online service life calculation
www.igus.com/iglide-expert

[54 s. 252]

Technical data

iglide® J350

General properties			Testing method
Density	g/cm³	1.44	
Color		yellow	
Max. moisture absorption at +73°F and 50% r.h.	% weight	0.3	DIN 53495
Max. moisture absorption	% weight	1.6	
Coefficient of friction, dynamic, against steel	μ	0.10 – 0.20	
pv value, max. (dry)	psi · fpm	13,000	
Mechanical properties			
Flexural modulus	psi	290,075	DIN 53457
Flexural strength at +68°F	psi	7,977	DIN 53452
Compressive strength	psi	8,702	
Max. recommended surface pressure (+68°F)	psi	8,702	
Shore D hardness		80	DIN 53505
Physical and thermal properties			
Max. application temperature long-term	°F	+356	
Max. application temperature short-term	°F	+428	
Min. application temperature	°F	-148	
Thermal conductivity	W/m · K	0.24	ASTM C 177
Coefficient of thermal expansion (at +73°F)	K[illegible] · 10[illegible]	7	DIN 53752
Electrical properties			
Specific contact resistance	Ωcm	> 10[illegible]	DIN IEC 93
Surface resistance	Ω	> 10[illegible]	DIN 53482

Table 01: Material properties

-148°F up to +356°F

8,702psi

iglide® J350 blends universally good wear resistance, flexibility and temperature resistance into a very versatile iglide® material with a broad application spectrum.

Moisture absorption
The moisture absorption of iglide® J350 is low and can be ignored when using standard plain bearings. Even when saturated with water, iglide® J350 does not absorb more than 1.6% weight of water (by weight).

Vacuum
In vacuum, any present moisture is released as vapor. Use in vacuum is only possible with dehumidified iglide® J350 bearings.

Radiation resistance
Plain bearings made from iglide® J350 are resistant up to a radiation intensity of $2 \cdot 10^2$Gy.

Resistance to weathering
iglide® J350 plain bearings are continuously resistant to weathering. The material properties are only slightly affected. Possible discolorations are only superficial.

Mechanical properties
With increasing temperatures, the compressive strength of iglide® J350 plain bearings decreases. Diagram 02 shows this inverse relationship. The maximum recommended surface pressure is a mechanical material parameter. No conclusions regarding the tribological properties can be drawn from this. iglide® J350 plain bearings are adequate for medium and high loads. Diagram 03 shows the elastic deformation of iglide® J350 at radial loads. It shows the material behavior submitted to a short-term load. The ambient temperatures are only noticeable at 8,702psi.

▶ Surface pressure, **Page 50**

Permissible surface speeds
iglide® J350 plain bearings are suitable for low and medium speeds in rotating and oscillating applications. The wear rates, however, are much better in the case of rotating applications. iglide® J350 is also excellent for linear movements.

▶ Surface speed, **Page 44**

[54, s. 253]

D Flow Divider Bi-directional Series MTDA

BUCHER hydraulics

Flow Divider

Bi-directional
Series MTDA

- robust, simple and reliable
- easy to service
- flows can be split or merged with accuracy (divide/combine functions).
- the flow division ratio can be altered to suit customer requirements.

6 Ordering code

6.1 MTDA08 / MTDA16

M T D A 0 8 - 0 0 4 M 3 0 / [ ]

Flow divider — MT

Bi-directional — D

Port thread — A

Nominal size — 08, 16

Control flow range [l/min]

MTDA08		MTDA08		MTDA16	
004	= 2-4	025	= 12-25	100	= 35-100
006	= 3-6	032	= 16-32	120	= 40-120
008	= 4-8	050	= 25-50	160	= 50-160
012	= 6-12	075	= 37-75	200	= 60-200
016	= 8-16	100	= 50-100	250	= 75-250

Port threads — Metric = M; Inch = R

Division ratio, see section 6.4 (no valid for division ratio 1:1)

Option (to be inserted by the factory)

[55, s.1,6]

E Resato P80 Datasheet

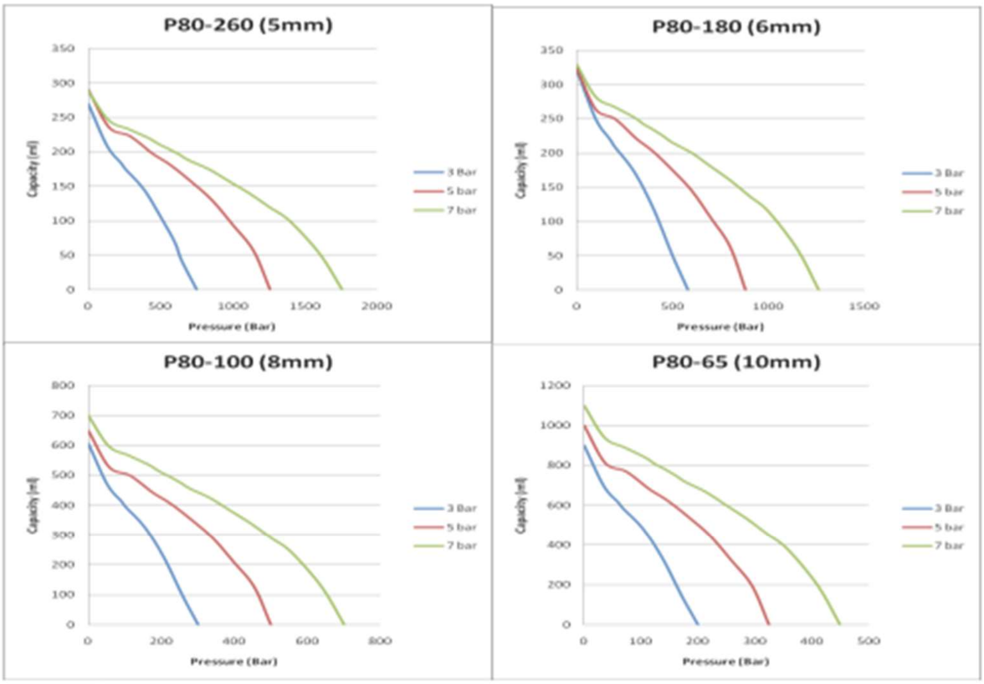
Resato
HIGH PRESSURE TECHNOLOGY

THE INNOVATORS
OUR COMMITMENT, YOUR SUCCESS!

Type table & flow curves Resato P80 series

P80 Pump type	Max. outlet Actual ratio	Volume pressure bar/psi	Max. per cycle cc	Flow l/min	Connections Suction	 Discharge
P80-65-1/U/N	65	455 / 6600	1.57	1.10	3/8" BSP	M16x1.5*
P80-100-1/U/N	100	700 / 10150	1.01	0.70	3/8" BSP	M16x1.5*
P80-180-1/U/N	180	1260 / 18275	0.57	0.33	3/8" BSP	M16x1.5*
P80-260-1/U/N	260	1820 / 26400	0.39	0.29	3/8" BSP	M16x1.5*

* High-pressure connections with conical sealing.



RESATO INTERNATIONAL BV
1e Energieweg 13 NL-9301 LK Roden – Holland Phone: +31 (0) 50 501 6877 E-mail: hpsales@resato.com
P.O. Box 30 NL-9300 AA Roden - Holland Fax: +31 (0) 50 501 2402 Internet: www.resato.com

[56, s.2]

F Parker O-Ring Handbook

[57, s.1]

2.1 Definition of design

O-rings can be used in static applications such as covers or pins. If the machine parts being sealed move relative to one another, the O-ring acts as a dynamic seal.
The seal type designs are defined as follows:

- When a **female gland** is cut in the outside machine part, it is regarded as a "rod seal".
- When a **male gland** is cut in the inside machine part, it is regarded as a "piston seal".
- When there is **axial compression**, it is regarded as a "face seal".

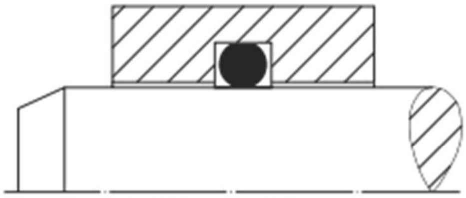


Fig. 2.1 Female gland ("rod seal"): O-ring with radial compression

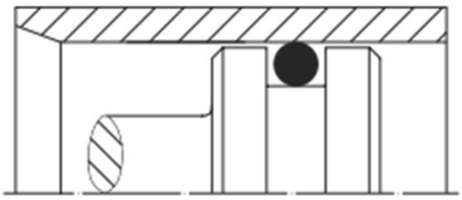


Fig. 2.2 Male gland ("piston seal"): O-ring with radial compression

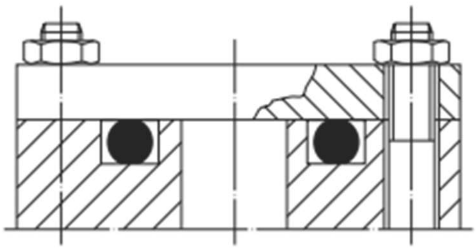


Fig. 2.3 Face seal: O-ring with axial compression

2.2 Static seals

O-rings are particularly suitable for use in static applications because the applied deformation produces a seal effect which increases with increasing system pressure. The effectiveness of the seal is influenced by both a correctly-designed gland and the choice of compound.

In all applications, it is correct to select an O-ring with the **largest possible cross-section** allowed by the design constraints. In general it can be said that an O-ring circumference should not be stretched more than 6 % nor compressed more than 1 to 3 % when installed (measured by the inner diameter of the O-ring).
The hardness of an O-ring is selected according to the applied pressure, the tolerances (and related gap widths) and the surface finish of the elements to be sealed.
The elastic elongation of metallic materials (e.g. lids, flanges, cylinder walls or screw joints) under pressure must be considered. Due to this, an oversized clearance gap can occur, which the O-ring must bridge.
The type of sealing point also depends on the mechanical processing. Economic processing methods can necessitate higher tolerances and therefore larger clearance gaps. Back-up rings can be used to protect radially-deformed O-rings against expected extrusion.

The Parbak® back-up ring size list gives the relevant continuous elastomer back-up rings suiting O-ring sizes 2-004 to 2-475 (for more information, see section "Parbak® back-up rings"). For silicone compounds, the allowable gap size is 50 % of that normally allowed with other elastomer compounds, as these materials have very poor extrusion and tear resistance properties.
High pulsating pressure and the resulting relative movement of machine parts promote are the causes of wear in an O-ring. Additionally, elastic elongation of the individual components can result in a larger sealing gap. If signs of wear are found on a static seal, we recommend improving the surface finish or using Ultrathan® (polyurethane) O-rings (see catalogue "Pneumatic Seals" or "Hydraulic Seals").

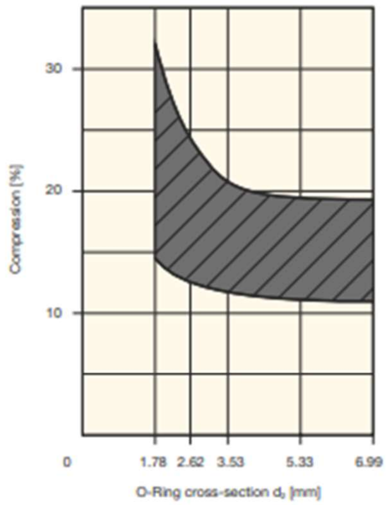


Fig. 2.4 Acceptable compression, dependent upon cross-section d_2 for static seal

[57, s.9]

3.1 Static seals

3.1.1 Compression and design dimensions

Piston seal – radial compression
O-ring assembly in inside element

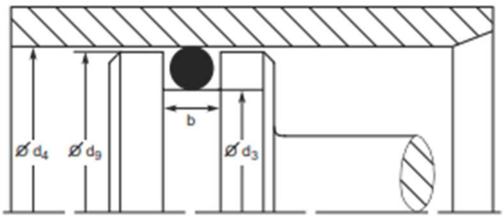


Fig. 3.1 Piston seal – radial compression

Rod seal – radial compression
O-ring assembly in outside element

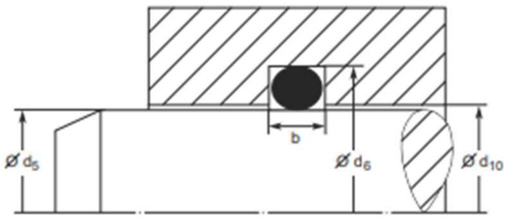


Fig. 3.2 Rod seal – radial compression

Flange seal – axial compression
Pressure from inside: O-ring outside diameter must be compressed

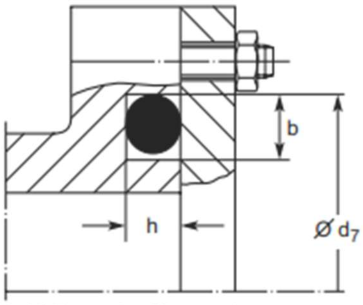


Fig. 3.3 Flange seal – axial compression

Pressure from outside: O-ring inside diameter must be stretched

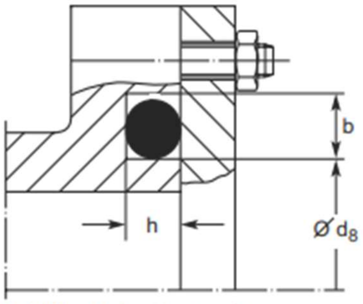
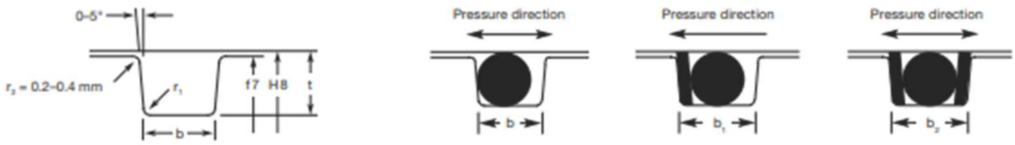


Fig. 3.4 Flange seal – axial compression



Cross-section d_2 [mm]	Gland depth t [mm]	Compres-sion [mm]	Compression [%]	Groove width b without back-up ring [mm]	Groove width b_1 one back-up ring [mm]	Groove width b_2 two back-up rings [mm]	Radius r_1 [mm]
1.78 $^{\pm0.08}$	1.40	0.26 - 0.58	15 - 31	2.40 - 2.60	3.50 - 3.70	4.60 - 4.80	0.20 - 0.40
2.62 $^{\pm0.09}$	2.20	0.26 - 0.64	10 - 23	3.60 - 3.80	4.70 - 4.90	5.80 - 6.00	0.20 - 0.40
3.53 $^{\pm0.10}$	2.90	0.40 - 0.85	11 - 23	4.80 - 5.00	5.80 - 6.00	6.80 - 7.00	0.40 - 0.80
5.33 $^{\pm0.13}$	4.50	0.57 - 1.08	11 - 20	7.20 - 7.40	8.70 - 8.90	10.20 - 10.40	0.40 - 0.80
6.99 $^{\pm0.15}$	5.90	0.80 - 1.35	11 - 19	9.60 - 9.80	12.00 - 12.20	14.40 - 10.60	0.40 - 0.80

Tab. 3.1 Design dimensions for O-rings – static seal

[57, s.21]

3 Design recommendations

3.1.2 Piston seal static

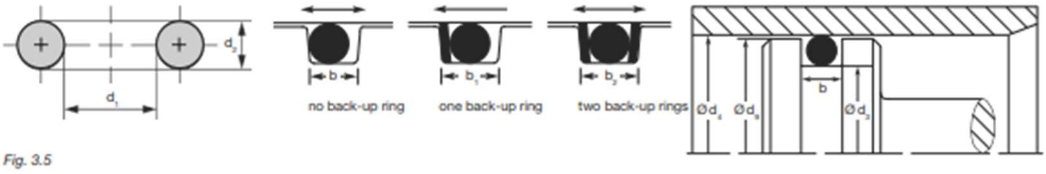


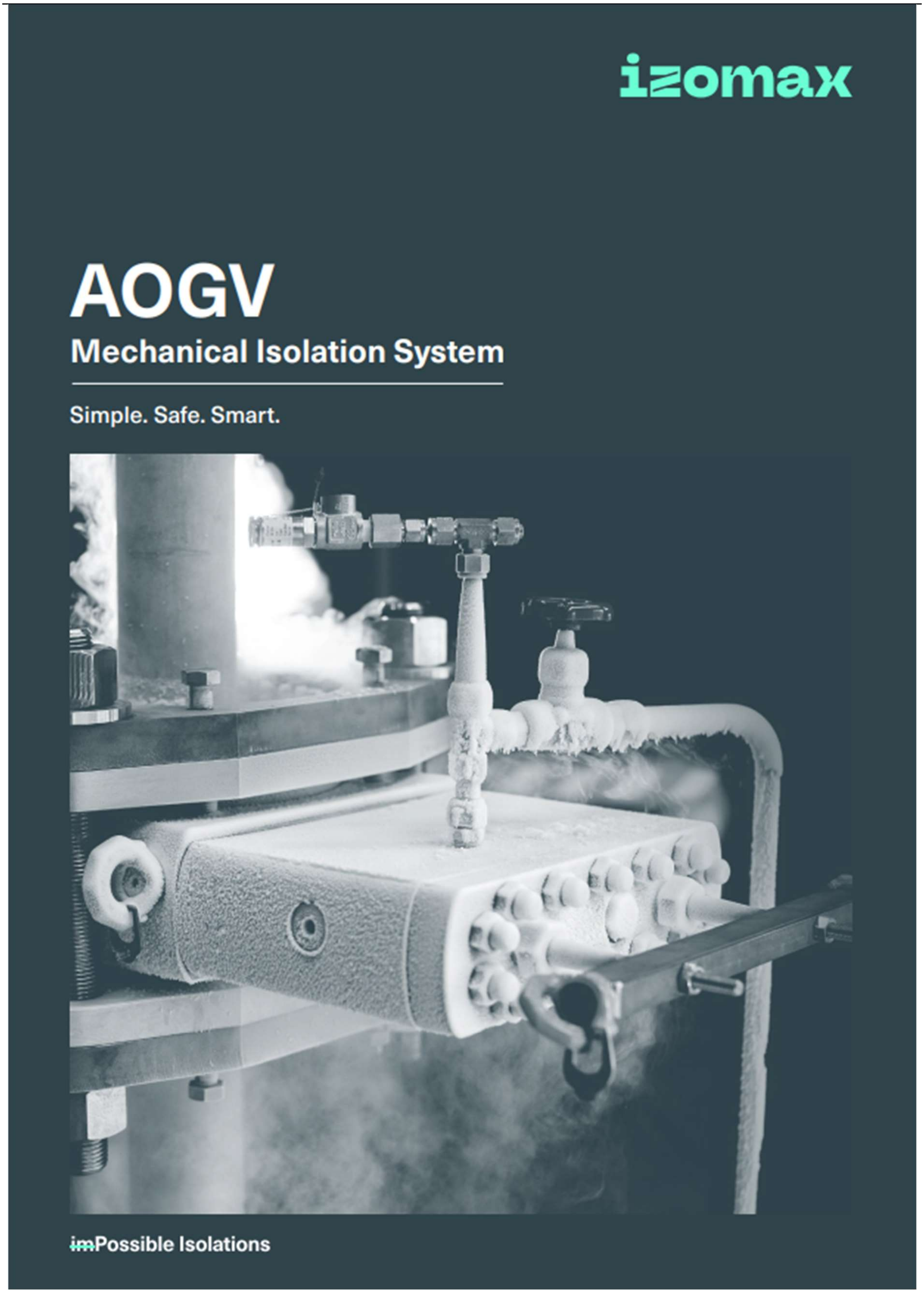
Fig. 3.5

Parker no.	d_1	d_2	b +0.2 0	b_1 +0.2 0	b_2 +0.2 0	d_3 h9	d_4 H8	d_9 f7
2-006	2.9	1.78	2.4	3.5	4.6	2.9	5.5	5.5
5-190	3.35	1.78	2.4	3.5	4.6	3.4	6	6
2-007	3.68	1.78	2.4	3.5	4.6	3.9	6.6	6.5
2-008	4.47	1.78	2.4	3.5	4.6	4.4	7	7
5-581	4.9	1.9	2.4	3.5	4.6	5	7.8	7.8
2-009	5.28	1.78	2.4	3.5	4.6	5.4	8	8
5-582	5.7	1.9	2.4	3.5	4.6	5.7	8.5	8.5
2-010	6.07	1.78	2.4	3.5	4.6	6.4	9	9
5-052	6.86	1.78	2.4	3.5	4.6	7.4	10	10
2-011	7.65	1.78	2.4	3.5	4.6	8.4	11	11
5-612	8.74	1.78	2.4	3.5	4.6	8.9	11.5	11.5
2-012	9.25	1.78	2.4	3.5	4.6	9.4	12	12
5-212	9.75	1.78	2.4	3.5	4.6	10.4	13	13
2-013	10.82	1.78	2.4	3.5	4.6	10.9	13.5	13.5
5-613	11.1	1.78	2.4	3.5	4.6	11.4	14	14
2-014	12.42	1.78	2.4	3.5	4.6	12.4	15	15
6-129	13.29	1.78	2.4	3.5	4.6	13.4	16	16
2-016	15.6	1.78	2.4	3.5	4.6	15.4	18	18
2-017	17.17	1.78	2.4	3.5	4.6	17.4	20	20
2-018	18.77	1.78	2.4	3.5	4.6	18.4	21	21
2-019	20.35	1.78	2.4	3.5	4.6	20.4	23	23
2-020	21.95	1.78	2.4	3.5	4.6	22.4	25	25
2-021	23.52	1.78	2.4	3.5	4.6	23.4	26	26
2-022	25.12	1.78	2.4	3.5	4.6	25.4	28	28
2-023	26.7	1.78	2.4	3.5	4.6	27.4	30	30
2-024	28.3	1.78	2.4	3.5	4.6	29.4	32	32
2-025	29.87	1.78	2.4	3.5	4.6	30.4	33	33
2-026	31.47	1.78	2.4	3.5	4.6	32.4	35	35
2-027	33.05	1.78	2.4	3.5	4.6	33.4	36	36
2-028	34.65	1.78	2.4	3.5	4.6	35.4	38	38
6-154	36.3	1.78	2.4	3.5	4.6	37.4	40	40
2-030	41	1.78	2.4	3.5	4.6	42.4	45	45
2-031	44.17	1.78	2.4	3.5	4.6	45.4	48	48
2-032	47.35	1.78	2.4	3.5	4.6	47.4	50	50
2-033	50.52	1.78	2.4	3.5	4.6	52.4	55	55
2-034	53.7	1.78	2.4	3.5	4.6	55.4	58	58
2-035	56.87	1.78	2.4	3.5	4.6	57.4	60	60
2-036	60.08	1.78	2.4	3.5	4.6	60.4	63	63

Parker no.	d_1	d_2	b +0.2 0	b_1 +0.2 0	b_2 +0.2 0	d_3 h9	d_4 H8	d_9 f7
2-037	63.22	1.78	2.4	3.5	4.6	65.4	68	68
2-038	66.4	1.78	2.4	3.5	4.6	67.4	70	70
2-039	69.57	1.78	2.4	3.5	4.6	69.4	72	72
2-040	72.75	1.78	2.4	3.5	4.6	75.4	78	78
2-041	75.92	1.78	2.4	3.5	4.6	77.4	80	80
2-042	82.27	1.78	2.4	3.5	4.6	82.4	85	85
2-043	88.62	1.78	2.4	3.5	4.6	89.4	92	92
2-044	94.97	1.78	2.4	3.5	4.6	97.4	100	100
2-045	101.32	1.78	2.4	3.5	4.6	102.4	105	105
2-046	107.67	1.78	2.4	3.5	4.6	107.4	110	110
2-047	114.02	1.78	2.4	3.5	4.6	117.4	120	120
2-048	120.37	1.78	2.4	3.5	4.6	122.4	125	125
2-049	126.72	1.78	2.4	3.5	4.6	127.4	130	130
2-050	133.07	1.78	2.4	3.5	4.6	135.4	138	138
2-110	9.19	2.62	3.6	4.7	5.8	9.3	13.5	13.5
5-614	9.93	2.62	3.6	4.7	5.8	9.8	14	14
2-111	10.77	2.62	3.6	4.7	5.8	10.8	15	15
5-615	11.91	2.62	3.6	4.7	5.8	11.8	16	16
2-112	12.37	2.62	3.6	4.7	5.8	12.8	17	17
5-616	13.11	2.62	3.6	4.7	5.8	13.3	17.5	17.5
2-113	13.94	2.62	3.6	4.7	5.8	14	18	18
5-239	14.48	2.69	3.6	4.7	5.8	14.6	19	19
5-243	15.34	2.62	3.6	4.7	5.8	15.8	20	20
2-114	15.54	2.62	3.6	4.7	5.8	16.8	21	21
2-115	17.12	2.62	3.6	4.7	5.8	17.8	22	22
5-256	17.96	2.62	3.6	4.7	5.8	18.8	23	23
2-116	18.72	2.62	3.6	4.7	5.8	19.8	24	24
2-117	203.29	2.62	3.6	4.7	5.8	20.8	25	25
2-118	21.89	2.62	3.6	4.7	5.8	21.8	26	26
2-119	23.47	2.62	3.6	4.7	5.8	23.8	28	28
2-120	25.07	2.62	3.6	4.7	5.8	25.8	30	30
2-121	26.64	2.62	3.6	4.7	5.8	27.8	32	32
2-122	28.24	2.62	3.6	4.7	5.8	28.8	33	33
2-123	29.82	2.62	3.6	4.7	5.8	30.8	35	35
2-124	31.42	2.62	3.6	4.7	5.8	31.8	36	36
2-125	32.99	2.62	3.6	4.7	5.8	33.8	38	38
2-126	34.59	2.62	3.6	4.7	5.8	35.8	40	40
2-127	36.17	2.62	3.6	4.7	5.8	36.8	41	41

[57, s.22]

G AOGV Mechanical Isolation System Brochure



[58, s.1]

AOGV

Mechanical Isolation Tool

Developed by Izomax, the AOGV Mechanical Isolation Tool can insert and remove an isolation spade on any live flange pair to create a zero-energy zone where inspection, modification and maintenance work can be performed safely and efficiently whilst production is maintained.

APPLICATIONS INCLUDE:

- On-site repair of valves and valve replacement
- Repair and modifications of parts of process facilities
- Retrospective installation of equipment
- Allow safe entry of vessels for maintenance, repair or cleaning

The AOGV tool has been approved and deployed by oil and gas supermajors and multinational NOCs across upstream, downstream, and integrated gas assets.

[58, s.2]

How does the AOGV work?

The Izomax AOGV Mechanical Isolation System is assembled in sections over any live flange pair, upstream or downstream of the pipework or equipment requiring intervention.

Sealing on the flange circumference and the flange bolt holes, the pipe pressure and inventory is contained within the AOGV housing. The flanges are separated, the gasket removed, and a spade is inserted for isolation purposes.

The AOGV tool is then disassembled and moved to the next location leaving the flange pair and pipework in the same condition as it was pre-intervention.

Positive Isolation

Positive Isolation is regarded as the most secure method for energy isolation and the use of the AOGV facilitates:

1. Spool removal: removal of a piped section or spool piece and blanking the live end – also called 'air gapping'.
2. Blind isolation: insertion of a blind between flanges (spade).

AOGV TIMELINE

2016

2016
AOGV concept

2017 MAR
First use of a prototype AOGV

2018 MAY
AOGV case jointly presented at OTC Houston by end client and Izomax

2018 AUG
First High pressure AOGV job successfully executed (150Bar/2175psi)

2019 AUG
AOGV Patent Awarded

2019 SEP
AOGV case jointly presented at Offshore Europe by BP and Izomax

2020 FEB
First AOGV application for a major maintenance contractor

2020 MAY
First frame agreement awarded by NOC for delivery of AOGV technology and services

2020 AUG
Finalist for the Offshore Northern Seas Innovation Award

2020 AUG
First operation of a G2.0 High pressure AOGV (155bar / 2250psi)

2021 MAY
First AOGV Global Technology Release (approval) by a "Supermajor"

2021 AUG
First cryogenic AOGV application

2021 OCT
2021 - AOGV Patent awarded in USA

2021 OCT
AOGV job # 50 Successfully Executed

2022 JAN
Awarded largest AOGV scope to date - 89 valve replacements

2022 MAR
Order received for the first G1.0 butterfly valve AOGV

2022 APR
Qualification of the first Compact Flange AOGV

2022
2023

[58, s.4]

Environmental impact

The AOGV reduces the isolated area, meaning that more of the process inventory is left in the plant, reducing the risk of spill and volume of emissions.

- AOGV enables reduced requirement for drainage, venting, purging and flushing
- Reduces volumes to be gas-freed and flared
- Minimises requirements for storing or transport of drained fluids
- Minimises disposal of unwanted fluids
- Minimises release of Volatile Organic Compounds to the environment

Minimise your isolation impact

Passing valves and leaking flanges is a challenge in any process plant. Built-in isolation points can require partial facility shutdowns and the ejection and flushing of large inventory volumes. Typically, this type of work must wait for – or trigger – a full or partial facility shutdown, leading to significant production loss and increased exposure to risk for personnel.

By bringing the isolation point closer to the point of interest, the AOGV reduces the area impacted by the work, negating the likelihood of shutdown and a large ejection of inventory.

The AOGV is designed to ensure facility downtime is kept to a minimum, asset integrity is maintained, and the risk is mitigated to "as low as reasonably practicable".

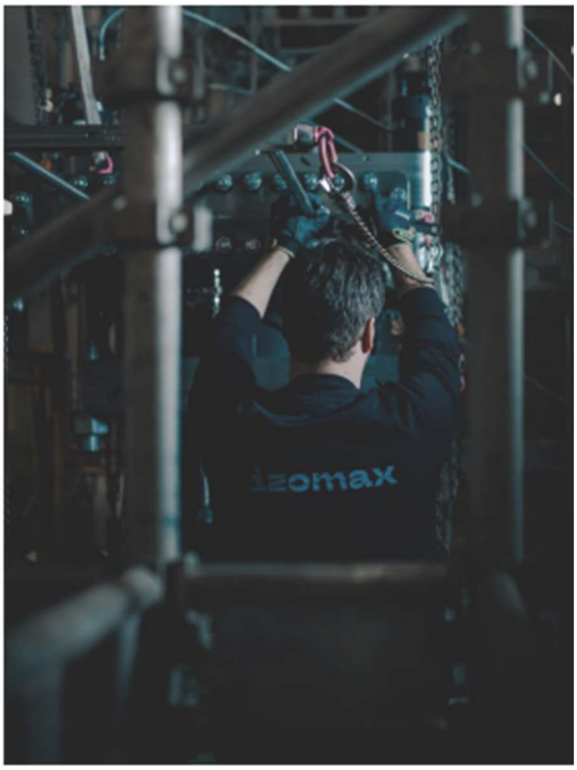
Increases maintenance flexibility

The AOGV can insert an isolation spade at any live flange pair, isolating individual pieces of equipment or sections of the process plant where no other isolation points are available. This makes it possible to execute inspection, modification and maintenance work, as and when needed, without interruption to production.

The AOGV technology provides quantifiable value by reducing the time spent "in-plant" and the area of the facility impacted. Compared to alternatives, the AOGV allows:

- Isolation of individual parts of equipment where no other means are provided or available
- Execution of work outside of a turnaround (TAR), increasing asset uptime (reliability)
- No requirement for "hot work"
- No permanent alteration to the pipework
- Reduction in maintenance schedules by minimising isolation impacts
- Reduction in drainage, venting, purging and flushing time and cost
- Maintained production through simplified isolation

The AOGV is tested to and complies with all relevant regulations and standards – PED2014/68/EU, EN 13445, ASME B31.3, and is CE marked by DNV.

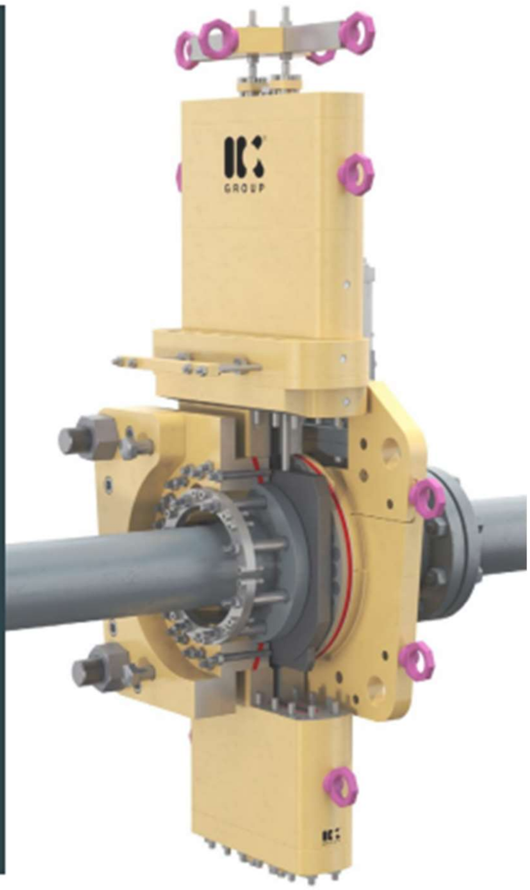


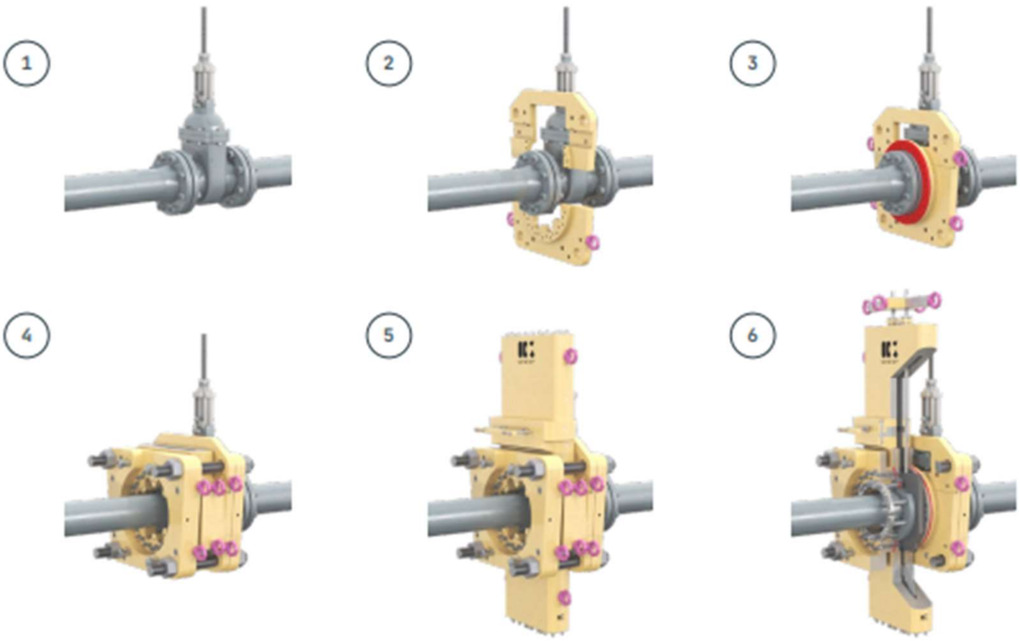
IZOMAX AOGV – MECHANICAL ISOLATION TOOL 5

[58, s.5]

Installation sequence

- Fits on any standard ASME flange
- Leave pipe medium in place
- Suspend the weight of the AOGV and clamp on the flange
- Transfer the compression force from the flange bolts to the AOGV & unbolt the flange using standard tools
- Plug the flange bolt holes
- Separate the pipe flanges and remove the gasket
- Insert a blind spade and compress the flanges to seal
- Perform the required work
- Release and retract the blind spade
- Insert new gasket and compress flanges to seal
- Install flange bolts and torque up flanges to reinstate the system





6 **IZOMAX AOGV** – MECHANICAL ISOLATION TOOL

[58, s.6]

Frequently asked question

1. **What temperature and pressure ranges can the AOGV be used for?**

   The temperature range is from -280 degrees to +400 degree Fahrenheit. Pressures of up to 2900psi have been achieved. Higher pressures are also feasible.

2. **What sizes of pipe and pressure class combinations can the AOGV be used for?**

   We have been focusing on the ASME class 150 & 300 in sizes 1" to 24" but have tools that can handle sizes up to 36" and up to class 2500. The AOGV can also accommodate other flange standards such as DIN, JIS and Compact Flanges. Please see our tool fleet at Izomax.com for availability of off the shelf sizes and class combinations.

3. **How much clearance does the AOGV need on either side of the flange to be able to be installed?**

   As a rule of thumb, for pipework from 1" to 4" the AOGV needs 2" of clearance and from 5" and upwards a 1/2 of the pipe diameter is needed, measured from the bolts and nut side of the flange.

4. **How does the AOGV seal on the circumference of the flange and the bolt holes?**

   The flange seal is pre-energized,and seals directly from the flange circumference to the inside of the AOGV. The bolt holes are also plugged with mechanical plugs bolted to the AOGV kit. The type of seal used is dependent on the application and process inventory but typically elastomer is used to make sure uneven surfaces will be sealed properly.

5. **Can the AOGV be fitted on the flange of a 3-piece ball valve?**

   The AOGV can be fitted on all flanged valve types and nozzles including 3-piece ball valves.

6. **Does the pipework have to bear the weight of the AOGV?**

   For some of the smaller sized AOGV's, the pipe can easily handle the weight. However, it is normal practice to suspend the weight of the AOGV in chain hoists attached to a super-structure or scaffolding above the AOGV.

7. **What about the condition of my flange face?**

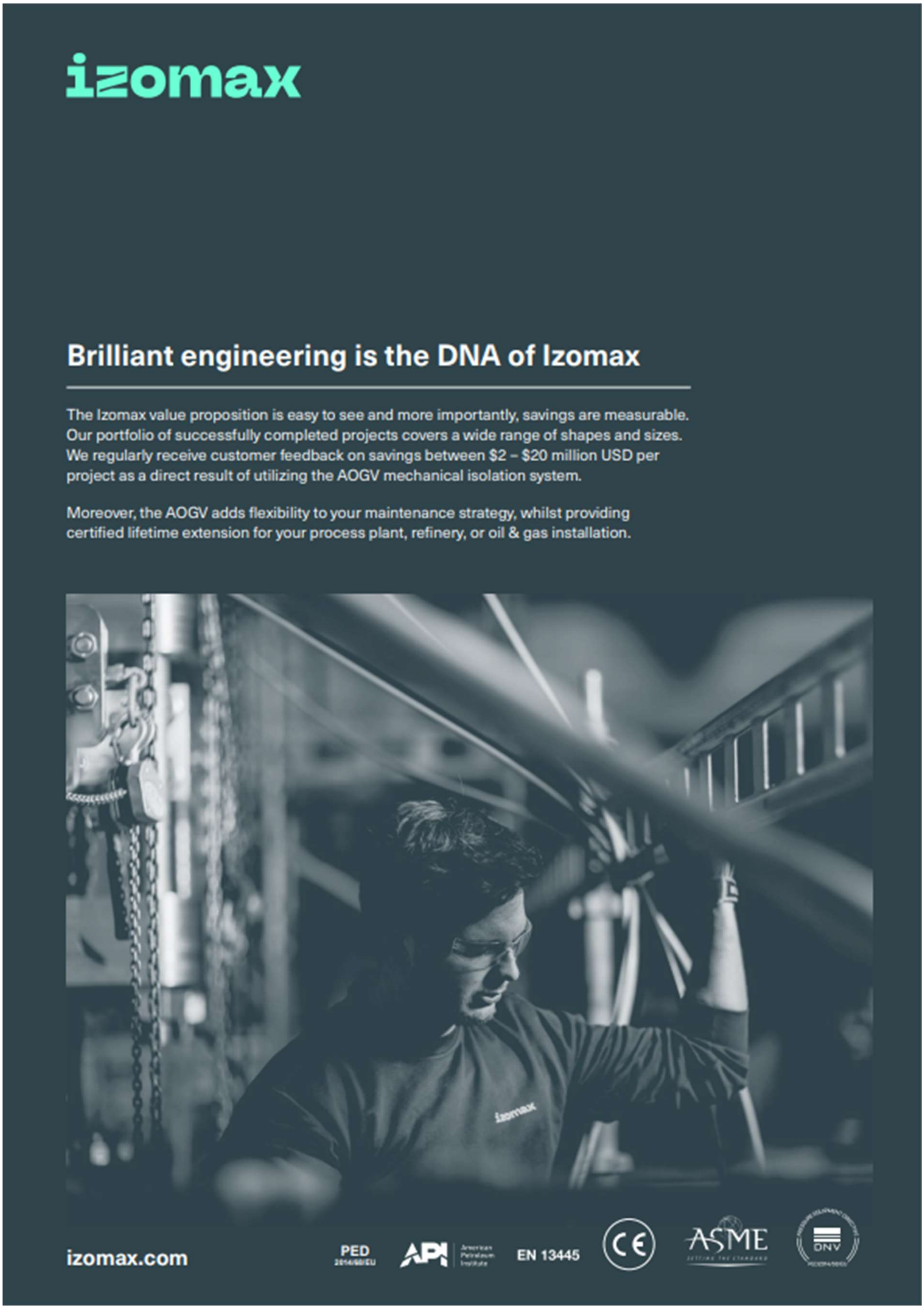
   As part of the AOGV operation, we remove the old seal/gasket at the beginning of the operation and replace it with a new seal/gasket at the end. We have been 100% successful in restoring the flange integrity.

8. **How do you perform the splitting of the flanges after the AOGV has been installed?**

   Either the system pressure is used for splitting the flanges or the pipe is gently moved by use of e.g., chain hoists to pull them apart. The stress tolerances are calculated for the displacements. The dispalcement itself is controlled by gradually releasing the compression exerted by the AOGV.

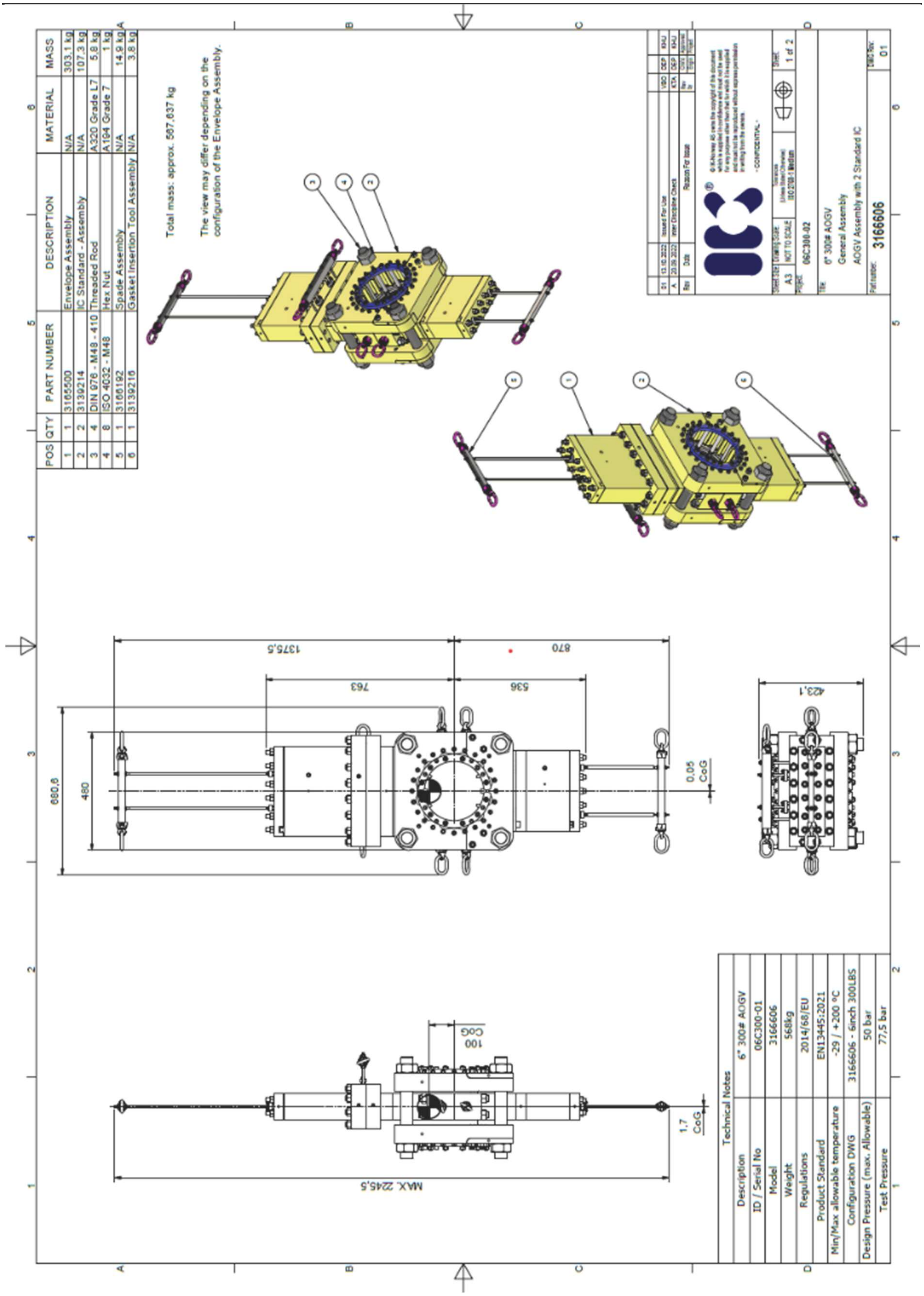
IZOMAX AOGV – MECHANICAL ISOLATION TOOL 7

[58, s.7]


izomax
Brilliant engineering is the DNA of Izomax
The Izomax value proposition is easy to see and more importantly, savings are measurable. Our portfolio of successfully completed projects covers a wide range of shapes and sizes. We regularly receive customer feedback on savings between $2 – $20 million USD per project as a direct result of utilizing the AOGV mechanical isolation system.
Moreover, the AOGV adds flexibility to your maintenance strategy, whilst providing certified lifetime extension for your process plant, refinery, or oil & gas installation.
izomax.com
PED
API
EN 13445
CE
ASME
DNV

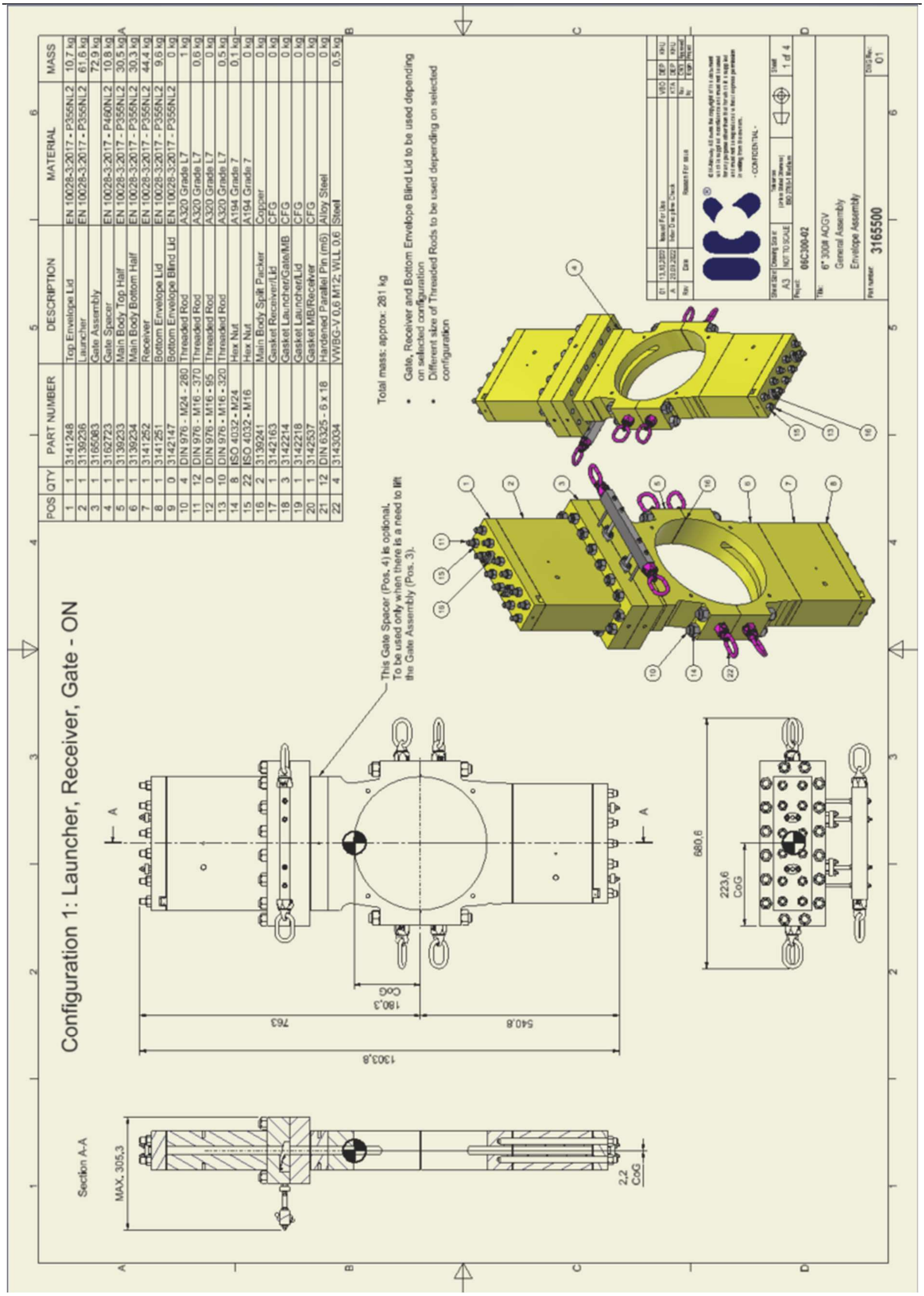
[58, s.8]

H 06” AOGV Assembly Drawing



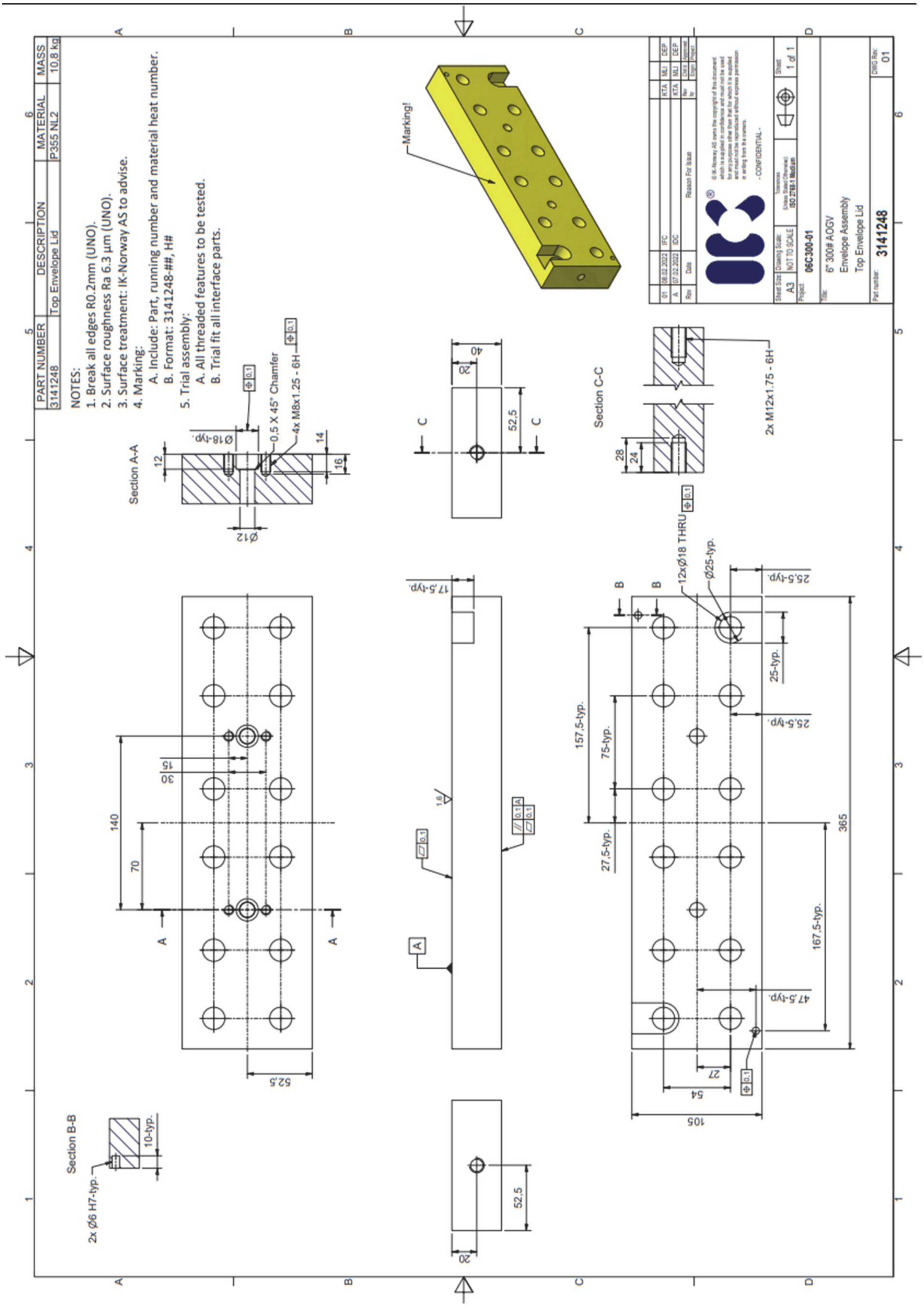
[59]

I Envelope Assembly Drawing



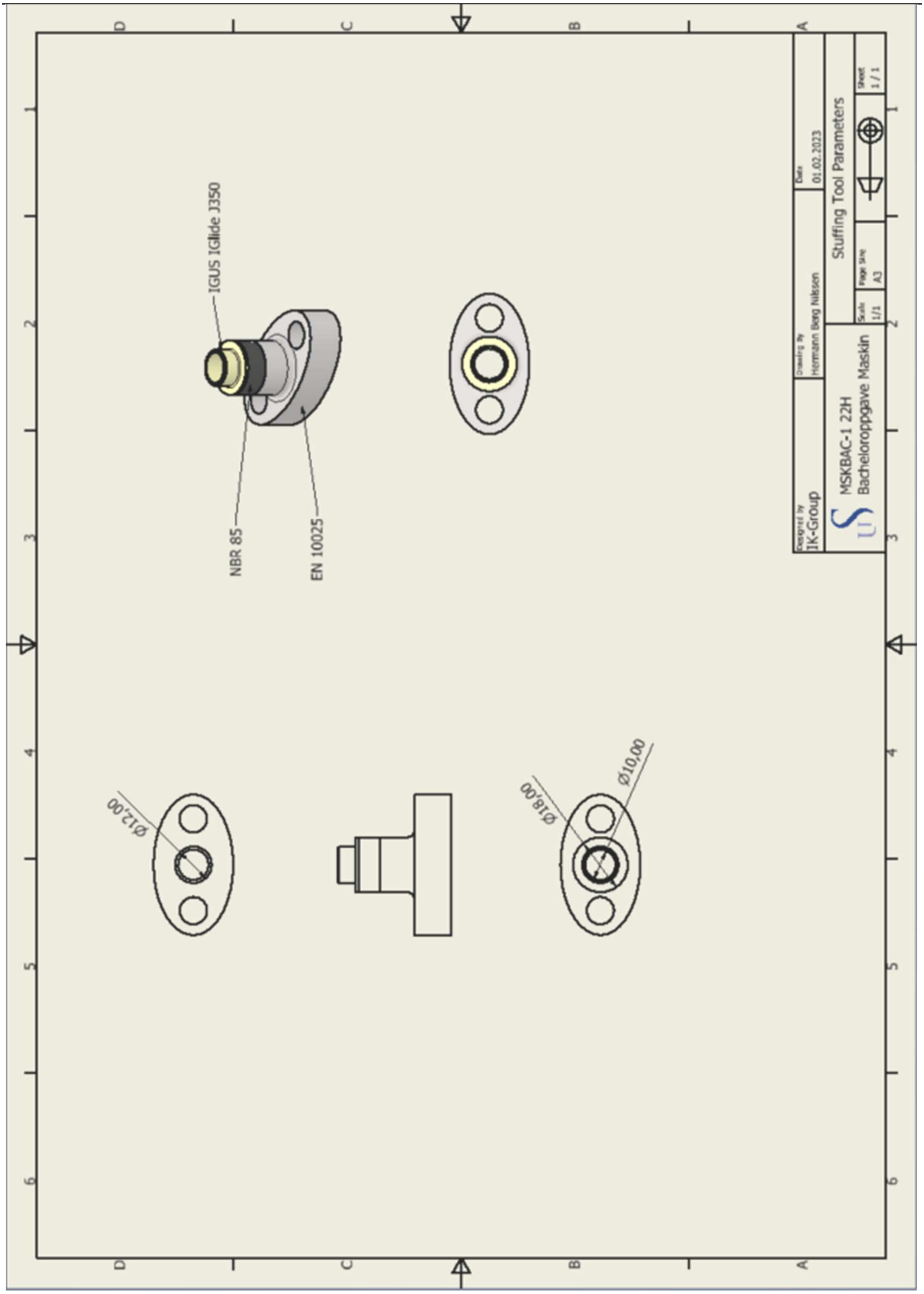
[60]

J Top Envelope Lid Drawing



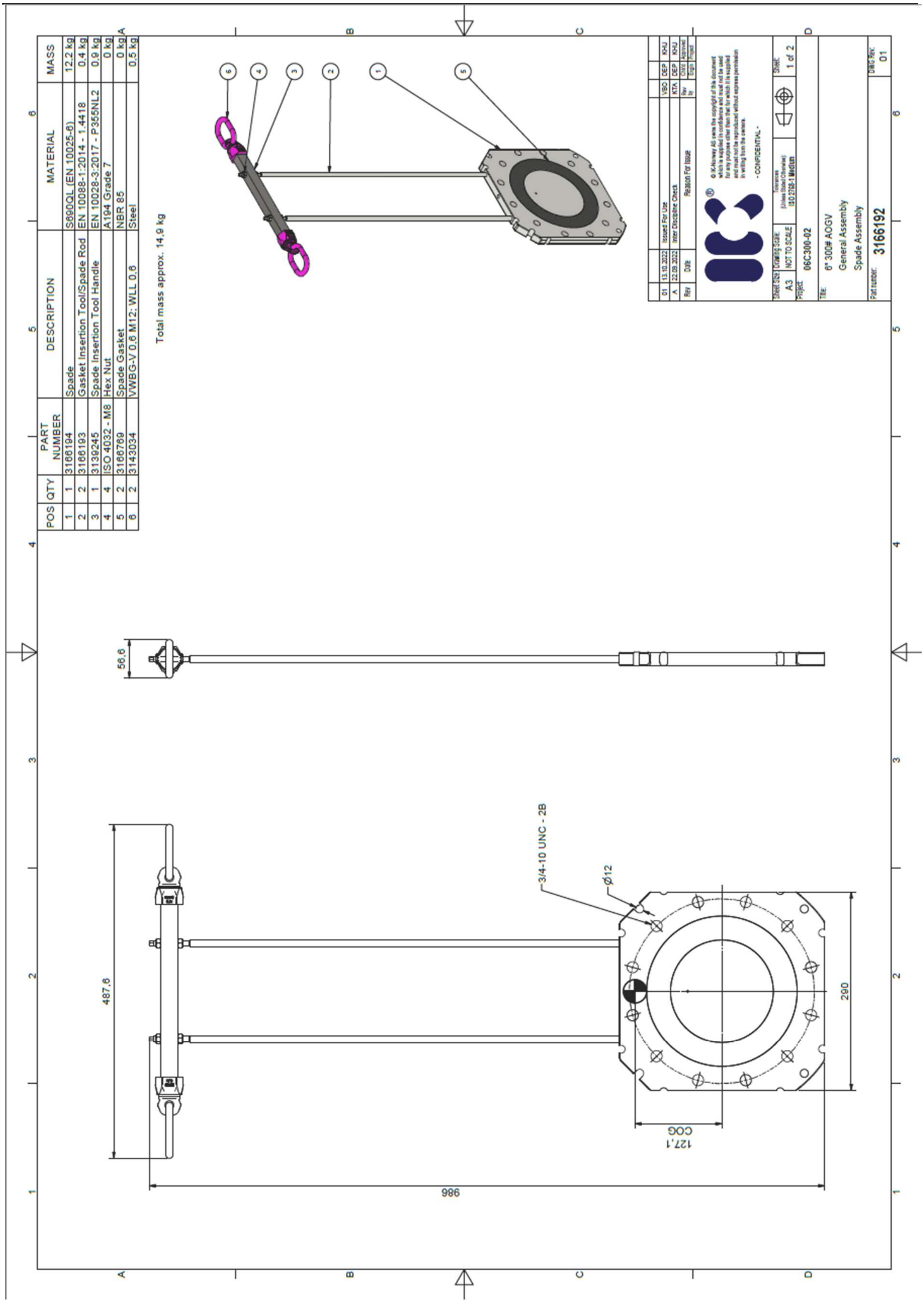
[61]

K Stuffing Box Drawing



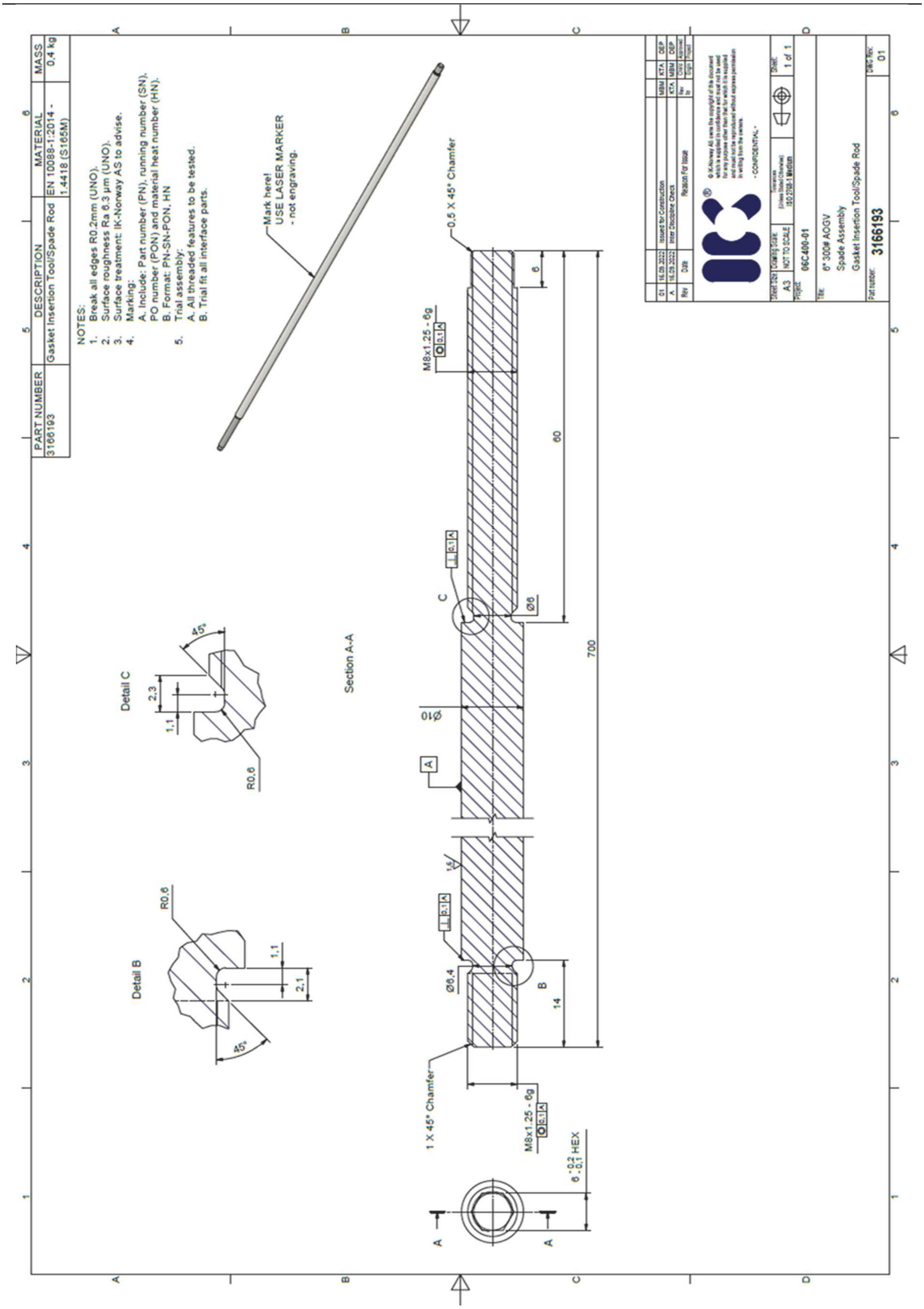
[62]

L Spade Assembly Drawing



[63]

M Rod Drawing



[64]